AD-A048 062 NAVAL RESEARCH LAB WASHINGTON D C SHOCK AND VIBRATIO--ETC F/G 20/11
THE SHOCK AND VIBRATION DIGEST. VOLUME 9, NUMBER 12, DECEMBER 7--ETC(U)
DEC 77 R L ESHLEMAN, J N ESHLEMAN

UNCLASSIFIED NL

1 OF 2
ADA048 062

AD A048062

(2)

(6)

THE SHOCK AND VIBRAT[illegible] DIGEST, Volum[illegible]

Number 12, Decem[illegible]

A PUBLICATION OF
THE SHOCK AND VIBRATION
INFORMATION CENTER
NAVAL RESEARCH LABORATORY
WASHINGTON, D.C.

D D [illegible]

JAN [illegible]

FOR RECORD AND ANNOUNCEMENT ONLY

NOT TO BE REPRODUCED FOR SALE
OR FREE DISTRIBUTION

may
be purchased from

Shock and Vibration Information Center
Naval Research Laboratory, Code 6020
Washington, D. C. 20390

(10) Ronald L. Eshl[illegible]
Judith Nagle-Eshl[illegible]
Milda Tamulion[illegible]

OFFICE
OF THE

DIRECTOR NOTES

Each year I am impressed by the interest and enthusiasm of those attending the shock and vibration symposia. This was particularly true of the 48th Symposium held at the Von Braun Civic Center in Huntsville, Alabama this past October. The convention facilities were excellent. The technical program was well received. Most important, the favorable feedback from the participants on the usefulness of the Symposium has been very gratifying. Our host, the U.S. Army Missile Research and Development Command, provided outstanding support resulting in a highly successful meeting. Mr. James Daniel, MIRADCOM member of the Technical Advisory Group to SVIC, was responsible for the support requirements. He deserves high commendation and our deepest thanks.

Mr. Daniel was also Chairman of an exceptional opening session. Following a gracious welcome by Major General Charles F. Means, Commander of MIRADCOM, Dr. John L. McDaniel gave an inspiring keynote address. Dr. McDaniel recently joined Hughes Aircraft Company following his retirement as Deputy/Technical Director of MIRADCOM. The three invited speakers all gave outstanding presentations. Thanks are due to Colonel John L. Cannon, Commander of the U.S. Army Waterways Experiment Station; to Mr. E.J. Kolb, Principal Technical Information Officer for the Army from the U.S. Army Materiel Development and Readiness Command; and to Dr. Robert M. Hamilton of the U.S. Geological Survey.

With the passing of the 48th Symposium and with this issue of the DIGEST, another year is completed. SVIC looks forward to continuing service to the technical community. For now, I extend my sincere best wishes to all our readers for a happy and prosperous holiday season.

H.C.P.

DISTRIBUTION/AVAILABILITY CODES
SPECIAL
A 21

EDITORS RATTLE SPACE

DECLINING ATTENDANCE AT TECHNICAL MEETINGS

It seems to me that attendance at technical meetings is continuing to decline. In fact, it is not uncommon for the authors in a session to talk only to each other! In some conferences the speakers and session chairmen outnumber the audience. The reasons for this decline, I believe, are overpublication, presentation of irrelevant material, reduced technical motivation, and economics.

In editorials during this past year I have stressed the problem of overpublication: much of the technical literature is no more than a rehash of previous work. Basic technology has been well established in many engineering areas; continued republication of the same material in slightly altered form does not motivate people to attend meetings. Publication of irrelevant material -- whether it is a super-technical treatise, technology with no practical application, or solutions to trivial problems -- is next to worthless.

Motivation for seeking new technology seems to be declining for two reasons. The first: engineers have discovered that they can solve given problems with the technical expertise they already have. The second reason also has to do with the engineer: a number of technically ill-equipped practicing engineers either are not aware of their problem or are not motivated to seek help -- until they have trouble.

Economics plays a big role in attendance at meetings. In a growing economy employers are more willing to spend money on "frills" such as technical meetings. When new technology is required to develop a product, employers are willing to support the learning process. However, in the absence of new development, they are reluctant to look at the long term education of an employee. It is unfortunate when an engineer has to perform at an optimum level on short notice -- the costs involved more often than not exceed those that would have been expended in a long-term educational program.

In order to stop the decline in meeting attendance, I believe we are going to have to select more carefully the material that is presented. This can be accomplished in part by establishing guidelines for the material to be presented at meetings and by upholding those guidelines in the review process. In addition, the effort to educate employers and employees (engineers) about the value of long-term learning should be intensified!

R.L.E.

SHIPBOARD SHOCK ENVIRONMENT AND ITS MEASUREMENT

M. W. Oleson and R. O. Belsheim*

Abstract - This paper contains a review and description of ship shock environments caused by adjacent explosions. The responses of a ship's structure and equipment to these environments are also discussed.

The ability to develop a wholly satisfactory characterization of the mechanical shock environment produced by a non-contact underwater explosion in proximity to a surface ship is limited. The shock environment of equipment is influenced by several factors. In addition to the obvious effects of charge size and distance of the explosion from the ship (attack geometry), the other effects are the response of the ship's structure to underwater shock and the dynamic properties of the equipment and the ship's structure. Reasonable experimental procedures for characterizing the free-field shock wave [1] and resulting motions of the ship's structure exist. A completely satisfactory characterization of the dynamic properties of the ship's structure has not yet been formulated.

SHOCK ENVIRONMENT

About 50 percent of the energy in an underwater explosion is propagated outward from the point of detonation in the form of an underwater shock wave. To an observer at some stationary point in the water, this wave, traveling at almost 5,000 feet per second, would appear as a pressure transient with an exponential waveshape and would be of very short duration.

The remaining energy released by the explosion is contained in a highly compressed gas bubble at the point of detonation. The bubble expands and contracts in an oscillatory fashion as it floats upward and ultimately vents at the water surface. Two effects are associated with bubble pulsation: first, water in the vicinity of the gas bubble undergoes oscillatory motions as a result of volume displacement; second, shock waves of successively lower energy are generated as the bubble contracts. Although these later effects may be important in overall ship strength computations, they are not usually significant factors of inboard shock environment. Inboard shock environment is affected by the size and position of the explosive with respect to the ship, however.

Conditions that would result in lethal hull damage are beyond the scope of this article, which concentrates on the effects of small conventional explosives at close range, large nuclear charges at long range, and various combinations of the two (Fig. 1).

Superficial equivalence between attack geometries might be based on pressure-time impulse at the target. The free-field impulse varies inversely with the distance of the explosive from the ship. The effective impulse at points near the water surface is also influenced by a surface-reflected rarefaction wave, which, in combination with the direct pressure wave, abruptly reduces the net pressure to zero. For large charges, the pressure decay is comparatively slow, and at shallow attack angles the surface cut-off effects a reduction in the free-field impulse.

Loading on a target ship varies as a function of the attack geometry. With small charges close to the ship, target loading tends to be localized -- decreasing in severity at points on the hull away from the point closest to the charge. With very large charges much farther from the ship, the shock wave is more nearly planar, and all points on the hull are loaded almost equally.

The energy in the shock wave that is transferred to the ship's hull is initially manifest as kinetic energy of motion. As the ship begins to move, restraining forces come into play. In the horizontal direction, motion is restrainted by the inertia of the water on the far side of the ship. In addition, an impulsive load of opposite phase occurs when the pressure wave has propagated to the far side. In the vertical direction, motion is restrained by gravity plus unbalanced air pressure due to cavitation beneath the ship's bottom as it moves upward in response to the initial velocity. Response in the vertical direction is

*Naval Research Laboratory, Washington, D.C. 20375

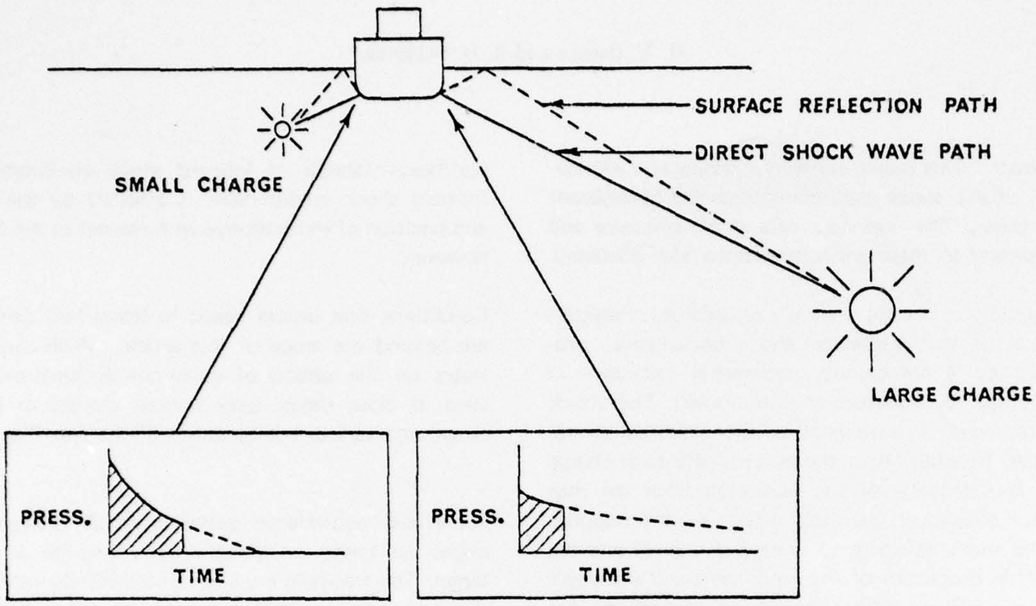

Figure 1. Pressure Time Impulses

usually greater, by a factor of two to four, than that in the horizontal direction.

SHIP RESPONSE

If a surface ship were truly rigid -- that is, without structural flexibility or structural modes -- it would respond to underwater shock as shown in Figure 2. An impulsive load from the shock wave would produce an initial-peak sawtooth velocity waveshape and a near-parabolic displacement waveshape.

Of course, surface ship's structures do have structural modes. The mode frequencies for a typical large ship range upward from one Hz, which is the first beam whipping mode [2]. Part of the kinetic energy initially transferred to the ship's bottom is manifest as rigid body motion; the remainder cause oscillatory distortions of the ship's structure at the various structural modes.

A two-mode representation of the midship's cross section amidships of a surface ship is shown in Figure 3. The response of each mass to an impulsive load applied to the lower mass (M1) could be represented by superimposing an oscillatory component on a sawtooth velocity waveshape similar to that of the rigid mass. In other words, some portion of the incident energy has been coupled to a non-rigid mode.

The response motions indicated for this simple model are not inconsistent with experimental measurements taken during shock tests. The waveshape of the lower mass (M1) is characteristic of waveshapes taken in the hold region of surface ships. The oscillatory motion of the upper mass (M2) is frequently seen at upper deck levels. Oscillatory motion in the hold region tends to be less obvious than that indicated in Figure 3, but spectral decomposition of actual records indicates that it is present in most cases.

This two-mass representation is of course a very simplified version of a ship's structure. Mass and elasticity in a ship are distributed in structural frame members, structural hull and deck plating, and attached machinery. Although the resulting modes

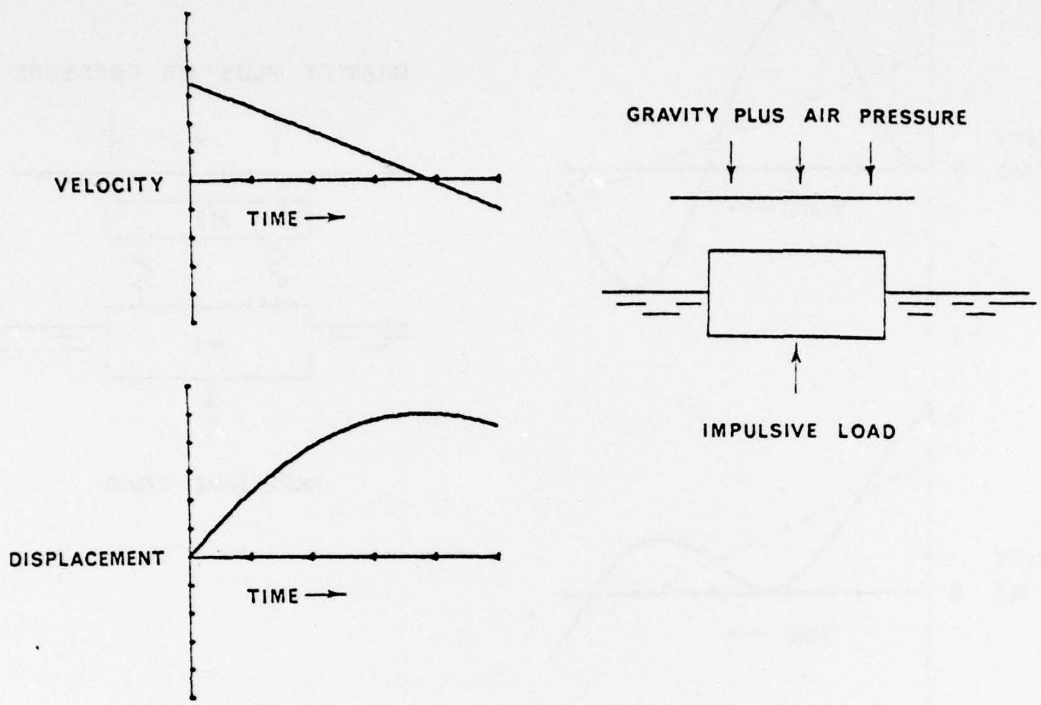

Figure 2. Response of a Rigid Structure to Underwater Shock

and their corresponding mode shapes are difficult to define both analytically and experimentally, experimental data tend to show at least one dominant lower frequency mode amidships in most surface vessels. It generally is in the range of 10 to 30 Hz and tends to have a nodal line spacing comparable to the beam of the ship.

Figure 4 is a structural schematic diagram of the cross section of amidships of an 18,000 ton combat support ship. Shock tests were conducted by placing a large conventional charge off the starboard beam. The response of the ship's structure was measured at port, centerline, and starboard positions below the main deck and at centerline positions above the main deck.

The velocity waveshapes shown in Figure 5 are positioned in approximately the same physical way as the gages in the ship. They show the first 100 milliseconds of response motion. Note the comparatively steep leading edges of the velocity waveshapes in the hold region. Note also that the initial steep rise is successively delayed at the centerline and port positions. The delay times correspond to the propagation time of the shock wave as it passed below the ship's hull. With respect to the transit time of the shock wave, therefore, the hold region was dynamically flexible.

As shock energy was propagated upward in the ship's structure, higher frequency motion components were attenuated by structural modes of the ship, and

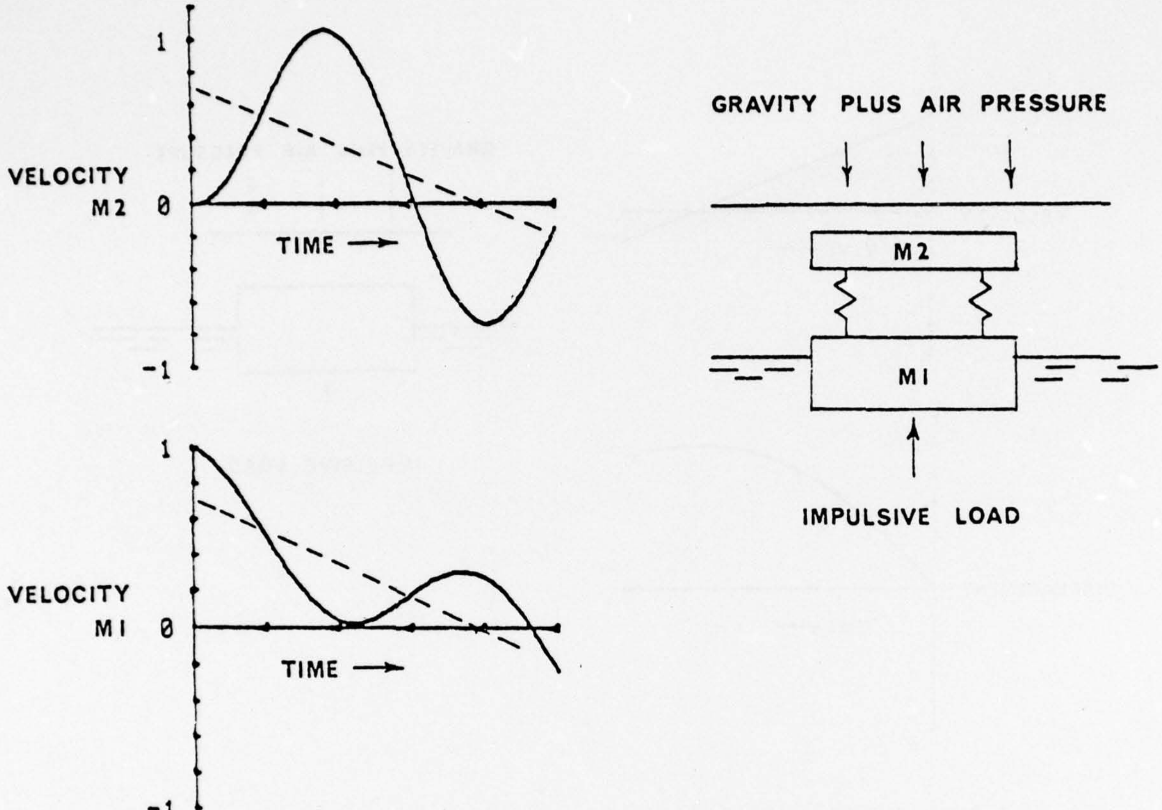

Figure 3. Two-Mode Representation of the Cross Section Amidships of a Surface Ship

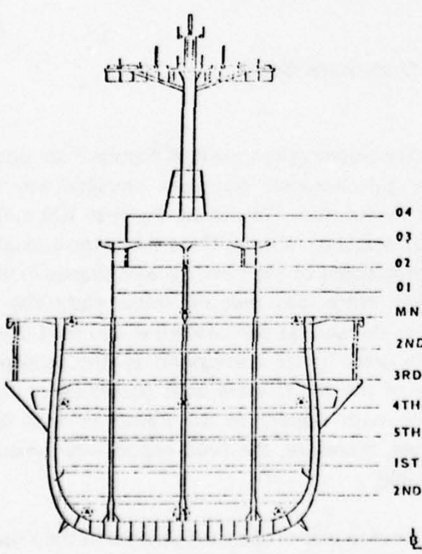

Figure 4. Structural Schematic Diagram of a Cross Section Amidships of an 18,000 Ton Combat Support Ship

responses at lower frequency modes became more prominent. As a matter of fact, if only the centerline gages are considered, it would not be difficult to justify a simple two-mode representation of the structural response of the ship. The upward velocity maximum measured at the 02 level and at the mast positions is approximately out of phase with a less obvious oscillatory component measured in the hold. The inadequacy of a two-mode representation would become evident if it were used to account for velocity waveshapes at port and starboard gage positions.

High frequency components of structural motion are most evident in the hold, as might be expected. Physically, this region of the ship is most affected by the incident shock wave. The shock wave loading is potentially capable of driving structural modes from the lowest value to the highest. However, the higher frequency modes tend to have comparatively closely

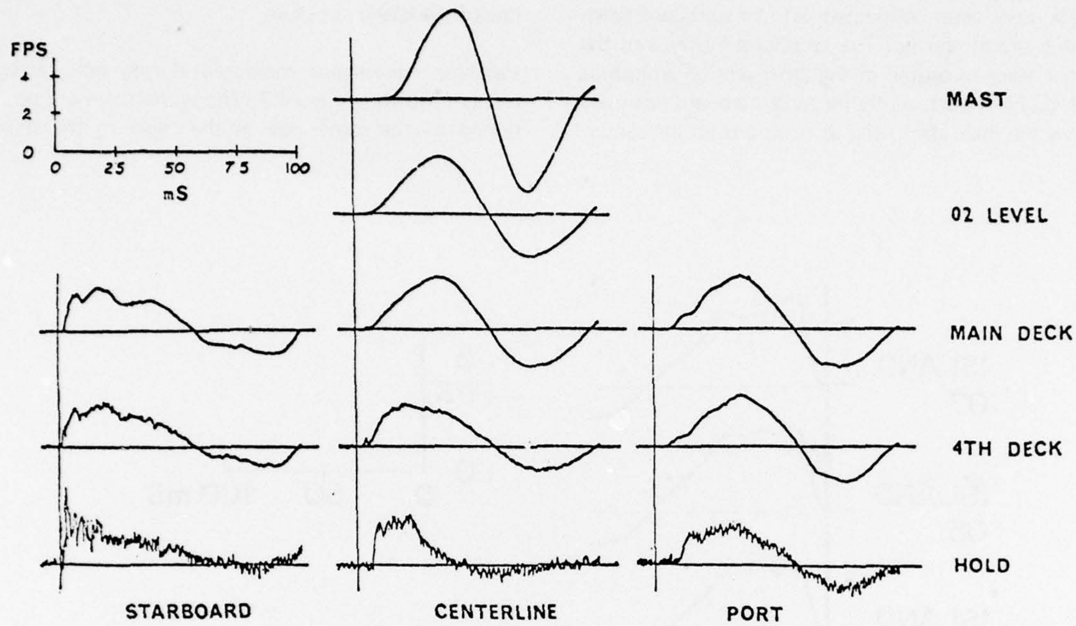

Figure 5. Velocity Waveshapes of a Ship's Response to Shock Tests

spaced nodal lines -- that is, they involve small regions of the ship -- and energy in these modes is not readily propagated over large regions of the ship. Thus, at instrumented positions on upper deck levels, the higher frequency modes were less vigorously excited, and higher frequency motion components appear progressively attenuated.

In a sense, a ship's structure can be viewed as a mechanical low-pass filter. Much equipment would experience a less severe shock environment at the upper deck levels.

Figure 6 is a structural schematic diagram of the cross section amidships of a 28,000 ton aircraft carrier of World War II vintage. It is only partially representative of modern carrier design. Typically, carrier design differs from that of smaller ships in several ways: the superstructure is displaced to one side; interior framing is interrupted at the main, or hanger deck, level; the cross section amidships is more nearly rectangular; and the multiple side tanks tend to increase the vertical stiffness of the port and starboard sides of the hull.

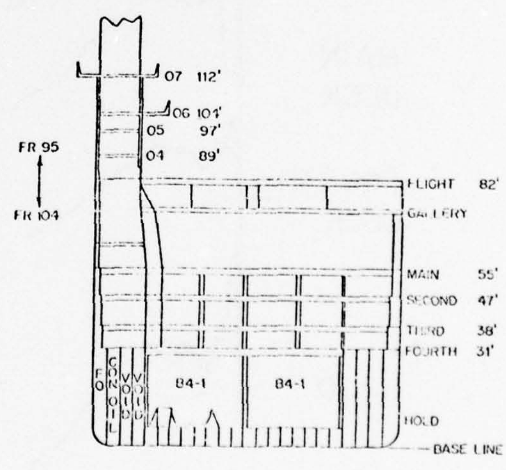

Figure 6. Structural Schematic Diagram of the Cross Section Amidships of a 28,000 Ton Aircraft Tanker

Shock tests were conducted off the starboard beam of this aircraft carrier. The structural motions of the carrier were measured in the cross section amidships with gages at port, centerline, and starboard positions below the main deck, and at positions up the centerline of the island structure.

Velocity waveshapes measured during one carrier test are shown in Figure 7. The waveshapes are positioned in the same way as the gages in the ship.

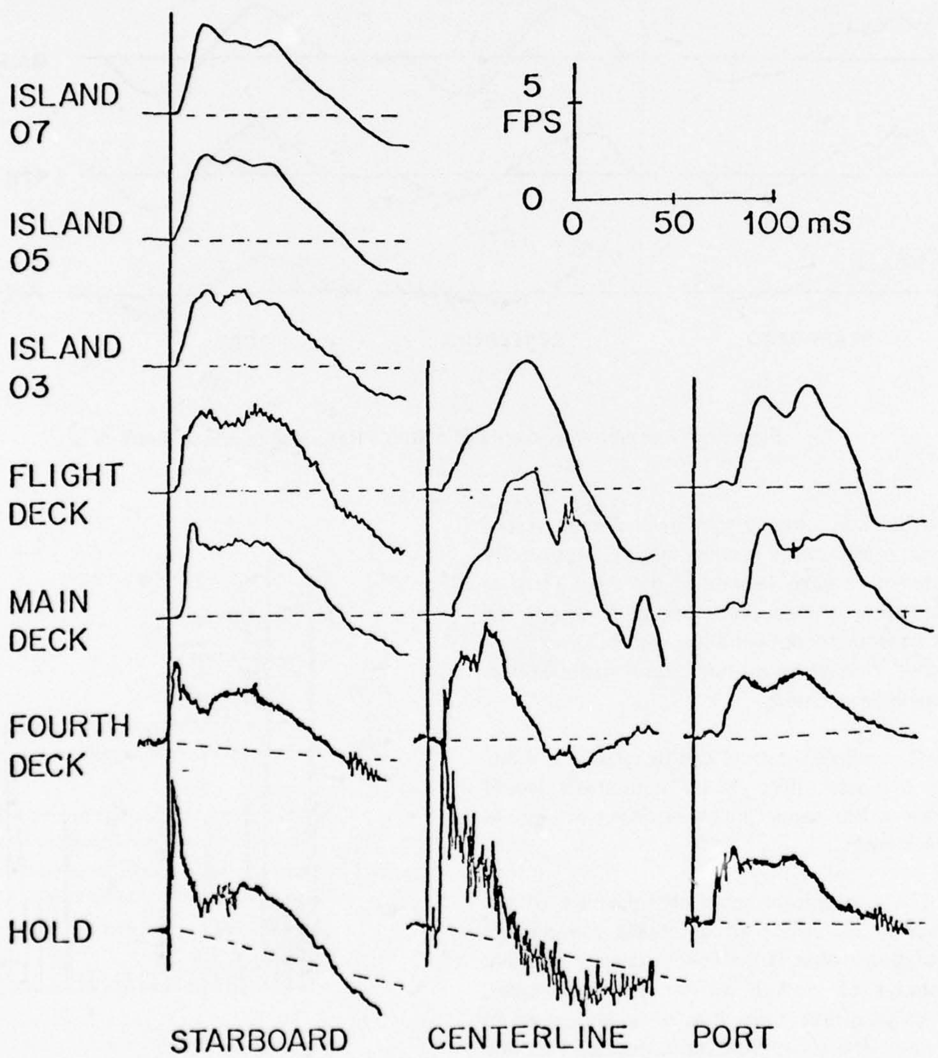

Figure 7. Velocity Responses of a Carrier to Shock Loading

The dominant features of these waveshapes are not unlike those of the smaller ship. Velocity waveshapes in the hold region have a steep leading edge and more high frequency motion than is evident at upper levels in the ship. At the centerline gage positions, motion at upper deck levels is almost sinusoidal, and would appear to be nearly out of phase with an oscillatory component of motion at the centerline gage position in the hold. This set of waveshapes has allowed an approximate experimental definition of a major structural mode of the ship.

At the flight deck level, the velocity waveshape at the centerline position is almost sinusoidal; however, the velocity waveshapes at the port and starboard edge positions might better be approximated as rectangles. The velocity waveshapes at positions on the main deck level are similar. The difference in waveshapes implies relative deflection of the centerline gage positions with respect to the port and starboard edge gage positions.

Recordings from each of three sets of gages were electronically combined and integrated to provide a time-history record of the relative deflection at deck centerline positions with respect to a line drawn between the two deck edge gage positions [3]. The time-history deflection records indicated sustained oscillatory deflections with frequencies in the neighborhood of 10 to 15 Hz.

This analysis, and other supporting data lead to the conclusion that the structural response mode involves vertical oscillation of the centerline region with respect to the sides. Such a mode might involve a significant fraction of the total mass of the ship. Because the mode was strongly excited by the incident shock, it would also contain a significant portion of the incident shock energy.

Another consequence concerns the shock environment of shipboard equipment. In most cases, damage to equipment can be related to shock-induced distortions within the equipment at natural frequencies of the equipment itself [4]. Energy to produce such distortions must necessarily be introduced via the ship structure. If a structural mode and a natural frequency of a piece of equipment were approximately the same, and if the structural mode contained significant energy, it would seem probable that the equipment damage would be enhanced.

A distinguishing feature among combatant ships of various classes regarding the shock environment of inboard equipment is associated with structural modes. In principle structural modes can be calculated. In practice such calculation has not proven adequate, and the characterization of inboard shock environment for various classes of combatant ship has been based on experimental data taken during ship-shock tests.

A broader concern is not with structural response motions of the ship alone but rather relates to the potential for damage to vital equipment.

A technological objective is to quantify and define, to engineering accuracy, shock-induced mechanical stress effects on arbitrary shipboard equipment. The engineering calculation as it pertains to the equipment is not especially difficult -- provided only an appropriate input motion (or design equivalent) can be stipulated at the equipment's foundation.

It is tempting to assume that an input motion could easily be synthesized by using a motion characteristic of the structural response of the ship. Unfortunately, such a formulation has limited validity.

The susceptibility of shipboard equipment to damage from shock is a function of natural frequencies of the equipment and of the ship's structure as well as the severity of the shock. If frequencies of the equipment are relatively high compared to the frequencies of contiguous motions of the ship's structure, stresses within the equipment can sometimes be approximated by using the weight of the equipment and measuring a peak acceleration value at the equipment foundation. Conversely, if equipment frequencies are relatively low -- a situation encountered with shock-mounted equipment -- stresses within the equipment can sometimes be approximated on the basis of excursions appropriate to the equipment foundation. For equipment whose structural frequencies are in the same general range as those of the structural modes of the ship, no single or simple parameter suitably characterizes the effective severity of the shock environment on the equipment.

In general, shock-induced response motions of installed equipment cause corresponding reaction forces at the equipment foundation. These reaction forces, in turn, tend to modify the input motion at the equipment foundation over that which would be observed were the equipment not in place -- usually in such a way as to reduce the response motions and corresponding stresses in the equipment [5].

In Figure 8, a measured shock velocity response has been transformed from the time domain to the frequency domain with a shock spectrum analysis. The ordinate is a measure of the response that a simple mechanical oscillator would exhibit at each frequency along the absicissa axis.

The velocity record from which this figure was derived was measured at the foundation of an 8,000 lb. mechanical mass-spring assembly attached to heavy deck plating on the third deck of a cruiser. The vertical arrow indicates the fundamental frequency of the mass-spring system, 47 Hz.

It can be shown that the maximum stress in the mass-spring system was directly related to the shock spectrum value measured at its natural frequency [6]. Yet the shock spectrum value at this frequency is greatly depressed with respect to the value at other frequencies. In effect, the assembly has reacted on foundation structure of the ship in such a way as to lessen the effect of the shock. The mass-spring system is artificial, however, because it was designed and installed to demonstrate the effect of structural interaction.

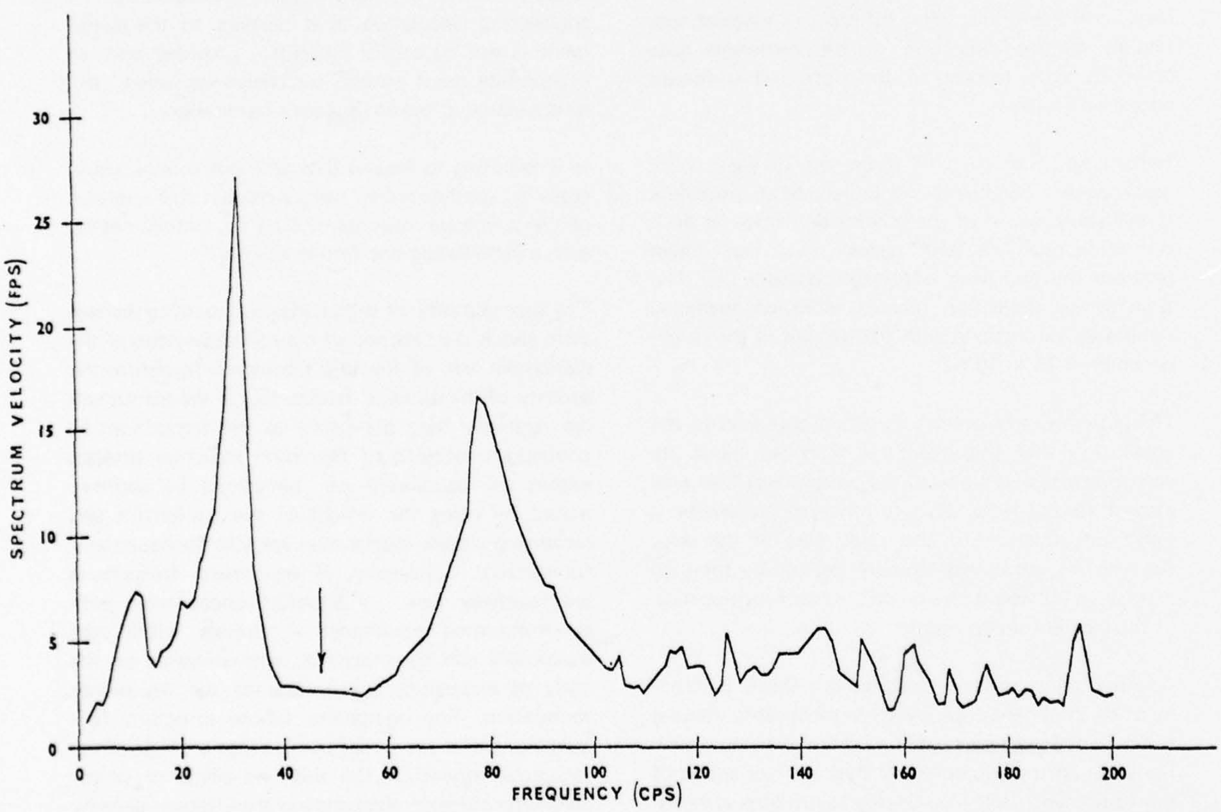

Figure 8. Shock Velocity Response Spectrum

The spectrum in Figure 9 was obtained from a velocity response measured at the foundation of an actual service turbo generator of a ship (SSTG). The SSTG weighed 33,000 pounds and was attached to the cruiser hull by vertical pipe stanchions. The SSTG installation exhibited a single dominant natural frequency at almost 30 Hz. Again there is clear evidence of structural reaction on the ship -- an obvious depression of the shock spectrum occurring at the natural frequency of the installed equipment.

In general, the effect of structural interaction is determined by modal frequencies and modal weights of both the equipment and the adjacent ship structure. Lightweight equipment attached to a heavy ship structure would probably cause little modification of the structural response of the unloaded ship.

Coversely, comparatively heavy equipment could be expected to produce a greater mitigation influence on its own environment.

The need to account for such structural interaction poses a substantial complication in the effort to characterize shock environments of ships. Structural dynamics as well as structural motion of the ship must be characterized.

