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RESPONSE OF A GRID SPACED FUEL ASSEMBLY TO SHIPPING ENVIRONMENT

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SUMMARY

The grid-spacer assembly is a viable method for providing lateral support to the fuel rods within the driver assemblies of Liquid Metal Fast Breeder Reactors. A honeycomb grid assembly has been used in the German and British LMFBR development programs and, more recently, by W-ARD as an alternate to the present wire-wrap fuel rod support system. A unique grid-spacer design for supporting fuel rods was conceived and is being developed at GE/FBRD.

Instead of the wire-wrapped lateral support system used with the driver fuel, the GE Grid-spaced Driver Assembly (GSDA) employs a series of staggered-grid spacer assemblies. The GE grid-spacer assembly design is based upon the bi-planar, line-of-sight support beam concept. The principle of fuel rod support in the GE system is to give compliance to the spacer beams by shaping them to a shallow triangular wave form to give point (rather than line) contact with the rods. Each grid assembly consists of two layers of parallel wave beams that are projection welded to form an assembly. The top layer of wave beams forms a 60° angle with the lower set of beams. Since beams in a layer occupy only every other available space ("staggered grids") a pair of grid assemblies is required to support each fuel rod on four sides. The main incentive for using staggered grids is to reduce the coolant pressure drop and hot spot temperatures within the assembly and to provide a more efficient means of accommodating fuel rod diametral growth. The grid-spacer assembly is mechanically attached to the inside of the hexagonal channel walls by tabs on the end of each wave beam, seating into recesses formed in the channel wall.

One of the important considerations in the design of grid spaced nuclear fuel assembly (GSA) is that it must not suffer any damage during shipping and handling. This is avoided first of all by designing secure but slip-tight interlocked fit between tabs of the grid spacers and recesses in the surrounding hexagonal channel, and secondly by insuring that dynamic forces which act on the hexagonal channel, and fuel rods during shipping and handling are not large enough as to result in disengagement of even a few spacer tabs from recesses in the channel. Strict adherence to limits regarding maximum acceleration during shipping and handling was felt imperative for truck transport of the GSA.

This paper presents a dynamic analysis approach for design verification of the GSA under conditions of shipping and handling using prototypical test examples of transportation by truck. Analytical response results were obtained using representative trailer bed acceleration time histories which had been recorded earlier under simulated shipping by truck transport environment. Important parameters affecting response are identified. Recognizing that the analytical estimate of the fundamental periods of the GSA cannot be very accurate owing to the complex geometry, upper bounds on acceleration reponses were established using the standard response spectrum technique. Finally, the GSA design was verified by laboratory tests—both static and dynamic. Static tests were used to calibrate displacement response of the grid spacer tabs relative to the fuel rods. Prototypical dynamic shaking table test were next employed in which the response of the GSA was determined in sinusoidal sweep type of excitation. Handling loads were represented by shock tests consisting of one-cycle impulses at various levels of acceleration.

The results of laboratory testing were seen to reinforce credance of design verification by analysis of the GSA subjected to shipping and handling environment.

1. Introduction

The grid-spacer assembly is a viable method for providing lateral support to the fuel rods within the driver assemblies of Liquid Metal Fast Breeder Reactors (LMFBR). A honeycomb grid assembly has been used by the Germans and the British in their LMFBR development programs and, more recently, by W-ARD as an alternate to the present wire-wrap fuel rod support system. A unique grid-spacer design for supporting fuel rods was conceived and is being developed at General Electric Fast Breeder Reactor Department (GE/FBRD).

Instead of the wire-wrapped lateral support system used with the driver fuel, the GE grid-spaced driver assembly (GSDA) employs a series of staggered-grid spacer assemblies. The GE grid-spacer assembly (GSA) design is based upon the bi-planar, line-of-sight support beam concept (Figure 1). The principle of fuel rod support in the GE system is to comply to the spacer beams by shaping them to a shallow triangular wave form to give point (rather than line) contact with the rods. Each grid assembly, consisting of two layers of parallel wave beams, forms a 60° angle with the lower set of beams. Because beams in a layer occupy only every other available space ("staggered grids"), a pair of grid assemblies is required to support each fuel rod on four sides. The main incentive for using staggered grids is to reduce the coolant pressure drop and hot spot temperatures within the assembly, and to provide a more efficient means of accommodating fuel rod diametral growth. The grid-spacer assembly is mechanically attached to the inside of the hexagonal channel walls by tabs on the end of each wave beam, seating into recesses preformed in the channel wall (Figure 2).

