Design Basis for Resistance to Shock and Vibration*

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INTRODUCTION

Sandia National Laboratories, in conjunction with its participation in the American National Standards Institute (ANSI) writing groups, has undertaken to provide an experimental and analytical basis for the design of components of radioactive materials packages to resist normal transport shock and vibration loads. Previous efforts have resulted in an overly conservative shock spectra description of the loads in the tie-downs and cask attachment points anticipated during normal shipment. The present effort is aimed at predicting the actual loads so that the design basis can be accurately determined. This goal is being accomplished with road simulator and over-the-road tests and the development of an analytical model. This model is used to parametrically evaluate and envelop the transportation systems' responses. The parameters to be varied include damping, stiffness, geometry, and cargo mass.

The over-the-road tests provide operational data that are used to validate the selection of environments for the road simulator tests. The road simulator tests provide verification for the model. This verification is accomplished since the road simulator tests provide not only the system response which can be measured in over-the-road tests but also the system input. Finally, when the model has been verified, it can be used to vary parameters to envelope a wide range of normal transport conditions.

ROAD SIMULATOR TESTS

The tests were conducted on the Fruehauf Research and Development Division's road simulator, which is equipped with six electro-hydraulic servo-controlled actuators. The hydraulic actuator capacity is a ± 5-inch (12.7 cm) stroke over a nominal road frequency range of 0 to 50 Hz.

During the tests, the actuators sequentially drove the tires of the tractor front axle, the tractor tandem, then the trailer tandem. Each actuator set was independent, and no correlation in time was present in the test. These separately driven responses were then combined to generate the total response of a given accelerometer location to the road.

These tests utilized two separate cask/trailer systems of a similar geometry and load but with varying suspension types to verify the ability of the analytical model to respond correctly to structural changes. The first system consisted of a White Freight Liner tractor with leaf-spring suspension and a Fruehauf drop frame trailer with air-sprung rear suspension. The load was a NuPac 7D-3.0 Type B cask weighing 26,000 pounds (116,000 N) which contained 5,400 (24,000 N) pounds of sand.

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The second system tested used an air-sprung tractor and a Transport Systems drop frame trailer with an air-sprung suspension. It carried a CNS 14-170 cask. This cask, which weighs 32,000 pounds (142,000 N), carried a contents load of 5,000 pounds (22,200 N). Three types of roads and four shock events were simulated for these tests. These included a smooth primary road, a spalled primary road, and a rough secondary road. Each road surface input function was obtained by matching the response of an over-the-road test with the input from the road simulator.

The smooth primary road was derived from a section of Interstate 75 south of Detroit, driven at a speed of 55 mph (88 kph). The surface was smooth asphalt over original concrete and contained slight undulations due to the subsurface.

The spalled primary road was derived from a section of Interstate 94 within the Detroit city limits. The road was driven at a speed of 45 mph (72 kph). The surface is concrete which was cracked and broken after a harsh winter.

The rough secondary road was derived from a section of 31 Mile Road driven at 40 mph (64 kph). This is a county road north of Detroit whose surface is rough, old, and weathered asphalt.

The transient or shock environments were derived from a section of the Bendix Test Track in South Bend, Indiana that was constructed and maintained for this type of test. The test track includes diagonal bumps, undulating road, chatter bumps, and a 4-inch (10-cm) deep chuck hole.

The condition of each cask/truck system was average in that the components were not new but were of good repair. The tire pressure used was the maximum recommended, and the air bags were inflated to their normal operating pressure.

OVER-THE-ROAD TESTS

The over-the-road tests were performed on three radioactive materials transport systems. These included the Transuranic Package Transporter (TRUPACT-I), the CNS 3-55, and the CNS 14-170. All trailers used in the over-the-road tests had air sprung suspension systems.

The TRUPACT system was designed to ship contact handled transuranic waste. The outside dimensions are 25 feet long by 8 feet wide by 9 feet tall (7.6 m x 2.4 m x 2.7 m), and its total weight is 50,000 pounds (222,000 N). The package is tied down to the trailer at four ISO frame locations.

The CNS 14-170 is designed to ship dewatered or solidified waste material. The outer diameter is 93.7 inches (2.4 m), and it is 84.25 inches (2.1 m) tall. The package is shipped vertically. It is tied down to the trailer using crossed wire ropes. The total weight was 47,000 pounds (209,000 N).