A similar complication in electrical network analysis is readily reduced by means of Thevenin's theorem -- one of several network theorems that apply to linear systems. On the basis of this theorem, a complicated electrical network of active sources and passive circuit components can frequently be represented by a single equivalent source and a single equivalent circuit impedance. A similar representation of a ship structure might involve an equivalent velocity-time history and an equivalent mechanical mobility at selected positions on the ship's structure. *Proposals to develop such a characterization have been made in* past years, but have not been implemented [7]. Indeed it is not clear that current technology is adequate to accomplish the task.

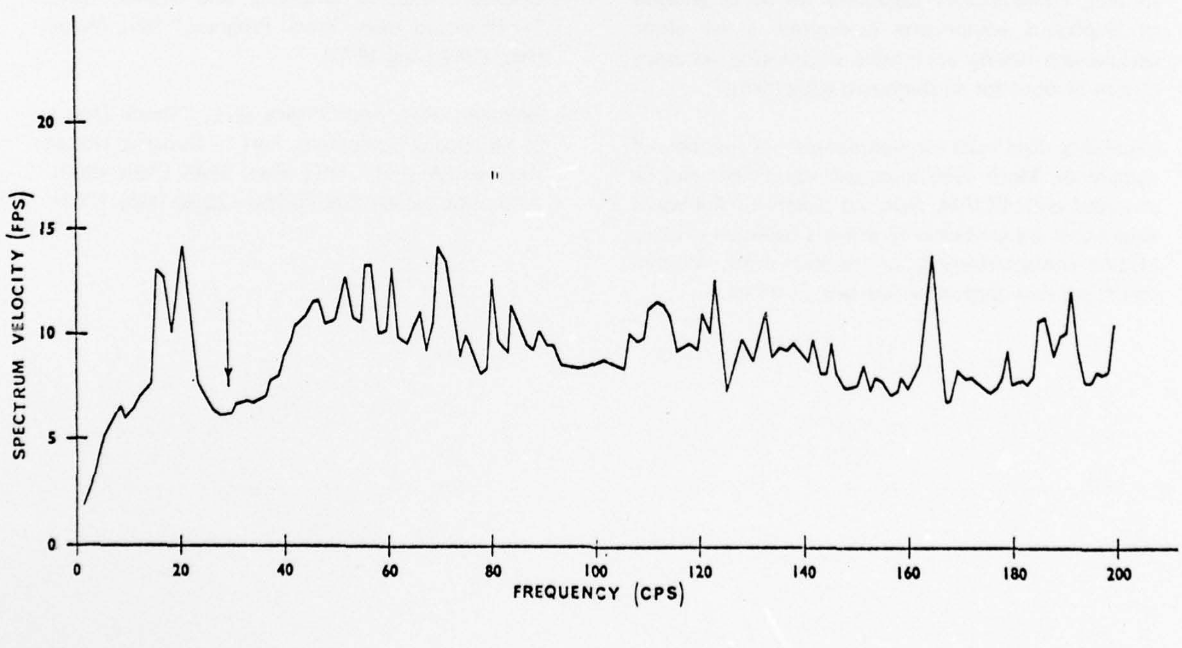

Figure 9. Velocity Response Spectrum at the Foundation of a Ship's Service Turbo Generator

Another approach to adequate characterization of the ship's structure is implicit in the Navy's Dynamic Design Analysis Method (DDAM). Numerical values used for DDAM calculations have been derived from motion measurements taken at the foundations of installed equipment [8]. Such measurements account for both structural interaction and basic structural response of the ship. Properly interpreted, the motion measurements are appropriate to other, similar installations.

DDAM in its present form, however, is not adequate for an engineering analysis of all classes of shipboard equipment. Any experimental measurement essentially characterizes the combined influences of structural response and structural dynamics of the ship and structural dynamics of the equipment. A large number of possible combinations exists, of course. The synthesis of a large number of experimental measurements has yielded a generalized design input for many of the more important combinations. But, for practical purposes, the existing data base is not extensive enough to characterize all combinations of engineering interest.

In fact, contemporary capability for shock analysis of shipboard equipments is limited; shock stress calculations having acceptable engineering accuracy cannot be done for all shipboard equipment.

Capability does exist for evaluating shock hardness of equipment. Much vital shipboard equipment can be analyzed with DDAM. Selected classes of shipboard equipment are amenable to analysis based on existing motion characterizations of the ship shock environment, and new approaches are being studied.

REFERENCES

1. Cole, R.H., Underwater Explosions, Princeton Univ. Press (1948). Reprinted, Dover (1965).

2. Vane, F., "Shock and Vibration in Ships," Ch. 46, Shock and Vibration Handbook, Harris, C.H. and Crede, C.E. (Eds.) McGraw-Hill (1961).

3. Oleson, M.W., "An Experimental Approach to Understanding Shock Response," Shock Vib. Bull, U.S. Naval Res. Lab., Proc., 40, Pt. 2, pp 1-19 (Dec 1969).

4. O'Hara, G.J., "Background for Mechanical Shock Design of Ships Systems," NRL Rep. 6267 (Mar 1965).

5. O'Hara, G.J., "Effect Upon Shock Spectra of the Dynamic Reaction of Structures," NRL Rep. 5236 (Dec 1958).

6. Blake, R.E. and Swick, E.S., "Dynamics of Linear Elastic Structures," NRL Rep. 4420 (Oct 1954).

7. Belsheim, R.O., O'Hara, G.J., and Oleson, M.W., "A Proposed Navy Shock Program," NRL Memo Rep. 2342 (Aug 1971).

8. Belsheim, R.O. and O'Hara, G.J., "Shock Design of Shipboard Equipment, Part 1 - Dynamic Design Analysis Method," NRL Rep. 5545 (Sept 1960). Also publ. as NAVSHIPS 250-423-30 (May 1961).

LITERATURE REVIEW

survey and analysis of the Shock and Vibration literature

The monthly Literature Review, a subjective critique and summary of the literature, consists of two to four review articles each month, 3,000 to 4,000 words in length. The purpose of this section is to present a "digest" of literature over a period of three years. Planned by the Technical Editor, this section provides the DIGEST reader with up-to-date insights into current technology in more than 150 topic areas. Review articles include technical information from articles, reports, and unpublished proceedings. Each article also contains a minor tutorial of the technical area under discussion, a survey and evaluation of the new literature, and recommendations. Review articles are written by experts in the shock and vibration field.

This issue of the DIGEST features a literature review on a new way to model mechanisms and machines by Dr. R.C. Winfrey. His article on the finite element method applied to the analysis of mechanisms and machines reflects a new way at looking at the problem.

Drs. Ross, Strickland and Sierakowski review experiments involving basic structural elements such as beams and plates subjected to blast loading. Responses and failures of these elements are described.

RESPONSE AND FAILURE OF SIMPLE STRUCTURAL ELEMENTS SUBJECTED TO BLAST LOADINGS

C.A. Ross*, W.S. Strickland**, and R.L. Sierakowski***

***Abstract** - This paper is a review of experiments involving basic structural elements such as beams, plates, and cylindrical shells that have been exposed to mild blast loadings. The response and subsequent failure of these structural elements are described in some detail.*

The response and failure of structural elements under dynamic loadings are complicated processes that are difficult to analyze. The responses of beams and plates to blast loadings are similar; the response of cylindrical shells tends to be much less predictable and more complicated. This paper describes the effects of mild blast loadings on these simple structural elements.

BEAMS

Aluminum beams, 0.0254 m wide, 0.454 m long, and 0.16 to 0.32 cm (0.063 - 0.125 in) thick were exposed to a fuel-air-explosive (FAE) device. This fuel-air device, which was used in all of the tests consists of a plate and beam test fixture fabricated from 2.54 cm steel plate and bolted to a concrete pad; a gas bag containing the fuel-air mixture is placed in series as shown in Figure 1. Polyurethene plastic is stretched over a waterpipe frame, and the assembly is sealed with plastic tape. A detonating charge of 100 gr of Data Sheet is placed at the end of the bag opposite the plate; the bag is filled with 0.91 kg of MAPP (methyl acetylene propadiene) gas and allowed to mix with air for ten minutes. Detonation of the Data Sheet creates a Chapman-Jouget wave as the fuel air mixture travels the length of the bag and impinges upon the test device. The device produces a wave of constant velocity and pressure; the reflected pressure on the test item can be varied, however, by changing the distance from the end of the bag to the test item (D of Fig. 1). Initial measurements were made on a thick non-deforming plate instrumented with piezoelectric transducers for recording peak pressure versus position for various distances between the bag and the test fixture. As a check, pressure was measured around the outside of the test section during both the plate and beam tests. Pressure and impulse data reported herein are based on pressure-time histories recorded on the flat non-deforming plate.

Both ends of the beams were held fixed against rotation and deflection. The load was applied normal to the 2.54-cm beam width by placing the bag in series with the test stand (see Fig. 1). Pressure was also measured on 4.90-cm thick steel beams fixed as shown in Figure 2. Deflection-time histories for

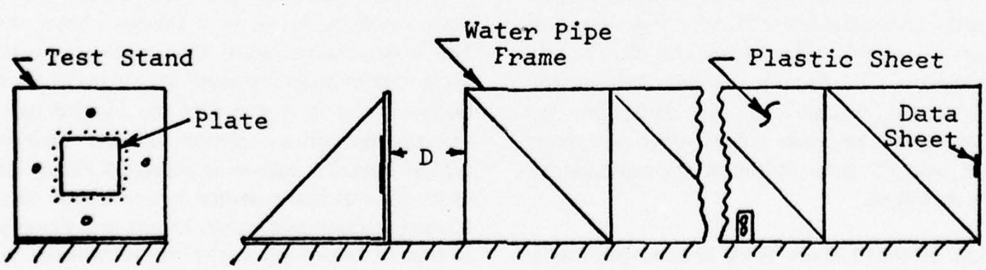

Figure 1. Gas Bag and Plate Test Fixture in Place

*Professor, University of Florida Graduate Engineering Center, Eglin AFB, FL 32542
**Project Engineer, USAF Armament Lab., Eglin AFB, FL 32542
***Professor, Engineering Sciences Dept., University of Florida, Gainesville, FL 32611

PRECEDING PAGE BLANK-NOT FILMED

the beam were obtained by placing a lined grid to the side and behind the beam (see Fig. 3) and using a high-speed camera to record the deflection.

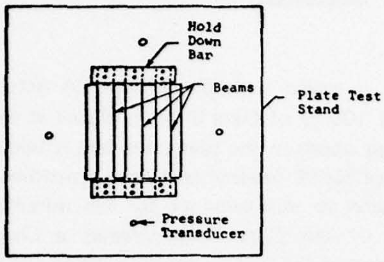

Figure 2. Beam Test Specimen Bolted in Place

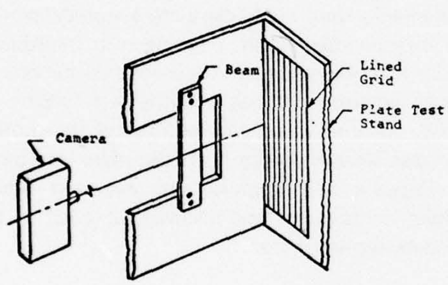

Figure 3. Test Fixture for Recording Time History of Beam Deflection

The responses of the beams can be separated according to beam thickness: with thick beams, large permanent deflection occurs with little or no rebound; with thinner beams, large deflections take place with considerable rebound. The response and failure of the thick beams (0.32 cm) can involve: (a) permanent deformation without failure; (b) failure at some critical load and deflection; (c) failure during the response mode before maximum deflection; and (d) shear failure at the edges before deformation begins.

Deflection of the 0.32-cm thick beams occurs as a traveling hinge motion (see Fig. 4a, b), which continues until the motion reaches the midpoint of the beam. If the loading is sufficient, failure can occur at the fixed ends; with smaller loads, some elastic rebound occurs, and the beam is permanently deformed (see Fig. 4c). Apparently, for a given beam a critical load exists at which maximum deflection for failure occurs. For all beams tested, failure occurred at one of the fixed ends. If the load was increased beyond the critical load where failure and maximum deflection are coincident, failure occurred at the fixed ends during the initial hinge motion. Continued increases in the loading could cause shear failure at the fixed ends before any noticeable deformation took place [1]. It would appear that the failure mode for the thicker beams changes from a tensile failure to a shear failure with increases in loading at constant thickness.

Based on tests on aluminum beams [2] the traveling hinge velocity -- approximately 3,000 m/sec -- indicates a shear wave. The tensile to shear failure transition can be explained with a critical shear particle velocity concept. The initial transverse velocities of the beam were calculated from known applied impulse values and compared to the critical shear particle velocity for the beam material [3]. The initial transverse velocity exceeded the critical shear particle velocity in each case of shear failure at the fixed ends.

The thinner beam elements show considerable rebound if failure does not occur. Film clips obtained according to the scheme in Figure 3 showed that the beams begin to deflect with a traveling hinge motion (see Fig. 4a, b). The hinge motion continues to the midpoint of the beam (Fig. 4c) at which time a reflection of the waves occurs and the beam begins to rebound in the same shape as that of the initial deformation (note flat midsection of Fig. 4e). If the beam fails, failure occurs when the two traveling hinges reach the midpoint of the beam. If failure does not occur, rebound continues toward the initial position of the beam in a traveling hinge motion. The plastic deformation that occurred during the initial deformation increased the length of the beam, however, and it is therefore too long to pass back through its initial position without buckling. A typical buckling pattern is shown in Figure 4f. The beam thus oscillates several times through its underformed position and comes to rest in a shape similar to that of Figure 4h. As the load is increased to some critical value, failure occurs in a deflected mode similar to thick beam failure. Continued load increases beyond this critical value ultimately produce the failure described for thicker beams.

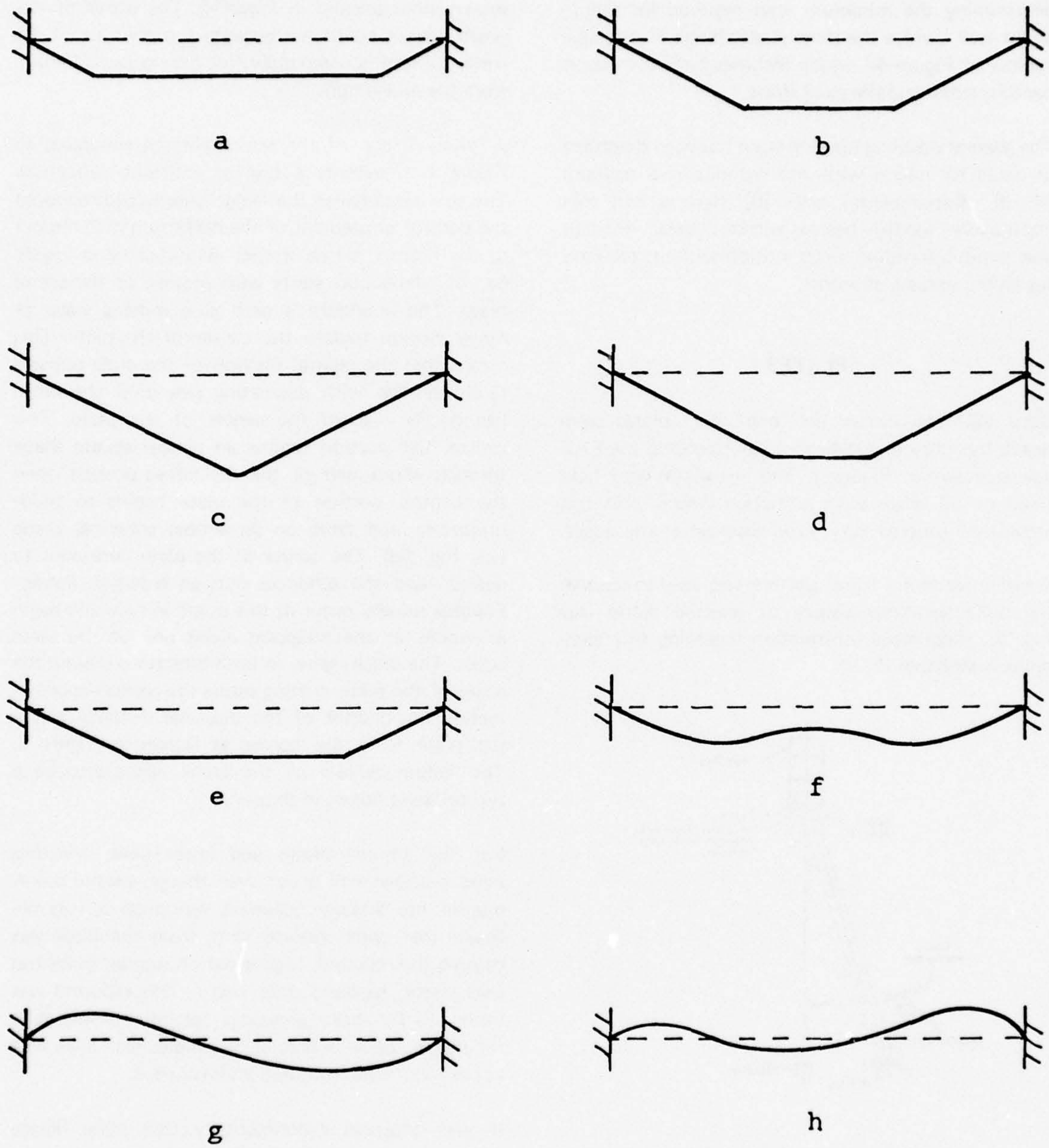

Figure 4. Typical Responses of Beam Elements

Based on these experimental results, a method for determining the minimum load required for failure might well involve the three plastic hinge mode shape typical of Figure 4c, which includes both the plastic bending stress and the axial stress.

The general traveling hinge motion has been described in detail for beams with and without axial restraint [4, 5]. Experimental tests [6] have shown that impulsively loaded beams without axial restraint also exhibit traveling hinge motion without rebound for all thicknesses of beams.

PLATES

Both 2024 aluminum and mild steel plates were tested by subjecting 0.46 m square plates to the FAE device shown in Figure 1. The test plates were held fixed on all edges with a friction device; post test inspection showed very little slippage at the edges.

A reflective Moire fringe pattern was used to observe the deflection-time history of selected plates (see Fig. 5). Additional information regarding this technique is available [7, 8].

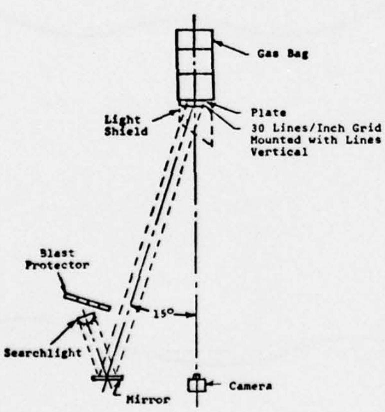

Figure 5. Schematic Diagram for Moire Pattern Experiments

Square plates 0.46 m (18 in.) wide and from 0.064 to 0.32 cm (0.025 - 0.125 in.) thick were tested at various blast pressures. The response modes for the aluminum and steel plates were similar. Reflective Moire fringe patterns photographed at 40,000 frames/sec revealed a hinge type motion for the deflection shown schematically in Figure 6. The actual photograph cannot be shown because the contrast of the fringe pattern is completely lost during reproduction from the movie film.

A single fringe of the Moire pattern simulated in Figure 6 represents a line of constant deflection. The spacing between the fringe lines would represent the density or gradient of the deflection with respect to the normal to the fringes. As indicated in Figure 6a, the deflection starts with motion of the entire plate. The boundary is seen as a moving wave or hinge motion toward the center of the plate. This means that the central portion of the plate remains relatively flat with decreasing size until the hinge has nearly reached the center of the plate. This central flat portion retains an almost square shape through about half of the delfection process; then the central portion of the plate begins to bulge uniformly and takes on an almost spherical shape (see Fig. 6d). The center of the plate continues to deflect, and the spherical portion enlarges slightly. Failures usually occur at this point in time and begin as cracks at the midpoint along one of the plate edges. The cracks grow in both directions around the edges of the plate, cutting across the corners approximately one-quarter of the diagonal distance across the plate from the corner as shown in Figure 7. The failure surface of the crack appears to be a typical sheet failure in tension.

For the thinner plates and lower peak pressures some rebound will occur even though plastic deformation has already occurred. Although it was believed that some reverse flow from the blast was causing the rebound, high-speed photography verifies that elastic rebound does occur. This rebound was found to be more prevalent for thin beams than for plates. Table 1 lists all the plates tested, as well as pertinent data measured and recorded.

It was observed experimentally that plate failure occurs in an almost fundamental mode for the loadings used in this study, even though the higher modes are active during the major portion of the deformation process. For more severe loadings failure begins as shear of the sheet at the edges before any deformation takes place. However, this type shear failure requires that the peak pressure be greater than that for any failure occurring from

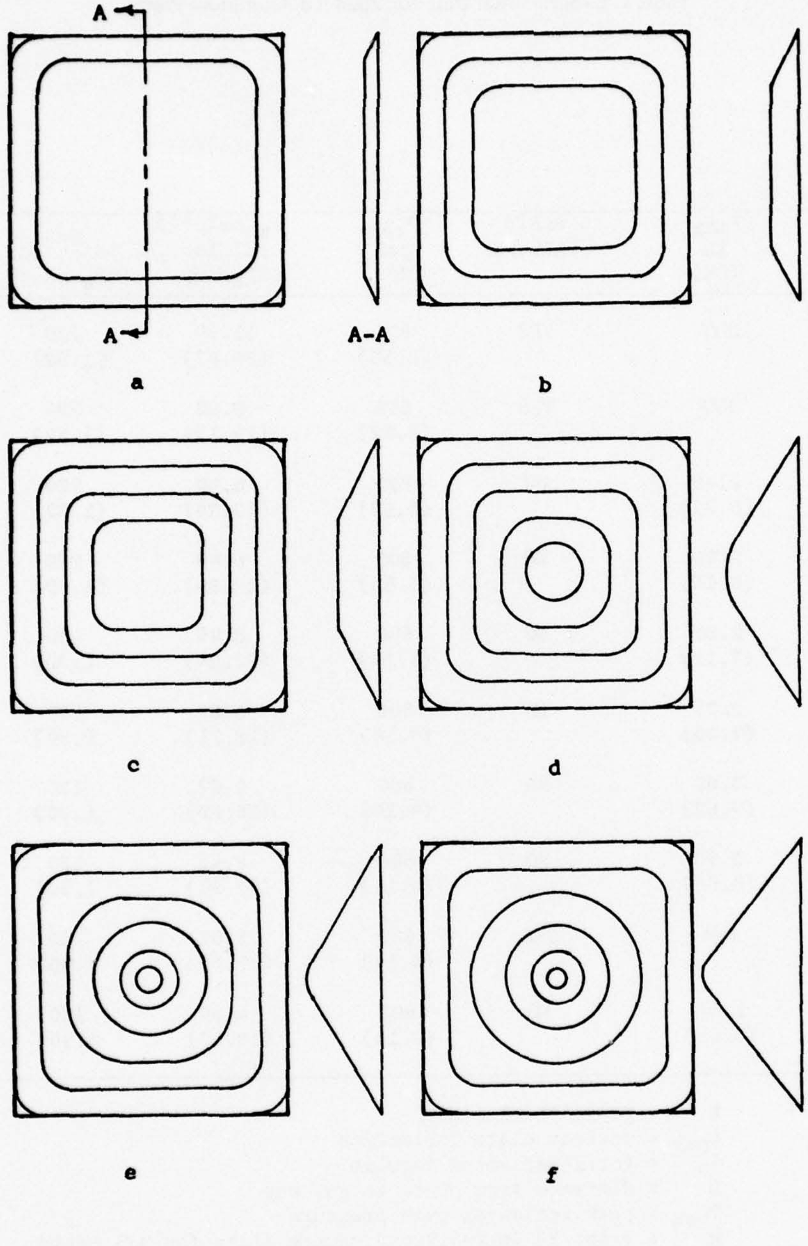

Figure 6. Sketch of Moire Fringe Patterns

The diagram to the right of each pattern represents the shape of the plate across centerline as shown at a typical section A-A of (a).

Table I. Experimental Data for 2024-T3 Aluminum Plates

h IN (CM)	w_{max} IN (CM)	PLATE FAILURE	P_{max} psi (MP_a)	($P_m/h \times 10^{-3}$) psi/in (MP_a/cm)	I_{max} psi-msec (MP_a-msec)	D FT. (M)
.071 (.180)	N/A	YES	800 (5.52)	11.30 (30.67)	220 (1.52)	0
.090 (.229)	N/A	YES	800 (5.52)	8.90 (24.17)	220 (1.52)	0
.125 (.318)	2.45 (6.22)	NO	800 (5.52)	6.40 (17.38)	220 (1.52)	0
.125 (.318)	2.60 (6.60)	NO	800 (5.52)	6.40 (17.38)	220 (1.52)	0
.071 (.180)	2.80 (7.11)	NO	600 (4.14)	8.45 (22.94)	130 (.90)	3 (.92)
.090 (.229)	2.75 (7.00)	NO	600 (4.14)	6.67 (18.11)	130 (.90)	3 (.92)
.063 (.160)	3.00 (7.62)	NO	600 (4.14)	9.52 (25.85)	130 (.90)	3 (.92)
.071 (.180)	2.70 (6.86)	NO	600 (4.14)	8.45 (22.94)	130 (.90)	3 (.92)
.050 (.127)	N/A	YES	600 (4.14)	12.00 (32.58)	130 (.90)	3 (.92)
.125 (.318)	1.98 (5.03)	NO	600 (4.14)	4.80 (13.03)	130 (.90)	3 (.92)

h = plate thickness
w_{max} = maximum plate deflection
I_m = total reflected impulse
D = distance from plate to gas bag
P_{max} = peak reflected over pressure
a = 9 in, 18 in (45.72cm) square plate for all tests
$p(t) = p_m(1-t/\tau)\exp(-\alpha t/\tau)$
α = decay constant
τ = positive pressure phase duration

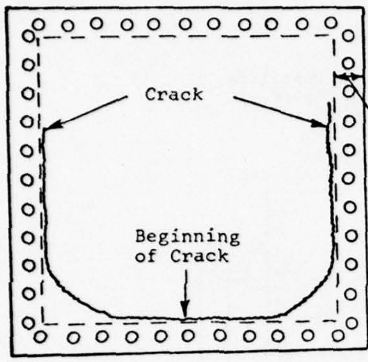

Figure 7. Typical Plate Failure

some deformation process.

The fact that, for the plates tested, failure occurs while the plate is in a fundamental mode shape supports the idea that an analysis could be based on a deformation to failure in a fundamental mode. The assumption that the energy to drive the plate to failure is independent of how it got there allows for a simple analysis. This analysis was applied to the plates tested with reasonable results for center point deflection for plates that did not fail [7]. This simple analysis also predicts failure at the midpoint of one edge when the ultimate strain, from the static stress-strain curve, is used as the failure criterion. Any strain rate effect or sensitivity is neglected, however. But, for the work hardened material and plates tested, this assumption is not unreasonable. Figure 8 is a comparison of analytical and experimental results.

CYLINDRICAL SHELLS

Aluminum cylindrical shells with fixed ends and subjected to both a fuel air explosive (FAE) and spherical pentolite (HE) device have been studied using the test fixture shown in Figure 9. For the FAE loading the plate test fixture was replaced with the cylindrical test fixture; for the HE loading the spherical charge was hung directly over the cylinder as shown in Figure 9.

For the cylinders tested, the internal diameter was held constant at 0.31 m. Length/diameter ratios of 1.89, 0.89, and 0.39 were matched with radius/thickness ratios of 188, 117, and 95 to give a nine point data base for comparison.

The coordinate system used in the description of the response and failure is shown in Figure 10. A circumferential mode number n and a longitudinal mode number m used in the expression for radial deflection w

$$w = \Sigma\Sigma w_{mn} \cos(n\theta) \sin(m\pi x/L)$$

have been used to describe the general response modes of the cylinders. The number of buckles per circumferential length for a given mode shape

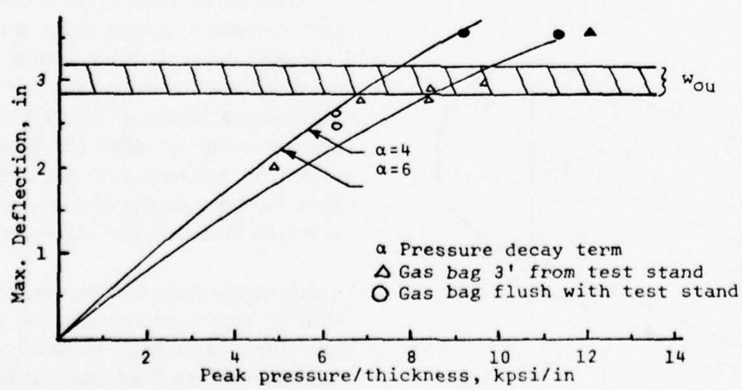

Figure 8. Maximum Plate Deflection Versus Pressure-to-Thickness Ratio
Plate failure range, w_{ou}, is based on an 18-20% ultimate strain.
Solid curves represent analytic results; solid symbols represent plate failure.

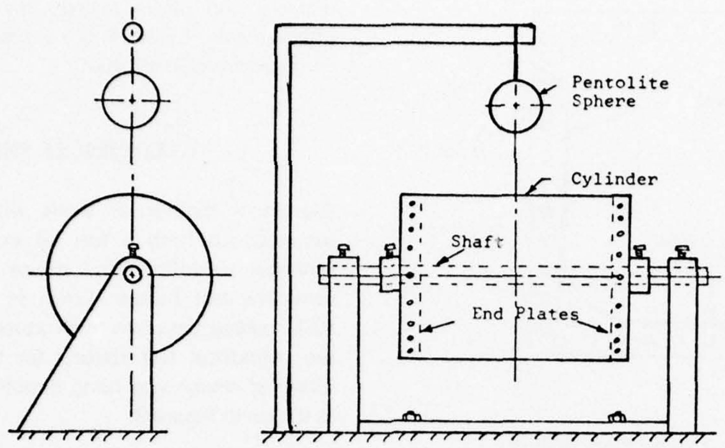

Figure 9. Cylindrical Shell Test Fixture

is n, and m is the number of half sine waves in the longitudinal direction. For all cases m was determined experimentally to be unity and is designated as the fundamental mode shape in the x direction. Experimentally n was calculated by dividing the number of buckles by the fraction of circumferential distance associated with the buckles. For example, for three buckled modes in only 25 percent of the cylinder (Fig. 11b), an experimental value of n = 12 is given. Attempts to photograph shell response were not successful, and all information from the FAE experiments was determined by post-test inspection. However, some high-speed photography have been obtained [9] for shock tube experiments on cylindrical shells.

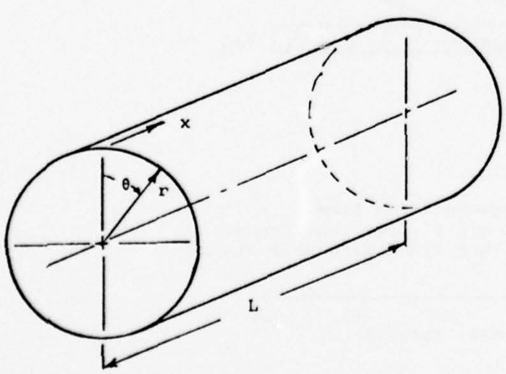

Figure 10. Coordinate System for Cylinders

With fixed end cylinders failure always began as a crack at the θ - 0 position of one of the fixed ends and spread circumferentially in two directions. The failure mode of those cylinders that failed was the same regardless of the response mode shape prior to the beginning of failure.

Experiments on cylinders loaded with planar blast waves have shown that buckling begins along the length of the cylinder at the θ - 0 position and spreads circumferentially to about the $\pm 45°$ positions. In all cases tested the average buckled area was only about 25 percent of the circumference. Deflection of the shell coincides with buckling and forms the fundamental mode in the x direction. The maximum deflection occurs along the mid-length and θ - 0 position as shown in Figure 11. In some cases circumferential buckling did not occur. The unbuckled cross section of Figure 11c is typical of this response, which is called the fundamental collapse mode. The occurrence of the various mode shapes before failure complicates the analysis and is unlike the results for beams and plates discussed previously.

Experimental data for the cylindrical shells (see Table 2) show that buckling can occur for a given shell and a given load. However, a change in the magnitude of the load can produce a collapse response (see data points 15 and 28 in Table 2). Such results suggest that, for a given cylindrical shell, some critical load provides a transition between the buckled and collapse patterns.