One of the important considerations in the design of a grid-spaced nuclear fuel assembly is that it must not be damaged during shipping and handling. This is avoided, first of all, by designing a secure but slip-tight interlocked fit between tabs of the grid spacers and recesses in the surrounding hexagonal channel (Figure 3), and secondly, by ensuring that dynamic forces which act on the hexagonal channel and fuel rods during shipping and handling are not large enough to disengage even a few spacer tabs from recesses in the channel. Minimum G values for design against shock and vibration are therefore required; RDT Standard F9-8T [1], for example, specifies such limits in highway transport by truck or tractor-trailer for fuel when package weight exceeds one ton. To minimize the effects of shock and vibration on fuel assemblies similar to the GSA, a few designs of packaging with or without a base and steel or nylon tiedown, using an air suspended tractor-trailer, had been tested earlier under representative field conditions [2]. Based on these test results and the RDT Standard requirements for air suspended trailer hauled by air suspended tractors, maximum package acceleration response limits of 6G's in the vertical direction and 2G's in the horizontal direction were adopted as shipping design bases. Strict adherence to these limits during shipping and handling in either horizontal or vertical mode was felt to be imperative for truck transport of the GSDA.

This paper presents a dynamic analysis approach to design verification of the GSA, under conditions of shipping and handling that uses prototypical tests of transportation by truck. Analytical response results were obtained by using representative trailer bed acceleration time histories which had been recorded earlier [2] under the environment of shipping by truck transport. Important parameters affecting response are identified. Recognizing that the analytical estimate of the fundamental periods of the GSA in the shipping mode (horizontal or vertical) cannot be very accurate because of the complex geometry, and other difficulties in precisely determining the dynamic properties of the GSA, the upper bounds in acceleration responses were established by using the standard response spectrum technique and the finite element method. Finally, the GSA design was verified by laboratory tests - both static and dynamic. Static tests were used to calibrate displacement of the grid spacer tabs relative to the fuel rods, for the purpose of determining the minimum limit of engagement between the spacer tabs and the channel recesses. Prototypical dynamic shaking table tests were used next to determine the GSA response to sinusoidal sweep type of excitation. Handling loads were represented by shock tests consisting of one-cycle impulses at various levels of acceleration.

2. Description of the Grid Spaced Assembly

The GSA is an LMFBR oriented, advanced design which has the potential for increased breeding ratios, better accommodation of swelling and creep, lower pressure drop, improved vibrational characteristics, and lower fabrication costs. At GE, a number of grid configurations have been conceptualized and one of the designs that will be included in the initial core of the Fast Test Reactor (FTR) now being constructed near Richland, Washington is the staggered grid design.

The assembly geometric parameters are:

Number of Rods	217
Fuel Rod Length (inches)	94
Rod Pitch (inches)	0.2905
Pitch to Diameter Ratio	1.263
Inner Channel Across Hexagonal (inches)	4.335
Channel Wall Thickness (inches)	0.120
Number of Grid Spacers	26
Fuel Rod Length (inches)	94
Rod Outside Diameter (inches)	0.230
Cladding Thickness (inches)	0.015

The rod bundle is supported in the channel by a series of staggered grid spacers (Figure 4). As noted earlier, the grid spacers are held into place by grid tabs which engage in channel recesses.

The grid spacers and rod bundle are loaded into the channel at the same time by sequentially installing a grid spacer and inserting the rod bundle to engage the appropriate grid cells as each spacer is installed. Installation of the rod bundle locks the grid-spacer tabs into the channel recesses.

3. Field Test Data

To determine transport response of the GSA analytically, a reasonable estimate of acceleration time history response at the trailer bed during haulage under representative transport conditions is necessary. Data obtained from prototypical field tests, which are reported in detail elsewhere [2], is selected and similar truck transport environment is assumed in this presentation. Analytical resulcs based on finite element and response spectrum techniques are presented in the next section.

In the field tests, the shipping container was strapped in the vertical direction to the trailer floor with nylon or steel ties and was anchored to the flat plywood bulkhead. The trailer floor was weighted with sandbags because experience indicated that this arrangement would mitigate shock and vibration effects (Figure 5).