The CNS 3-55 is designed to ship irradiated components and solidified wastes. The cask is a steel-encased lead cylinder that is 133.75 inches (3.4 m) long and 50.5 inches (1.3 m) in diameter. The cask is shipped horizontally in a steel cradle with steel strap tie-downs. The total weight was 57,000 pounds (254,000 N).

The CNS casks were tested in Barnwell, South Carolina and the TRUPACT-I was tested in Albuquerque, New Mexico. The type of roads and shock events are listed in Table I.

TABLE I

Shock and Vibration Events

Casks	Event
CNS, TRUPACT	Smooth asphalt primary
CNS, TRUPACT	Railroad grade crossing
CNS, TRUPACT	Bridge approach
CNS, TRUPACT	Rough asphalt primary
CNS, TRUPACT	Rough concrete primary
CNS, TRUPACT	Rough asphalt secondary
CNS, TRUPACT	Spalled asphalt secondary
CNS	Turning
CNS	Braking

INSTRUMENTATION

All road simulator and over-the-road tests were instrumented to determine the loads acting on the packages. Accelerometers were used to obtain vertical, longitudinal, and transverse accelerations. These were mounted at tie-down locations, on the packages, and on the trailer bed. Load cells were used in the CNS 14-170 and NuPac 7D-3.0 tie-downs to directly monitor tie-down loads. Strain gages were used on the CNS 3-55 tie-down straps so that tie-down loads could be calculated.

TEST RESULTS

A sample of the test data is given in Table II and Table III. Table II is the peak vibration data for the CNS 14-170 cask. Table III lists the root-mean-square (RMS) response for the same events. Data from all the tests is included in SAND88-0589, "Truck Cask Response to Normal Environments," K. W. Gwinn and R. E. Glass, Sandia National Laboratories.

TABLE II

Peak Response for Road Surface Events

Transducer Location	Event: Smooth Asphalt	Rough Asphalt	Rough Concrete	Secondary Asphalt	Spalled Asphalt
Cask top					
Transverse (g)	0.17	0.21	0.12	0.13	0.22
Vertical (g)	0.23	0.32	0.20	0.35	0.58
Longitudinal (g)	0.17	0.38	0.22	0.65	0.88
Trailer, mid					
Vertical (g)	0.21	0.37	0.07	0.07	0.08
Trailer, rear					
Vertical (g)	0.46	1.4	0.95	1.68	3.1
Longitudinal (g)	0.14	0.37	0.22	0.43	0.85
Trailer, front					
Vertical (g)	0.73	1.7	1.3	2.7	4.5
Front tiedown (lb)	430	580	220	350	460
(N)	1900	2600	980	1600	2000
Rear tiedown (lb)	220	360	150	280	650
(N)	980	1600	670	1200	2900

TABLE III

RMS Response for Road Surface Events

Transducer Location	Event: Smooth Asphalt	Rough Asphalt	Rough Concrete	Secondary Asphalt	Spalled Asphalt
Cask top					
Transverse (g)	0.042	0.043	0.025	0.027	0.054
Vertical (g)	0.041	0.096	0.050	0.066	0.125
Longitudinal (g)	0.041	0.057	0.055	0.143	0.227
Trailer, mid					
Vertical (g)	0.040	0.093	0.010	0.011	0.011
Trailer, rear					
Vertical (g)	0.135	0.211	0.233	0.401	0.718
Longitudinal (g)	0.030	0.042	0.059	0.088	0.180
Trailer, front					
Vertical (g)	0.201	0.294	0.403	0.571	1.03
Front tiedown (lb)	141	156	55.4	87.5	137.
(N)	630	690	250	390	610
Rear tiedown (lb)	70	93	36.3	72.5	150.
(N)	310	410	160	320	670

It can be seen for both peak and RMS values that the cask response was less than 1 g and that the tie-down loads were small. This is due to the cask and trailer acting essentially as a single unit with the trailer bending about the cask. This does result in relatively high g loads (up to 4.5) acting at points on the trailer away from the cask. The frequency content and mode shapes of the trailer that result in these loads will be discussed in the analysis sections.

ANALYTICAL MODEL

The analytical model is used for parametric studies of the cask and trailer components. Parameters varied in the analysis were tie-down stiffness, trailer main beam stiffness and damping, cask mass, and suspension stiffness and damping. This allows the model to represent a range of cask/trailer systems. The model, as shown in Fig. 1, represents the truck/trailer components in the pitch plane. Two important points to note from the figure are: (1) the inclusion of the tractor geometry and inertia which are required to obtain the complete set of deformation modes and (2) the separate spring and damper elements for the tires and suspension components for the trailer and tractor. This model allows translation fore and aft with rotation normal to the direction of travel only. This model is used because the tractor/trailer response is dominated by the vertical inputs as described in "Truck Ride Improvement Using Analytical and Optimization Methods," J. M. Baum et al., SAE Technical Paper No. 770609, 1978. For similar reasons lateral motion and other three-dimensional modes of the truck have been shown in tests to account for less than 10 percent of the responses in the plane shown.