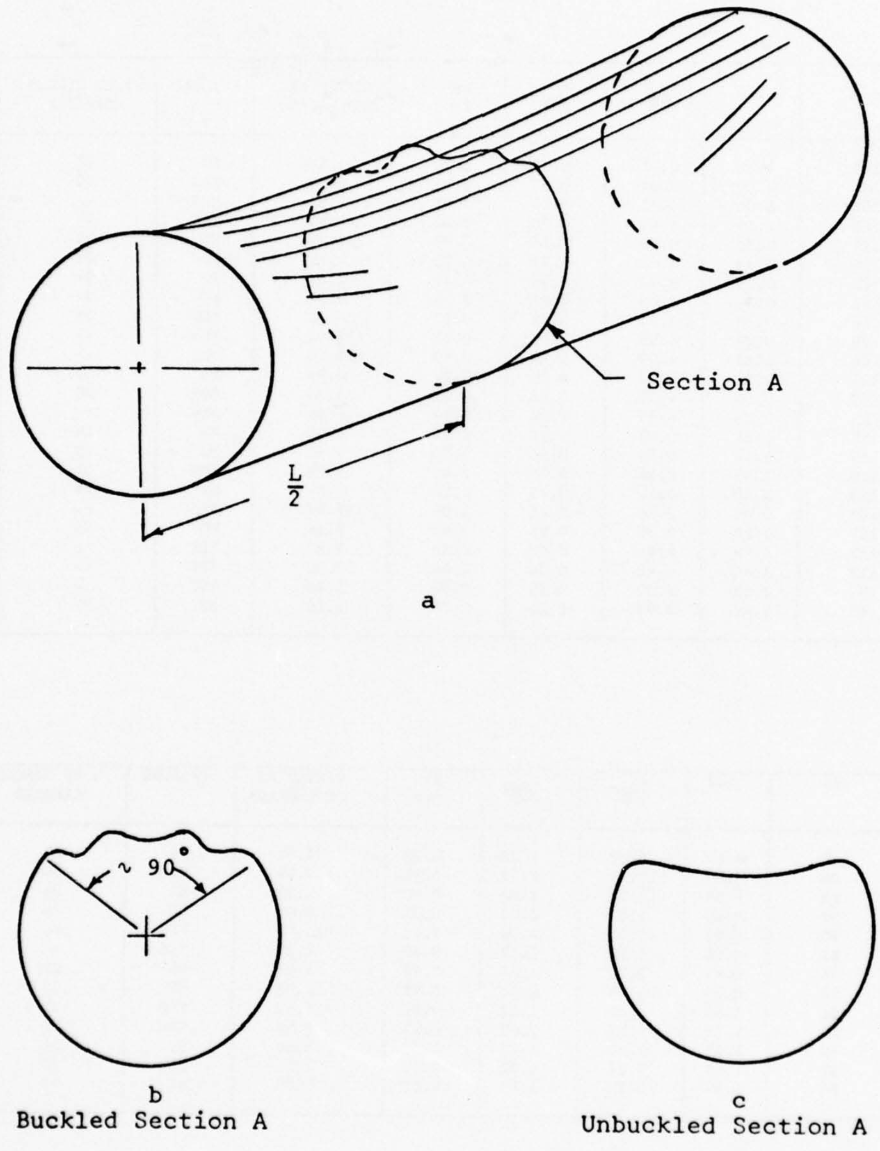

Figure 11. Buckling Patterns and Modes of Cylindrical Shells

Table 2. Summary of Cylindrical Shell Tests

Fuel-Air Device

DATA POINT	a/h	L/D	P_{max} $\theta=0$	I_{max} $\theta=0$	Δt $\theta=0$	CENTER PT. DEFLECTION	FAILURE	% OF CIRCUM. DAMAGED	n
1	188	0.39	2.41	0.52	0.93	1.63	NO	30	28
2	188	0.39	2.97	0.59	0.87	2.03	YES	30	38
3	188	0.39	3.65	0.67	0.83	2.87	YES	--	34
4	188	0.39	4.48	0.76	0.80	10.41	YES	--	36
5	117	0.39	4.48	0.76	0.80	2.24	NO	32	22
6	117	0.39	6.03	0.90	0.70	2.31	NO	34	25
7	95	0.39	6.03	0.90	0.70	1.02	NO	24	26
8	95	0.39	6.03	0.90	0.70	0.74	NO	24	26
9	188	0.89	2.97	0.59	0.87	6.10	YES	34	32
10	188	0.89	3.65	0.67	0.83	>11.43	YES	--	32
11	188	0.89	4.48	0.76	0.80	>11.43	YES	--	33
12	117	0.89	4.48	0.76	0.80	3.96	NO	32	26
13	117	0.89	6.90	0.83	0.75	6.05	YES	36	34
14	117	0.89	6.03	0.90	0.70	8.26	YES	--	22
15	95	0.89	6.03	0.90	0.70	3.05	NO	34	18
16	85	0.89	6.03	0.90	0.70	2.41	NO	32	19
17	188	1.89	1.28	0.37	1.40	4.32	YES	34	13
18	188	1.89	1.52	0.40	1.20	--	NO	34	13
19	188	1.89	1.86	0.45	1.00	>11.43	YES	--	19
20	117	1.89	2.97	0.59	0.87	3.66	NO	29	1
21	117	1.89	3.65	0.67	0.83	7.87	YES	--	1
22	117	1.89	4.48	0.76	0.80	7.32	YES	--	1
23	95	1.89	6.03	0.90	0.70	6.99	YES	--	10
24	85	1.89	6.03	0.90	0.70	6.50	NO	37	10

DATA POINT	a/h	L/D	P_{max} $\theta=0$	I_{max} $\theta=0$	Δt $\theta=0$	CENTER PT. DEFELCTION	FAILURE	% OF CIRCUM. DAMAGED	n
25	117	0.39	8.27	1.28	0.72	3.02	NO	25	30
26	85	0.39	23.44	2.33	0.40	2.64	NO	30	23
27	85	0.39	17.24	1.93	0.46	1.37	NO	34	25
28	95	0.89	5.52	1.03	0.91	0.86	NO	24	1
29	95	0.89	10.34	1.45	0.61	>11.43	YES	--	1
30	85	0.89	6.90	1.17	0.80	8.26	YES	--	1
31	85	0.89	8.27	1.31	0.72	1.60	NO	22	26
32	48	0.89	23.44	1.93	0.46	1.83	NO	24	11
33	95	1.89	5.52	1.03	0.91	>11.43	YES	32	1
34	95	1.69	5.52	1.03	0.91	5.56	YES	32	1
34	85	1.89	6.90	1.17	0.80	3.56	NO	32	1
36	85	1.89	8.27	1.31	0.72	7.21	YES	30	25
37	85	1.89	10.34	1.45	0.63	7.06	NO	33	25

P_{max} ($\theta=0$) = Normally reflected pressure in megapascals (MPa), (1.0MPa=145psi)
I_{max} ($\theta=0$) = Normally reflected impulse in megapascals- m sec (MPa-msec)
Center Pt. Deflection in centimeters (cm)
L/D = Length to diameter ratio
a/h = Radius to thickness ratio
n = Circumferential mode number
Δt = Positive pressure time phase in milliseconds (msec)

L/D values less than one have a decreasing mode number n for increasing thickness (see Table 2). This trend holds reasonably well for an L/D value of 0.89. For L/D values greater than one, however, the influence of change in thickness is less apparent.

Schuman [10] tested several sizes of cylinders subjected to blast loads but gave no response modes. His experimental results and those in Table 2 are generally in good agreement, but the analysis by Greenspon [11, 12] of Schuman's shells do not verify the results of Table 2. Another analysis [9] showed very good agreement for shells tested by the authors, but it predicted higher mode numbers than those in Table 2. The lack of correlation may be due to differences in the manner of loading and in calculated impulse values. Application of a modal type analysis [13] provided reasonable predictions of the final mode shape, but the method lacks appropriate criteria for predicting failure.

Determination of the load distribution for analysis is a major problem. A series of blast loads were imposed on a non-deforming cylinder using the loading methods described for the FAE and HE cylinder test. Experimental determination of the peak radial pressure distribution, as a function θ, approximated the expression

$$p_m = p_s + (p_r - p_s)\,(\cos\theta)^{1.8}$$

p_r and p_s are, respectively, the normal reflected pressure and the static pressure of a plane shock wave in air. Pressure measurements made by Lindberg [9] showed closer agreement to a $(\cos\theta)^2$ form. Time variations due to engulfment and decay were

$$p(\theta, t) = p_m[1 - (t - t_o)/\tau]\ \exp\,[-\alpha(t - t_o/\tau]$$

where t_o is the engulfment time based on shock wave speed, α is the approximate decay rate of the plane wave, and τ is the time of the positive pressure phase of the plane wave.

SUMMARY

Beams and plates respond similarly to blast loadings. Initially, at lower or mild blast loads, both beams and plates respond with a hinge type motion that traverses the entire width or length of the element; failure or rebound then follows. For more severe loadings failure occurs early in the initial hinge motion. For intense loadings failure occurs as complete edge shear before any deflection takes place. For all plates and beams tested, failure occurred at the fixed ends or edges.

Cylindrical shell response to blast loading tends to be much less predictable than that for plates and beams and is complicated by a buckling phenomena that is dependent upon loading characteristics as well as the geometric and material properties of the cylindrical shell. In general, for a given cylindrical shell there exists a critical load which governs the response mode shape for the cylinder. Transverse blast loaded cylinders respond circumferentially in either a buckled or collapse mode coupled with a fundamental mode shape in the axial direction. For the cylinders tested, almost all of the damage occurred over only one-fourth of the circumference. The damage was centered around the leading edge of the cylinder; failure began as a crack at the fixed ends of the leading edge.

ACKNOWLEDGEMENT

Many of the experiments described in this paper were conducted as a part of an in-house test and research program by the U.S. Air Force Armament Laboratory, Eglin AFB, Florida.

REFERENCES

1. Menkes, S.B. and Opat, J.J., "Broken Beams," Exptl. Mech., 13 (11), pp 480-486 (Nov 1973).

2. Ross, C.A., Sierakowski, R.L., and Malvern, L.E., "Dynamic Response of Structures and Materials to Impulsive Loads," AFATL-TR-74-120, USAF Armament Lab., Eglin AFB, FL (July 1974).

3. Division 2, Natl. Defense Res. Comm., Effect of Impact and Explosion, AO 221586, Summary Technical Rept., 1 (1946).

4. Symonds, P.S. and Mentel, T.J., "Impulsive Loading of Plastic Beams with Axial Constraint," J. Mech. Phys. Solids, 6 (1958).

5. Symonds, P.S. and Jones, N., "Impulsive Loading of Fully Clamped Beams with Finite Plastic Deflections and Strain Rate Sensitivity," Intl. J. Mech. Sci., 14 (1972).

6. Abrahamson, R.G., Florence, A.L., and Lindberg, H.E., "Radiation Damage Studies, Vol. XIII - Dynamic Response of Beams, Plates, and Shell to Pulse Loads," BSD TR-66-372 (Sept 1966).

7. Ross, C.A. and Strickland, W.S., "Response of Flat Plates Subjected to Mild Impulsive Loads," Shock Vib. Bull., U.S. Naval Res. Lab., Proc., No. 45, Pt. 4, pp 105-116 (June 1975).

8. Baynet, P. and Plunkett, R., "Plate Impact and Plastic Deformation by Projectiles," Exptl. Mech., II (2), pp 64-70 (Feb 1971).

9. Lindberg, H.E., Anderson, D.L., Firth, R.D., and Parker, L.V., "Response of Reentry Vehicle-Type Shells to Blast Loads," Stanford Res. Inst., Menlo Park, CA, USAF Contract AF04 (694)-655 (Sept 1965).

10. Schuman, W.J., Jr., "The Response of Cylindrical Shells to External Blast Loadings," BRL Memo Rept. 1461, Ballistic Res. Lab., Aberdeen Proving Ground (Mar 1963).

11. Greenspon, J.E., "Collapse, Buckling and Post Failure Behavior of Cylindrical Shells under Elevated Temperature and Dynamic Loads," J.G. Greenspon Res. Assoc., TR-6, DDC No. AD 630-269 (Nov 1965).

12. Greenspon, J.E., "Theoretical Calculation of Iso-Damage Characteristics," J.G. Greenspon Eng. Res. Assoc., TR-10, DDC No. AD-869093 (Feb 1970).

13. Strickland, W.S., Milton, J.E., Ross, C.A., and Mente, L.J., "Failure of Aluminum Cylindrical Shells Subjected to Transverse Blast Loadings," 47th Shock and Vibration Symp., Albuquerque, NM (Oct 1976).

THE FINITE ELEMENT APPLIED TO THE ANALYSIS OF MECHANISMS AND MACHINES

R.C. Winfrey*

Abstract - This review contains a survey of some approaches to the analysis of mechanisms. Complex models are described, as are various problems associated with the use of finite elements in such analyses.

Improvements in machinery frequently involve conflicting design goals -- for instance, both higher operating speeds and improved positioning accuracy. In this context, "high speed" is taken to mean any speed at which inertial forces are of sufficient magnitude that they cannot be ignored. If such inertial forces are ignored, stresses can increase because of resonance build-up, or failure can occur because of premature fatigue; at the least, overall performance is generally less than expected.

During the design process of a machine, it is convenient to use a simple mechanism as a model. A two- or three-dimensional model with simple finite elements can be used to analyze variable systems with such nonlinearities as damping, backlash, and clearances. The information obtained from studying a simple mechanism can be of great value in solving a complex problem.

This review is limited to a discussion of mechanisms. Various approaches to the analysis of mechanisms and the role of the finite element are described. Complex models containing clearances at the joints of the linkages are discussed, as well as problems associated with analyses of mechanisms and directions for future work.

Early attempts to include elastic effects in the analysis of mechanisms [1-7] were generally based on the slider-crank mechanism (Fig. 1) because of its simplicity. To further simplify the problem, elasticity was usually ignored in all members except the connecting rod (member 2 in the figure), and analog and/or digital computers were used to solve the derived equations of motion. More recent investigations [8] have made use of this simple model to study various effects of interest.

*Principal Engineer, Digital Equipment Corp., Maynard, MA 01754

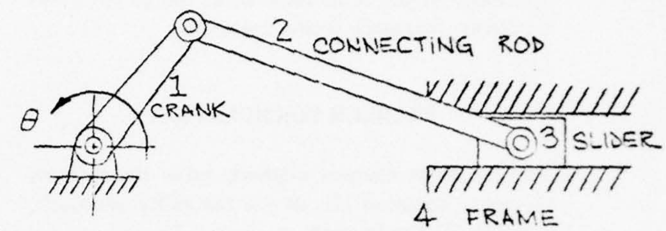

Figure 1. Slider-Crank Mechanism

The plane, four-bar mechanism (Fig. 2) was also given early consideration [9-10], and in 1969 the finite element method was used in general analyses of this and other mechanisms [11-15, 17]. Finite difference techniques were also applied to the analysis of mechanisms at about this time [16]; a unique method using an undulating elastica [7] was also introduced. The finite element method has become well established in engineering and it can be used to model two- and three-dimensional systems. A significant library of finite elements is now available in the literature.

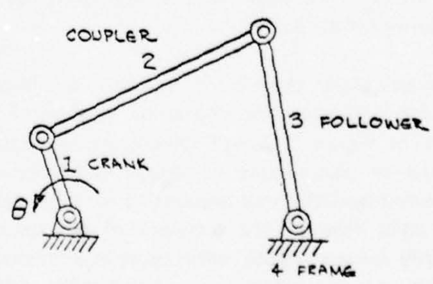

Figure 2. Four-Bar Mecha ism

Early investigations were limited to analytical studies with little or no experimental verification. Among the first attempts at experimental verification were those by Alexander and Lawrence [18, 19], who used a slider crank model to confirm connecting rod resonances. They also verified the occurrence of significant fatigue stress reversals at five to ten times the driving frequency (crank speed).

PROBLEM FORMULATION

With the finite element method, either the stiffness approach, equation (1), or the flexibility approach, equation (2), can be used.

$$(F) = [k]\ (x) \tag{1}$$

$$(X) = [a]\ (F) \tag{2}$$

where $[a] = [k]^{-1}$

By definition, a mechanism allows rigid body deformations; therefore, the stiffness matrix [k] is singular, and [a] does not exist. The flexibility approach can be used by introducing artificial constraints, but the stiffness approach is more direct.

The simplest approach to modeling a four-bar mechanism is to use three classical beam elements and assume a rigid ground, as shown in Figure 3. It should be emphasized, however, that the rigid ground assumption is not made because of any limitations but for convenience. More coordinates could just as easily be added, and the frame included in the analysis, as shown in Figure 4.. The added coordinates create more work for the computer, but not necessarily for the analyst.

Ten elastic link degrees of freedom, q_1 through q_{10}, and one rigid link degree of freedom, θ, are shown in Figure 3. Small deflections are usually assumed in calculations of elastic deflections. It has been shown [6] that accounting for large deflections adds little to the accuracy of the solution, primarily because elastic deflections in a functional machine are considered as second-order effects. Machine failure would occur long before the deflections increased appreciably. Because the elastic deflections are assumed to be small, they can thus be superimposed directly upon the rigid link mechanism.

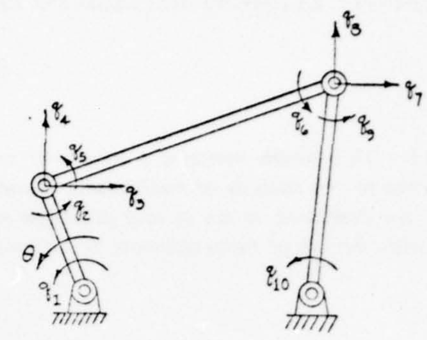

Figure 3. Four-Bar Mechanism - Rigid Base

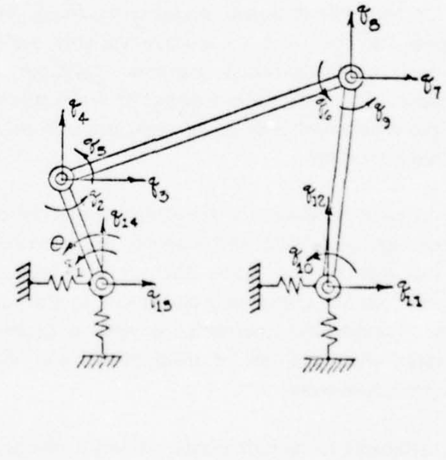

Figure 4. Four-Bar Mechanism - Elastic Base

A typical solution would include the following general steps:

- calculate the rigid body position, velocity, and acceleration of each link in the mechanism for a specific value of θ.
- use classical finite element methods to construct the dynamic equations of motion as if the mechanism were a stable structure.

$$[m]\left(\ddot{q}\right)+[k]\left(q\right) = \left(F\right) \quad (3)$$

- solve the equations of motion. Use as the initial conditions the results obtained as the final conditions of the previous elastic solution; superimpose the results upon the rigid body solution.
- [m] and [k] were obtained by assuming a fixed geometry but are actually functions of θ. Return to the first step and repeat when θ has changed enough -- perhaps one or two degrees, depending on the mechanism.

There are two major differences between finite element solutions for structures and for mechanisms. The obvious difference is that the geometry changes, so that [m] and [k] must be continuously recalculated - a significant task. The more subtle difference lies in the calculation of (F) in equation (3). Not only must (F) account for the usual external forces but also for the rigid body inertial forces. One approach [11, 14] used to obtain an expression for (F) is shown in equation 4.

$$(F) = (F)_{external} + (F)_{inertial} + (F)_{relative} \quad (4)$$

In the equation $(F)_{external}$ is the conventional set of externally applied loads; $(F)_{inertial}$ is somewhat analogous to a set of D'Alembert forces. These inertial forces arise from the rigid body accelerations of each link in the mechanism. The final term, $(F)_{relative}$, is like a Coriolis term. It arises because of the variable geometry and is a second order term, compared with $(F)_{inertial}$, for small elastic deflections. Thus, for the overall mechanism, $(F)_{relative}$ is essentially a second order effect and can be ignored. This is fortunate because its calculation can be cumbersome. Procedures for calculating $(F)_{inertial}$ can be found in the references [11].

METHODS OF SOLUTION

After equation (3) has been formulated, it must be solved to obtain the elastic deformations. The solution can be piecewise.

Modal Analysis

Modal analysis requires an eigenvalue routine for computing the eigenvalues and eigenvectors. The eigenvectors are then used to transform (q) into a set of modal coordinates (n) [24].

$$(q) = [\Phi]\,(n)$$

where

$$[\Phi] = [\,(\phi)_1\,(\Phi)_2\,...(\phi)_{10}\,] \quad (5)$$

and $(\phi)_j$ is the jth eigenvector.

Equation (5) is then applied to equation (3); the result is premultiplied by $[\Phi]^T$ to obtain the set of uncoupled differential equations shown in equation (8).

$$[\Phi]^T[m][\Phi](\ddot{n}) + [\Phi]^T[k][\Phi](n) = [\Phi]^T(F)$$

or,

$$\lceil M \rfloor(\ddot{n}) + \lceil K \rfloor\,(n) = (N) \quad (8)$$

The solution to equation (8) for a step response is well known.

$$n_R(t) = \frac{N_R}{M_{RR}w_R^2}(1-\cos w_Rt) + n_R(o)\cos w_Rt + \frac{\dot{n}_R(o)}{w_R}\sin w_Rt \quad (9)$$

Equation (9) is used to find the system response during the short period of time, t that both the geometry and the forcing function are assumed to remain fixed.

Final values of (n) are transformed back to (q) with equation (5). The eigenvectors are an orthogonal set; if each vector is reduced to unit length, therefore,

$$[\Phi]^T = [\Phi]^{-1}$$

and the inverse transform is easily made, Thus, the initial values at step i+1 are found from the previous final values at step i by

$$\left(n_o\right)_{i+1} = [\Phi]^T_{i+1} \left(q_f\right)_i$$

Modal damping can be included in equation (9), or some other classical form of damping can be introduced at an earlier stage.

One advantage of modal analysis is that relatively large steps can be taken as the mechanism rotates. A method for gaining more insight into how large a step can be taken under conditions of constant geometry has been described [14]. The major disadvantage to modal analysis is the time required to determine all the eigenvectors. A clever approach to reducing solution time -- supposedly by a factor of three -- was to estimate the rate of change in eigenvectors [15], thereby prolonging the calculation of new mass and stiffness properties. Another approach might be to use only one eigenvector, depending on the mechanism, because higher frequencies are usually less important than lower ones. The lowest frequency can be calculated quickly [24(pp 77, 78)].

Modal analysis is important in the dynamic analysis of linear elastic structures. For variable geometry problems, however, modal analysis has given way to direct, numerical integration techniques. One reason for this change is the long computation time required for modal analysis. Another is that research is being directed toward such highly nonlinear effects as clearance between members at their joints.

Numerical Integration

Numerical integration is an efficient way to solve both the older and the newer problems. The Runge-Kutta method [24] has been widely used; other schemes include the Newmark method [20] and the Wilson-θ method [21]. The Newmark method is simple because a linearly varying acceleration is assumed. The Wilson-θ method is somewhat more complex but can be shown to be unconditionally stable.

The major difficulty in using numerical integration to analyze mechanisms is that the links are essentially beam members. Frequencies associated with axial motion are therefore usually several orders of magnitude higher than frequencies associated with bending motion. Thus, even though axial motion is of little concern, it must be accounted for in the determination of an integration time step. The problem can be avoided by eliminating axial, elastic degrees of freedom, but this must be done with care to avoid interference with the rigid body axial motion and with the bending modes of adjacent links.

ADVANCED TOPICS AND FUTURE TRENDS

One of the most exciting and challenging topics to come out of the application of the finite element method to mechanisms has been the study of impact and the effect of clearance at the joints between links. A great deal of effort is also being directed at gearing applications. The impact between two bodies has been studied for some time, of course, but the application of the finite element method is new.

The coefficient of restitution adequately accounts for the loss of energy and general behavior of such simple systems as a bouncing ball. Its main failing is that motion before impact is related to motion after impact; what happens during impact is ignored. A pin in a practical mechanism joint will have a close fit with its bearing; the time of impact is thus a significant part of the total time. A better model of impact is needed.

The impact damper is an example of a simple mechanism with joint clearance. It consists of a box, or enclosure, containing a ball that is allowed to roll back and forth through a small, carefully controlled distance. The idea is not new [25] and has in fact been studied for the past ten years [26-30]. Dubowsky proposed a model for an impact pair [31] and later made experimental studies [32]. The clever experiment was a quasi-inversion of the box/ball configuration; both the acceleration of the freely moving mass (ball) and the box could be directly measured. More recently, other have reported experimental and theoretical work on similar configurations [33-35]. Obviously, much is to be learned from this simple device.

The finite element method was first applied to simple, one-dimensional impact models such as the cam/follower mechanism [30, 37, 38]. One-dimensional impact implies that impact occurs along a single line -- as opposed to the much more complex two-dimensional impact situation which occurs, for example, between a pin and its mating hole. The study of two-dimensional impact was recently applied to rigid link mechanisms [39, 40]; the more difficult concept of elasticity has also been included in the links [41].

Modeling of large deflections has not received much attention. An undulating elastica [7] and the finite element method in a nonlinear, piecewise fashion [42] have been used.

The finite element models described above are complex, and, of course as the complexity of both the model and its nonlinear elements increases, so must the computer time required to solve the equations of motion. For the conventional structural analysis of linear systems, a large problem may have from 1,000 to 50,000 degrees of freedom or more. Even though the simple mechanisms discussed in this review are typically modeled with 10 or 15 degrees of freedom, computer times tend to be excessive for repetitive parameter studies. A few attempts have been made to reduce computer time with simplified models [14, 43] and more efficient coding [15, 44], but a method for significantly reducing solution times without affecting accuracy has not yet been developed.

Another problem facing the analyst using complex models is the proper display of the voluminous data produced by the computer. It is difficult enough to understand what is actually happening to a machine as a pin "rattles" around in a bearing. When a number of joints are rattling at the same time, it is almost impossible to determine if the vibrations can be reduced -- by changing the size of a clearance or by slightly adjusting the geometry or mass distribution. Yet, these are the types of solutions that must be sought in order to build faster, more precise machinery.

CONCLUDING REMARKS

The application of the finite element method to the analysis of mechanisms has been a challenging task during the 1970s. Considerable insight has accumulated with regard to techniques for efficiently analyzing sets of highly nonlinear equations. The results of recent studies will be manifest as a capability to more accurately predict the behavior of new machines before they are built, and will also serve as a guide for the trouble-shooting of existing machines.

REFERENCES

1. Meyer zur Capellan, W., "Bending Vibrations in the Coupler of an Oscillating Crank Mechanism," Osterreichisches Ing. - Arch., XVI (4), p 341 (1962).

2. Khotin, B.M., "A Kinematic Analysis of Mechanisms with Consideration of Linkage Elasticity," Sbornik Trudy, Leningradshii Institut Inzhenerov Zheleznodorozhnogo Transporta, No. 218, pp 214-219 (1964) (In Russian).

3. Neubauer, A.H., Jr., Cohen, R., and Hall, A.S., Jr., "An Analytical Study of the Dynamics of an Elastic Linkage," J. Engr. Indus., Trans. ASME, 88 (3), pp 311-317 (Aug 1966).

4. Jasinski, P.W., Lee, H.C., and Sandor, G.N., "Stability and Steady-State Vibrations in a High-Speed Slider-Crank Mechanism," J. App. Mech., 37 (4), pp 1069-1076 (Dec 1970).

5. Jasinski, P.W., Lee, H.C., and Sandor, G.N., "Vibration of Elastic Connecting Rod of a High-Speed Slider-Crank Mechanism," J. Engr. Indus., Trans. ASME, 93 (2), pp 336-344 (May 1971).

6. Viscomi, B.V. and Ayre, R.S., "Nonlinear Dynamic Response of Elastic Slider-Crank Mechanism," J. Engr. Indus., Trans. ASME, 93 (1), pp 251-262 (Feb 1971).

7. Shoup, T.E., "On the Use of the Undulating Elastica for the Analysis of Flexible Link Mechanisms," J. Engr. Indus., Trans. ASME, 93 (1), pp 263-267 (Feb 1971).

8. Golebiewski, E.P. and Sadler, J.P., "Analytical and Experimental Investigation of Elastic Slider-Crank Mechanisms," ASME Paper No. 76-DET-18 (to be published in J. Engr. Indus.).

9. Gayfer, J.R. and Mills, B., "Small-Amplitude Vibrations of the Four-Bar Linkage Chain," J. Mech. Engr. Sci., 7 (3), pp 252-258 (1965).

10. Burns, R.H., "The Kinetostatic Synthesis of Flexible Link Mechanisms," Dissertation, Yale University (1964).

11. Winfrey, R.C., "Dynamics of Mechanisms with Elastic Links," Doctoral Dissertation, UCLA (1969).

12. Winfrey, R.C., "Elastic Link Mechanism Dynamics," J. Engr. Indus., Trans. ASME, 93 (1), pp 268-272 (Feb 1971).

13. Erdman, A.G., Sandor, G.N., and Oakberg, R.G., "A General Method for Kineto-Elastodynamic Analysis and Synthesis of Mechanisms," J. Engr. Indus., Trans. ASME, 94, pp 1193-1205 (Nov 1972).

14. Winfrey, R.C., "Dynamic Analysis of Elastic Link Mechanisms by Reduction of Coordinates," J. Engr. Indus., Trans. ASME, 94, pp 577-582 (Nov 1972).

15. Imam, I., Sandor, G.N., and Kramer, S.N., "Deflection and Stress Analysis in High Speed Planar Mechanisms with Elastic Links," J. Engr. Indus., Trans. ASME, 95 (2), pp 541-548 (May 1973).

16. Sadler, J.P. and Sandor, G.N., "A Lumped Parameter Approach to Vibration and Stress Analysis of Elastic Linkages," J. Engr. Indus., Trans. ASME, 95 (2), pp 549-557 (May 1973).

17. Imam, I., "A General Method for Kento-Elastrodynamic Analysis and Design of High Speed Mechanisms," Doctoral Dissertation, Rensselaer Polytechnic Institute, Troy, NY (Aug 1973).

18. Alexander, R.M. and Lawrence, K.L., "Dynamic Strains in a Four-Bar Mechanism," Proc. Third Appl. Mech. Conf., Stillwater, OK, Paper No. 26 (Nov 1973).

19. Alexander, R.M. and Lawrence, K.L., "An Experimental Investigation of the Dynamic Response of an Elastic Mechanism," J. Engr. Indus., Trans. ASME, 96 (1), pp 268-274 (Feb 1974).

20. Golebiewski, E.P. and Sadler, J.P., "Analytical and Experimental Investigation of Elastic Slider-Crank Mechanisms," ASME Paper No. 76-DET-18 (to be published in J. Engr. Indus).

21. Newmark, N.M., "A Method of Computation for Structural Dynamics," ASCE J. Engr. Mech. Div., 85 (EM3), pp 67-94 (July 1959).

22. Bathe, K.J. and Wilson, E.L., "Stability and Accuracy Analysis of Direct Integration Methods," Intl. J. Earthquake Engr. Struc. Dynam., 1 (2) (1973).

23. Hurty, W.C. and Rubenstein, M.F., Dynamics of Structures, Prentice-Hall, Englewood Cliffs, NJ (1964).

24. Hovanessian, S.A. and Pipes, L.A., Digital Computer Methods in Engineering, McGraw-Hill (1969).

25. Lieber, P. and Jensen, D.P., "An Acceleration Damper: Development, Design, and Some Applications," Trans. ASME, 67, pp 523-530 (1945).

26. Masri, S.F. and Caughey, T.K., "On the Stability of the Impact Damper," J. Appl. Mech., Trans. ASME, 88, pp 586-592 (1966).

27. Egle, D.M., "An Investigation of the Impact Vibration Absorber," J. Engr. Indus., Trans. ASME, 89 (4), pp 653-657 (Nov 1967).

28. Masri, S.F. and Ibrahim, A.H., "A Hybrid Electromechanical Analog Computer Technique for Optimizing Vibration Systems," J. Engr. Indus., Trans. ASME, 94 (2), pp 381-387 (May 1972).

29. Masri, S.F., "Steady-State Response of a Multidegree System with an Impact Damper," ASME Paper No. 72-APM-39.

30. Winfrey, R.C., "Multidegree-of-Freedom Elastic Systems Having Multiple Clearances," Shock Vib. Bull., U.S. Naval Res. Lab., Proc., 43, Pt. 2, pp 23-30 (June 1973).

31. Dubowsky, S. and Freudenstein, F., "Dynamic Analysis of Mechanical Systems with Clearances, Part I: Formation of Dynamic Model; Part II: Dynamic Response," J. Engr. Indus., Trans. ASME, 93 (1), pp 305-316 (Feb 1971).

32. Dubowsky, S. and Young, S.C., "An Experimental and Analytical Study of Connection Forces in High-Speed Mechanisms," J. Engr. Indus., Trans. ASME, 97 (4), pp 1166-1174 (1975).

33. Veluswami, M.A. and Crossley, F.R.E., "Multiple Impacts of a Ball between Two Plates, Part 1: Some Experimental Observations," J. Engr. Indus., Trans. ASME, 97 (1) (Feb 1975).

34. Veluswami, M.A., Crossley, F.R.E., and Horvay, G., "Multiple Impacts of a Ball between Two Plates; Part 2: Mathematical Modelling," J. Engr. Indus., Trans. ASME, 97 (1) (Feb 1975).

35. Beckett, R.E., Pan, K.C., and Chu, S.C., "A Numerical Method for the Dynamic Analysis of Mechanical Systems in Impact," ASME Paper No. 76-DET-1 (to be published in J. Engr. Indus.).

36. Herbert, R.G. and McWhannell, D.C., "Shape and Frequency Composition of Pulses from an Impact Pair," ASME Paper No. 76-DET-36 (to be published in J. Engr. Indus.).

37. Winfrey, R.C., Anderson, R.V., and Gnilka, C.W., "Analysis of Elastic Machinery with Clearances," J. Engr. Indus., Trans. ASME, 95 (3), pp 695-703 (Aug 1973).

38. Dubowsky, S. and Gardner, T.N., "Dynamic Interactions of Link Elasticity and Clearance Connections," J. Engr. Indus., Trans. ASME, 97 (2), pp 652-661 (May 1975).

39. Mansour, W.M. and Townsend, M.A., "A Pendulating Model for Mechanisms with Clearances in the Revolute," J. Engr. Indus., Trans. ASME, 97 (1), pp 354-358 (Feb 1975).

40. Mansour, W.M. and Townsend, M.A., "Impact Spectra and Intensities for High-Speed Mechanism," J. Engr. Indus., Trans. ASME, 97 (1), pp 347-353 (Feb 1975).

41. Dubowski, S. and Gardner, T.N., "Design and Analysis of Multilink Flexible Mechanisms with Multiple Clearance Connections," ASME Paper No. 76-DET-9 (to be published in J. Engr. Indus.).

42. Sevak, N.M. and McLarnan, C.W., "Optimal Synthesis of Flexible Link Mechanisms with Large Static Deflections," J. Engr. Indus., Trans. ASME, 97 (1) (Feb 1975).

43. Midha, A., Erdman, A.G., and Frohrib, D.A., "An Approximate Method for the Dynamic Analysis of Elastic Linkages," ASME Paper No. 76-DET-49 (to be published in J. Engr. Indus.).

44. Dubowsky, S. and Grant, J.L., "Application of Symbolic Manipulation to Time Domain Analysis of Nonlinear Dynamic Systems," J. Engr. Indus., Trans. ASME, 97 (1), pp 60-68 (Mar 1975).

ANNUAL ARTICLE INDEX

FEATURE ARTICLES

LITERATURE REVIEWS

	ISSUE	PAGES
Mitchell, W.S. **Shock and Vibration Instrumentation: Accelerometers**	1	15-18
Derby, T.F. **Computer Programs: Shock and Vibration Isolation**	1	19-26
Gibson, R.F. and Plunkett, R. **Dynamic Stiffness and Damping of Fiber-Reinforced Composite Materials**	2	9-17
Krishna Murty, A.V. **Finite Element Modeling of Natural Vibration Problems**	2	19-37
Rao, J.S. **Turbine Blading Excitation and Vibration**	3	15-22
Chen, F.Y. **A Review of the Literature on the Dynamics of Cam Mechanisms**	3	23-36
Jensen, J.J. and Madsen, N.Fl. **A Review of Ship Hull Vibration. Part 1: Mathematical Models**	4	13-22
Chen, L.H. and Pierucci, M. **Underwater Fluid-Structure Interaction. Part I: Introduction and Scope**	4	23-24
Chen, L.H. and Pierucci, M. **Underwater Fluid-Structure Interaction. Part II: Mechanically-Applied Forces**	5	17-23
Jensen, J.J. and Madsen, N.Fl. **A Review of Ship Hull Vibration. Part II: Modeling Physical Phenomena**	5	25-38
Chen, L.H. and Pierucci, M. **Underwater Fluid-Structure Interaction. Part III: Acoustically-Applied Forces**	6	13-17
Jensen, J.J. and Madsen, N.Fl. **A Review of Ship Hull Vibration. Part III: Methods of Solution**	6	19-27
Jensen, J.J. and Madsen, N.Fl. **A Review of Ship Hull Vibration. Part IV: Comparison of Beam Models**	7	13-28
Chen, L.H. and Pierucci, M. **Underwater Fluid-Structure Interaction. Part IV: Hydrodynamically Applied Forces (Moving Medium)**	7	29-37
Ward, H.S. **The Characteristics of Dynamic Loads and Response of Buildings**	8	13-20

BOOK REVIEWS

HANDBOOK OF PYROTECHNICS

K.O. Brauer

Chemical Publishing Co., Inc. (1974)

The Handbook of Pyrotechnics is a fascinating book that seems to fulfill the expressed intent of the author:

> "It is the purpose of this handbook to provide useful data and information about theory and practical application of pyrotechnics for engineers, designers, technicians and students."

No previous knowledge of the subject is assumed and the material is presented in an almost "popular" way. Thus it can either be scanned rapidly for basic ideas or individual mechanisms can be studied in more detail.

A quotation from the author's introduction outlines *the contents:*

> "The contents of this handbook are divided into six parts: Explosive Materials, Explosive-Actuated Devices, Pyrotechnic Systems, Reliability and Testing, Explosive Production Methods, and Appendix.
>
> The handbook contains numerous charts, graphs, and illustrations as useful aids. Theory, data, and practical applications are explained in detail. Valuable new information is presented in this handbook, as for example data about the effects of extreme environmental conditions on pyrotechnic materials and devices, hints and data for qualification testing, hints for the design and application of pyrotechnic systems, and data for the application of explosive methods in manufacturing processes.
>
> It is recommended to use this handbook together with the book Military and Civilian Pyrotechnics by Dr. Herbert Ellern, published by the Chemical Publishing Company, which contains more detailed information about the properties, and production of pyrotechnic materials and an extensive manufacturing formulary."

The book is filled photographs, sectioned drawings, schematics, sequence diagrams, tables, and graphs that describe the operation and construction of specific devices and systems. The information can be very useful for a designer attempting to solve a problem. It is not detailed enough for him to complete a design solution but is a good source of possible approaches.

The book contains a wealth of descriptive material on spacecraft systems and a lesser amount on aircraft and missile systems. Manufacturing uses are covered briefly but well. The book contains a reasonable glossary and 95 references, most of which are from open literature periodicals and books. Although credits are given for the many photographs and diagrams, company literature, which must have provided sources, is not mentioned. Some way for readers to contact producers and developers would be a useful addition to the book.

Although readers of the DIGEST might find this handbook interesting and useful, they will no doubt realize that one subject has not been included: that of the shocks produced by the various pyrotechnic devices. DIGEST readers would find such information useful -- even rudimentary typical descriptions. A classification of devices according to shocks produced would also be a helpful tool for designers.

Ronald L. Woodfin
Head, Environmental Engineering Branch
Naval Weapons Center
China Lake, California

VIBRATION OF BEARINGS
(Vibratsiya Podshipnikov)

K. Ragulskis, A. Jurkauskas, V. Atstupenas, A. Vitkute, and A. Kulvec
Leidykla Mintis, Vilnius, Lithuania (1974)
(In Russian)

The vibration of bearings became important with the invention of the wheel. No one knows exactly when it was invented or by whom. However, archaeologists have found evidence of the existence of wheels in graves that date from 5,500 years ago. A large body of literature has accumulated pertaining to the vibration of bearings; it is scattered in many publications throughout the world. In general each publication is concerned with study of only a part of the "total problem of vibration of bearings,"

The authors of this book are associated with the Kaunas Polytechnic Institute in Lithuania. They were assisted by A.B. Palionis, R.P. Atstunenene, R.V. Kanapenas, V.I. Zdanavichyus, V.N. Augutis, and I.R. Zhitkevichyus, all of whom are also associated with the Kaunas Institute. Their book is a welcome addition to the literature of vibration of bearings, especially because it treats the "total problem of vibration of bearings." The book provides an excellent summary of the current state-of-the-art of vibration of bearings in Eastern European countries.

The book is concerned with analytical and experimental investigation of bearings and with the design of bearings and bearing units. It contains the following chapters:

1. Analytical determination of rotational resisting moments.
2. Determination of the elastic and damping characteristics of oscillating bearings.
3. Analysis of radial vibration of bearings and bearing units.
4. Methods and equipment for measurement of dynamic characteristics of bearings.
5. Errors in measurement of rotational resisting moments and means for reducing them.
6. Method for statistical treatment of experimental results obtained from investigation of the dynamics of precision bearings.
7. Experimental investigation of the dynamic characteristics of precision bearings and their units.
8. Methods and schemes to reduce the rotational resisting moments and the vibration of bearings.

The analytical determination of rotational resisting moments is based on theory that has evolved during the past 15 years. The theory assumes that this total moment is composed of a sum of eight components, all of which are multiplied by a single corrective coefficient to account for factors that cannot be accounted for analytically. Thus, a complete analytical theory for rotational resisting moments of bearings that properly accounts for all relevant factors remains to be developed.

Of especial interest in the book are the experimental data on rotational resisting moments versus rotational speed for various elevated temperatures. The last chapter in the book will also be of interest to designers who sometimes state "Don't bother me with the *theory and the experimental results -- just tell me in* plain English how I can *reduce* the rotational resisting moments and the vibration of bearings,"

References in this book by number were: Eastern European countries (470), German (61), English (33), and Italian (1). In some chapters the text refers to references that are not listed in the bibliography following the chapter. A number of typographical errors can be found in most of the bibliographies following each chapter. Only 1,000 copies of the book have been printed.

M. Dublin
General Dynamics/Pomona Division
Pomona, California 91766

NONLINEAR AND LINEAR TRANSIENT DEFORMATION WAVES IN THERMO-ELASTIC AND ELASTIC BODIES

(Nelineinye i lineinye perekhodnye volnovye protsessy deformatsii termouprugikh i uprugikh tel)

U.K. Nigul and Yu. K. Engel'brekht
Tallin, Akademiya Nauk Estonskoi SSR
Institut Kibernetiki (1972)

This monograph presents a systematic treatment of transient wave processes in continuous media from a differential equation approach. A rather extensive bibliography is included and takes up almost two-fifths of the entire manuscript. Aside from summary treatment of works by other authors, it is primarily a reiteration of the work of the first author.

Although a pretense in rigor was attempted, nothing more than academic solution was advanced. In particular, the authors made only a passing remark in the introduction of artificial damping in the solution of a moving shock front (Section 7.2, p. 84) and expressed their doubts as to its validity, apparently being unaware of the rather important contribution by von Neumann using artificial damping to enable rapid and efficient numerical integration of wave problems with high gradients. Finite-element treatment, likewise, received only scanty passing mention. Boley's treatment of Timoshenko-type beams, using solutions for separate regions (1955, 1956), was cited as a starting point for some new development in USSR by Slepiana (Section 9.2, p. 91). However, no description of Slepiana's work was included in the monograph, consequently tending to leave this reviewer completely in the dark as to what is the real improvement in the technique.

The monograph is divided into three parts. Part 1 examines the general equations governing thermoelasticity with due consideration of nonlinear equations of thermal conductivity. It is shown that for typical structural materials, the effects due to geometrical nonlinearities are of the same order of magnitude as those due to physical nonlinearities. In part 2, methods of analysis of transient wave processes caused by mechanical inputs are examined and classified. In part 3, one-dimensional example is treated to exhibit nonlinear and thermal effects.

This reviewer considers that this monograph, although lacking in some degree a complete survey of contemporary techniques, nevertheless provides a clear and concise systematic identification of the problems involved; *i.e.*, it is a good primer to the subject, but not very useful for anything else.

C.C. Wan, USA
Courtesy of Applied Mechanics Reviews

ROTOR VIBRATIONS AND BALANCING

(Kolebaniya i uravnoveshchivanie rotorov)

Izdatelstvo "Nauka", Moscow (1973)
($2.50)

The book consists of 18 papers on some recent problems in vibrations of high-speed rotors and their balancing.

The first six consider vibrations of flexible rotors and deal with: vertical rotors under gravitational forces, (M. F. Zeitman); excitation of counterprecession, (G. I. Anikiejev); influence of disk dimensions on natural frequencies of excited vibrations of step rotors under concentrated forces, (A. A. Gusarow); dynamic deflections of eccentric rotors, (N. G. Samarow), and determination of optimal parameters, (M. F. Zeitman and R. B. Statnikov).

Papers dealing with balancing problems can be grouped as those concerned with: (1) the influence of balancing weight distribution, (N. G. Samarov and L.N. Kudriaszew), their flexible mounting insensitive balancing speeds, (A. A. Gusarov); (2) applications of amplitude-phase characteristics to rotor balancing (L. Ia. Banach, M. D. Piermikov, and L. N. Shatalov), (A. A. Gurarov and L. N. Ghatalov); (3) automatic balancing, (W. I. Susanin), (M. D. Genkin and others). Other special and more general problems of balancing and measurement are also considered by M. E. Levit, A. I. Maximienko, Iu. A. Samsaev, K. W. Frolov, T. P. Kozlanikov, and Iu. A. Pietrov.

The general introduction is given by A. A. Gusarov.

Z. Parszewski, Poland
Courtesy of Applied Mechanics Reviews

SHORT COURSES

JANUARY

EARTHQUAKE SIMULATION AND RESPONSE

Dates: January 9 - 13, 1978
Place: Long Island, New York
Objective: Safe shut-down of a nuclear power generating station following an earthquake will be the main topic of this course, to be held at the facilities of Dayton T. Brown, Inc., Bohemia, Long Island, New York -- one of the few laboratories in the world capable of earthquake simulation. This course, aimed at test and quality engineers, will stress interpreting standards and specifications and conducting tests (including the proper mounting of test specimens).

Contact: Wayne Tustin, Tustin Institute of Technology, 22 East Los Olivos St., Santa Barbara, CA 93105 (805) 953-1124.

MAINTAINABILITY ENGINEERING

Dates: January 9 - 13, 1978
Place: UCLA Extension
Objective: This course is designed to help participants to determine the following: the distribution of times-to-repair components and times-to-restore equipment, the equipment mean-time-to-restore, the mean man-hours neeeded to restore, the optimum preventive maintenance schedules for minimum total corrective and preventive maintenance cost, spare parts requirments with a specified assurance and their optimization, the reliability, maintainability and availability (both instantaneous and steady state) of maintained equipment and systems, interpret and use MIL/STB-471 and MIL/DBK-472. The course is intended for those involved in the conception, design, operation and maintenance of any equipment in today's mechanical society. A Bachelor's degree in engineering, mathematics, or equivalent is required.

Contact: Continuing Education in Engineering and Mathematics, Short Courses, 6266 Boelter Hall, UCLA Extension, Los Angeles, CA 90024 (213) 825-3344 or 825-1295.

FEBRUARY

VIBRATION SURVIVABILITY

Dates: February 6 - 10, 1978
Place: Fullerton, California
Objective: This course, which will be held at the facilities of Hughes Aircraft, Malvern at Gilbert Sts., Fullerton, Calif., is designed to provide basic education in resonance and fragility phenomena, in environmental vibration and shock measurement and analysis, also in vibration and shock environmental testing to prove survivability. This course will concentrate upon techniques and equipments rather than upon mathematics and theory.

Contact: Wayne Tustin, Tustin Institute of Technology, 22 E. Los Olivos St., Santa Barbara, CA, 93105 (805) 963-1124.

MACHINERY VIBRATION MONITORING AND ANALYSIS SEMINAR

Dates: February 13, 14, & 15, 1978
Place: Houston, Texas
Objective: This seminar will be devoted to the understanding and application of vibration technology to machinery vibration monitoring and analysis. Basic and advanced techniques with illustrative case histories and demonstrations will be discussed by industrial experts and consultants. Topics to be covered in the seminar include preventive maintenance, measurements, analysis, data recording and reduction, computer monitoring, acoustic techniques, misalignment effects, balancing, turbomachinery blading, bearing fault diagnosis, torsional vibration problems and corrections, and trend analysis. An instrumentation show will be held in conjunction with this seminar.

Contact: Dr. R. L. Eshleman, Vibration Institute, Suite 206, 101 W. 55th St., Clarendon Hills, IL, 60514 (312) 654-2254.

NEWS BRIEFS

news on current and Future Shock and Vibration activities and events

CALL FOR PAPERS
1979 Fifth World Congress on the Theory of Machines and Mechanisms

The Fifth World Congress on the Theory of Machines and Mechanisms, to be held at Concordia University, Montreal, Canada, during July 8 - 13, 1979, will be a forum to discuss all aspects of problems related to the theory of machines and mechanisms and applied problems.

Delegates from all over the world are expected and papers are solicited in the areas of kinematic analysis and synthesis; dynamics of machines and mechanisms; gearing and transmissions; preventive maintenance and reliability control; rotor-dynamics; vibrations and noise in machines; biomechanisms; technology transfer; robots, manipulators and man-machine systems; computer-aided design and optimization; pneumatics, hydraulics and electro-dynamics; industrial applications for special machines and mechanisms; experimental and teaching methods.

For further information, please contact:

Dr. Seshadri Sankar
Papers Review and Program Chairman
IFToMM Congress
Dept. of Mechanical Engineering
Concordia University
1455 de Maisonneuve W.
Montreal, Canada H3G 1M8
Telephone (514) 879-5839

INSTITUTE OF ENVIRONMENTAL SCIENCES
Shock and Vibration Test Problems Subcommittee

The Test Problems Subcommittee of the IES Shock and Vibration Committee under the chairmanship of Wayne Tustin, Tustin Institute of Technology, will compile a list and describe the most critical problems in shock and vibration testing practice. The subcommitee expects the list will be complete by December 31, 1977 and published in draft shortly thereafter.

The subcommittee is soliciting input and assistance.

Subcommittee members are:

Wayne Tustin, Chairman
John Losse, Delco
Dick Shelby, Hughes Aircraft
Darrell Dickey, Raytheon

Contact Wayne Tustin at Tustin Institute of Technology, 22 E. Olivos St., Santa Barbara, CA 93105 (805) 963-1124.

SECOND WORLD CONGRESS ON FINITE ELEMENT METHODS
Bournemouth, Dorset, England
23rd to 27th October, 1978

The Second World Congress on Finite Element Methods is to be held at the Royal Bath Hotel, Bournemouth, Dorset, England, 23rd to 27th October, 1978. A Finite Element Method Exhibition will also be held at the same event. The theme of the Congress is finite element methods in the commercial environment and Professor R. H. Gallagher, Cornell University, will deliver the main invited lecture.

For further information, please contact:

Dr. John Robinson
Robinson and Associates
Horton Road, Woodlands, Wimborne
Dorset BH21 6NB England

ABSTRACT CATEGORIES

ANALYSIS AND DESIGN

Analogs and Analog Computation
Analytical Methods
Dynamic Programming
Impedance Methods
Integral Transforms
Nonlinear Analysis
Numerical Analysis
Optimization Techniques
Perturbation Methods
Stability Analysis
Statistical Methods
Variational Methods
Finite Element Modeling
Modeling
Digital Simulation
Parameter Identification
Design Information
Design Techniques
Criteria, Standards, and Specifications
Surveys and Bibliographies
Tutorial
Modal Analysis and Synthesis

COMPUTER PROGRAMS

General
Natural Frequency
Random Response
Stability
Steady State Response
Transient Response

ENVIRONMENTS

Acoustic
Periodic
Random
Seismic
Shock
General Weapon
Transportation

PHENOMENOLOGY

Composite
Damping
Elastic
Fatigue
Fluid
Inelastic
Soil
Thermoelastic
Viscoelastic

EXPERIMENTATION

Balancing
Data Reduction
Diagnostics
Equipment
Experiment Design
Facilities
Instrumentation
Procedures
Scaling and Modeling
Simulators
Specifications
Techniques
Holography

COMPONENTS

Absorbers
Shafts
Beams, Strings, Rods, Bars
Bearings
Blades
Columns
Controls
Cylinders
Ducts
Frames, Arches
Gears
Isolators
Linkages
Mechanical
Membranes, Films, and Webs
Panels
Pipes and Tubes
Plates and Shells
Rings
Springs
Structural
Tires

SYSTEMS

Absorber
Acoustic Isolation
Noise Reduction
Active Isolation
Aircraft
Artillery
Bioengineering
Bridges
Building
Cabinets
Construction
Electrical
Foundations and Earth
Helicopters
Human
Isolation
Material Handling
Mechanical
Metal Working and Forming
Off-Road Vehicles
Optical
Package
Pressure Vessels
Pumps, Turbines, Fans, Compressors
Rail
Reactors
Reciprocating Machine
Road
Rotors
Satellite
Self-Excited
Ship
Spacecraft
Structural
Transmissions
Turbomachinery
Useful Application

ABSTRACTS FROM THE CURRENT LITERATURE

Copies of articles abstracted in the DIGEST are not available from the SVIC or the Vibration Institute (except those generated by either organization). Inquiries should be directed to library resources. Government reports can be obtained from the National Technical Information Service, Springfield, VA 22151, by citing the AD-, PB-, or N- number. Doctoral dissertations are available from University Microfilms (UM), 313 N. Fir St., Ann Arbor, MI; U.S. Patents from the Commissioner of Patents, Washington, D.C. 20231. Addresses following the authors' names in the citation refer only to the first author. The list of periodicals scanned by this journal is printed in issues 1, 6, and 12.

ABSTRACT CONTENTS

ANALYSIS AND DESIGN

ANALYTICAL METHODS

77-2044

The Effect of Delay on the Behavior of a Nonlinear Vibration System (Uber den Einfluss von Totzeiten auf nichtlineare Schwingungssysteme)

J.A. Mitropolskij

Math. Inst. of the Academy of Sciences; Uliza Repina 3, 25 260 I Kiew, USSR, Ing. Arch., 45 (5/6), pp 387-392 (1976) 8 refs

(In German)

Key Words: Nonlinear systems, Vibrating structures

The effect of delay on the behavior of a nonlinear oscillating system is investigated. Qualitative analysis has been carried out for some practically important problems and the influence of delay effects on the oscillation properties; namely, type of oscillation, stability, nature and intensity of damping have been examined.

IMPEDANCE METHODS

(See No. 2080)

NONLINEAR ANALYSIS

77-2045

Parametric Vibration of a Non-Linear System

A. Tondl

National Research Inst. for Machine Design, 25097 Praha 9 - Bechovice, CSSR, Ing. Arch., 45 (5/6), pp 317-324 (1976) 9 figs, 3 refs

Key Words: Nonlinear systems, Single degree of freedom systems, Parametric response

An analysis is presented of a non-linear system with one degree of freedom, in which the restoring force is expressed by the product of a periodic function of time and a non-linear function of deflection. In such a system there can occur not only the expected parametric resonances of the order n (n = 1, 2, ...) but resonances of the order 1/N (N = 2, 3, ...) as well.

NUMERICAL ANALYSIS

77-2046

Analysis and Design of Numerical Integration Methods in Structural Dynamics

H.M Hilber

Ph.D. Thesis, Univ. of California, Berkeley, 111 pp (1976)

UM 77-15,718

Key Words: Numerical analysis, Dynamic structural analysis

The objective of this work is to develop one-step methods for the integration of the equations of structural dynamics which are unconditionally stable, have an order of accuracy not less than two, and possess numerical dissipation which can be controlled by a parameter other than the time step size. In particular, no numerical dissipation is included. Four new families of algorithms are discussed from this point of view, and compared with algorithms, such as the Newmark, Wilson and Houbolt methods, which are commonly used in structural dynamics and do not achieve these requirements.

77-2047

A Splitting Method for Computing Coupled Hydrodynamic and Structural Response

J.E. Ash

Argonne National Lab., Argonne, IL 60439, Appl. Math. Modeling, 1 (6), pp 333-338 (Sept 1977) 4 figs, 5 refs

Key Words: Numerical analysis, Nuclear reactor containment, Underwater explosion, Hydrodynamic excitation

A numerical method is developed for application to unsteady fluid dynamics problems. In particular to the mechanics following a sudden release of high energy. Solution of the initial compressible flow phase provides input to a power-series method for the incompressible fluid motions. The system is split into spatial and time domains which lead to the convergent computation of a sequence of elliptic equations. Two sample problems are solved.

OPTIMIZATION TECHNIQUES

77-2048

Reliability-Based Optimization for Dynamic Loads

J.W. Davidson, L.P. Felton, and G.C. Hart

Ameron, South Gate, CA., ASCE J. Struc. Div., 103 (ST10), pp 2021-2035 (Oct 1977)

Key Words: Minimum weight design, Shock response, Probability theory

A general formulation is presented for weight optimization of indeterminate structures subject to transient dynamic loads and reliability constraints. Two distinct methods of structural analysis are examined and compared for use in the optimization algorithm: Numerical integration of equations of motion and shock spectra. Details of the essential computation of standard derivation of response quantities associated with each analysis technique are also examined. The formulations are illustrated by design examples of a rigid frame subjected to an acceleration impulse applied to its base.

77-2049

Structural Properties of Linear Dynamic Systems: Application to Optimal Control and Filtering

O.L. Mercier

Office National d'Etudes et de Recherches Aerospatiales, Paris, France, Rept. No. ONERA-NT-1977-4, FR-ISSN-0078-3781, 26 pp (Mar 1977) refs (In French)

N77-25859

Key Words: Optimum control theory, Dynamic systems

The major results concerning the modern concepts of controllability, observability, reconstructibility, stability, stabilizability, and detectability of linear dynamic systems are presented. These concepts, developed during the 1960 to 1972 period, are of prime importance for the control of dynamic systems, especially to design feedback controls and to synthesize the filters, state reconstructors, and observers usually associated with these controls.

STABILITY ANALYSIS

77-2050

Energy Expressions as Stability Criteria in Linear Differential Equations with Periodic Coefficients (Energieausdrücke als Stabilitätskriterien bei linearen Differentialgleichungen mit Periodischen Koeffizienten)

E. Brommundt

Mechanik-Zentrum, Lehrstuhl A für Mechanik, Technische Universität Braunschweig, Postfach 3329, D-3300 Braunschweig, Federal Rep. of Germany, Ing. Arch, 45 (5/6), pp 325-330 (1976) 2 refs

(In German)

Key Words: Stability, Turbomachinery, Perturbation technique

Starting from the principle of virtual work the stability of the trivial solution is investigated by means of a perturbation technique. The stability conditions have the form of energy expressions which, in general, cannot be interpreted as energy flows.

STATISTICAL METHODS

(See Nos. 2087, 2125)

FINITE ELEMENT MODELING

(Also see No. 2140)

77-2051

Solutions to Initial Value Problems by Use of Finite Elements -- Unconstrained Variational Formulations

J.J. Wu

Benet Weapons Lab., Watervliet Arsenal, Watervliet, NY 12189, J. Sound Vib., 53 (3), pp 341-356 (Aug 8, 1977) 2 figs, 5 tables, 15 refs

Key Words: Boundary value problems, Finite element technique, Forced vibration

This paper presents a variational formulation which treats initial value problems and boundary problems in a unified manner. The basic ingredients of this theory are adjoint variable and unconstrained variations. It is an extension of the finite element unconstrained variational formulation used previously in solving several non-conservative stability problems. The technique which makes this extension possible is described. This formulation thus enables one to adapt such numerical techniques as the finite element method, which has had great success and popularity for solution of boundary value problems, for solutions of initial value problems as well.

PARAMETER IDENTIFICATION

(Also see No. 2177)

77-2052

Maximum Likelihood Parameter Identification of Linear Dynamic Systems

F. Chen

Ph.D. Thesis, Northeastern Univ., 114 pp (1977)

UM 77-17,784

Key Words: Linear systems, Parameter identification

This dissertation develops and compares two maximum-likelihood methods for parameter estimation. It includes: Formulation and comparison of the performance criteria for two maximum-likelihood methods, denoted as ML1 and ML2, Derivation of an equivalent ML2 criterion and a numerical procedure to provide the estimation of the state and the unknown parameter vector separately, Investigation and comparison of the estimation properties of the ML1 and ML2 methods with numerical examples included.

77-2053
Correction of the Theoretical Model of an Elastomechanical System by Means of Measured Forced Vibrations (Die Korrektur des Rechenmodells eines elastomechanischen Systems mittels gemessener erzwungener Schwingungen)
H.G. Natke
Lehrstuhl für Schwingungs- und Messkunde und Curt-Risch-Institut, Technische Universität Hannover, Callinstr. 32, D-3000 Hannover, Federal Rep. of Germany, Ing. Arch., 46 (3), pp 169-184 (1977) (In German)

Key Words: Mathematical models, Parameter identification

The system analysis of elastomechanical systems results in a theoretical model as an approximation of the real structure. The system identification leads to the uncomplete experimental model. The quality criterion applied to the theoretical model may be the accordance of the eigencharacteristics of the theoretical model with the eigencharacteristics of the experimental model or the accordance of their frequency responses.

77-2054
Dynamic Data System: A New Modeling Approach
S.M. Wu
Dept. of Mech. Engrg., Univ. of Wisconsin, Madison, WI, J. Engr. Indus., Trans. ASME, 99 (3), pp 708-714 (Aug 1977) 4 figs, 45 refs

Key Words: Mathematical models, Parameter identification

The dynamic data system is a modeling technique that uses dynamic data in the form of a time series to develop physically meaningful stochastic difference/differential equations. The general mathematical formulation and background of the dynamic data system methodology are given, and the modeling procedure evolved in this approach is illustrated by an example pertaining to neutron flux data. An example of a machine tool system analysis is presented to show the physical interpretation and the subsequent exploitation of the mathematical models. Various applications of the technique are also described, and the future development is envisaged.

DESIGN TECHNIQUES
(See Nos. 2068, 2069)

CRITERIA, STANDARDS, AND SPECIFICATIONS
(See No. 2154)

SURVEYS AND BIBLIOGRAPHIES
(Also see No. 2104)

77-2055
Acoustic Holography (Citations from the Engineering Index Data Base)
W.E. Reed
National Technical Information Service, Springfield, VA., Rept. No. NTIS/PS-77/0579/1GA, 218 pp (July 1977)

Key Words: Acoustic holography, Bibliography

Worldwide research on acoustic holography is covered. Theory, uses, equipment design, and imaging techniques are presented. Most of the studies are general and not applied to a specific use of acoustic holography. However, there are citations which do discuss its use in medicine, nuclear reactors, and nondestructive testing. (This updated bibliography contains 211 abstracts, 50 of which are new entries to the previous edition.)

77-2056
Acoustic Holography (Citations from the NTIS Data Base)
W.E. Reed
National Technical Information Service, Springfield, VA., Rept. No. NTIS/PS-77/0578/3GA, 130 pp (July 1977)

Key Words: Acoustic holography, Bibliography

All aspects of acoustic holography are covered in this bibliography of Federally-funded research. Theory, equipment design, uses, and imaging techniques are presented. The applications include underwater and underground object locating, structural geology and tectonics, sonar imaging, non-destructive testing, antenna radiation patterns, nuclear reactor inspection, remote sensing, and use in medical examinations. (This updated bibliography contains 125 abstracts, 23 of which are new entries to the previous edition.)

77-2057

Environmental Pollution: Noise Pollution - Sonic Boom

Defense Documentation Center, Alexandria, VA., Rept. No. DDC/BIB-77/06, 201 pp (June 1977) AD-A041 400/3GA

Key Words: Sonic boom, Bibliographies

This bibliography contains citations of studies and analyses covering a wide range of the parameter of sonic boom and noise pollution, as well as damages caused by it. Corporate Author-Monitoring Agency, Subject, Title and Personal Author are provided.

77-2058

The Characteristics of Dynamic Loads and Response of Buildings

H.S. Ward

School of Engrg. Science, Plymouth Polytechnic, Plymouth PL4 8AA, UK, Shock Vib. Dig., 9 (8), pp 13-20 (Aug 1977) 3 figs, 42 refs

Key Words: Buildings, Seismic response, Reviews

This paper is concerned with structural dynamic problems involving buildings. Ground-borne disturbances including earthquakes, nuclear explosions, construction activities and vehicular traffic are discussed. Air-borne disturbances including wind and overpressures due to explosions are reviewed. Finally, thermal loads are included in the paper.

77-2059

Beam Vibrations - A Review

H. Wagner and V. Ramamurti

Indian Inst. of Tech., Madras, India, Shock Vib. Dig., 9 (9), pp 17-24 (Sept 1977) 115 refs

Key Words: Beams, Vibration response, Reviews

Most structural elements encountered in practice can be treated as beams sacrificing little accuracy. For this reason, this review article summarizes work on the vibration of beams since 1973.

77-2060

Turbomachinery Vibration

J.F. Traexler

Steam Turbine Div., Lester Branch, Westinghouse Electric Corp., Philadelphia, PA 19113, Shock Vib. Dig., 9 (8), pp 3-10 (Aug 1977) 8 figs

Key Words: Turbomachinery, Steam turbines, Vibration response, Rotors, Reviews

This article is concerned with turbomachinery vibrations, particularly those that occur in large steam turbines at central station power plants. Rotor dynamics and blading are reviewed.

77-2061

Exhaust Noise and Its Control - A Review

M.L. Munjal

Dept. of Mech. Engrg., Indian Inst. of Science, Bangalore - 12, India, Shock Vib. Dig., 9 (8), pp 21-32 (Aug 1977) 5 figs, 41 refs

Key Words: Mufflers, Noise reduction, Reviews

This article describes recent developments in the field of analysis and design of exhaust mufflers. The article is concerned only with exhaust noise.

77-2062

Acoustic Vibration of Structures in Liquids

D. Firth

Risley Engrg. and Materials Lab., United Kingdom Atomic Energy Authority, Risley, Warrington WA3 6AT, UK, Shock Vib. Dig., 9 (9), pp 3-7 (Sept 1977) 33 refs

Key Words: Submerged structures, Fluid-induced excitation, Acoustic excitation, Plates, Ducts, Reviews

This article outlines the physics of the vibration of an elastic structure excited by sound waves in a liquid in contact with the structure. The historical background is summarized, and some recent literature is described. Examples include plates, ducts, and complicated engineering systems. Possible future developments are suggested.

77-2063

Some Transient Problems of Structures Interacting with Fluid

D. Krajcinovic

Dept. of Materials Engrg., Univ. of Illinois at Chicago Circle, Chicago, IL, Shock Vib. Dig., 9 (9), pp 9-16 (Sept 1977) 5 figs, 29 refs

Key Words: Interaction: structure-fluid, Transient response, Reviews

This paper is a general r w of transient interaction problems involving either a nstant wetted surface or an expanding or receding wetted surface.

MODAL ANALYSIS AND SYNTHESIS

(See No. 2072)

COMPUTER PROGRAMS

GENERAL

77-2064

A FORTRAN IV Computer Program for the Time Domain Analysis of the Two-Dimensional Dynamic Motions of General Buoy-Cable-Body Systems

H.T. Wang

David W. Taylor Naval Ship Res. and Dev. Center, Bethesda, MD., Rept. No. DTNSRDC-77-0046, 95 pp (June 1977)

AD-A041 049/8GA

Key Words: Computer programs, Buoys, Cables, Dynamic response

The present report gives a detailed description of Program CABUOY, which analyzes in the time domain the two-dimensional dynamic behavior of general ocean cable systems consisting of a surface buoy, connecting cable, and intermediate bodies. The equations which model the motions of the surface waves and the various components of the cable system are presented, and the subroutines of the program are briefly outlined. Instructions on the use of the program include a listing of the input READ statements, definitions of the input variables, and a number of comments on the entering of input data. Several sample problems are given to illustrate use of the program, the output of the program, and computer costs for a range of cases. The listing of the program is given in the appendix.

77-2065

Computer Programs for the Calculation of Flexural Vibration of Turbomachinery Shafts (Programmsystem zur Berechnung von Biegeschwingungszuständen an Turbomaschinenwellen)

E. Thomas and K.-H. Schubert

VEB Bergmann Borsig/Görlitzer Maschinenbau, West Berlin, German Democratic Republic, Maschinenbautechnik, 26 (7), pp 322-326 (July 1977) 6 figs, 6 refs

(In German)

Key Words: Computer programs, Turbomachinery, Shafts

The article describes computer programs for the calculation of vibration behavior of turbomachinery shafts, available at the VEB Bergmann Borsig/Görlitzer Maschinenbau. The aim of the calculations in recent years has been to achieve a high degree of automation of the turbomachinery shaft vibration calculation taking the actual conditions as much as possible into consideration.

77-2066

Nonlinear Analysis of Frame Structures Subjected to Blast Overpressures

W. Stea, G. Tseng, D. Kossover, S. Weissman, and N. Dobbs

Ammann and Whitney, New York, NY, Rept. No. ARLCD-CR-77008, 440 pp (May 1977)

AD-A040 708/0GA

Key Words: Computer programs, Frames, Buildings, Blast resistant structures

In modern day explosive manufacturing and LAP facilities, many of the structural steel buildings will be required to provide protection for personnel and/or equipment against the effects of HE-type explosions. Therefore, computer program entitled 'Dynamic Nonlinear Frame Analysis' (DYNFA) has been developed whereby the responses of frame structures subjected to blast loadings can be determined. This report contains the background for the development of DYNFA as well as the equations and procedures necessary for its use. The report also contains example problems illustrating the use of DYNFA for the design of blast-resistant frame structure.

77-2067

First Report on Capabilities of Dynamic Structural Analysis by the Strudl Program (Primo Rapporto Sulle Capacita Di Analisi Dinamica Dello Strudl)

B. Atzori and F. Fresa

Ist. di Construzione di Macchine, Bari Univ., Italy, Rept. No. 76-2, 18 pp (Oct 1976)
(In Italian)
N77-26551

Key Words: Computer programs, Frames, Dynamic structural analysis

The capabilities of dynamic structural analysis by the STRUDL 2 program were studied. The case of frame analysis was examined for checking the validity of the results. Several factors, such as the influence of the number of elements on the approximation of the results and the CPU time necessary to solve some typical cases, were also investigated.