The next step in ensuring generality of the model response is to select an input which is vehicle independent. This is accomplished by loading the model with the intrinsically random road surface at the tire. Many different road surfaces can be analyzed using the same structural model just as the same truck is driven over many different road types.

To model a complex structure with several massive constituents, such as a trailer/tractor, a multiple degree-of-freedom model is required. Because this model resembles the actual structure and comparisons with actual test data are direct, the contributions due to any specific mode of deformation is easily discerned and understood.

The details of the finite element model are as follows. Tractor and trailer frame members were represented as beam elements, while the suspensions were modeled as discrete damping and elastic elements. Each suspension was modeled using a pair of elements to represent the spring components and a pair to represent the tires. The cask centerline was modeled as a beam and was connected to the tie-down points using rigid link elements. In this manner any geometric rotations of the cask will be correctly transformed to the tie-down response. The tie-downs were modeled using truss elements which can have no moments at the ends, thus, simulating a wire rope connection to the cask. Tie-downs were modeled at the top and bottom of the cask. Consistent mass was used for all elements so that rotational inertias are incorporated in the analysis. The inertia and center of gravity of the tractor components were also included. The finite element code MSC/Nastran, as described in MSC/Nastran User's Guide, Version 65, 1985, was used for the analysis.

RANDOM RESPONSE DESCRIPTION

The response of a linear system in the frequency domain is related to the input by the frequency response function (FRF), $H(\omega)$. This is shown mathematically in Eq. 1 as described in An Introduction to Random Vibrations and Spectral Analysis, D. E. Newland, 1981,

$$So(\omega) = H(\omega)^2 \, Si(\omega) \tag{1}$$

where $Si(\omega)$ and $So(\omega)$ are the input and output spectral density functions, respectively.

This operation is completed in the frequency domain for several reasons. First, the mathematics are very straightforward as is seen in Eq. 1. Second, in conjunction with the mode shapes and frequencies, the contributions of any particular mode to the total system response is easily distinguished from the FRF's. An example of an FRF with the corresponding mode shapes is shown in Fig. 2. Third, the formulations of the input and output spectral densities are random functions and can be compared directly with measured road surface roughness descriptions and measured response data from tests.

The input spectral density gives an idea how the input energies are distributed by frequency. The output spectral density shows the relative contribution of each mode to the total response of the system. Examples of an output spectral densities are shown in Figs. 3 and 4. Fig. 3 shows the response of the cask vertically to the rough road input. The tie-down force spectral response density shown in Fig. 4 demonstrates that the majority of response is due to the first bounce mode shown in Fig. 2, which occurs at 1.8 Hz. Very low contributions from other modal deformations occur up to 15 Hz, but the response is less than one-fifth the peak at 1.8 Hz.

The square root of the area under the spectral density response curve is defined as the root mean square (RMS). This value is used to envelop the probability of an occurrence. Three times the RMS value will be greater than 99.9 percent of all expected values for a Gaussian process. For example, the RMS value of the tie-down force obtained in traveling down a rough road was determined from Fig. 4 to be 248 pounds (1,100 N). The force gage used in this test data was zeroed out after the preload was applied, so the 3 RMS value is 744 pounds (3,300 N), which is added to the preload of 2,000 pounds (8,900 N). Therefore, 99.9 percent of the time the force in the tie-down will be less than 2,744 pounds (12,200 N). The absolute maximum measured will always be greater than 3 RMS. The maximum responses for all tests were normally in the range of 4 to 5 times the RMS.

For fatigue assessment the RMS response is used. Because the responses at 1.8 Hz dominate the tie-down force shown, fatigue analysis requirement should use a 1.8 Hz loading cycle.

MODEL VERIFICATION

Road simulation was used to verify the analytical model since both the input and output are known. By matching the output quantities from the road simulation test, the verification of the model is completed.