77-2068

A Sparsity-Oriented Approach to the Dynamic Analysis and Design of Mechanical Systems -- Part 1

N. Orlandea, M.A. Chace, and D.A. Calahan
Dept. of Mech. Engrg.., Iowa State Univ., Ames, IA, J. Engr. Indus., Trans. ASME, 99 (3), pp 773-779 (Aug 1977) 7 figs, 2 tables, 14 refs

Key Words: Computer programs, Computer-aided design, Suspension systems (vehicles), Landing gear

The work described herein is an extension of sparse matrix and stiff integrated numerical algorithms used for the simulation of electrical circuits and three-dimensional mechanical dynamic systems. By applying these algorithms big sets of sparse linear equations can be solved efficiently, and the numerical instability associated with widely split eigenvalues can be avoided. The new numerical methods affect even the initial formulation for these problems. In this paper, the equations of motion and constraints (Part 1) and the force function of springs and dampers (Part 2) are set up, and the numerical solutions for static, transient, and linearized types of analysis as well as the modal optimization algorithms are implemented in the ADAMS (automatic dynamic analysis of mechanical systems) computer program for simulation of three-dimensional mechanical systems (Part 2). The paper concludes with two examples: computer simulation of the front suspension of a 1973 Chevrolet Malibu and computer simulation of the landing gear of a Boeing 747 airplane. The efficiency of simulation and comparison with experimental results are given in tabular form.

77-2069

A Sparsity-Oriented Approach to the Dynamic Analysis and Design of Mechanical Systems -- Part 2

N. Orlandea, D.A. Calahan, and M.A. Chace
Dept. of Mech. Engrg., Iowa State Univ., Ames, IA, J. Engr. Indus., Trans. ASME, 99 (3), pp 780-784 (Aug 1977) 3 figs, 2 tables, 9 refs

Key Words: Computer programs, Computer-aided design, Suspension systems (vehicles), Landing gear

The work described herein is an extension of sparse matrix and stiff integrated numerical algorithms used for the simulation of electrical circuits and three-dimensional mechanical dynamic systems. By applying these algorithms, big sets of sparse linear equations can be solved efficiently, and the numerical instability associated with widely split eigenvalues can be avoided. The new numerical methods affect even the initial formulation for these problems. In this paper, the equations of motion and constraints (Part 1) and the force function of springs and dampers (Part 2) are set up, and the numerical solutions for static, transient, and linearized types of analysis as well as the model optimization algorithms are implemented in the ADAMS (automatic dynamic analysis of mechanical systems) computer program for simulation of three-dimensional mechanical systems (Part 2). The paper concludes with two examples: computer simulation of the front suspension of a 1973 Chevrolet Malibu and computer simulation of the landing gear of a Boeing 747 airplane. The efficiency of simulation and comparison with experimental results are given in tabular form.

77-2070

Torsional Vibration Calculations of Machine Tool Drives (Berechnung des Torsionsschwingungsverhaltens von Werkzeugmaschinenantrieben)

R. Böhm
Konstruktion, 29 (7), pp 259-264 (July 1977) 13 figs, 4 refs
(In German)

Key Words: Computer programs, Torsional vibration, Machine tools, Gear drives

Gear drives - especially spur gear drives - are the most commonly used main drives in machine tools. Earlier investigations have shown that the main drive has a very strong effect on the stability of machine tool. In the article a computer program BEIGE for calculation of torsional frequency and the shape of vibration is described, which requires as input only data taken from construction drawings. Experimental data confirm a sufficient accuracy of the method.

77-2071

Modal Frequency and Random Response of the Airbus A300B Antenna

H. Goedel and F. Weiss
Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, West Germany, Rept. No. UFE-1242-0, 14 pp (Apr 27, 1976) refs

N77-25378

Key Words: Antennas, Computer programs, Frequency response, Shells

Using the NASTRAN program system a computation of vibration and response was carried out for the ADF (Automatic Direction Finder) of the Airbus A300B in order to estimate the stress level within the scope of service life considerations. Using RIGID FORMAT 3 for normal mode computations and RIGID FORMAT 11 for power spectral density analysis, it was possible to achieve the actually obtained results for frequency responses in a simple way by means of the NASTRAN system.

77-2072

Stiffness Coupling Application to Modal Synthesis Program. Users Guide

E.J. Kuhar

General Electric Co., Philadelphia, PA., Rept. No. NASA-CR-145197, 26 pp (1976)

N77-25575

Key Words: Computer programs, Modal synthesis, Stiffness methods, Matrix methods

A FORTRAN IV computer program used to perform modal synthesis of structures by stiffness coupling, using the dynamic transformation method is described. The program was named SCAMP (Stiffness Coupling Approach Modal-Synthesis Program). The program begins with the entry of a substructure's physical mode shapes and eigenvalues or a substructure's mass and stiffness matrix. If the mass and stiffness matrices are entered, the eigen problem for the individual substructure is solved. Provisions are included for a maximum of 20 substructures which are coupled by stiffness matrix springs.

77-2073

A FORTRAN Program to Extract Static and Dynamic Moments from Free Oscillations in a Wind Tunnel

R.L. Pope

Weapons Research Establishment, Salisbury, Australia, Rept. No. WRE-TN-1729(WR/D), 42 pp (Dec 1976) refs

N77-25093

Key Words: Computer programs, Parameter identification, Wind tunnel tests

A FORTRAN program was developed using the parameter estimation technique to extract the static pitching moment and the dynamic pitch damping moment from incidence measurements taken during planar oscillations of a model in a wind tunnel. The advantage of the parameter estimation method of analysis in this particular case is its ability to treat highly nonlinear forms of the static pitching moment. Comparisons are made with other wind tunnel measurements. A listing of the program and a sample run are included.

ENVIRONMENTS

ACOUSTIC

(Also see Nos. 2055, 2056, 2108, 2153, 2158, 2160)

77-2074

Acoustic Diffraction. Part 1. Plane Diffractors and Wedges

E.J. Skudrzyk, S.I. Hayek, and A.D. Stuart

Applied Research Lab., Pennsylvania State Univ., University Park, PA., Rept. No. TM-73-109-Pt-1, 160 pp (May 14, 1973)

AD-A040 668/6GA

Key Words: Acoustic diffraction

This memorandum documents the theoretical investigations in the Acoustic Diffraction Program. This report discusses the acoustic diffraction and backscattering phenomena for plane and wedge scatterers which are insonified by plane or point sources. The theories of diffraction used in this report are those of the approximate integral representations of Kirchoff-Rubinowicz. Those were compared with the geometrical theory of diffraction (GTD) which is developed by J.B. Keller, and is based on the ray theory.

77-2075

Noise Due to the Interaction of Boundary Layer Turbulence with a Marine Propulsor or an Aircraft Compressor

N. Moiseev, B. Lakshminarayana, and D.E. Thompson

Applied Research Lab., Pennsylvania State Univ., University Park, PA., Rept. No. TM-76-258, 122 pp (Oct 11, 1976)

AD-A040 946/6GA

Key Words: Noise generation, Rotor blades, Compressors, Propulsion systems

The sound generated by the interaction of inlet boundary layer turbulence with a rotating blade row is investigated. To experimentally study this radiated sound, an existing aeroacoustic facility was modified to produce the inflows desired. The rotor was operated in air with different blade space-to-chord ratios, different flow coefficients and different anisotropic, nonhomogeneous turbulent inflows. The inflows ingested are: natural boundary layer on hub and annulus wall, a tripped boundary layer on the hub, and a fully developed boundary layer on the hub. The turbulence intensities and length scales were altered by placing a grid at the inlet.

77-2076

Industrial Noise Control: Putting it all Together

T.D. Miller

Donley, Miller and Nowikas, Inc., 56 State Highway 10, East Havover, NJ 07936, Noise Control Engr., 9 (1), pp 24-31 (July/Aug 1977) 7 figs, 1 table, 5 refs

Key Words: Noise control, Industrial facilities, Human response, Regulations

Industrial noise control has two fundamental objectives: to meet the requirements of federal law and to protect employees' hearing. The author outlines a total noise control program, and details some of the steps necessary to ensure that these goals are successfully met at minimum cost.

77-2077

Shielding Highway Noise

Z. Maekawa

Environmental Acoustics Lab., Kobe Univ., Rokko, Kobe, 657, Japan, Noise Control Engr., 9 (1), pp 38-44 (July/Aug 1977) 12 figs, 14 refs

Key Words: Noise barriers, Traffic noise

One of the most widespread problems in environmental acoustics is the control of road traffic noise. In urban areas and in the vicinity of residential districts especially, this has become an extremely serious issue. The author reviews typical methods of noise shielding, presents new results of experimental studies, and introduces some theoretical approaches.

77-2078

Two Experiments on the Perceived Noisiness of Periodically Intermittent Sounds

I. Pollack and R.M. Garrett

Dept. of Architecture, Muroran Inst. of Tech., 27 Mizumoto-cho, Muroran, Hokkaido, Japan 050, Noise Control Engr., 9 (1), pp 16-23 (July/Aug 1977) 10 figs, 4 tables, 15 refs

Key Words: Noise tolerance, Human response

The author describes a study aimed at clarifying the nature of the perceived noisiness of intermittent sounds, in order to establish an efficient method of assessment. Experimental results indicate that loudness and noisiness are different qualities. Further research delineates the structure of human response to these sounds.

PERIODIC

77-2079

A New Method for Predicting Response in Complex Linear Systems II

J.L. Bogdanoff, K. Kayser, and W. Krieger

School of Aeronautics and Astronautics, Purdue Univ., West Lafayette, IN 47907, J. Sound Vib., 53 (4), pp 459-469 (1977) 8 figs, 2 tables, 6 refs
Sponsored by NASA, Marshall Space Flight Center

Key Words: Linear systems, Random excitation, Steady state excitation, Lumped parameter method

A new method is presented for response estimation in complex lumped parameter linear systems under random or deterministic steady state excitation. The essence of the method is the use of relaxation procedures with a suitable error function to find the estimated response; natural frequencies and normal modes are not computed. For a 45 degree of freedom system, and two relaxation procedures, convergence studies are made. Frequency response estimates are made.

RANDOM

(Also see No. 2079)

77-2080

A Probabilistic Model for a Randomly Excited Flow

Y.K. Gayed, M.R. Haddara, and A.H.A. Baghdadi

Dept. of Mech. Engrg., Cairo Univ., Cairo, Egypt, Appl. Math. Modeling, 1 (6), pp 299-309 (Sept 1977) 9 figs, 1 table, 11 refs

Key Words: Hydroelectric power plants, Transient response, Random response, Mathematical models, Probability theory

This work concerns a probabilistic model of the random problem, whose solution gives the distribution and probability density functions of the variables involved, namely the pressures, velocities and surge tank oscillation. Order statistical methods were also used to estimate the probability of occurrence of extreme head fluctuations.

SEISMIC

(Also see Nos. 2097, 2151, 2169, 2170, 2171, 2185)

77-2081

Learning from Earthquakes. 1977 Planning and Field Guides

Earthquake Engrg. Research Inst., California Univ., Los Angeles, CA., Rept. No. NSF/RA-770081, 221 pp (1977)
PB-268 083/3GA

Key Words: Earthquake damage

The aim is to maximize the learning to be gained from investigations following future destructive earthquakes. The Guides are meant for use in the planning and field execution of such investigations. Through their use, both the afflicted communities and the investigators can understand how to participate in the investigation and what information is of greatest value.

77-2082

The Earthquake Response of Deteriorating Systems

N.C. Gates
Ph.D. Thesis, California Inst. of Tech., 140 pp (1977)
UM 77-19,980

Key Words: Linear systems, Earthquake response, Approximation methods, Stiffness methods, Energy methods

This thesis is concerned with the earthquake response of deteriorating systems. A model for stiffness degrading or deteriorating systems is used to describe six different single-degree-of-freedom systems. A numerical investigation of the response of these six systems is performed using an ensemble of twelve earthquakes. The response is studied at nine nominal periods of oscillation. The numerical results are presented as response spectra corresponding to six different ductilities. An approximate analytical method for calculating the earthquake response of deteriorating systems from a linear response spectrum is presented. The method, called the average stiffness and energy method, is based upon the premise that a linear system may be defined which is in some sense equivalent to the deteriorating system. The criterion for equivalence in this method is that the average stiffness of the deteriorating system be equal to the stiffness of the linear system and the average energy dissipated by the linear system be the same as the average energy dissipated by the deteriorating system. The new analytical method is compared to existing methods. Comparison with the numerical results is also made. Based upon these comparisons, it is concluded that the average stiffness and energy method represents a significant improvement over currently available methods for predicting the earthquake response of deteriorating and nondeteriorating systems.

77-2083

Investigation of the Inelastic Characteristics of a Steel Frame Using System Identification and Shaking Table Experiments

V.C. Matzen
Ph.D. Thesis, Univ. of California, Berkeley, 127 pp (1976)
UM 77-15,782

Key Words: Framed structures, Seismic response, System identification, Experimental results

In this dissertation, system identification is used to formulate a realistic nonlinear mathematical model to represent the seismic behavior of a single story steel structure. With this model and the parameters established for it, the energy absorbing characteristics of the structure are investigated. During this study, system identification itself is examined to determine how it can be better utilized in structural engineering. There are three major parts to this research. The first is the mathematical development of system identification to meet the particular needs of this problem. The second part of the research involved shaking table experiments in which a single story steel frame was subjected to several earthquake excitations. The last part of the research is the use of test data in the identification program to establish the four parameters in the mathematical model. When different values are used for T, parameter sets are established which give the best model response for that amount of test data. The resulting sets of parameters reflect the way in which the properties of the structure change during the excitation.

77-2084

Performance and Analysis of Earth Dams During Strong Earthquakes

F.I. Makdisi
Ph.D. Thesis, Univ. of California, Berkeley, 248 pp (1976)
UM 77-15,778

Key Words: Dams, Earthquake response

An investigation into the behavior of a number of earth dams that were severely shaken during the San Francisco 1906 earthquake was undertaken to identify the factors contributing to their adequate performance. It was found that the majority of these embankments consisted of predominantly clay soils. On the basis of the knowledge of the behavior of clays under cyclic loading conditions, it is shown that the clayey nature of these embankments was the significant factor contributing to their stability during the earthquake. In addition, the contrasting behavior of sandy embankments is demonstrated by studying the failure and near failure of a number of embankments during four other earthquakes in California and Japan.

SHOCK

(Also see Nos. 2048, 2066, 2150, 2186, 2188, 2189)

77-2085

Surface Waves Generated by Shallow Underwater Explosions

A. Falade

Ph.D. Thesis, Univ. of California, Berkeley, 93 pp (1976)

UM 77-15,679

Key Words: Underwater explosions, Explosion effects

In this report, surface water waves generated by surface and near surface point explosions are calculated. Taking impulse distribution imparted at the water surface by the explosion as the overriding mechanism for transferring energy of the explosive to surface wave motion, the linearized theory of Kranzer and Keller is used to obtain the wave displacement in the far field.

GENERAL WEAPON

77-2086

Parametric Resonance in Gun Tubes

T.E. Simkins

Watervliet Arsenal, NY, Rept. No. WVT-TR-77009, 70 pp (Feb 1977)

AD-A040 677/7GA

Key Words: Gun barrels, Parametric resonance

This work examines the likelihood of encountering parametric resonance in gun tubes. The resonance is induced conceptually by the periodic changes in transverse stiffness induced by the axial vibrations resulting from a single application of ballistic pressure - 'single round parametric resonance', the periodic applications of ballistic pressure such as encountered in an automatic weapon - 'multiple round parametric resonance'.

TRANSPORTATION

(Also see Nos. 2186, 2188, 2189)

77-2087

Experimental Designs and Psychometric Techniques for the Study of Ride Quality

M.D. Havron and R.A. Westin

ENSCO, Inc., Springfield, VA., Rept. No. DOT-TSC-OST-76-54, 301 pp (May 1977)

PB-268 584/0GA

Key Words: Transportation vehicles, Ride dynamics, Human response, Statistical analysis

A major variable in both the cost of any new transportation system and rider acceptance of the system is the ride quality of its vehicles. At this time, there exists no set of objective criteria which would allow the transportation system designer to determine what level of ride quality would be considered acceptable by a wide variety of potential passengers. The purpose of the study was to establish statistically acceptable techniques for the development of methods for relating physical measures of vehicle vibration to passenger estimates of ride quality.

PHENOMENOLOGY

DAMPING

(Also see Nos. 2107, 2192)

77-2088

Tuned Mass Dampers for Buildings

R.J. McNamara

Gillum-Colaco Consulting Struct. Engrs., Cambridge, MA., ASCE J. Struc. Div., 103 (ST9), pp 1785-1798 (Sept 1977) 13 figs, 14 refs

Key Words: Tuned dampers, Buildings, Single degree of freedom systems

Tuned mass dampers attached to single degree-of-freedom systems representing tall buildings are studied. System equations are formulated and solved for various input forcing functions. Design parameters of the damper are varied to study the response reduction. Experimental wind tunnel results are presented, and a practical application of a large-scale damper is illustrated.

77-2089

The Damping of Structural Vibration by Rotational Slip in Joints

C.F. Beards and J.L. Williams

Dept. of Mech. Engrg., Imperial College of Science and Tech., London SW7 2BX, UK, J. Sound Vib., 53 (3), pp 333-340 (Aug 8, 1977) 5 figs, 3 tables, 9 refs

Key Words: Slip joints, Coulomb friction, Computer programs

Interfacial slip in joints is the major contributor to the inherent damping of most fabricated structures. By fastening joints tightly enough to prohibit translational slip, but not tightly enough to prohibit rotational slip (thereby making only a small sacrifice in static stiffness), it is shown, both experimentally and theoretically, that a useful increase in the inherent damping in a structure can be achieved, provided an optimum joint load is maintained. The analysis is simplified by using a general dynamic analysis computer program with a sub-program to model the friction joint.

77-2090

Some Comments on the Estimation of Resonant Peak Amplitudes

R.E.D. Bishop

Dept. of Mech. Engrg., Univ. College London, Torrington Place, London WC1E 7JE, UK, Ing. Arch., 45 (5/6), pp 331-336 (1976) 4 figs, 5 refs

Key Words: Resonant response, Damped structures, Forced vibration

In a recently published paper, a way of estimating resonant responses of a damped system by means of calculations for the undamped system was suggested. No reference was made to the existing literature on the theory of forced vibration. The object of the present paper is to show how his approach fits in and, in particular, to illustrate what it implies in terms of polar response plots.

77-2091

Subsynchronous Resonance in Power Systems: Damping of Torsional Oscillations

K.T. Khu

Ph.D. Thesis, Iowa State Univ., 154 pp (1977)
UM 77-16,962

Key Words: Electric generators, Vibration resonance, Torsional vibrations, Self-excited vibrations, Hunting, Vibration damping

Studies of subsynchronous resonance phenomena are conducted in a power system composed of a tandem-compound steam turbo-generator set connected to an infinite bus via a series capacitor compensated transmission line. Complete detailed representation of the electromechanical system has confirmed the existence of (n-1) modes of oscillation, where n is the number of lumped masses of the shaft, as well as the existence of super- and subsynchronous components in the electrical network. The eigenvalue method of analysis is used to study the interaction between the mechanical and electrical networks under small perturbations, and to identify the conditions in which the system would be subjected to torsional interaction, self-excitation, and hunting. Transient analysis is carried out on an analog computer to observe the electrical quantities and the torques of the various sections of the shaft before, during, and after a three-phase fault is applied.

ELASTIC

77-2092

Dynamic Stresses Produced in an Elastic Half Space by Reciprocally Moving Surface Loads

T. Ohyoshi

Mining College, Akita Univ., Akita, Japan, Bull. JSME, 20 (145), pp 777-784 (July 1977) 10 figs, 5 refs

Key Words: Elastic properties, Half space, Moving loads

In studies of moving load problems, Galilean or Laplacian transformations have been commonly used by several previous investigators to construct the solutions. In this paper analytical techniques of superposition of harmonic vibrations are available because the elements composing an elastic half space are excited periodically by reciprocating surface loads.

77-2093

Elastodynamic Analysis of a Completely Elastic System

D. Kohli, D. Hunter, and G.N. Sandor

Univ. of Wisconsin, Milwaukee, WI., J. Engr. Indus., Trans. ASME, 99 (3), pp 604-609 (Aug 1977) 3 figs, 1 table, 16 refs

Key Words: *Slider crank mechanisms*, Elastodynamic response, Transverse shear deformation effects, Rotatory inertia effects

The completely elastic system considered for this vibration analysis consists of an offset slider-crank mechanism having elastic supports and mountings of the mechanism permitting translational vibrations of the shafts and supports, elastic shafts permitting torisonal vibrations, elastic links of the mechanism which deform due to external or internal body forces and allow flexural and axial vibrations. Both the effect of the deformations caused by the inertia forces in the mechanism links, shafts, and supports and the effect of change in the inertia forces due to these deformations are taken into account in constructing a general mathematical model for conducting elastodynamic analysis.

FLUID
(See No. 2062)

SOIL
(Also see No. 2185)

77-2094
Dynamic Torsional Response of Foundations on Layered Media
A. Prodanovic
Ph.D. Thesis, Rice Univ., 278 pp (1977)
UM 77-19,285

Key Words: Footings, Foundations, Torsional response, Layered materials

A study is made of the steady-state harmonic torsional response of a rigid circular footing perfectly bonded to the surface of a layered elastic or viscoelastic medium, the footing being excited either kinematically or under the action of a torque. The supporting medium is assumed to consist of a finite number of horizontal layers of constant thickness overlying a homogeneous half-space. Primary attention is given to the problems involving a single layer over a homogeneous half-space and a stratum over a rigid base; the homogeneous half-space is also considered as a limiting case.

77-2095
Dynamics of Certain Structure-Foundation Interacting Systems
J.B. Valdivieso
Ph.D. Thesis, Rice Univ., 227 pp (1977)
UM 77-19,297

Key Words: Interaction: structure-foundation

An analytical investigation of three interrelated problems in the general area of structure-foundation interaction is conducted. The effects of the presence of a substantial foundation mass on the response of interacting systems is initially studied. The foundation medium is assumed to be a halfspace with elastic or viscoelastic properties. Attention is given to the effects of foundation mass on the magnitude of the forces developed during motion since these generally govern the structural design. The applicability of the use of a Single Degree of Freedom equivalent oscillator to predict the dynamic behavior of a soil-structure interacting system with a finite foundation mass is assessed.

77-2096
Dynamic Response of Friction Piles
C.-S. Chon
Ph.D. Thesis, The Univ. of Michigan, 232 pp (1977)
UM 77-17,968

Key Words: Interaction: soil-structure, Pile structures

The influence of several "soil-pile interaction" parameters on the dynamic and static response of single friction piles to lateral loads were studied by performing model pile tests and comparing the results with theoretical analyses. Both dynamic and static model pile tests were performed in a specially constructed facility which was designed to operate as a "quick-sand" tank. The quicksand operation provided for rapid and easy reconstitution of the fine, uniform sand to preselected conditions before each test.

77-2097
Seismic Response of Axisymmetric Soil-Structure Systems
E. Berger
Ph.D. Thesis, Univ. of California, Berkeley, 189 pp (1976)
UM 77-15,607

Key Words: Interaction: soil-structure, Seismic response, Finite element techniques, Computer programs, Nuclear power plants

The accuracy of seismic response computations made with two-dimensional finite element methods of analysis applied to three-dimensional soil-structure systems is investigated. The three-dimensional soil-structure system is modeled by an axisymmetric finite element model while the equivalent two-dimensional system is represented by a plane strain model. A finite element computer code ALUSH is developed which computes the seismic response of axisymmetric soil-structure systems subjected to horizontal, vertical and rotational earthquake input motions. The nonlinear stress-strain behavior of soil masses subjected to strong earthquake motions and the frequency independent nature of the damping characteristics of soils are considered in the method of analysis by use of equivalent linear method and the complex response method, respectively.

77-2098
Unified Boundary for Finite Dynamic Models
W. White, S. Valliappan, and I.K. Lee
Dept. of Civil Engrg. Materials, The Univ. of New South Wales, Kensington, New South Wales, Australia, ASCE J. Engr. Mech. Div., 103 (EM5), pp 949-964 (Oct 1977) 4 figs, 3 tables, 8 refs

Key Words: Soils, Dynamic response, Finite element technique, Energy absorption

The finite element analysis of dynamic problems in an infinite, isotropic medium is examined. To simulate the physically infinite system by a finite model, an energy absorbing boundary is proposed. This boundary is frequency independent and proves to be very efficient in absorbing stress waves. The boundary constants are calculated for the particular cases of plane strain and axisymmetry for isotropic materials.

77-2099
Hydrodynamic Pressure in Semicylindrical Reservoir
F.J. Sanchez-Sesma and E. Rosenblueth
Instituo de Ingeniería, Universidad Nacional Autónoma de México, México, ASCE J. Engr. Mech. Div., 103 (EM5), pp 913-919 (Oct 1977) 4 figs, 3 tables, 11 refs

Key Words: Dams, Modal analysis, Seismic design, Hydrodynamic excitation

Solutions are presented for modal analysis of hydrodynamic pressures generated by the three translational seismic components - longitudinal, vertical, and transverse - on a dam limiting a semicircular cylindrical reservoir. The main purpose is to show the influence of the cross-sectional shape of the reservoir in the hydrodynamic responses. Results are compared with those for rectangular cross section.

VISCOELASTIC

77-2100
Design of a Viscoelastic Dynamic Absorber for Machine Tool Applications
G.L. Nessler, D.L. Brown, D.C. Stouffer, and K.C. Maddox
Appl. Dynamics & Acosutics Section, Battelle Columbus Labs., Columbus, OH, J. Engr. Indus., Trans. ASME, 99 (3), pp 620-623 (Aug 1977) 5 figs, 11 refs

Key Words: Machine tools, Viscoelastic damping

The design equations are developed for a viscoelastic dynamic absorber in uniaxial compression. The dependence of mechanical properties of the absorber on frequency, temperature, and preload are developed through an extension of the thermorheologically simple theory of linear viscoelasticity. An approximation of the exact boundary value problem is made in order to develop practical design criteria for the size and shape of the absorber element. The results of the experimental program for the constitutive equation are included. A dynamic absorber is designed to control a self-excited lathe chatter problem and a significant improvement is demonstrated.

EXPERIMENTATION

DIAGNOSTICS

(Also see No. 2126)

77-2101
Increase Plant Availability with Trend Monitoring
E.G. Filetti and P.R. Trumpler
Energy Technology, Inc., West Chester, PA, Hydrocarbon Processing, 56 (9), pp 233-240 (Sept 1977) 5 figs, 2 refs

Key Words: Diagnostic techniques, Machinery vibration, Critical speed, Whirling

Trend monitoring is a modern engineering method designed to minimize unscheduled process plant shutdowns by anticipating malfunctions in on-line machines. The onset of machine problems is usually detected as an increase in vibration level. Two particularly important machine characteristics, lateral critical speeds and whirl, are discussed in some detail. Several applications are also described.

77-2102
A Survey of Design Methods for Failure Detection in Dynamic Systems
A.S. Willsky

Electronic Systems Lab., Massachusetts Inst. of Tech., Cambridge, MA., In: AGARD Integrity in Electron. Flight Control Systems, 14 pp (Apr 1977) refs (N77-25055)
N77-25060

Key Words: Diagnostic techniques, Dynamic systems, Nonlinear systems

A number of methods for the detection of abrupt changes (such as failures) are surveyed in stochastic dynamical systems. The class of linear systems is concentrated, but the basic concepts, if not the detailed analyses, carry over to other elements of systems. The methods range from the design of specific failure-sensitive filters, to the use of statistical tests on filter innovations, to the development of jump process formulations. Tradeoffs in complexity versus performance are discussed.

77-2103
What Can Mini-Computers do for Machinery Reliability?
R.G. Harker
Bently Nevada Corp., Minden, NV, Hydrocarbon Processing, 56 (8), pp 137-143 (Aug 1977) 11 figs

Key Words: Diagnostic techniques

As major turbomachinery trains become more complex and critical, condition monitoring for maximum reliability becomes more important. Dedicated mini-computer systems appear to be the coming way to perform this task.

FACILITIES

77-2104
An Historical View of Dynamic Testing
H.C. Pusey
Naval Research Lab., Shock and Vibration Information Center, Washington, D.C., J. Environ. Sci., 20 (5), pp 9-14 (Sept/Oct 1977) 83 refs

Key Words: Dynamic testing, Reviews

Developments in the field of dynamic testing over the past thirty years are examined. Assessment of present capabilities and future needs leads to the conclusion that the problems to be solved are more managerial than technical. Some controversial questions are posed with respect to dynamic tests and specifications.

INSTRUMENTATION

77-2105
New Electromagnetic Transducers for Recording Translations and Vibrations
B.Z. Kaplan
Dept. of Electrical Engrg., Ben Gurion Univ. of the Negev, Be'er Sheva, Israel, Israel J. Tech., 14 (4/5), pp 187-195 (1976) 10 figs, 12 refs

Key Words: Transducers, Measuring instruments, Recording instruments, Vibration measurement, Vibration recording

The paper discusses new instrumentation developed for measuring translations and vibrations of mechanical parts. Electromagnetic fields are employed for these measurements, and mechanical loading is, therefore, avoided. At first one-sided capacitive transducers are treated. Secondly, differential capacitive transducers are discussed. An electronic method is investigated by which the operation of such differential bridges can be maintained linear even if the deviation of the moving member from its central position was large. The last parts of the paper deal with microwave interferometric bridges. It is shown that movements of remote objects with amplitudes in the micrometer region can be recorded from distances of several meters.

TECHNIQUES
(Also see Nos. 2087, 2159)

77-2106
Application of Modal Testing Techniques to Solve Vibration Problems in Machinery Supporting Structures
J.W. Martz and T. Leist
Structural Dynamics Research Corp., Cincinnati, OH, ASME Paper No. 77-DE-16

Key Words: Testing techniques, Modal testing, Machine foundations

This paper describes the use of state-of-the-art testing techniques to solve vibration problems that result from design incompatibility between machinery and the machinery supporting structures. The general techniques of "mechanical impedance," or "modal" testing described herein have become widely used in the laboratory over the past several years to solve vibration problems in machine tools, automotive vehicles, construction, and agricultural machinery.

77-2107

A Forced-Vibration Technique for Measurement of Material Damping

R.F. Gibson and R. Plunkett
Dept. of Engrg. Science and Mech., and Engrg. Res. Inst., Iowa State Univ., Ames, IA 50011, Exptl. Mech., 17 (8), pp 297-302 (Aug 1977) 9 figs, 18 refs

Key Words: Measurement techniques, Material damping

This article describes a technique for measuring material damping in specimens under forced flexural vibration. Although the method was developed for testing fiber-reinforced composite materials, it could be used for any structural material. The test specimen is a double-cantilever beam clamped at its midpoint and excited in resonant flexural vibration by an electromagnetic shaker. Under steady state conditions, material damping is defined in terms of the ratio of input energy to strain energy stored in the specimen. If external losses are negligible, the input energy must equal the energy dissipated in the specimen. Input energy and strain energy are found from measured specimen dimensions, resonant frequency, input acceleration and bending strain. Problems associated with minimization of external energy losses in the apparatus and verification of measurements are discussed in detail. Measured damping of aluminum-alloy calibration specimens shows good agreement with calculated thermoelastic damping. Examples of measured damping showing amplitude and frequency dependence in fiber-reinforced plastic materials are presented.

77-2108

Characteristics and Calibration of Reference Sound Sources

P. Francois
Electricité de France, 1 Avenue General de Gaulle, 92141 Clamart, France, Noise Control Engr., 9 (1), pp 6-15 (July/Aug 1977) 9 figs, 3 tables, 8 refs

Key Words: Noise measurement, Measurement techniques

The reference sound source - a source of known acoustic power output - was developed in the United States in the mid-1950s. Several new devices to simplify the determination of sound power have emerged since 1970, and standards for the characteristics, calibration, and usage of these instruments are now being developed. Current sources and some proposed techniques for calibration are discussed by the author.

COMPONENTS

ABSORBERS

(See No. 2100)

BEAMS, STRINGS, RODS, BARS

(Also see Nos. 2059, 2064, 2065)

77-2109

The General Solution to the Classical Problem of Finite Euler Bernoulli Beam

M.Y. Hussaini and C.L. Amba-Rao
Ames Research Center, NASA, Moffett Field, CA., Rept. No. NASA-TM-X-73253; A-7076, 13 pp (June 1977)
N77-26533

Key Words: Beams, Bernoulli theory, Free vibration, Forced vibration, Winkler foundations, Viscous damping

An analytical solution is obtained for the problem of free and forced vibrations of a finite Euler Bernoulli beam with arbitrary (partially fixed) boundary conditions. The effects of linear viscous damping, Winkler foundation, constant axial tension, a concentrated mass, and an arbitrary forcing function are included in the analysis. No restriction is placed on the values of the parameters involved, and the solution presented here contains all cited previous solutions as special cases.

77-2110

Thin-Walled Curved Beam Finite Element

S.K. Chaudhuri and S. Shore
ASCE J. Engr. Mech. Div., 103 (EM5), pp 921-937 (Oct 1977) 9 figs, 5 tables, 21 refs

Key Words: Curved beams, Bridges, Moving loads

The generalized displacements and forces at the two nodes of the beam elements are: three translations and their corresponding forces, three rotations and their corresponding moments, the out-of-plane warping of the end cross section and its corresponding bi-moment. The solutions to the homogeneous differential equations governing the static deformation of curved beams along with kinematical boundary conditions are given. The stiffness matrix is formed by evaluating the stress resultants at the two ends of the element

corresponding to each unit generalized displacement. The method using the principle of virtual work to obtain the equivalent nodal forces due to external loading and the consistent mass matrix is outlined. Several examples are presented and comparisons made to demonstrate the accuracy and the usefulness of the element. This element has been successfully used in the finite element discretization of curved girders of horizontally curved highway bridges in studying the response of the bridges subjected to moving loads.

77-2111

Response of Beam to Stochastic Boundary Excitation

S.F. Masri and A. Aryafar
Dept. of Civil Engrg., Univ. of Southern California, Los Angeles, CA., ASCE J. Engr. Mech. Div., 103 (EM5), pp 807-822 (Oct 1977) 14 figs, 7 refs

Key Words: Beams, Boundary condition effects, Bernoulli-Euler method, Stochastic processes

A closed-form solution is presented for the covariance kernel of the transient response of a damped Bernoulli-Euler beam with arbitrary boundary conditions to correlated stochastic excitation applied at the boundaries. The analytical results are applied to the case where the autocorrelation function of the excitation resembles that of a wide class of input functions including earthquake excitations. The mean-square transient response at arbitrary locations along the beam is evaluated, and the effects of various system parameters are determined.

77-2112

Dynamic Responses of Viscoelastic Continuous Beams on Elastic Supports

K. Nagaya and Y. Hirano
Faculty of Engrg., Yamagata Univ., Yonezawa, Japan, Bull. JSME, 20 (145), pp 785-792 (July 1977) 10 figs, 12 refs

Key Words: Continuous beams, Viscoelastic properties, Elastic foundations

This paper deals with the vibration and the transient response problems of a viscoelastic continuous beam on non-periodic elastic supports. In the analysis, the restoring forces of the elastic supports are regarded as unknown external forces applied to the beam. The solution for the viscoelastic beam is obtained from the correspondence principle by applying the Laplace transform to the constitutive equation and the equation of motion for the elastic beam in terms of these unknown forces.

77-2113

Experimental Assessment of the Mindlin-McNiven Rod Theory

H.D. McNiven and Y. Mengi
Univ. of California, Berkeley, CA 94720, J. Acoust. Soc. Amer., 62 (3), pp 589-594 (Sept 1977) 8 figs, 6 refs

Key Words: Rods, Axisymmetric vibrations, Approximation methods, Experimental data

The three-mode theory due to Mindlin and McNiven, [J. Appl. Mech. 27, 145-151 (1960)] governing axisymmetric motions in a circular rod, is appraised by comparing responses predicted by it with experimental data obtained by Miklowitz and Nisewanger [J. Appl. Mech. 24, 240-244 (1957)]. *The problem studied involves a semi-infinite rod,* made of 24S-T aluminum alloy, subjected to pressure applied to the end of the rod. The two sets of responses are compared at various stations along the rod. To make the comparisons meaningful, it was necessary to recognize that the pressure applied experimentally had a finite rise time, however short; to make an estimate from the responses of what that rise time might be; and then apply this time distribution of pressure in evaluating the theoretical responses.

BEARINGS

77-2114

Consideration of the Negative Pressure Field at the Computation of Dynamic Loaded Radial Sliding Bearings. Model of a Fluid-Gas-Mixture in the Lubrication Gap (Einbeziehung des Unterdruckgebietes in die Berechnung dynamisch belasteter Radialgleitlager. Modell eines Flüssigkeits-Gas-Gemischs im Schmierspalt)

R. Wegmann
Wilhelm-Pieck-Universität Rostock, German Dem. Republic, Maschinenbautechnik, 26 (7), pp 320-321 (July 1977) 2 figs, 8 refs
(In German)

Key Words: Slider bearings, Dynamic response

The article shows that for the calculation of dynamically loaded sliding bearings it is necessary to consider the negative pressure field. The behavior of lubricants at low pressures is described and a model for the fluid-gas-mixture is set up.

77-2115

A Cantilever Mounted Resilient Pad Gas Bearing

I. Etsion

Lewis Res. Center, NASA, Cleveland, OH, Rept. No. NASA-CASE-LEW-12569-1, 12 pp (Apr 28, 1977) PAT-APPL-792 069/GA

Key Words: Gas bearings

The patent application relates to a gas-lubricated bearing employing at least one pad mounted on a rectangular cantilever beam to produce a lubricating wedge between the face of the pad and a moving surface. The load-carrying and stiffness characteristics of the pad are related to the dimensions and modulus of elasticity of the beam. The invention is applicable to a wide variety of types of hydrodynamic bearings.

BLADES

(Also see No. 2174)

77-2116

Wind Tunnel Tests of a Two Bladed Model Rotor to Evaluate the TAMI System in Descending Forward Flight

R.P. White, Jr.

Rasa Div., Systems Research Labs., Inc., Newport News, VA., Rept. No. NASA-CR-145195, 53 pp (May 1977) refs

N77-25080

Key Words: Rotor blades, Noise reduction, Vortex induced excitation

A research investigation was conducted to assess the potential of the Tip Air Mass Injection system in reducing the noise output during blade vortex interaction in descending low speed flight. In general it was concluded that the noise output due to blade vortex interaction can be reduced by 4 to 6 db with an equivalent power expenditure of approximately 14 percent of installed power.

77-2117

Flap/Lag Torsion Dynamics of a Uniform, Cantilever Rotor Blade in Hover

W. Johnson

Ames Res. Center, NASA, Moffett Field, CA., Rept. No. NASA-TM-73248; A-7063, 19 pp (May 1977) Sponsored in part by the U.S. Army Air Mobility Res. and Dev. Lab., Moffett Field, CA

N77-26068

Key Words: Rotor blades, Dynamic stability

The dynamic stability of the flap/lag/torsion motion of a uniform, cantilever rotor blade in hover is calculated. The influence of blade collective pitch, lag frequency, torsional flexibility, structural coupling, and precone angle on the stability is examined. Good agreement is found with the results of an independent analytical investigation.

77-2118

Unsteady Hovering Wake Parameters Identified from Dynamic Model Tests. Part 1

K.H. Hohenemser and S.T. Crews

Dept. of Mech. Engrg., Washington Univ., St. Louis, MO., Rept. No. NASA-CR-152022, 120 pp (June 1977)

N77-26077

Key Words: Rotor blades, Parameter identification, Perturbation theory

The development of a 4-bladed model rotor is reported that can be excited with a simple eccentric mechanism in progressing and regressing modes with either harmonic or transient inputs. Parameter identification methods were applied to the problem of extracting parameters for linear perturbation models, including rotor dynamic inflow effects, from the measured blade flapping responses to transient pitch stirring excitations. These perturbation models were then used to predict blade flapping response to other pitch stirring transient inputs, and rotor wake and blade flapping responses to harmonic inputs. The viability and utility of using parameter identification methods for extracting the perturbation models from transients are demonstrated through these combined analytical and experimental studies.

DUCTS

(Also see No. 2062)

77-2119

Transmission of Sound Through Nonuniform Circular Ducts with Compressible Mean Flows

A.H. Nayfeh, B.S. Shaker, and J.E. Kaiser

Dept. of Engrg. Science and Mech., Virginia Polytechnic Inst. and State Univ., Blacksburg, VA., Rept. No. NASA-CR-145126, 66 pp (May 1977)

N77-25914

Key Words: Ducts, Sound transmission, Sound attenuation, Computer programs

An acoustic theory is developed to determine the sound transmission and attenuation through an infinite, hard-walled or lined, circular duct carrying compressible, sheared, mean flows and having a variable cross section. The theory is applicable to large as well as small axial variations, as long as the mean flow does not separate.

77-2120
Sound Attenuation in Multiply Lined Rectangular Ducts Including Effects of the Wall Impedance Discontinuities. Part 2: Liners in Parallel
W. Koch
Deutsche Forschungs- und Versuchsanstalt für Luft- und Raumfahrt, Goettingen, West Germany, Rept. No. ESA-TT-399, DLR-FB-76-58, 42 pp (Nov 11, 1976) refs
(In German)
N77-25917

Key Words: Ducts, Acoustic liners, Noise reduction

The problem of sound attenuation by a combination of two acoustic liners of finite length and of different wall impedance on opposite walls in an infinitely long rectangular duct was formulated as a Wiener-Hopf problem for zero mean flow. A coupled system of two generalized Wiener-Hopf equations was obtained and solved. Numerical results are given for a realistic wall impedance model. The influence of several liner parameters on sound attenuation is displayed graphically.

FRAMES, ARCHES

77-2121
The Steady State Response of Geometrically Non-Linear Shallow Arches
D. Hitchings and P. Ward
Dept. of Aeronautics, Imperial College of Science and Tech., London, UK, Intl. J. Numer. Methods Engr., 11 (8), pp 1261-1269 (1977) 5 figs, 9 refs

Key Words: Arches, Periodic response, Finite element technique

The non-linear steady state response of structures with curvature is investigated through the expository example of a shallow circular arch. A consistent mass finite element formulation is employed to derive the governing non-linear differential equations. These equations are solved by assuming a single mode expansion reducing the governing equations to the single degree-of-freedom Duffing's equation with a quadratic term. The non-symmetric amplitude-frequency curve is derived and compared with results previously obtained by direct integration of the equations of motion.

GEARS

77-2122
Digital Simulation of Impact Phenomenon in Spur Gear Systems
R.C. Azar and F.R.E. Crossley
Dept. of Mech. Engrg., Western New England College, Springfield, MA., J. Engr. Indus., Trans. ASME, 99 (3), pp 792-798 (Aug 1977) 11 figs, 17 refs

Key Words: Gears, Shafts, Impact pairs, Digital simulation, Torsional vibrations

A digital simulation model is developed to represent a lightly geared torsional system consisting of a drive unit, spur gear pair and load connected by flexible shafts. A clearance model called an Impact Pair is used to represent the gear pair and includes the effects of backlash, time-varying stiffness and damping of the gear teeth and tooth-form error. Experimentally determined frequency spectra of the torsional oscillations of a gear-driven shaft have been plotted and reported on earlier. Similar frequency plots are obtained from the simulation study, and data from these plots are compared with the experimental results for a variety of parameter changes including shaft speed, backlash and load. Results indicate that the simulation model portrays reasonably well the torsional behavior of the output shaft.

77-2123
Dynamic Stability of a Two-Stage Gear Train Under the Influence of Variable Meshing Stiffnesses
G.V. Tordion and R. Gauvin
Dept. of Mech. Engrg., Laval Univ., Quebec, P.Q., Canada, J. Engr. Indus., Trans. ASME, 99 (3), pp 785-791 (Aug 1977) 11 figs, 4 refs

Key Words: Gears, Parametric excitation, Dynamic stability

In a two-stage gear train, the two meshing stiffnesses acting on the intermediate shaft produce parametric vibrations. Equations to find the principal and secondary regions of instability are given. Results showing the influence of the phase angle between both meshing stiffnesses are presented. An easy way to determine whether a certain operating condition lies in a stability or instability region is also suggested.

77-2124

Measurement and Evaluation of Geared Engine Noises (Messung und Beurteilung der Geräusche von Getriebemotoren)

H. Greiner

Industrie-Anz., 98 (72), pp 1281-1284 (1976)

(In German)

Key Words: Gears, Engine noise

The article describes the causes of gear noises in engines. Gear sizes, speed reduction, skew angle, profile offset, profile correction, concentricity of the shaft end, gear material and hardness, lubrication, relative loading are considered. The article also describes how the measured noises are evaluated, analysis and evaluation of noise frequencies, ISO noise rating, determining factors for noise intensity of geared motors, and decline of noise level with distance.

77-2125

Statistical Analysis of Dynamic Loads on Spur Gear Teeth

T. Tobe and K. Sato

Faculty of Engrg., Tohoku Univ., Sendai, Japan, Bull. JSME, 20 (145), pp 882-888 (July 1977) 15 figs, 11 refs

Key Words: Gears, Dynamic loads, Statistical analysis

Analysis of transmission error curve of a pair of gears measured by a single flank meshing tester shows that the error can be separated into harmonic components and random ones. In this paper the effect of the random components of the error on dynamic loads is analyzed theoretically. One example of numerical result is shown.

LINKAGES

(Also see No. 2089)

77-2126

The Theory of Torque, Shaking Force, and Shaking Moment Balancing of Four Link Mechanisms

J.L. Elliott and D. Tesar

Dept. of Mech. Engrg., Univ. of Florida, Gainesville, FL., J. Engr. Indus., Trans. ASME, 99 (3), pp 715-722 (Aug 1977) 9 figs, 10 tables, 30 refs

Key Words: Linkages, Mechanisms, Balancing

A method for the driving torque, shaking moment, and shaking force balancing is given as individual or combined problems for all of the four-link mechanisms: the four-bar, the slider-crank, the inverted slider-crank, and the oscillating block mechanism.

77-2127

A Numerical Method for the Dynamic Analysis of Mechanical Systems in Impact

R.E. Beckett, K.C. Pan, and S.C. Chu

Gen. Thomas J. Rodman Lab., Rock Island Arsenal, Rock Island, IL., J. Engr. Indus., Trans. ASME, 99 (3), pp 665-673 (Aug 1977) 15 figs, 1 table, 16 refs

Key Words: Mechanisms, Linkages, Dynamic response, Numerical analysis

A general procedure is developed for solving mechanism problems where intermittent separations and impacts can occur between mating parts. The numerical technique employed to solve the problem identifies the onset of separation and gives the behavior of the mechanism during separation and impact.

77-2128

Shape and Frequency Composition of Pulses From an Impact Pair

R.G. Herbert and D.C. McWhannell

Dept. of Mech. Engrg., Univ. of Southampton, UK, J. Engr. Indus., Trans. ASME, 99 (3), pp 513-518 (Aug 1977) 10 figs, 9 refs

Key Words: Impact pairs, Linkages, Mechanisms, Noise generation

With the need to improve the reliability and noise emmissions from real mechanisms, an impact in the classical impact pair configuration is investigated. The impact pulse level and its frequency composition as possible sources of high-frequency energy in articulated systems is considered.

77-2129

Dynamic Response of a Cam-Actuated Mechanism with Pneumatic Coupling

F.Y. Chen

Dept. of Mech. Engrg., Ohio Univ., Athens, OH, J. Engr. Indus., Trans. ASME, 99 (3), pp 598-603 (Aug 1977) 7 figs, 6 refs

Key Words: Cam followers, Pneumatic equipment, Stability, Dynamic response, Lumped parameter method

The dynamic characteristics of a cam-actuated system whose follower mass is coupled with a nonlinear pneumatic mechanism of hysteretic type are investigated using a lumped-parameter model. The dynamic response of the cam follower is obtained from the solution of the formulated system equation by the Krylov-Bogliubov method of variation of parameters. The stability of the system is investigated.

77-2130

A Survey of the State of the Art of Cam System Dynamics

F.Y. Chen

Dept. of Mech. Engrg., Ohio Univ., Athens, OH 45701, Mech. and Mach. Theory, 12 (3), pp 201-224 (1977) 17 figs, 128 refs

Key Words: Cams, Dynamic properties

The primary goal of this report is to present a comprehensive survey of the state of knowledge on the kinematic and dynamic aspects of the cam driven mechanisms and systems. The kinematics deals with the geometry of the cam curve, its continuity, curvature and boundary conditions as well as the mathematical derivatives of the curve which govern the velocity and acceleration characteristics of the motion. The dynamic problem areas concern physical modeling, formulations of the equations of motion, solution techniques, presentation of system's responses and the influence of design parameters.

PIPES AND TUBES

77-2131

An Experimental Investigation of Flow in an Oscillating Pipe

M. Clamen and P. Minton

Dept. of Civil Engrg., Imperial College, London, UK, J. Fluid Mech., 81 (3), pp 421-431 (July 13, 1977) 8 figs, 13 refs

Key Words: Pipes, Fluid-induced excitation

The hydrogen-bubble technique has been used to measure the velocities of pulsating water flow in a rigid circular pipe. Mean flows with Reynolds numbers between 1275 and 2900 were superimposed on an oscillating flow produced by moving the pipe axially with simple harmonic motion. While the velocities in the oscillating boundary layers on the pipe wall were found to be close to those predicted by laminar flow theory, at the higher Reynolds numbers the velocities near the center of the pipe were lower than those predicted and more uniformly distributed. The intermittency of the periodic bursts of turbulent motion at the higher Reynolds numbers was measured. At each mean-flow Reynolds number the turbulent intermittency of the flow was found to be a function of a single parameter: the harmonic-flow Reynolds number.

77-2132

Vibration of Tubes Conveying Fluids

V.A. Svetlitsky

Moscow Higher Tech. School, Moscow, USSR, J. Acoust. Soc. Amer., 62 (3), pp 595-600 (Sept 1977) 4 figs, 18 refs

Key Words: Pipes (tubes), Fluid-induced excitation

General, nonlinear equations are derived for the vibration of rectilinear tubes conveying incompressible fluid. From these equations are obtained the equations for small vibrations. If values of tube frequencies and critical flow parameters are to be predicted accurately, the initial state of stress must be taken into account. A numerical example is considered.

77-2133

Bifurcations to Divergence and Flutter in Flow-Induced Oscillations: A Finite Dimensional Analysis

P.J. Holmes

Inst. of Sound and Vibration Res., Univ. of Southampton, Southampton SO9 5NH, UK, J. Sound Vib., 53 (4), pp 471-503 (1977) 16 figs, 35 refs

Key Words: Pipes (tubes), Flutter, Flow-induced excitation

The behavior of a pipe conveying fluid and a fluid loaded panel are studied from the veiwpoint of differentiable dynamics. Non-linear terms are included. A general approach for solution is illustrated by analysis of two mode models of a pipe and of a panel and some important omissions in previous treatments of linear and undamped systems are discussed.

77-2134

A Preliminary Study of Flow and Acoustic Phenomena in Tube Banks

J.A. Fitzpatrick and I.S. Donaldson

Univ. of Glasgow, Glasgow, UK, ASME Paper No. 77-FE-7

Key Words: Tubes, Acoustic response, Wind tunnel tests

Experiments have been performed in a low-turbulence wind tunnel to investigate the effect of tube pitch to diameter ratios, depth of bank and Reynolds number on the parameters associated with resonant acoustic vibration in in-line tube banks.

77-1235

Experimental Data on the Natural Frequency of a Tubular Array

B.T. Lubin, K.H. Haslinger, A. Puri, and J. Goldberg
Combustion Engineering, Inc., Windsor, CT, ASME Paper No. 77-FE-10

Key Words: Tubes, Natural frequency

Data from experiments on an array of tubes in water showed that the tubes vibrated over a range of frequencies centered about an isolated single tube frequency. The concept of a motion dependent hydrodynamic mass has been successfully used to explain the observed results.

77-2136

Exchanger Design Cuts Tube Vibration Failures

W.M. Small and R.K. Young
Phillips Petroleum Co., Bartlesville, OK, Oil and Gas J., 75 (37), pp 77-80 (Sept 5, 1977) 5 figs, 1 table

Key Words: Tubes, Heat exchangers, Vibration reduction

Rod-baffle heat exchanger design is described which solves the problem of tube failures due to vibration and provides a low pressure drop across the bundle.

77-2137

Flow-Induced Vibrations of a Hydraulic Valve and Their Elimination

D.S. Weaver, F.A. Adubi, and N. Kouwen
McMaster Univ., Hamilton, Ontario, Canada, ASME Paper No. 77-FE-24

Key Words: Hydraulic valves, Fluid-induced excitation

The flow-induced vibrations of a check valve with a spring damper to prevent slamming have been studied experimentally. Both prototype and two-dimensional model experiments were conducted to develop an understanding of the mechanism of self-excitation. The phenomenon is shown to be caused by the high rate of change of discharge at small angles of valve opening and the hysteretic hydrodynamic loading resulting from fluid inertia.

PLATES AND SHELLS

(Also see Nos. 2062, 2071)

77-2138

An Analogy Between Free Vibration of a Plate and of a Particle of Mass

Z. Celep
Faculty of Engrg. and Architecture, Technical Univ., Istanbul, Turkey, J. Sound Vib., 53 (3), pp 323-331 (Aug 8, 1977) 5 figs, 10 refs

Key Words: Plates, Free vibration, Flexural vibration

In this paper, the free flexural vibration of an elastic circular thin plate with an initial imperfection is investigated. Approximate solution of the problem for the fundamental frequency of vibration, of large amplitude and with the plate imperfection, leads to a nonlinear ordinary differential equation with time as the independent variable.

77-2139

Measurement of Mechanical Vibration Damping in Orthotropic, Composite and Isotropic Plates Based on a Continuous System Analysis

N. Basavanhally and R.D. Marangoni
Dept. of Mech. Engrg., Univ. pf Pittsburgh, Pittsburgh, PA 15261, Intl. J. Solids Struc., 13 (8), pp 669-707 (1977) 8 figs, 9 refs

Key Words: Plates, Vibration damping, Measurement techniques

The problem of free and forced transverse vibration of an orthotropic, composite, and isotropic thin square plates with uniformly distributed damping and simply supported boundary conditions has been solved, using a modal expansion technique. A load of the type $P_0 \cos \Omega t$ applied at the center of plate has been considered and the phase angle between the forcing function and the vibration response at the center, as a function of the forcing frequency and the damping parameter determined. This theoretical relationship together with the experimentally measured phase angle between the applied mechanical forcing and the resulting vibration response at various forcing frequencies was used to determine an equivalent viscous damping parameter. This technique has been found to be particularly useful for the measurement and comparison of the relative damping in composite or orthotropic materials. Also, a theoretical relation for the energy loss due to viscous damping in vibrating plates has been developed and the theoretical energy loss at various frequencies has been compared with the experimentally measured energy loss at the same frequencies. Typical damping results are presented for aluminum, steel and aluminum/graphite-fiber composite materials.

77-2140

Variable Order Finite Elements for Plate Vibration

J.R. Hutchinson and J.J. Benitou
Dept. of Civil Engrg., Univ. of Calif., Davis, CA,
ASCE J. Engr. Mech. Div., 103 (EM5), pp 779-787 (Oct 1977) 3 tables, 12 refs

Key Words: Plates, Finite element technique, Natural frequencies, Mode shapes

Rectangular finite elements with a variable number of degrees-of-freedom per element are developed for thin elastic plates. The displacement field for the element is described by a fixed series of polynomial terms plus a variable number of trigonometric terms.

77-2141

Loss Factor for a Rectangular Plate of Parabolic Thickness Variation

S.P. Nigam, G.K. Grover, and S. Lal
Mech. Engrg., Government Engrg. College, Jabalpur, India,
J. Engr. Indus., Trans. ASME, 99 (3), pp 799-801 (Aug 1977) 2 tables, 4 refs

Key Words: Rectangular plates, Variable cross section, Fundamental mode, Internal damping

The importance of the internal damping and of the evaluation of the fundamental mode loss factor of structural members subjected to multiaxial stress system is well known. It appears that little work has been done on vibrations of rectangular plates of variable thickness, though such cases are of interest in the aeronautical field since they approximate to wing sections. In the present work, the fundamental mode loss factors for a simply supported rectangular plate with parabolic thickness variation in X direction have been evaluated for different combinations of the aspect ratios and the taper parameters. An approximate relationship has been obtained which correlates the loss factor for the plate of variable thickness with that of a plate of uniform thickness.

77-2142

Stability of Elastic Plates via Liapunov's Second Method

H.H.E. Leipholz
Solid Mech. Div., Faculty of Engrg., Univ of Waterloo, Waterloo, Ontario N2L 3G1, Canada,
Ing. Arch., 45 (5/6), pp 337-345 (1976) 4 figs, 3 refs
Sponsored by the National Res. Council of Canada

Key Words: Rectangular plates, Follower forces, Liapunov's method

Stability of a rectangular elastic plate is investigated by means of a Liapunov's Second Method. It is assumed that the plate is subjected to tangential follower forces which are parallel to one edge of the plate, that the plate has internal viscous damping, and that it is simply supported and/or clamped along its contour. The main result is that only for sufficiently large damping, stability is ensured for reasonably large follower forces.

77-2143

Large Amplitude Radial Oscillations of Layered Thick-Walled Cylindrical Shells

A. Ertepinar
Dept. of Engrg. Sciences, Middle East Technical Univ., Ankara, Turkey,
Intl. J. Solids Struc., 13 (8), pp 717-723 (1977) 4 figs, 1 table, 9 refs

Key Words: Cylindrical shells, Oscillation

Finite breathing motions of multi-layered, long, circular cylindrical shells of arbitrary wall thickness are investigated on the basis of the theory of large elastic deformations. The materials of the layers are assumed to be isotropic, elastic, homogeneous and incompressible. The governing non-linear ordinary differential equation is solved partially to give the frequencies of oscillations in an integral form. An approximate solution technique based on Galerkin's orthogonalization process is also formulated to give complete solutions. A tube consisting of two layers of new-Hookean materials is solved both by exact and approximate methods. An excellent agreement is observed between the two sets of results.

77-2144

Axially Symmetric Vibrations of Finite Cylindrical Shells of Various Wall Thicknesses - II

J. Chandra and R. Kumar
Systems Engrg., Div., Defence Science Lab., Delhi-110054, India,
Acustica, 38 (1), pp 24-29 (July 1977) 9 figs, 2 refs

Key Words: Cylindrical shells, Resonant frequency, Axisymmetric vibration

Using the exact three-dimensional equations of linear elasticity, the vibrational characteristics of circular cylindrical shells of various wall thicknesses and finite length have been studied. The motion of the shell is assumed to be axially symmetric but anti-symmetric about its central plane. The stress-free conditions on the lateral surface of the shell have been satisfied exactly and the real, imaginary and complex branches of the dispersion spectra have been superposed to satisfy the stress-free conditions at the flat ends of the shell to a good degree of accuracy. The aspect ratio curves, the residual stresses at the flat ends and the displacements have been given for various wall thicknesses.

77-2145

Analysis of a Cylindrical Shell Vibrating in a Cylindrical Fluid Region

H. Chung, P. Turula, T.M. Mulcahy, and J.A. Jendrzejczyk

Components Tech. Div. Argonne National Lab., IL, Rept. No. ANL-76-48, 24 pp (Aug 1976)

N77-26544

Key Words: Cylindrical shells, Nuclear reactor components, Natural frequencies, Mode shapes, Computer programs

Analytical and experimental methods are presented for evaluating the vibration characteristics of cylindrical shells such as the thermal liner of the Fast Flux Test Facility (FFTF) reactor vessel. The NASTRAN computer program is used to calculate the natural frequencies, mode shapes, and response to a harmonic loading of a thin, circular cylindrical shell situated inside a fluid-filled rigid circular cylinder. Solutions in a vacuum are verified with an exact solution method and the SAP IV computer code. Comparisons between analysis and experiment are made, and the accuracy and utility of the fluid-solid interaction package of NASTRAN is assessed.

77-2146

Vibrations of Prolate Spheroidal Shells of Constant Thickness

C.B. Burroughs

Ph.D. Thesis, The Catholic Univ. of America, 35 pp (1977)

UM 77-17,514

Key Words: Spherical shells, Fluid-induced excitation, Transverse shear deformation effects, Rotatory inertia effects

The general displacement-equilibrium equations, which include the effects of transverse shear and rotary inertia, have been derived for a fluid-loaded prolate spheroidal shell of constant thickness subject to an harmonically time-varying, arbitrary spatially distributed force normal to the shell surface. The solution is formulated for the axisymmetric motion of a shell that is immersed in an inviscid fluid of infinite extent. The approximate solutions for the two nontorsional displacements of the shell middle surface and the nontorsional rotation of the shell cross-section are obtained by using an extension of Galerkin's variational method developed by Chi and Magrab.

77-2147

Vibration of Complex Structures by Matching Spatially Dependent Boundary Conditions of Classical Solutions. Specifically Vibration Characteristics of Hollow Symmetrical Blades Based on Thin Shell Theory

A.M. Al-Jumaily

Ph.D. Thesis, The Ohio State Univ., 230 pp (1977)

UM 77-17,072

Key Words: Blades, Shells, Plates, Beams, Resonant frequencies, Mode shapes

The mathematical formulation and solution methods for dynamics problems of continuous structures composed of beam, plate, and shell elements are investigated by developing and using the Matching of Continuous Boundary Conditions Technique. This technique results in a closed form functional solution for the resonant frequencies and corresponding mode shapes of the composite structure. A hollow symmetrical turbomachinery blade is used to illustrate the general method. The blade is composed of two-co-linear open profile circular cylindrical shell elements connected at their straight edges. Experimental investigations are performed to support the results of the theories. In the course of formulating the blade problem, two new simplified shell solution techniques are introduced. One is based on Yu's assumptions for shells with small radius to length ratios; the second theory is derived from basic principles based on different assumptions gathered from the literature. The results of using the simplified shell solution technique, the Matching of Continuous Boundary Conditions method, and the experimental investigations are compared. Other methods of solution for dynamic problems of continuous structures, such as the Point Matching Technique, are investigated.

77-2148

The Effect of Creep Deformation on the Vibration and Stability Characteristics of Axisymmetric Thin Shells

A.P. Gelman

Ph.D. Thesis, Univ. of Southern California (1977)

Key Words: Shells, Natural frequencies, Computer programs, Stiffness methods

An analysis and a computer program have been developed for calculating the changes in the natural frequencies of axisymmetric thin shells when they are subjected to axisymmetric loads and are permitted unrestricted creep. The method of solution is an extension of the direct stiffness method. The shell is replaced by a system of discrete finite elements consisting of conical frustra; these elements are interconnected along circumferential nodal circles. The dynamical equations of equilibrium are obtained from the principles of minimum potential energy. The Sanders nonlinear strain displacement relations are utilized to obtain a linear stiffness matrix, a stress dependent geometric stiffness matrix, a nonlinear large displacement matrix, and a consistent mass matrix.

STRUCTURAL

77-2149

Earthquake Response of Coupled Shear Wall Buildings

T. Srichatrapimuk

Ph.D. Thesis, Univ. of California, Berkeley, 122 pp (1976)

UM 77-15,866

Key Words: Buildings, Walls, Earthquake response

An efficient analytical technique for determining linear and nonlinear response of coupled shear wall structures is developed. Walls are assumed to be nonyielding with all inelastic action confined to coupling beams. Structural displacements are then represented as a linear combination of the first few natural mode shapes in both lateral and longitudinal (vertical) vibration of individual walls which are treated as independent cantilevers. The effectiveness and flexibility of this general approach in reducing the number of degrees of freedom are demonstrated. The analytical technique is implemented in earthquake response analyses of two coupled shear wall systems; analytical results are then correlated with observations of earthquake damage in these structures. The earthquake response of coupled shear walls is then interpreted, and design considerations for efficient earthquake resistant shear wall systems are suggested.

77-2150

Air Blast Effects on Concrete Walls

C.A. Kot and P. Turula

Argonne National Lab., IL, Rept. No. ANL-CT-76-50, 67 pp (July 1976)

N77-26540

Key Words: Walls, Concrete construction, Blast effects

Estimates are obtained both for the spalling of the back-face of the concrete wall and for the overall wall response produced by the total impulsive load of the air blast. Assuming elastic wave propagation in the concrete wall, it is found that as spall thickness increases, the spall velocity decreases. This holds for normal as well as oblique wave incidence on the back-face of the wall. Therefore, for debris which has significant mass, the ejection velocity produced by spalling action alone is quite moderate. Plastic yield-line analysis of the wall segment subjected to the impulsive loading of the air blast indicates that for sufficiently large explosions substantial displacements and peak velocities can occur in typical shield walls.

77-2151

Seismic Response of a Periodic Array of Structures

H. Murakami and J.E. Luco

Dept. of Applied Mech. & Engrg. Sci., Univ. of Calif. at San Diego, La Jolla, CA, ASCE J. Engr. Mech. Div., 103 (EM5), pp 965-977 (Oct 1977) 6 figs, 12 refs

Key Words: Walls, Buildings, Earthquake response

A simplified two-dimensional model of the dynamic interaction, through the soil, among adjacent structures in a densely built area is presented. The model consists of an infinite number of identical parallel infinitely long shear walls placed on equally spaced rigid semi-cylindrical foundations. The steady-state response of the shear walls to obliquely incident plane SH waves is evaluated and compared with the response of an isolated structure.

SYSTEMS

ABSORBER

(Also see No. 2088)

77-2152

Design of Viscous Torsional Vibration Absorbers (Auslegung von Viskositätsdreh-Schwingungsdämpfern)

R. Mehner

Tech. Univ. Dresden, German Democratic Republic, Maschinenbautechnik, 27 (7), pp 326-329 (July 1977) 8 figs, 5 refs

(In German)

Key Words: Optimization, Vibration absorbers

An exact method for the optimization of vibration absorbers is obtained from the relationship of single mass systems with the viscosity torsional vibration absorbers. The method is based on electronic data processing.

NOISE REDUCTION

(Also see Nos. 2061, 2163, 2194)

77-2153

Machinery Noise Reduction. Correct Design Improves Efficiency (Lärmabschirmungen an Maschinen. Richtiges Gestalten erhöht die Wirksamkeit.)

J. Thoma

Techn. Rdschau (Bern), 68 (38), p 33 (1976) 1 fig, 3 refs

Key Words: Machinery noise, Noise reduction

The topics discussed are active and passive measures, simplified physics of noise, reflection, absorption and transmission of noise, noise amplification by means of reflection of sound in protective housing, harmful effects of small holes , absorption and stiffening for increasing the effectiveness of housing.

77-2154

Reducing Machinery Noise

R.L, Hershey

Booz, Allen & Hamilton, Inc., Indus. Res., 19 (9), pp 118-121 (Sept 1977) 6 refs

Key Words: Machinery noise, Noise reduction, Regulations

Considerable research has been devoted to reducing the noise from industrial machinery, such as circular saws, punch presses, textile spinning frames, and typewriters. This article describes some of the research areas and the regulations that have provided impetus toward quieting these machines.

77-2155

Systems for Noise and Vibration Control

W.E. Purcell

S/V, Sound Vib., 11 (8), pp 4-30 (Aug 1977)

Key Words: Noise reduction, Acoustic absorption, Noise barriers, Vibration control

Systems for noise and vibration control are finished products or components generally designed for specific purposes. For his discussion the author classifies such systems into: silencers, sound absorptive systems, sound barrier systems, and vibration/shock control systems.

77-2156

Acoustical Scale Model Study of the Attenuation of Sound by Wide Barriers

E.S. Ivey and G.A. Russell

Dept. of Physics, Smith College, Northampton, MA 01060, J. Acoust. Soc. Amer., 62 (3), pp 601-606 (Sept 1977) 8 figs, 15 refs

Key Words: Noise barriers, Acoustic attenuation, Model testing

Acoustical scale model experiments carried out with building-size barriers are described. The results of experiments conducted with the barrier in a free field and on a reflecting surface are presented. The free field measurements are compared to several theoretical models and discrepancies between the theoretical and experimental results are discussed. Also presented is a simple expression which relates the excess attenuation obtained with the barrier situated on the ground to that of the same barrier in the free field. This expression predicts excess attenuations which agree quite closely with those actually measured in the scale model experiments.

77-2157

OSHA and the Noise of Pneumatic Systems

R.C. Potter

Bolt Beranek and Newman, Inc., Cambridge, MA, ASME Paper No. 77-DE-49

Key Words: Pneumatic equipment, Noise reduction

Pneumatic systems produce high-level sounds in that part of the frequency spectrum that has the most influence on human hearing. OSHA requires that the hearing of individual workers be protected, and it is often the pneumatics of a machine that will control the sound levels received. Descriptions are given of the noise produced by the compressors that supply the air, the pipes and valves that transmit and control the air, and the devices, mechanisms, and tools that use the air. Methods are discussed for reducing the noise, and it is concluded that both management and employees will benefit from consideration of the problem of pneumatic system noise in present plants and in the design of future installations.

AIRCRAFT

(Also see No. 2197)

77-2158

Supersonic Jet Exhaust Noise Investigation. Volume IV. Acoustic Far-Field/Near-Field Data Report

P.R. Knott and J.F. Brausch
Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R74-AEG452-Vol-4, AFAPL-TR-76-68-Vol-4, 504 pp (July 1976)
AD-A040 894/8GA

Key Words: Jet noise, Aircraft noise

This report is an acoustic data report presenting a series of parametric acoustic far-field and near-field results for subsonic and supersonic heated flow conditions for a simple conical nozzle (thin lip and thick lip) and a convergent-divergent nozzle at design and off-design conditions.