The analytical model was able to simulate the natural frequencies and mode shapes observed in the data to 20 Hz. The first several mode shapes, shown in Fig. 2, dominate the response of the trailer and cask system. The first mode is the bounce mode of the trailer and tractor, bouncing on the suspension in unison. This first mode occurs at 1.8 Hz. The second mode, which occurs at 4.5 Hz, is the pitching of the trailer and tractor with deformation still limited to the suspension elements. The next several modes involve deformation of the structure with combinations of bending of the main frame members in the trailer and tractor. The pattern of mode shapes and frequencies has been found to be common to a variety of trailer/cask configurations as described in "TRUPACT-I Over-The-Road Test," R. E. Glass and K. W. Gwinn, Sandia National Laboratories, 1987, and "Truck Cask Response to Normal Environments," K. W. Gwinn and R. E. Glass, Sandia National Laboratories, to be published. Once the mode shapes and frequencies of the structure were matched, the tie-down forces from the analytical model were matched to those measured during the road simulation tests. This was accomplished by varying the damping of the suspension elements.

ANALYSIS RESULTS

The responses of the cask and tie-downs were investigated using the baseline analytical model described above to determine the effects of varying cask/tie-down/trailer components.

Varying the tie-down stiffness, which would represent altering the sizes of the tie-downs at the base and top of the cask, produced no practical effect on the tie-down forces. Varying the stiffness by an order of magnitude resulted in a 15 percent change in tie-down force.

The system was also insensitive to changes to the trailer frame stiffness. Varying the frame stiffness by 50 percent produced a 2 percent change in tie-down force. It should be noted that the responses at other locations on the trailer were significantly affected by these variations. This is of interest to trailer designers.

As was seen in Fig. 2, the majority of the cask/tie-down response was due to the first bounce mode of the trailer at 1.8 Hz. Changes to the damping of the modes below 3 Hz produced the greatest effect. The nominal damping to match the road simulation was between 2 and 3 percent of critical for the suspension elements, while changes to the frame damping had little effect on the tie-down response at these frequencies. It is interesting to note that there was little difference between the RMS force response in the tie-downs for the air-ride or leaf spring suspensions. There was a slight difference in the peak response between the two types of suspensions.

The way to affect the tie-down force is to change the response of the cask. This is because the cask inertia dominates the response of the trailer, resulting in little or no motion of the trailer in the vicinity of the cask. Relative displacements between the cask and trailer tie-down points relate directly to changes in tie-down force. It is obvious from the modal shapes that deformation of the trailer is small at frequencies above 12 Hz. At these higher frequencies the energy in the road surface has dropped by an order of magnitude, as is shown in Fig. 3. Correspondingly, the energy to drive these modes has dropped an order of magnitude, and the driving force on these higher modes is low.

The model was used to investigate the effect of changes in the cask mass. The first bounce mode for the system tested was in the 1.8 Hz range. This cask weighed 49,000 pounds (218,000 N). For a cask weighing 3,500 pounds (15,600 N), the first mode increases to 2.14 Hz. Because of this small change in frequency in the dominant mode, the nature of the cask/tie-down responses remains the same. The complete list of results are shown in Table IV. The maximum RMS tie-down force per unit of cask weight is shown to be 0.0074 for the 14,000- to 21,000-pound cask weights. This RMS force per tie-down was for a system which had four tie-downs. The total tie-down force would be approximately 0.1 g for three times the RMS.

TABLE IV
Analytical Extension to Lighter Packaging

Cask Weight lbs.	(N)	RMS Force per Tiedown lbs.	(N)	RMS Force/Cask Weight	Bounce Mode (Hz)
49000	(218,000)	210	(934)	.0045	1.7
42000	(187,000)	187	(832)	.0045	1.8
35000*	(156,000)	150	(667)	.0043	1.9
28000	(125,000)	119	(529)	.0043	2.0
21000	(93,000)	155	(689)	.0074	2.1
14000	(62,000)	104	(462)	.0074	2.1
7000	(31,000)	47.7	(212)	.0067	2.1
3500	(16,000)	20.8	(93)	.0061	2.1

*Experimentally defined in road simulation test.

CONCLUSIONS

Relating the total tie-down force in terms of the rigid body acceleration of the cask mass produces loads below 0.1 g. The maximum tie-down design values proposed for the ANSI N14.23 are shown in Table V. These values are a factor of 25 times greater than the calculated values. These design levels are adequate and represent a very safe design margin for casks.