77-2159

Recommended Procedures for Measuring Aircraft Noise and Associated Parameters

A.H. Marsh
DyTec Engrg., Inc., Huntington Beach, CA., Rept. No. NASA-CR-145187, 164 pp (Apr 1977) refs
N77-25912
N77-25912

Key Words: Aircraft noise, Noise measurement

Procedures are recommended for obtaining experimental values of aircraft flyover noise levels (and associated parameters). Specific recommendations are made for test criteria, instrumentation performance requirements, data-acquisition procedures, and test operations. The recommendations are based on state-of-the-art measurement capabilities available in 1976 and are consistent with the measurement objectives of the NASA Aircraft Noise Prediction Program. The recommendations are applicable to measurements of the noise produced by an airplane flying subsonically over (or past) microphones located near the surface of the ground. Aircraft types covered by the recommendations are fixed-wing airplanes powered by turbojet or turbofan engines and using conventional aerodynamic means for takeoff and landing. Various assumptions with respect to subsequent data processing and analysis were made (and are described) and the recommended measurement procedures are compatible with the assumptions. Some areas where additional research is needed relative to aircraft flyover noise measurement techniques are also discussed.

77-2160

Problems in Predicting Aircraft Noise Exposure

A.H. Odell
Port Authority of New York and New Jersey, One World Trade Ctr. 65S, New York, NY 10048, Noise Control Engr., 9 (1), pp 32-37 (July/Aug 1977) 9 figs, 21 refs

Key Words: Aircraft noise, Noise prediction, Human response

For more than twenty years, the aviation industry has tried to develop a single universal rating method which would accurately describe the noise produced by aircraft operations in terms of the subjective reaction of the exposed population. Some of the basic assumptions involved in this procedure are examined by the author. Also offered are suggestions for improvement in the methodology and potential areas of study.

77-2161

On the Growth Rate of Bending Induced Edge Cracks in Acoustically Excited Panels

K.P. Byrne
Dept. of Mech. and Industrial Engrg., Univ. of New South Wales, Kensington, NSW 2033, Australia, J. Sound Vib., 53 (4), pp 505-528 (1977) 16 figs, 1 table, 9 refs

Key Words: Aircraft, Acoustic excitation, Acoustic fatigue

The emphasis of the work described in this paper is on examining the growth rate of edge cracks in acoustically excited panels. A single panel with an edge crack is considered and this structural element is modelled as a flat plate clamped on three edges and part of the fourth. The crack is represented by the unclamped part of the fourth edge. Fracture mechanics principles are used to predict the crack growth rates associated with the first two modes of vibration of the edge cracked panel. The crack tip stress intensity factors associated with these panel modes are estimated by a technique based on finding the nominal bending stresses at the crack tips. The nominal bending stresses are in turn found from mode shapes determined by the Rayleigh Principle. The validity of the various assumptions is assessed by comparing the predicted crack growth rates with measured growth rates in panels representative of those used in aircraft construction.

77-2162

Non-Linear Effects in Aircraft Ground and Flight Vibration Tests

G. Haidl
Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, Fed. Rep. Germany, Rept. No. MBB-UFE-1273-0, 16 pp (Sept 16, 1976) refs
N77-25153

Key Words: Aircraft, Resonance tests, Vibration tests, Flutter

Examples of nonlinear vibration behavior in ground resonance tests of an aircraft are shown. Model tests for a simplified system with nonlinear properties were performed to study the effects of friction and backlash with respect to ground resonance test and flight flutter test. With symmetric and asymmetric nonlinear stiffness characteristics effects of amplitude dependent frequencies, mode coupling, mode asymmetries, and the consequences in parameter identification in vibration tests are pointed out and discussed. In case of flutter critical modes the problems of apparent damping caused by nonlinear system properties are shown, and recommendations are given to reach a representative flutter clearance with respect to this nonlinear system behavior.

77-2163

Supersonic Transport Noise Reduction Technology Program - Phase II. Volume I

S.B. Kazin, E.J. Stringas, J.T. Blozy, V.L. Doyle, and R.B. Mishler

Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R75AEG362-Vol-1, FAA-SS-73-29-1, 478 pp (Sept 1975)

AD-B010 468/7GA

Key Words: Supersonic aircraft, Noise reduction

The Supersonic Transport Noise Reduction Technology Program, sponsored by the Federal Aviation Administration, was conducted as a follow-on effort after cancellation of the SST Program to finalize selected noise technology areas and summarize results of the SST Program. The overall program objective was to provide additional acoustic technology necessary, to design high speed aircraft systems, recognizing future acceptable noise levels. General Electric's effort was divided into the acoustic technology areas of jet noise reduction, turbomachinery noise reduction, and aircraft system integration. Jet noise reduction technology work was achieved through analytical studies, model tests, and J79 engine tests. Selected suppresssion systems identified during the SST Program were further refined (multispoke/chute suppressors or annular plug nozzles). Novel advanced concepts of suppression were identified, and extensive aerodynamic (static and wind-on) performance tests and hot-jet acoustic tests were performed.

77-2164

Airframe, Wing, and Tail Aerodynamic Characteristics of a 1/6-Scale Model of the Rotor Systems Research Aircraft with the Rotors Removed

R.E. Mineck and C.E. Freeman

Army Air Mobility Res. and Dev. Lab., Hampton, VA, Rept. No. NASA-TN-D-8456, 141 pp (May 1977)

N77-26082

Key Words: Aircraft, Helicopters, Airframes, Aircraft wings, Wind tunnel tests

A wind-tunnel investigation was conducted to determine the aerodynamic characteristics of the rotor systems research aircraft (RSRA) as the helicopter and the compound helicopter with the rotors removed. Data were obtained over ranges of angles of attack and angle of sideslip. Results are presented for the total loads on the airframe as well as the loads on the wing and the tail.

77-2165

Treatment of the Nonlinear Vibration of a Variable Sweep Aircraft Wing with its Drive Using a Simplified Wing Model (Behandlung des nichtlinearen Schwingungsverhaltens eines schwenkbaren Flugzeugfluegels mit seinem Verstellantrieb mittels eines vereinfachten Schwingungsmodells)

B. Schoen

Unternehmensbereich Flugzeuge-Entwicklung, Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, W. Germany, Rept. No. MBB-UFE-1191(0)), 155 pp (Aug 1, 1975)

(In Georgian)

N77-26156

Key Words: Aircraft wings, Vibration response, Mathematical models

A wing vibration model was constructed to investigate the vibration behavior of a variable sweep wing with its pivot drive. The model provides for simulation of the clearance, the static friction, and damping proportional to velocity. The physical vibration behavior was investigated by variation of these parameters. The complex phenomenon was also studied theoretically by approximation solutions, and the dependence on parameter variations indicated. Experimental and theoretical results are combined to provide a complete picture of the vibration phenomenon.

77-2166

Flutter Analysis of an All-Movable Horizontal Tail with Geared Elevator on a Supersonic Transport

J.L. Stelma

Boeing Commercial Airplane Co., Seattle, WA, Rept. No. D6-60293, FAA-SS-73-16, 60 pp (June 1974)

AD-B000 285/7GA

Key Words: Flutter, Supersonic aircraft

This document presents symmetric flutter analyses conducted on the all-movable horizontal tail of the Boeing-designed SST. Interaction effects on flutter speed that are produced by the wing, fuselage, control systems and elevator gear ratio are included. Failure conditions of the horizontal-tail actuators are covered.

77-2167

A Low Speed Model Analysis and Demonstration of Active Control Systems for Rigid-Body and Flexible Mode Stability

R.A. Gregory, A.D. Ryneveld, and R.S. Imes

Boeing Commercial Airplane Co., Seattle, WA, Rept. No. D6-60295, FAA-SS-73-18, 203 pp (June 1974) AD-B000 286/5GA

Key Words: Supersonic aircraft, Flutter, Wind tunnel tests, Stability analysis

An existing low-speed SST flutter model was modified to include two hydraulic aileron control systems and a horizontal stabilizer system. Wing mode flutter suppression systems were analyzed and wing tunnel tested, using wing strain gages and the aileron systems in the active control feedback loops. Rigid-body stability systems were theoretically analyzed and experimentally synthesized using body-mounted sensors. Variable rigid-body stability was achieved through a remote-transfer water ballast system. The results of parallel analysis and wind tunnel tests, the methods of approacn, the problems encountered, and a list of recommendations for the advancement of the active controls technology are reported in this document.

BRIDGES

(Also see No. 2110)

77-2168

Motion of Suspended Bridge Spans under Gusty Wind

R.H. Scanlan and R.H. Gade

ASCE J. Struc. Div., 103 (ST9), pp 1867-1883 (Sept 1977) 5 figs, 17 refs, 5 tables

Key Words: Suspension bridges, Wind-induced excitation

The buffeting response of suspended-span bridges can be calculated if certain wind-tunnel section model data, plus wind spectral information, are provided. The needed wind tunnel data are the self-excited aerodynamic (flutter) coefficients. The meteorological data required are vertical and horizontal gust spectra of the natural wind at the bridge site. The natural mechanical modes of vibration in bending (vertical and lateral) and torsion are assumed known, and the response of each of these with postulated negligible aerodynamic coupling between modes, is calculated. Some examples are then given of the calculated vertical and torsional buffeting responses of a flexible long-span bridge (Golden Gate type) and a stiff, medium-span type (Sitka Harbor). The wind velocity range covered is 60 mph to 90 mph (27 m/s to 40 m/s).

77-2169

Effects of Uniform and Non-Uniform Seismic Disturbances on a Long Multi-Span Highway Bridge

R.E. Hamati

Ph.D. Thesis, Univ. of Calif., Berkeley, 397 pp (1976) UM 77-15,710

Key Words: Bridges, Seismic design

Criteria were developed for the seismic design of a long multi-span highway bridge. The criteria are for requirements of seismic strength to resist inertia effects, and provisions for sufficient ductility to absorb the displacements and deformations caused by uniform and non-uniform distributions of ground motions. Criteria were also developed for determining the ductilities and capacities of elements of the bridge to absorb the maximum relative displacements that may be caused by residual deformations of the soils. In developing the criteria, various parameters were considered. Among the parameters are those related to bridge types, articulations, soil conditions, and spatial distributions of ground motions. The effects of soil-structure interaction are included.

BUILDING

(Also see Nos. 2083, 2088, 2149, 2151)

77-2170

Inelastic Earthquake Response of Three-Dimensional Buildings

R. Guendelman-Israel

Ph.D. Thesis, Univ. of Calif., Berkeley, 130 pp (1976) UM 77-15,705

Key Words: Buildings, Earthquake response, Computer programs

A computational procedure and computer program for the inelastic dynamic response analysis of three-dimensional buildings of essentially arbitrary configuration is described. The building is idealized as a series of independent plane substructures interconnected by horizontal rigid diaphragms. Each substructure can be of arbitrary geometry and include structural elements of a variety of types.

77-2171
Inelastic Response to Site-Modified Ground Motions
R.V. Whitman and J.N. Protonotarios
Mass. Inst. of Tech., Cambridge, MA, ASCE J. Geotech. Engr. Div., 103 (GT10), pp 1037-1053 (Oct 1977) 16 figs, 1 table, 12 refs

Key Words: Buildings, Earthquake response

A building with a period equal to that of a site may be more susceptible to yielding during a moderate earthquake, but the larger yielding during a major earthquake is much the same as for a building having a different period. This conclusion results from analyzing one-degree-of-freedom, elastoplastic structures using ground motions (both real and calculated) whose elastic response spectra have peaks attributable to site conditions. Inelastic response spectra for site-modified motions do not show pronounced peaks at the period of the site; rather, they are as "smooth" as inelastic spectra computed from motions unaffected by site conditions. Inelastic spectra for design may be based upon the same ratios of spectral acceleration to peak acceleration and spectral velocity to peak velocity as for normal motions. Thus, the amount by which a site modifies peak acceleration and peak velocity is important, and the period of a site is not significant by itself.

77-2172
Review of Literature on Earthquake Damage to Single-Family Masonry Dwellings
R.D. Benson
Applied Tech. Council, Palo Alto, CA, 31 pp (Apr 29, 1977)
PB-267 947/0GA

Key Words: Earthquake damage, Buildings, Masonry, Reviews

The report contains a review and evaluation of information concerning the behavior of single-family masonry dwellings in Zone 2 earthquake areas of the United States (1973 Uniform Building Code classification). In general, reinforced masonry has exhibited satisfactory performance, sustaining little or no damage in moderate earthquakes. Reported damage is often associated with poor workmanship/inspection. Unreinforced masonry (old and new) and masonry chimneys have exhibited poor performance. Available data has been found to be limited and general in nature.

FOUNDATIONS AND EARTH
(See Nos. 2084, 2106)

HELICOPTERS
(Also see No. 2164)

77-2173
Aeroelastic Stability of Complete Rotors with Application to a Teetering Rotor in Forward Flight
J. Shamie and P. Friedmann
Mechanics and Structures Dept., School of Engrg. and Applied Science, Univ. of Calif., Los Angeles 90024, J. Sound Vib., 53 (4), pp 559-584 (1977) 12 figs, 23 refs

Key Words: Helicopter rotors, Dynamic stability

The derivation of a set of non-linear coupled flap-lag-torsion equations of motion for moderately large deflections of an elastic, two-bladed teetering helicopter rotor in forward flight is concisely outlined.

77-2174
Effect of Production Modifications to Rear of Westland Lynx Rotor Blade on Sectional Aerodynamic Characteristics
P.G. Wilby
Aerodynamics Dept., Royal Aircraft Establishment, Farnborough, UK, Rept. No. ARC-CP-1362; RAE-TR-73043; ARC-34835, 21 pp (1977) refs
N77-25101

Key Words: Helicopter rotors, Rotary wings, Aerodynamic response

The RAE (NPL) 9615 airfoil was accepted, on the basis of wind tunnel tests, as the basic blade section for the Westland WG 13 Lynx helicopter rotor; however, production methods necessitated a modification to the rear profile of the blades which was considered sufficient to produce changes in the aerodynamic characteristics of the airfoil. Thus, the modified profile was tested in the wind tunnel and the experimental data compared with those for the original profile.

77-2175
Application of System Identification to Analytic Rotor Modeling from Simulated and Wind Tunnel Dynamic Test Data. Part 2
K.H. Hohenemser and D. Banerjee
Dept. of Mech. Engrg., Washington Univ., St. Louis, MO, Rept. No. NASA-CR-152023, 194 pp (June 1977)
N77-26078

Key Words: Helicopters, Aircraft, Parameter identification, Rotors, Mathematical models

An introduction to aircraft state and parameter identification methods is presented. A simplified form of the maximum likelihood method is selected to extract analytical aeroelastic rotor models from simulated and dynamic wind tunnel test results for accelerated cyclic pitch stirring excitation. The dynamic inflow characteristics for forward flight conditions from the blade flapping responses without direct inflow measurements were examined.

HUMAN

(Also see No. 2087)

77-2176

Hand-Arm Vibration Part II: Vibrational Responses of the Human Hand

J.W. Mishoe and C.W. Suggs

Agricultural Research and Education Ctr., Dept. of Agricultural Engrg., Univ. of Florida, Belle Glade 33430, J. Sound Vib., 53 (4), pp 545-558 (1977)
14 figs, 6 refs

Key Words: Human hand, Vibration response, Mathematical models, Mechanical impedance

When vibration is applied to the hand in the vertical (dorsal-to-ventral) and transverse direction, the hand arm system can be modeled by a three-mass model with each of the masses connected by a parallel spring and damper. For vibration input directed into the long axis of the forearm the model requires an additional parallel spring and damper to connect the last mass to an infinite base.

ISOLATION

77-2177

Equation Error Identification of Vehicle Suspension Parameters

D.M. Brueck

Ph.D. Thesis, Purdue, Univ., 200 pp (1976)
UM 77-15,384

Key Words: Suspension systems (vehicles), Parameter identification

A simplified method for the identification of vehicle suspension parameters is developed. Increased use of computer simulations in the design, development, and testing of vehicles requires that the various vehicle parameters be easily obtainable. Methods to obtain the vehicle sprung mass and sprung mass moments of inertia are available; however, a simplified method to obtain the vehicle suspension spring rates, damping characteristics, and the unsprung mass inertia properties is needed. The technique that was developed in this thesis to obtain these suspension parameters requires a test of short duration, less than three seconds, and avoids vehicle disassembly. The parameters are identified from suspension force and displacement data, eliminating the need for complex calculations using detailed information concerning the characteristics and placement of each of the many components making up the suspension.

MECHANICAL

77-2178

Active Electromagnetic Vibration Control in Rotating Discs

R.W. Ellis

Ph.D. Thesis, Univ. of Calif., Berkeley, 81 pp (1976)
UM 77-15,673

Key Words: Disks, Rotating structures, Saws, Vibration control

This thesis introduces a promising new technique for improving saw performance using an electronic feedback control system. The system consists of a non-contacting position sensor placed alongside the lateral surface of the saw, some control circuitry, and a pair of electromagnets placed alongside the saw, one on each side. The position sensor measures deviations from a normal undeflected condition and the control has produced significantly increased lateral stiffness and vibration damping characteristics in laboratory experiments, and it shows every indication of proving applicable to production situations.

METAL WORKING AND FORMING

77-2179

A Stability Analysis of Single-Point Machining with Varying Spindle Speed

J.S. Sexton, R.D. Milne, and B.J. Stone

Dept. of Mech. Engrg., Univ. of Bristol, Queens Bldg., Univ. Walk, Bristol BS8 1TR, UK, Appl. Math. Modeling, 1 (6), pp 310-318 (Sept 1977)
8 figs, 1 table, 8 refs

Key Words: Machine tools, Stability analysis, Chatter

The rate at which metal can be removed by a machine tool is often limited by the onset of an instability commonly called 'chatter.' It has been suggested that greater widths of cut could be achieved without chatter on a given machine by modulating the spindle speed continuously. A stability analysis is presented which gives, for any mean spindle rotation speed and degree of modulation, the limiting width of cut for chatter-free cutting.

77-2180

Study on Optimum Design of Machine Structures with Respect to Dynamic Characteristics (Approach to Optimum Design of Machine Tool Structures with Respect to Regenerative Chatter)

M. Yoshimura

Faculty of Engrg., Kyoto Univ., Yoshida Sakyo-ku, Kyoto, Japan, Bull. JSME, 20 (145), pp 811-818 (July 1977) 10 figs, 3 tables, 5 refs

Key Words: *Machine tools, Chatter*

In order to attain dynamically optimum design of machine tools which would have minimum chance of machining chatter, an approach based on energy balances of a mathematical system at the resonance is developed and analyzed theoretically. This method aims that the maximum compliance of the tool-work relative displacement in the direction normal to cut across all frequency ranges. Using the computer simulations of machine tool structures, modal flexibilities are computed, by the magnitude of which the chance of regenerative chatter is judged.

77-2181

Identification and Active Adaptive Control of Chatter in *Single-Point* Machining Operations (Vol. I and II)

K. Srinivasan

Ph.D. Thesis, Purdue Univ., 883 pp (1976)

UM 77-15,476

Key Words: *Machine tools, Chatter*

Three areas of relevance to the active control of machine-tool chatter are considered in this thesis: identification of machining system dynamics; controller design for machining systems; identification and controller adaptation for traverse machining operations.

77-2182

A New Approach to the Analysis of Machine-Tool System Stability under Working Conditions

F.A. Burney, S.M. Pandit, and S.M. Wu

Mech. Engrg. Dept., Univ. of Wisconsin, Madison, J. Engr. Indus., Trans. ASME, 99 (3), pp 585-590 (Aug 1977) 6 figs, 2 tables, 20 refs

Key Words: *Machine tools, Stability, Cutting, Mathematical models*

A new stochastic approach is developed in this paper for analyzing the machine-tool system stability under working conditions. Mathematical models are fitted to the relative longitudinal cutter-workpiece displacement data recorded under different cutting conditions during the face-milling operation on a milling machine. The stability of the system is judged from the characteristic roots of these models. The variation in stability is examined versus both the cutting speed and the feed, and good results are obtained. It is shown that not only the dynamic but also the static stability can be ascertained. Furthermore, the stability of subsystems can also be determined. The significance of these results is discussed with special reference to on-line chatter control.

PUMPS, TURBINES, FANS, COMPRESSORS

(Also see No. 2157)

77-2183

Solve Vertical Pump Vibration Problems

R.J. Meyer

Industrial Pump Div., Allis-Chalmers Corp., Cincinnati, OH, Hydrocarbon Processing, 56 (8), pp 145-149 (Aug 1977) 6 figs

Key Words: *Pumps, Vibration monitoring*

Because of their long, slender structure, vertical pumps can have severe vibration problems. Possible causes of vibration and how to verify these causes by testing are discussed.

RAIL

77-2184

Reduction of Railway Noise with Composite Concrete Rails

J. Halpenny

Earth Physics Branch of the Dept. of Energy, Mines and Resources, Ottawa, Ontario, Canada, High-Speed Ground Transp. J., 11 (2), pp 173-175 (Summer 1977) 4 refs

Key Words: *Railroad tracks, Noise reduction*

Noise due to high speed trains can be greatly reduced by the use of a suitable track structure. A rail with increased stiffness and mass allows the use of much more flexible mountings than are possible with conventional rails. Vibration of the ground and track structure, the most difficult type of sound to handle, is isolated at source. The track will hold a more precise alignment longer, and demands on the foundation are less severe. The technique requires advances in concrete technology, but will make rail systems much quieter.

REACTORS

(Also see No. 2145)

77-2185

Seismic Soil-Structure Interaction Effects at Humboldt Bay Power Plant

J.E. Valera, H.B. Seed, C.F. Tsai, and J. Lysmer
Dames & Moore, San Francisco, CA, ASCE J. Geotech. Engr. Div., 103 (GT10), pp 1143-1161 (1977) 15 figs, 4 tables, 10 refs

Key Words: Nuclear power plants, Earthquake response, Seismic design, Interaction: soil-structure

The results of a study of the distribution of ground motions and structural response in the Humboldt Bay Nuclear Power Plant during the Ferndale earthquake of June 7, 1975 are presented. Based on a knowledge of the motions recorded at the ground surface in the free-field, computations are made to determine the characteristics of the motions likely to develop at the base of the buried reactor caisson at a depth of 85 ft below the ground surface and within the Refueling Building at the ground surface level.

ROAD

(Also see No. 2087)

77-2186

Crash Testing of Experimental Safety Vehicles. Volume II. Renault Basic Research Vehicle

N.J. DeLeys
Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5857-V-2-Vol-2, COT-HS-802 380, 185 pp (May 1977)
PB-267 966/0GA

Key Words: Collision research (automotive), Crashworthiness, Test data

Results from two crash tests of the Renault Basic Research Vehicle (BRV) are presented. The tests were a left front oblique impact with a rigid 30-degree angled barrier at a speed of 42.5 MPH, and a 75-degree right side impact of the same BRV by the front of a production. Renault R-12 automobile at a speed of 31.3 MPH. The objective of the tests was to evaluate the safety performance of the Renault BRV from the vehicle and dummy occupant responses measured in the crashes.

77-2187

Application of Military Vibration Standards to Public Transport Vehicles

G.F. Capponi
ATM Public Transport of Milan, Italy, J. Environ. Sci., 20 (5), pp 25-28 (Sept/Oct 1977) 7 figs

Key Words: Vibration tests, Buses (vehicles), Standards and codes

The objective was to establish a tentative vibration test specification for the ticket machines used on ATM buses. A vibration simulation criterion is described, developed following MIL-STD-810B and considering acceleration measurements made on ATM buses (Public Transport of Milan).

77-2188

An Investigation of Some Responses of an Out-of-Position Driver in an ACRS-Equipped Oldsmobile during Crash Induced Bag Deployment

D.J. Bliss
Office of Vehicle Systems Res., National Highway Traffic Safety Admin., Washington, D.C., Rept. No. DOT-HS-802 315, 69 PP (May 1977)
PB-267 951/2GA

Key Words: Collision research (automotive), Air bags (safety restraint systems), Test data

A study was conducted to investigate the undesirable side effects of inflating a driver air bag system against a forward positioned occupant. The study was at least suggested by an accident which occurred in February 1976 in Memphis, TN, in which the driver of an ACRS-equipped Oldsmobile died as the car struck a utility pole at a speed below the 30 mph design speed of the system. A series of curb rideover tests and a pole impact test were conducted to consider the general problem of occupants positioned forward against inflating air bags and specifically to note any similarities with the Memphis accident.

77-2189

Crash Testing of Experimental Safety Vehicles. Volume I. British Leyland Marina Safety Research Vehicle

N.J. DeLeys

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5857-V-2-Vol-1, DOT-HS-802 379, 189 pp (May 1977)

PB-267 965/2GA

Key Words: Collision research (automotive), Crashworthiness, Test data

Results from two crash tests of Phase I Marina Safety Research Vehicles (SRV) developed by British Leyland Motor Corp. are presented. The tests were a central head-on collision of a Marina SRV with an AMF experimental safety vehicle at a closing speed of 60 MPH, and a 90-degree side impact of another Marina SRV by a modified production Marina automobile at a speed of 30 MPH. The objective of the tests was to evaluate the safety performance of the Marina SRVs from the vehicle and dummy occupant responses measured in the crashes.

ROTORS

(Also see No. 2118)

77-2190

Finite Element Stability Analysis for Coupled Rotor and Support Systems (Part 3)

K.H. Hohenemser and S.K. Yin

Dept. of Mech. Engrg., Washington Univ., St. Louis, MO, Rept. No. NASA-CR-152024, 47 pp (June 1977)

N77-26079

Key Words: Rotors, Supports, Stability, Finite element technique

The effects of fuselage motions on stability and random response were analytically assessed. The feasibility of adequate perturbation models from non-linear trim conditions was studied by computer and hardware experiments. Rotor wake-blade interactions were assessed by using a 4-bladed rotor model with the capability of progressing and regressing blade pitch excitation (cyclic pitch stirring), by using a 4-bladed rotor model with hub tilt stirring, and by testing rotor models in sinusoidal up to side flow.

77-2191

Effect of Inertia Moment on Critical Speed Calculation of Rotating Shafts (Effetto del Momento Raddrizzante sul Calcolo Delle Velocita Critiche di Alberi Rotanti)

B. Atzori

Ist. de Costruzione di Macchine, Bari Univ, Italy, Rept. No. HC A02/MF A01, 12 pp (Oct 16, 1976) refs

(In Italian)

N77-25544

Key Words: Rotors, Shafts, Critical speed, Inertial forces

The effect of taking into account the lateral inertia in the computation of critical speeds of rotating shafts was analyzed. The power method, Von Borowicz's method, Dunkerley's method, and the matrix displacement and force methods were considered. Some procedures for extending the validity of the examined methods are described after analyzing the mathematical implications due to the presence of negative eigenvalues.

77-2192

The Effect of Nonlinear Internal Damping on the Stability of Simply Loaded Shafts (Zur Stabilität einfach besetzter Wellen mit nichtlinearer innerer Dämpfung)

P. Hagedorn, H. Kühl, and W. Teschner

Institut für Mechanik, Technische Hochschule Darmstadt, Hochschulstrasse 1, D-6100 Darmstadt, Fed. Rep. of Germany, Ing. Arch., 46 (3), pp 203-212 (1977) 3 refs

(In German)

Key Words: Rotors, Internal damping, Stability

The destabilizing effect of linear internal damping on rotating shafts with a single disc is well-known. Internal damping forces can however in general not be well described by linear functions, but may only be produced with some accuracy with nonlinear terms. In this paper, nonlinear internal damping as well as nonlinear restoring forces are considered, the stability of the vertical and of the horizontal shaft are discussed and non-trivial stationary solutions are also examined. The obtained results confirm to a certain extent the behavior of rotating shaft found by Tondl.

77-2193

A Method for Estimating the Condition that a Rotor Can Pass Through Resonance

K. Matsuura

Hitachi Res. Lab., Hitachi, Ltd., Hitaschisi, Japan, Bull. JSME, 20 (145), pp 801-810 (July 1977) 14 figs, 9 refs

Key Words: Rotors, Critical speed

A rotor accelerated across a resonance, which possesses linear properties with a single degree of freedom, excited by an unbalanced rotating mass is considered. It is said that by investigating the non-stationary transitions of motion of a rotor under the critical condition, it can be found whether or not a rotor can pass through resonance or not. It is possible to formulate the condition; and an expression for estimation.

SPACECRAFT

77-2194

Noise Reduction Evaluation of Grids in a Supersonic Air Stream with Application to Space Shuttle

J.M. Seiner, J.C. Manning, P. Nystrom, and S.P. Rao
Langley Res. Ctr., NASA, Langley Station, VA., Rept. No. NASA-TM-X-74034, 36 pp (May 1977) refs
N77-25913

Key Words: Spacecraft, Launching, Noise reduction

Near field acoustic measurements were obtained for a model supersonic air jet perturbed by a screen. Noise reduction potential in the vicinity of the space shuttle vehicle during ground launch when the rocket exhaust flow is perturbed by a grid was determined. Both 10 and 12 mesh screens were utilized for this experiment, and each exhibited a noise reduction only at very low frequencies in the near field forward arc.

77-2195

An Evaluation of Reaction Wheel Emitted Vibrations for Space Telescope

Sperry Flight Systems, Phoenix, AZ, Rept. No. NASA-CR-150303; Publ-71-0989-00-00, 108 pp (Mar 1977)
N77-26181

Key Words: Spacecraft components, Vibration measurement

Emitted forces and moments characteristics of the Space Telescope Reaction Wheel Assembly (ST RWA) were measured under room temperature and pressure, thermal extremes, and vibratory conditions. The RWA/Emitted Vibration Measurement Fixture was calibrated statically and dynamically, and background noise was measured with ST RWA not operating. A base line set of forces and moments of the ST RWA along and about three mutually perpendicular axes were recorded at room ambient.

77-2196

Identification of Natural Frequencies and Modal Damping Ratios of Aerospace Structures from Response Data

C.D. Michalopoulos
Dept. of Mech. Engrg., Houston Univ., TX, Rept. No. NASA-CR-151419; TR-NC-1, 36 pp (Nov 1976)
N77-26532

Key Word: Spacecraft, Natural frequencies, Modal damping

An analysis of one and multidegree of freedom systems with classical damping is presented. Definition and minimization of error functions for each system are discussed. Systems with classical and nonclassical normal modes are studied, and results for first order perturbation are given. An alternative method of matching power spectral densities is provided, and numerical results are reviewed.

TURBOMACHINERY

(Also see Nos. 2050, 2060, 2065)

77-2197

Supersonic Transport Noise Reduction Technology Program - Phase II, Volume 2

S.B. Kazin, E.J. Stringas, J.T. Blozy, V.L. Doyle, and R.B. Mishler
Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R75AEG362-Vol-2, FAA-SS-73-29-2, 470 pp (Sept 1975)
AD-B010 469/5GA

Key Words: Turbomachinery noise, Noise reduction, Supersonic aircraft

Both compressor and turbine noise were studied in the turbomachinery noise reduction areas. A 3-stage low pressure compressor with variable-flap inlet guide vanes was tested at General Electric's outdoor test site. A hybrid inlet, which employs airflow acceleration suppression in combination with wall acoustic treatment, was investigated as the suppression device for all three noise monitoring point operating conditions. The effect of auxiliary inlets on noise leakage and suppression was studied for takeoff mode. Also, variable inlet guide vane flaps were used to reduce area and generate high passage Mach numbers of another means of compressor noise suppression. Turbine noise was studied using a J85 engine with massive inlet suppressor and open nozzle to unmask the turbine. Second-stage turbine blade/ nozzle spacing and exhaust acoustic treatment were investigated as means of turbine noise suppression.

ANNUAL AUTHOR INDEX

- C -

- G -

- L -

- Mc -

- M -

- N -

- O -

- P -

- S -

- T -

AD-A048 062 NAVAL RESEARCH LAB WASHINGTON D C SHOCK AND VIBRATIO--ETC F/G 20/11
THE SHOCK AND VIBRATION DIGEST. VOLUME 9, NUMBER 12, DECEMBER 7--ETC(U)
DEC 77 R L ESHLEMAN, J N ESHLEMAN

UNCLASSIFIED NL

2 of 2

ADA048 062

END
DATE
FILMED
1 -78
DDC

- U -

- V -

ANNUAL SUBJECT INDEX

- A -

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9 Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9 Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

- G -

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Helmholtz Resonators
200 1779

High Frequency Resonance Technique
715

High Speed Transportation
1381

Hitches
use Drawbars

Hoists
367

Hole-Containing Media
1481 932 933 764 1645 758
1822 1124
1644

Holographic Techniques
1151 66 737 958 1799
1281 1456
1976

Holonomic Systems
854

Holzer Method
1039

Honeycomb Laminates
1124

Honeycomb Structures
847 39

Hopkinson Bar Technique
1642 196

Household Appliances
902

Housings
1556

Hovercraft
use Ground Effect Machines

Human Factors Engineering
391 994 1028

Human Hand
1355 1356 1357
2176

Human Head
use Head (Anatomy)

Human Organs
use Organs (Biological)

Human Response
570 2001 212 1743 1504 205 26 1517 1368 569
1190 252 1913 1744 995 206 1687 1518 1369
1230 1512 2014 1745 1016 2087 1688 1689
1400 2034 1746 2078
2160 2001 2076

Human Tolerance
1857

Hunting Motion
1210 1211 1212
2091

Hydraulic Dampers
1162

Hydraulic Equipment
1660 344 46 1737 1738 1739
1376
1736

Hydraulic Systems
1700

Hydraulic Valves
2137

Hydrodynamic Excitation
305 2047 179
1409
2099

Hydroelectric Power Plants
2080 801

Hydrofoil Craft
434

Hydrostatic Bearings
1805 1617

Hydrostatic Drives
1723

Hyperbolic Parabolic Shells
964 965 1139

Hysteretic Behavior
85

Hysteretic Damping
1751 653 276 1268 1229
1981 1926

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Manuals and Handbooks
1991

Marine Propellers
2024

Masonry
2172

Mass Half-Space Systems
1265

Mass-Beam Systems
1873 1445 1878

Mass-Spring Systems
530 81 522 333 1265 529
1873

Material Damping
1780 1424 1076 2107
1924

Materials Handling Equipment
1692 1693

Mathematical Modeling
use Mathematical Models

Mathematical Models
190 261 382 163 314 165 86 147 128 29
200 351 412 183 754 365 646 467 188 179
260 411 422 193 1314 875 656 607 218 379
480 551 452 273 1394 885 976 647 378 489
490 601 522 573 2054 1105 1096 857 468 629
590 961 532 583 1245 1106 1017 628 879
600 1031 622 1013 1355 1376 1177 698 969
800 1061 642 1353 1395 1476 1417 958 1049
880 1451 692 1423 1695 1696 2007 998 1069
1030 1012 1483 1845 1706 1058 1369
1050 1252 1953 1965 1756 1068 1399
1380 1352 2033 2025 1846 1108 1649
1520 1422 2053 2165 2176 1258 1819
1570 1832 2175 1508 1869
1580 2182 1628 1879
1610 1898 1889
1680 2039
1740
1750
1920
2080

Mathematical Programming
8

Matrix Methods
1290 2072 749

Maximum Response
1860

Maxwell Fluid
128

Measurement Instruments
use Measuring Instruments

Measurement Techniques
70 361 292 513 904 65 986 387 308 69
1190 371 512 603 1094 385 1056 577 1198 289
1440 1441 1092 733 1594 395 1597 2108 1199
1530 1722 1063 1704 515 2107 1599
1760 1093 1945 1689
1950 1443 1789
1743 1949
2139

Measuring Instrumentation
use Measuring Instruments

Measuring Instruments
1190 1441 292 603 504 385 286 287 288 1439
1722 903 904 905 1056 1597 1438
1433 1754 1945 1286
1943 1944 2105

Measuring Techniques
use Measurement Techniques

Mechanical Admittance
1878

Mechanical Elements
1130 531 1129
1939

Mechanical Filters
1

Mechanical Impedance
364 495 2176 857 728

Mechanical Reliability
use Reliability

Mechanical Systems
1860 255 1196 1197

Mechanical-Acoustic Systems
1733

Mechanisms
531 1132 1133 1045 2126 2127 2128
1131

Membranes
240 1154 245 1326 937 938
1560 1325 1466 1298
1638

Metal Working
370 1374 187
1200 1694

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Moving Strips

					1955				

Mufflers

200	451		143			146			969
	2061								

Multidegree of Freedom Systems

				1264	1885	686	1447	248	
						1566			

Multistory Buildings

1680		1102	1833	1834		686	797	358	169
							1177	798	799
							1677	1358	1179
									1359
									1789
									1849

Musical Instruments

	231						257		

- N -

Nacelles

	1671			1704					

Narrow-Band Excitation

				244					

NASTRAN (Computer Programs)

1560	1561	1552	1553	1554	1555	1556	1557	638	1559
		1562						1558	

Natural Frequencies/cy

120	91	322	773	104	245	16	277	88	79
310	121	612	1033	414	955	326	657	598	539
430	541	1332	1043	484	1605	336	777	778	639
540	741	1802	1293	534	1645	756	1007	1338	849
650	911	1822	1323	1364	1685	1446	1037	1448	989
990	1301	1832	1363	1444	2135	1966	1157	1628	1299
1340	1611			1644	2145	2196	1877	1638	1779
1810	1721			1974			1977	2148	
2020	1821								
2140	1871								
	2091								

Noise Barriers

970	971		543	144	545	1486	1487		29
	1991				2155	2156	1987		
							2077		

Noise Control

use Noise Reduction

Noise Generation

20	581	102	183	184	1005	846	157	28	819
100	661	202	203	204	1115	876	187	818	1219
170	1011	352	213	324	1195	1116	1167	948	1509

Noise Generation (Continued)

200	1511	462	933	734	1515	1216	1517	1248	
1220	1531	1512	1203	1004	1545	1516		1518	
1640	1841	1942	1373	1064	1705			2128	
1700	1861	2022	1493	1204	2075				
			1563	1514					
			1573	1524					
			1703	1674					

Noise Measurement

390	371	32	543	24	205	156	27	158	159
840	1441	292	833	64	515	206	227	1198	1199
1010		862	1063	504	835		387	1398	1689
1230		902	1093	524	1995		577	1438	1949
1440		1092	1543	734			1667	1858	2159
		1442	1863	1094				1998	
		1912	1993	1734				2108	
		2012	2013	1994					

Noise Meters

use Sound Level Meters

Noise Path Diagnostics

			23						

Noise Prediction

840	441	352	353	354	25		547	548	879
880	1911	1062	1843	464	1385		817		1879
1520		1992							
2160									

Noise Propagation

	581							878	

Noise Reduction

60	191	62	153	54	145	176	57	368	99
260	671	372	323	144	155	216	177	758	159
1170	761	402	353	154	185	346	207	788	189
1510	791	672	573	184	345	546	347	988	199
1520	831	792	663	344	665	1216	367	1218	259
2000	1001	832	813	354	815	1316	497	1488	389
2120	1051	842	843	544	975	1396	767	1988	789
	1221	932	1123	734	1015	1486	837	2018	1469
	1351	972	1233	794	1495	1496	877		1489
	1361	1002	1283	934	1545	1616	927		1519
	1491	1052	1403	1484	1665	1626	1217		1539
	1661	1462	1463	1494	1675	1666	1247		1699
	1691	1482	1623	1544	1855	1996	1397		1989
	1701	1542	1663	1564	2155	2116	1487		1999
	1891	1702	1693	1624	1485	2076	1667		
	2001	1842	1893	1664	1565		1817		
	2021	1892	2023	1714			1837		
	2061	1962	2153	1874			1867		
		2022	2163	1924			1997		
				1984			2017		
				2154			2037		
				2184			2157		
				2194			2197		

Noise Source Identification

	291	182			215		877	908	189
		352					1217		999
		722					1377		1589
		1222					1867		
		1262							

Noise Tolerance

	371			2014	475	566	1687	2078	

Nondestructive Tests

		1772	1923					898	39
									859

Nondestructive Testing
use Nondestructive Tests

Nonholonomic Systems

				854					

Nonlinear Analysis
use Nonlinear Theories

Nonlinear Damping

		1412					1067		

Nonlinear Porgramming

150									

Nonlinear Response

	1761		243	864	595	296	237	238	
				1244					

Nonlinear Systems

10	881	2102		2044	5	6	1727		1099
1860					655				
					2045				

Nonlinear Theories

	1501					126			
						1046			
						1886			

Normal Modes

960				44		16	7		1169
				684		806	47		
							427		

Nozzles

		202							

Nulcear Explosion Damage

		642							139

Nuclear Explosion Detection

		1072							

Nuclear Explosion Effects

	471	472	673	574	575		637		699
		1182		674					1569

Nuclear Explosions

510	511			494					

Nuclear Fuel Elements

		592							

Nuclear Power Plants

590	2011	822	593	124	585	596	197	198	1059
830		1382	823	194	705	706	827	768	1919
1560		1522	1363	704	785	826	1967	828	
			1383	824	825		2097		
				1384	1215				
					2185				

Nuclear Powered Ships

					435	436			

Nuclear Reactor Components

	591	592	1313	1214	2145			588	589
	821								829

Nuclear Reactor Containment

380	381	382	383			586	587	588	379
	591	592					2047		589
		1982							

Nuclear Reactors

		812		594	195	196	197		
					595	1866	707		

Nuclear Weapons Effects

				1754	1755				

Numerical Analysis

1730	11		123	1344	75	2046	1417		319
	1471						2047		429
							2127		1239

- O -

Oceans

									279

Off-Highway Vehicles

							1697		899

Off-Shore Structures

					1315				299
									1699

Optical Measuring Instruments

						906	1437		
						1946			

Optimization

1240	1241	12	893	424	855	76	997	608	1549
	1831	882	2033	654	1045	576	2027	968	
		2152			1145	836			
					1475	1536			

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9 Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Vibration Control

1840	811	552	1283	1714	185	166	1337	318	319
	1541		1523	1844	985	366		838	
			1713		1365	946		1188	
					1505	1086		1568	
					1535	1686		2178	
					2155	2136			

Vibration Dampers

				224		1986	1057		
							1987		

Vibration Damping

	1641	432			1425	46	987	78	2139
	2091	482				1006	1577		
		1312				1206			
						2006			

Vibration Detectors

1090							907		
							1947		

Vibration Effects

		1112				716	1867	1078	909
						786			
						906			

Vibration Energy Method

								608	

Vibration Excitation

570				1054		446			
1790									
1910									

Vibration Frequencies

						956			

Vibration Isolation

1690	201	552		2034		1616	1187	1058	189
	1191	1682				2016	1697	1188	1129
	1371						2017		1519
	1691								

Vibration Isolators

1350		1192	1193	654			997		809
1370				804			1057		
							1987		

Vibration Measurement

1230	721	832	403	304	65	376	1667	288	859
1760	861		513	904	395	1056		568	1789
	1281		903	1864	1345	1086		1398	
	1521		1083		1715	1356		1858	
			1443		2015				
					2035				
					2105				
					2195				

Vibration Meters

	1091						1947		

Vibration Monitoring

		1582	2183				1937		

Vibration Prediction

									1879

Vibration Recording

	1781				2105				

Vibration Reduction
use Vibration Control

Vibration Resonance
use Natural Frequencies

Vibration Response

80	231	602	203	4	75	166	167	18	119
170	321		313	364	425	316	747	308	129
1150	631		603	424	565	746	977	528	439
1640	801		893	564	585	1926	1357	1968	599
2060	931		953	744	1305	2176			689
	1151		1503		1355				749
					1555				1969
					1875				2059
					2165				

Vibration Signatures

				1584					
				1774					

Vibration Spectra
use Vibration Response Spectra

Vibration Tests

500	1731	1202	723	724	505	726	557	68	499
730		1232	1783	764	725	1456	727	168	639
1410		1592		1284	1175		1717	728	729
2010		1782		1574	1405		2187	1358	
		2162		2004					

Vibrators (Machinery)

490		1692		164	1775			508	359
				1684					489
									729

Vibratory Conveyors
use Vibrators (Machinery)
and Materials Handling Equipment

Vibratory Techniques

	1851	1532	1533	164			447		
			2043	1684					

Vibratory Tools

				1534					

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Abstract Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

PERIODICALS SCANNED

PUBLICATION AND ADDRESS	ABBREVIATION
ACTA MECHANICA Springer-Verlag New York Inc. 175 Fifth Ave. New York, NY 10010	Acta Mech.
ACUSTICA S. Hirzel Verlag, Postfach 347 D-700 Stuttgart 1, W. Germany	Acustica
AERONAUTICAL JOURNAL Royal Aeronautical Society 4 Hamilton Place London W1V OBQ UK	Aeronaut. J.
AERONAUTICAL QUARTERLY Royal Aeronautical Society 4 Hamilton Place London W1V OBQ, UK	Aeronaut. Quart.
AIAA JOURNAL American Institute of Aeronautics and Astronautics 1290 Ave. Americas New York, NY 10019	AIAA J.
APPLIED MATHEMATICAL MODELING IPC House, 32 High Street Guildford Surrey GU1 3EW, UK	Appl. Math. Modeling
ARCHIVE FOR RATIONAL MECHANICS AND ANALYSIS Springer-Verlag New York Inc. 175 Fifth Ave. New York, NY 10010	Archive Rational Mech. Anal.
ARCHIVES OF MECHANICS (ARCHIWUM MECHANIKI STOSOWANEJ) Export and Import Enterprise Ruch UL Wronia 23 Warsaw, Poland	Arc. Mech Strosowanej
ATM MESSTECHNISCHE PRAXIS R. Oldenbourg Verlag GmbH Rosenheimer Str. 145 8 München 80, W. Germany	Messtech-nishe Praxis
AUTOMOBILTECHNISCHE ZEITSCHRIFT Franckh'sche Verlagshandlung Abteilung Technik 7000 Stuttgart 1, Pfizerstrasse 5-7 W. Germany	Automobil-tech. Z.
AUTOMOTIVE ENGINEER P. O. Box 24, Northgate Ave. Bury St. Edmunds Suffolk 1P32 GBW, UK	Auto. Engr.
BALL BEARING JOURNAL (English Edition) SKF (U.K.) Ltd. Luton Bedfordshire LU3 1JF, UK	Ball Bearing J.
BAUINGENIEUR S. Hirtzel Verlag, Postfach 347 D-700 Stuttgart 1, W. Germany	Bauingen-ieur

PUBLICATION AND ADDRESS	ABBREVIATION
BROWN BOVERI REVIEW Brown Boveri and Co., Ltd. CH-5401, Baden, Switzerland	Brown Boveri Rev.
BULLETIN DE L'ACADEMIE POLONAISE DES SCIENCES, SERIES DES SCIENCES TECHNIQUES Ars Polona-Ruch 7 Krokowskie Przedmiescie, Poland	Bull. Acad. Polon. Sci., Ser. Sci. Tech.
BULLETIN OF THE FACULTY OF ENGINEERING, YOKAHOMA NATIONAL UNIVERSITY Yokahoma National University OHKA-MACHI, Minami-ku Yokahoma, Japan	Bull. Fac. Engr. Yokahoma Natl. Univ.
BULLETIN OF JAPAN SOCIETY OF MECHANICAL ENGINEERS Japan Society of Mechanical Engineers Sanshin Hokusei Bldg. H-9 Yoyogi 2-chome Shibuya-ku Tokyo 151, Japan	Bull. JSME
BULLETIN OF SEISMOLOGICAL SOCIETY OF AMERICA Bruce A. Bolt Box 826, Berkeley, CA 94705	Bull. Seismol. Soc. Amer.
CIVIL ENGINEERING (NEW YORK) ASCE Publications Office 345 E. 47th St., United Engr. Ctr. New York, NY 10017	Civ. Engr. (N.Y.)
CLOSED LOOP MTS Systems Corp. P. O. Box 24012 Minneapolis, MN 55424	Closed Loop
COMPUTERS AND STRUCTURES Pergamon Press Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Computers and Struc.
DESIGN NEWS Cahners Publishing Co., Inc. 221 Columbus Ave. Boston, MA 02116	Design News
DIESEL AND GAS TURBINE PROGRESS Diesel Engines, Inc. P. O. Box 7406 Milwaukee, WI 53213	Diesel and Gas Turbine Progress
ENGINEERING MATERIALS AND DESIGN IPC Industrial Press Ltd. 33-40 Bowling Green Lane London EC1R, UK	Engr. Matl. Des.
ENVIRONMENTAL QUARTERLY Environmental Publications, Inc. 252-46 Leeds Rd. Little Neck, NY 11362	Environ. Quart.

PUBLICATION AND ADDRESS	ABBREVIATION
EXPERIMENTAL MECHANICS Society for Experimental Stress Analysis 21 Bridge Sq., P.O. Box 277 Westport, CT 06880	Exptl. Mech.
FORSCHUNG IM INGENIEURWESEN Verein Deutscher Ingenieur, GmbH Postfach 1139, Graf-Recke Str. 84 4 Duesseldorf 1, W. Germany	Forsch. Ingenieurw.
GAS TURBINE INTERNATIONAL Gas Turbine 80 Lincoln Ave. Stamford, CT 06904	Gas Turbine Intl.
HIGH-SPEED GROUND TRANSPORTATION JOURNAL Planning Transportation Assoc., Inc. P. O. Box 4824, Duke Station Durham, NC 27706	High-Speed Ground Transp. J.
HYDROCARBON PROCESSING Gulf Publishing Co. Box 2608 Houston, TX 77001	Hydrocarbon Processing
IBM JOURNAL OF RESEARCH AND DEVELOPMENT International Business Machines Corp. Armonk, NY 10504	IBM J. Res. Dev.
INDUSTRIAL RESEARCH Dun-Donnelley Publishing Corp 222 S. Riverside Plaza Chicago, IL 60606	Indus. Res.
INGENIEUR ARCHIV Springer-Verlag New York Inc. 175 Fifth Ave. New York, NY 10010	Ing. Arch.
INSTITUTION OF MARINE ENGINEERS, TRANSACTIONS Marine Media Management Ltd. Memorial Bldg., 76 Mark Lane London EC3R 7JN, UK	Instn. Mar. Engrs., Trans.
INSTITUTION OF MECHANICAL ENGINEERS, (LONDON), PROCEEDINGS Institution of Mechanical Engineers 1 Birdcage Walk, Westminster, London SW1, UK	Instn. Mech. Engr. Proc.
INSTRUMENT SOCIETY OF AMERICA, TRANSACTION Instrument Society of America 400 Stanwix St. Pittsburgh, PA 15222	ISA Trans.
INTERNATIONAL JOURNAL OF CONTROL Taylor and Francis Ltd. 10-14 Macklin St. London WC2B 5NF, UK	Intl. J. Control
INTERNATIONAL JOURNAL OF EARTHQUAKE ENGINEERING AND STRUCTURAL DYNAMICS John Wiley and Sons Ltd. 650 Third Ave. New York, NY 10016	Intl. J. Earthquake Engr. Struc. Dynam.
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES Pergamon Press Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Engr. Sci.
INTERNATIONAL JOURNAL OF MACHINE TOOL DESIGN AND RESEARCH Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Mach. Tool Des. Res.
INTERNATIONAL JOURNAL OF MECHANICAL SCIENCES Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Mech. Sci.
INTERNATIONAL JOURNAL OF NONLINEAR MECHANICS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Nonlinear Mech.
INTERNATIONAL JOURNAL FOR NUMERICAL METHODS IN ENGINEERING John Wiley and Sons, Ltd. 605 Third Ave. New York, NY 10016	Intl. J. Numer. Methods Engr.
INTERNATIONAL JOURNAL OF SOLIDS AND STRUCTURES Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Solids Struc.
ISRAEL JOURNAL OF TECHNOLOGY Weizmann Science Press of Israel Box 801, Jerusalem, Israel	Israel J. Tech.
JAPAN SHIPBUILDING AND MARINE ENGINEERING Technical Information Service, Inc. 2-8 Kanda-Kagi-cho, Chiyoda-ku Tokyo, Japan	J. Shipbldg. Mar. Engr.
JOURNAL DE MÉCANIQUE Gauthier-Villars 55 Quai des Grands Augustines, Paris 6, France	J. de Mécanique
JOURNAL OF THE ACOUSTICAL SOCIETY OF AMERICA American Institute of Physics 335 E. 45th St. New York, NY 10010	J. Acoust. Soc. Amer.
JOURNAL OF AIRCRAFT American Institute of Aeronautics and Astronautics, 1290 Ave. Americas, New York, NY 10019	J. Aircraft
JOURNAL OF THE AMERICAN CONCRETE INSTITUTE American Concrete Institute P. O. Box 4754, Redford Station Detroit, MI 48219	J. Amer. Concrete Inst.
JOURNAL OF THE AMERICAN HELICOPTER SOCIETY American Helicopter Soceity, Inc. 30 E. 42nd St. New York, NY 10017	J. Amer. Helicopter Soc.

PUBLICATION AND ADDRESS	ABBREVIATION
JOURNAL OF BALLISTICS 1339 Brandywine St. Philadelphia, PA 19123	J. Ballistics
JOURNAL OF COMPOSITE MATERIALS Technomic Publishing Co., Inc. 265 Post Road West Westport, CT 06880	J. Composite Matl.
JOURNAL OF ENGINEERING MATHEMATICS Academic Press 198 Ash Street Reading, MA 01867	J. Engr. Math.
JOURNAL OF ENVIRONMENTAL SCIENCES Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056	J. Environ. Sci.
JOURNAL OF THE FRANKLIN INSTITUTE Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	J. Franklin Inst.
JOURNAL OF THE INSTITUTE OF ENGINEERS, AUSTRALIA Science House 157 Gloucter Sidney, Australia 2000	J. Inst. Engr. Australia
JOURNAL OF MECHANICAL ENGINEERING SCIENCE Institution of Mechanical Engineers 1 Birdcage Walk, Westminster London SW1 H9, UK	J. Mech. Engr. Sci.
JOURNAL OF MECHANICAL LABORATORY OF JAPAN (English Edition) The Government Mechanical Lab., Agency of Industrial Science and Technology, 4-12 Igusa Suginami-ku Tokyo, Japan	J. Mech. Lab. Japan
JOURNAL OF THE MECHANICS AND PHYSICS OF SOLIDS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	J. Mech. Phys. Solids
JOURNAL OF PHYSICS E. (SCIENTIFIC INSTRUMENTS) American Institute of Physics 335 E. 45th St. New York, NY 10017	J. Phys. E. (Sci. Instr.)
JOURNAL OF SHIP RESEARCH Society of Naval Architects and Marine Engineers 20th and Northhampton Sts. Easton, PA 18042	J. Ship Res.
JOURNAL OF THE SOCIETY OF ENVIRONMENTAL ENGINEERS The Moding Press Ltd. 6 Conduit St. London W1R 9TG, UK	J. Soc. Environ. Engr.
JOURNAL OF SOUND AND VIBRATION Academic Press 111 Fifth Ave., New York, NY 10019	J. Sound Vib.

PUBLICATION AND ADDRESS	ABBREVIATION
JOURNAL OF SPACECRAFT AND ROCKETS American Institute of Aeronautics and Astronautics, 1290 Ave. Americas New York, NY 10019	J. Spacecraft and Rockets
JOURNAL OF TESTING AND EVALUATION American Society for Testing & Materials 1916 Race St. Philadelphia, PA 19103	J. Test Eval.
LUBRICATION ENGINEERING American Society of Lubrication Engineers, 838 Busse Highway Park Ridge, IL 60068	Lubric. Engr.
MACHINE DESIGN Penton Publishing Co. Penton Bldg., Cleveland, OH 44113	Mach. Des.
MASCHINENBAUTECHNIK VEB Verlag Technik Oranienburger Str. 13/14 102 Berlin, E. Germany	Maschinenbautechnik
MÉCANIQUE APPLIQUÉE Editions de l'Academie De La Republique Socialiste de Roumanie 3 Bis Str., Gutenberg Bucarest, Romania	Mécanique Appliquée
MECCANICA Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Meccanica
MECHANICAL ENGINEERING American Society of Mechanical Engineers 345 E. 47th St. New York, NY 10017	Mech. Engr.
MECHANICS RESEARCH AND COMMUNICATIONS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Mech. Res. and Comm.
MECHANISM AND MACHINE THEORY Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Mech. and Mach. Theory
MEMOIRES OF THE FACULTY OF ENGINEERING, KYOTO UNIVERSITY Kyoto University Kyoto, Japan	Mem. Fac. Engr., Kyoto Univ.
MEMOIRES OF THE FACULTY OF ENGINEERING, NAGOYA UNIVERSITY Library, Nagoya University The Faculty of Engineering Furo-Cho, Chikusa-ku Nagoya, Japan	Mem. Fac. Engr., Nagoya Univ.
MTZ MOTORTECHNISCHE ZEITSCHRIFT Frankh'sche Verlagshandlung 7 Stuttgart 1, Pfizerstrasse 5-7 W. Germany	MTZ Motortech. Z.

PUBLICATION AND ADDRESS	ABBREVIATION
NAVAL ENGINEERS JOURNAL American Society of Naval Engineers Inc. Suite 507 Continental Bldg. 1012 14th St., N.W. Washington, D.C. 20005	Naval Engr. J.
NOISE CONTROL, VIBRATION AND INSULATION Trade and Technical Press Ltd. Crown House, Morden Surrey SM4 5EW, UK	Noise Control, Vib. and Insul.
NOISE CONTROL ENGINEERING P.O. Box 3206 - Arlington Branch Poughkeepsie, NY 12603	Noise Control Engr.
NUCLEAR ENGINEERING AND DESIGN North Holland Publishing Co. P.O. Box 3489 Amsterdam, The Netherlands	Nucl. Engr. Des.
OIL AND GAS JOURNAL The Petroleum Publishing Co. 211 S. Cheyenne Tulsa, OK 74101	Oil and Gas J.
OSAKA UNIVERSITY, TECHNICAL REPORTS Faculty of Technology Osaka University Miyakojima, Osaka, Japan	Osaka Univ., Tech. Rept.
PACKAGE ENGINEERING 5 S. Wabash Ave. Chicago, IL 60603	Package Engr.
POWER TRANSMISSION DESIGN Industrial Publishing Co. Division of Pittway Corp. 812 Huron Rd., Cleveland, OH 44113	Power Transm. Des.
PROCEEDINGS OF THE AMERICAN SOCIETY OF CIVIL ENGINEERS Publications Office, ASCE United Engineering Center, 345 E. 47th St. New York, NY 10017	
JOURNAL OF THE ENGINEERING MECHANICS DIVISION	ASCE J. Engr. Mech. Div.
JOURNAL OF THE GEOTECHNICAL ENGINEERING DIVISION	ASCE J. Geotech. Engr. Div.
JOURNAL OF THE STRUCTURAL DIVISION	ASCE J. Struc. Div.
POWER Power P. O. Box 521 Hightston, NJ 08520	Power
PRODUCT ENGINEERING (NEW YORK) McGraw-Hill Book Co. P. O. Box 1622, New York, NY	Product Engr. (N.Y.)
ROYAL INSTITUTION OF NAVAL ARCHITECTS, TRANSACTIONS Royal Institution of Naval Architects 10 Upper Belgrave St. London SW1X 8BQ, UK	Roy. Instn. Naval Arch., Trans.

PUBLICATION AND ADDRESS	ABBREVIATION
SAE PREPRINTS Society of Automotive Engineers Two Pennsylvania Plaza New York, NY 10001	SAE Prepr.
SHIPBUILDING AND MARINE ENGINEERING INTERNATIONAL Whitehall Technical Press, Ltd. Earl House, 27 Earl St., Maidstone Kent ME 1PE, UK	Shipbldg. Mar. Engr. Intl.
SIAM JOURNAL ON APPLIED MATHEMATICS Society for Industrial and Applied Mathematics, 33 S. 17th St. Philadelphia, PA 19103	SIAM J. Appl. Math.
SIAM JOURNAL ON NUMERICAL ANALYSIS Society for Industrial and Applied Mathematics, 33 S. 17th St. Philadelphia, PA 19103	SIAM J. Numer. Anal.
SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, NEW YORK, TRANSACTIONS Society of Naval Architects and Engineers, 20th and Northhampton St. Easton, PA 18042	Soc. Naval Arch. Mar. Engr., Trans.
S/V, SOUND AND VIBRATION Acoustic Publications, Inc. 27101 E. Oviat Rd. Bay Village, OH 44140	S/V, Sound Vib.
TRANSACTIONS OF THE AMERICAN SOCIETY OF LUBRICATING ENGINEERS Academic Press 111 Fifth Ave., New York, NY 10017	Trans. Amer. Soc. Lubric. Engr.
TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS United Engineering Center, 345 E. 47th St. New York, NY 10017	
JOURNAL OF APPLIED MECHANICS	J. Appl. Mech., Trans. ASME
JOURNAL OF DYNAMIC SYSTEMS, MEASUREMENT AND CONTROL	J. Dyn. Syst., Meas. and Control, Trans. ASME
JOURNAL OF ENGINEERING FOR INDUSTRY	J. Engr. Indus., Trans. ASME
JOURNAL OF ENGINEERING FOR POWER	J. Engr. Power, Trans. ASME
JOURNAL OF LUBRICATION TECHNOLOGY	J. Lubric. Tech., Trans. ASME
TRANSACTIONS OF THE INSTRUMENT SOCIETY OF AMERICA Instrument Society of America 400 Standix St. Pittsburgh, PA 15222	Trans. Instr. Soc. Amer.

PUBLICATION AND ADDRESS	ABBREVIATION
TRANSACTIONS OF THE NORTH EAST COAST INSTITUTION OF ENGINEERS AND SHIPBUILDERS North East Coast Institution of Engineers Bolbec Hall, Newcastle Upon Tyne 1 UK	Trans. North East Coast Inst. Engr. Shipbldg.
VDI ZEITSCHRIFT Verein Duetscher Ingenieur GmbH Postfach 1139, Graf-Recke Str. 84 4 Duesseldorf 1, Germany	VDI Z.
VEHICLE SYSTEMS DYNAMICS Swets and Zeitlinger N.V. 347 B Herreweg Lisse, The Netherlands	Vehicle Syst. Dyn.
VIBROTECHNIKA Kauno Polytechnikos Institutas Kaunas, Lithuania	Vibro-technika
WEAR Elsevier Sequoia S.A. P. O. Box 851 1001 Lausanne 1, Switzerland	Wear
ZEITSCHRIFT FÜR ANGEWANDTE MATHEMATIK UND MECHANIK Akademie Verlag GmbH Liepsiger Str. 3-4 108 Berlin, Germany	Z. angew. Math. Mech.
ZEITSCHRIFT FÜR FLUGWISSENSCHAFTEN DFVLR D-3300 Braunschweig Flughafen, Postfach 3267, W. Germany	Z. Flugwiss

ANNUAL PROCEEDINGS SCANNED

PUBLICATION	ABBREVIATION
INTERNATIONAL CONGRESS ON ACOUSTICS, ANNUAL PROCEEDINGS	Intl. Cong. Acoust., Proc
INSTITUTE OF ENVIRONMENTAL SCIENCES, ANNUAL PROCEEDINGS Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056	Inst. Environ. Sci., Proc.
MIDWESTERN CONFERENCE ON SOLID MECHANICS. ANNUAL PROCEEDINGS	Midw. Conf. Solid Mech., Proc.
THE SHOCK AND VIBRATION BULLETIN, UNITED STATES NAVAL RESEARCH LABORATORIES, ANNUAL PROCEEDINGS Shock and Vibration Information Ctr. Naval Research Lab., Code 8404 Washington, D.C. 20375	Shock Vib. Bull., U.S. Naval Res. Lab., Proc.
UNITED STATES CONGRESS ON APPLIED MECHANICS, ANNUAL PROCEEDINGS	U.S. Cong. Appl Mech., Proc.
WORLD CONGRESS ON APPLIED MECHANICS, ANNUAL PROCEEDINGS	World Cong. Appl. Mech., Proc.

CALENDAR

MARCH 1978

25-27 **Applied Mechanics Western and J.S.M.E. Conference,** Honolulu, Hawaii ***(ASME Hq.)***

APRIL 1978

3-5 **Structures, Structural Dynamics and Materials Conference,** [ASME] Bethesda, MD ***(ASME Hq.)***

9-13 **Gas Turbine Conference & Products Show,** [ASME] London ***(ASME Hq.)***

17-20 **Design Engineering Conference & Show** [ASME] Chicago, IL ***(R.C. Rosaler, Rice Assoc., 400 Madison Ave., N.Y., NY 10017)***

17-20 **24th Annual Technical Meeting and Equipment Exposition** [IES] Fort Worth, TX ***(IES Hq.)***

24-28 **Spring Convention** [ASCE] Pittsburgh, PA ***(ASCE Hq)***

MAY 1978

4-5 **IX Southeastern Conference on Theoretical and Applied Mechanics** [SECTAM] Nashville, TN ***(Dr. R.J. Bell, SECTAM, Dept. of Engrg. Sci. & Mech., Virginia Polytechnic Inst. & State Univ., Blacksburg, VA 24061)***

8-10 **Inter-NOISE 78,** San Francisco, CA ***(INCE, W.W. Lang)***

8-11 **Offshore Technology Conference,** Houston, TX ***(SPE, Mrs. K. Lee, Mtgs. Section, 6200 N. Central Expressway, Dallas, TX 75206)***

14-19 **Society for Experimental Stress Analysis,** Wichita, KS ***(SESA, B.E. Rossi)***

16-19 **Acoustical Society of America, Spring Meeting,** [ASA] Miami Beach, FL ***(ASA Hq.)***

JUNE 1978

30 **Eighth U.S. Congress of Applied Mechanics,** [ASME] Los Angeles, CA ***(ASME)***

SEPTEMBER 1978

24-27 **Design Engineering Technical Conference,** [ASME] Minneapolis, MN ***(ASME Hq.)***

OCTOBER 1978

49th Shock and Vibration Symposium, Washington D.C. ***(H.C. Pusey, Director, The Shock and Vibration Info. Ctr., Code 8404, Naval Res. Lab., Washington, D.C. 20375 Tel. (202) 767-3306)***

1-4 **Design Engineering Technical Conference,** [ASME] Minneapolis, MN ***(ASME Hq.)***

8-11 **Diesel and Gas Engine Power Conference and Exhibit,** [ASME] Houston, TX ***(ASME Hq.)***

8-11 **Petroleum Mechanical Engineering Conference,** [ASME] Houston, TX ***(ASME Hq.)***

17-19 **Joint Lubrication Conference,** [ASME] Minneapolis, MN ***(ASME Hq.)***

26-Dec 1 **Acoustical Society of America, Fall Meeting,** [ASA] Honolulu, Hawaii ***(ASA Hq.)***

DECEMBER 1978

10-15 **Winter Annual Meeting,** [ASME] San Francisco, CA ***(ASME Hq.)***

CALENDAR ACRONYM DEFINITIONS AND ADDRESSES OF SOCIETY HEADQUARTERS

AFIPS:	American Federation of Information Processing Societies 210 Summit Ave., Montvale, NJ 07645
AGMA:	American Gear Manufacturers Association 1330 Mass. Ave., N.W. Washington, D.C.
AHS:	American Helicopter Society 1325 18 St. N.W. Washington, D.C. 20036
AIAA:	American Institute of Aeronautics and Astronautics, 1290 Sixth Ave. New York, NY 10019
AIChE:	American Institute of Chemical Engineers 345 E. 47th St. New York, NY 10017
AREA:	American Railway Engineering Association 59 E. Van Buren St. Chicago, IL 60605
AHS:	American Helicopter Society 30 E. 42nd St. New York, NY 10017
ARPA:	Advanced Research Projects Agency
ASA:	Acoustical Society of America 335 E. 45th St. New York, NY 10017
ASCE:	American Society of Civil Engineers 345 E. 45th St. New York, NY 10017
ASME:	American Society of Mechanical Engineers 345 E. 47th St. New York, NY 10017
ASNT:	American Society for Nondestructive Testing 914 Chicago Ave. Evanston, IL 60202
ASQC:	American Society for Quality Control 161 W. Wisconsin Ave. Milwaukee, WI 53203
ASTM:	American Society for Testing and Materials 1916 Race St. Philadelphia, PA 19103
CCCAM:	Chairman, c/o Dept. ME, Univ. Toronto, Toronto 5, Ontario, Canada
ICF:	International Congress on Fracture Tohoku Univ. Sendai, Japan
IEEE:	Institute of Electrical and Electronics Engineers 345 E. 47th St. New York, NY 10017
IES:	Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056
IFToMM:	International Federation for Theory of Machines and Mechanisms, US Council for TMM, c/o Univ. Mass., Dept. ME Amherst, MA 01002
INCE:	Institute of Noise Control Engineering P.O. Box 3206, Arlington Branch Poughkeepsie, NY 12603
ISA:	Instrument Society of America 400 Stanwix St. Pittsburgh, PA 15222
ONR:	Office of Naval Research Code 40084, Dept. Navy Arlington, VA 22217
SAE:	Society of Automotive Engineers 400 Commonwealth Drive Warrendale, PA 15096
SEE:	Society of Environmental Engineers 6 Conduit St. London W1R 9TG, UK
SESA:	Society for Experimental Stress Analysis 21 Bridge Sq. Westport, CT 06880
SNAME:	Society of Naval Architects and Marine Engineers, 74 Trinity Pl. New York, NY 10006
SPE:	Society of Petroleum Engineers 6200 N. Central Expressway Dallas, TX 75206
SVIC:	Shock and Vibration Information Center Naval Research Lab., Code 8404 Washington, D.C. 20375
URSI-USNC:	International Union of Radio Science - US National Committee c/o MIT Lincoln Lab., Lexington, MA 02173