The g levels shown in Table V represent the maximum level expected due to very rough road operation. Using a high factor of safety on strength is prudent for the design of these components. For fatigue analysis the actual operating environment should be used. Because of the nature of the fatigue assessment for a structural component, actual vibration data needs to be used to provide a realistic basis for design. Since strength or capacity is adequately covered by the levels in Table V, "normal" road environments should be used for the vibration analysis. In this manner a desired factor of safety can be chosen for the particular application.

Data from Table IV can be used to describe the RMS force in the cask and tie-down components for truck-transported packaging with flexible tie-downs. As an example, two multipliers can be used to envelop the response. From the test data shown above, a value of .0045 lbs. RMS per tie-down can be used for heavier casks, and from the trends shown in Table II, twice that value for casks under 25,000 pounds. For steel strap/skid systems, the multiplier would be .020 RMS/pound for each strap. The load cycle necessary for fatigue would be 2 Hz for all casks and tie-down components.

TABLE V
Design Loads

Direction	Inertial Load (g)
Forward	2.3
Backward	2.3
Sideward	1.6
Upward	2.0

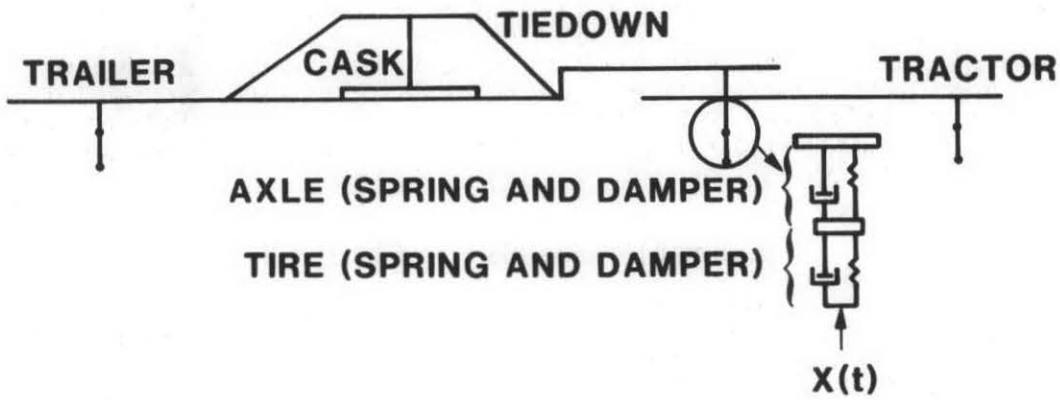

Figure 1. Truck/Trailer Finite Element Model

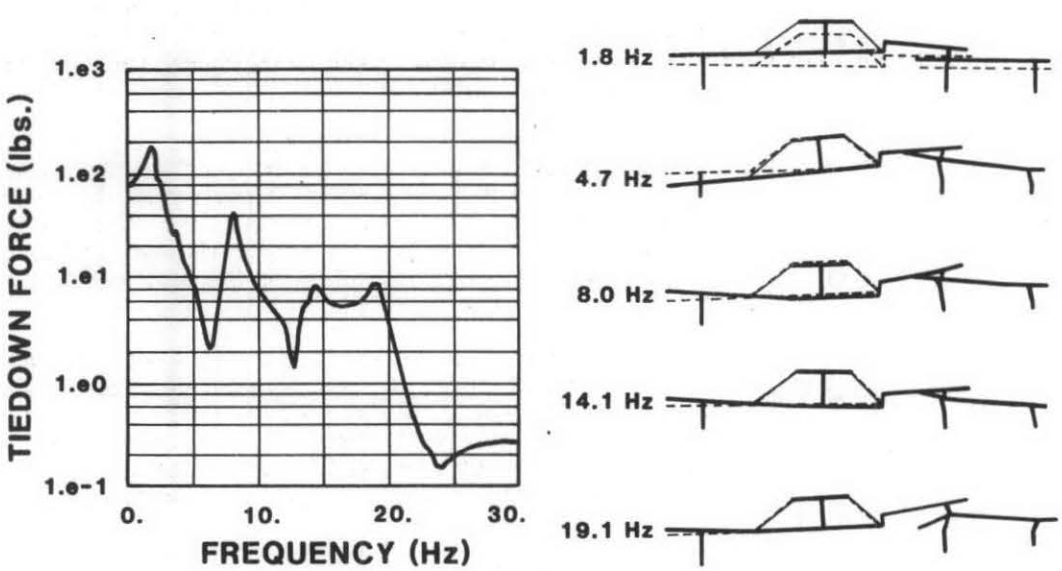

Figure 2. Frequency Response Function with Corresponding Mode Shapes

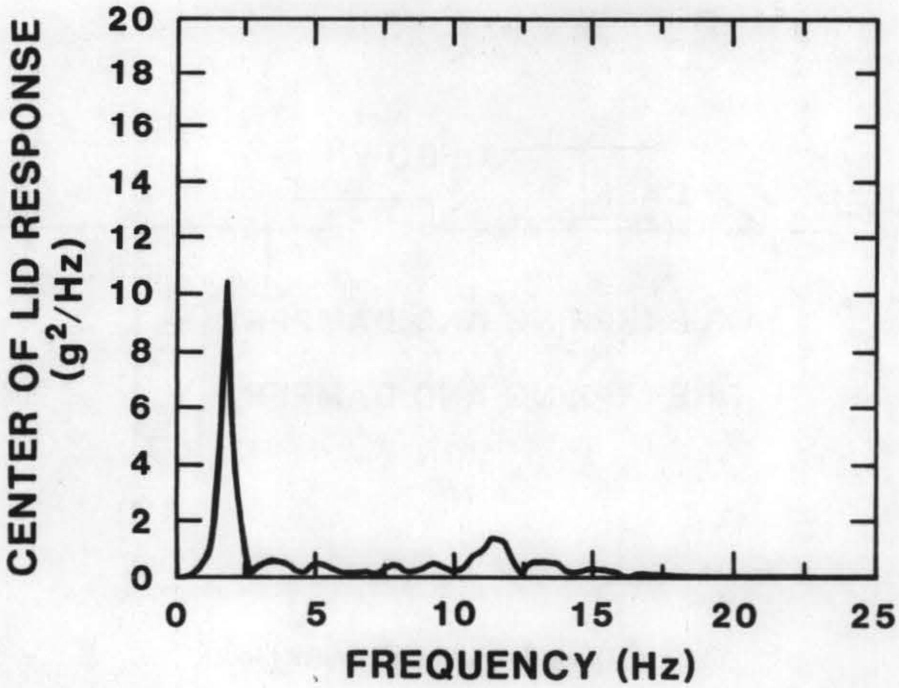

Figure 3. Power Spectral Density of Vertical Cask Response

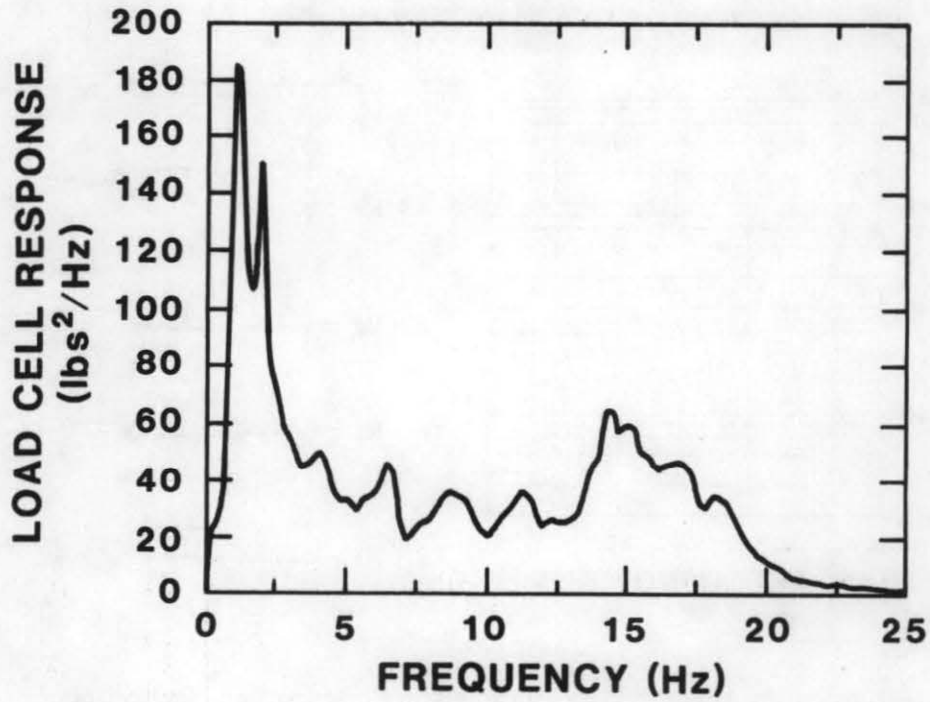

Figure 4. Power Spectral Density of of Tie-Down Load

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