Several test runs were made under conditions which were fairly representative of the transport environment. For this purpose, a 9-mile route comprising of approximately 1 mile of Interstate Route 55 and 8 miles of Illinois Highway 83 was selected. About half the run of Highway 83 was smooth, 4-lane, divided highway and the other half was 2-lane, relatively bumpy road. Vertical, lateral, and transverse acceleration responses of the trailer floor were recorded on magnetic tape and thus, with this arrangement, readings of the accelerometers implicitly accounted for the influence of dynamic interaction effects between the shipping container, the trailer-tractor assembly, and rocking vibrations, if any.

Data obtained from these tests suggested that use of nylon tiedown straps was beneficial and was therefore adopted. Furthermore, to examine the influence of padding on the ride characteristics, two additional test runs designated test #4 and test #6 were conducted. Test #6 had no padding in contrast to test #4, in which approximately 2-inch-thick Pimcore foam rubber padding was used under the container.

In both tests, the maximum acceleration response occurred in the vertical direction. This trend is similar to observations reported elsewhere [3]. Furthermore, the differences in the records of the extreme accelerometers during the same run appeared to be negligible. This meant that the rocking effects of the trailer bed were insignificant.

A limited history of vertical accelerations recorded on the middle accelerometer, which was judged to be the most intense for both horizontal and vertical directions, is presented in Figure 6a from test #4 and Figure 6b from test #6. From these figures differences in acceleration time history, including shock effects caused by road bumps are apparent, indicating the importance of the effect, beneficial or otherwise, of using padding under the container.

4. Dynamic Analyses

4.1 Transportation Response Spectra

To estimate the maximum response of the GSDA to the kind of excitations presented in Figures 6a and 6b requires analysis that is based on either the time history response or the response spectrum approaches. Because the stiffness properties of the padding were not known precisely, the response spectrum approach was preferred because this afforded a convenient and direct estimation of the influence of change in the fundamental frequency on the maximum response to the prescribed excitation of the GSA package.

Response spectra are plots of the maximum response - displacement, velocity, or acceleration - of a simple oscillator to a prescribed excitation, plotted as functions of the natural period (or natural frequency) and damping factor of the oscillator. Response spectra are compiled by a step-wise solution of the equations of motion of the oscillator and by monitoring the maximum values of the response parameters at each step.

Motion of a viscously damped, simple oscillator subjected to the base acceleration is described by

$$m\ddot{u}_{(t)} + c\dot{u}_{(t)} + ku(t) = -ma(t) \qquad (1)$$

in which m = mass of the oscillator

c = damping coefficient

k = stiffness of the restoring elements

u, $\dot{u}$, $\ddot{u}$ are respectively time dependent displacement velocity and acceleration of the oscillator mass.

Dividing Eq. (1) by m and setting $\omega^2 = k/m$ and $c = 2\xi m\omega$, Eq. (1) can be written as

$$\ddot{u}(t) + 2\xi\omega\dot{u}(t) + \omega^2 \dot{u}(t) = -a(t) \qquad (2)$$

in which ξ = damping factor expressed as a fraction of critical damping and
ω = the natural frequency (rad/sec) of the oscillator.

Efficient computer programs are available to solve Eq. (2). Computer program SPECEQ/SPEUQ [4] was used in this presentation. Using the standard response spectrum technique and preassigned fractions - 0.5, 1, 2 and 5% - of the critical damping, truck transport response spectra were developed and are presented in Figure 7 for the GSA subjected to the trailer floor motions defined in Figures 6a and 6b, respectively. Reference 1 notes that inherent resistance of shipping containers and contents, is nearly always at least 3% of critical damping and usually more than 5%. Nevertheless, a wide range of damping values was considered because the damping properties, of the GSA package including tie-down, were not available from test data reported in Reference 2. A minimum damping value of 0.5% of the critical was considered to be the lower bound.

From these spectra, it is apparent that the maximum acceleration response is higher for those systems with undamped fundamental periods less than 0.2 second (fundamental frequency greater than 5 Hz). Container assemblies having fundamental periods between approximately 0.2 to 0.4 second are preferable because their maximum acceleration response is smaller than that for assemblies having fundamental periods outside the range of 0.2 and 0.4 second. This points to the need of selecting the most appropriate padding in the design of the shipping container.

Furthermore, from Figure 7, it is apparent that the maximum vertical acceleration response under the test environment would be much less than the design limits. Moreover, because the vertical accelerations of the trailer floor were observed [2] to be of similar form, but significantly more intense than horizontal accelerations, the maximum horizontal acceleration responses of the GSA would be well within the design limit of 2G's in the lateral and transverse directions.

4.2 Dynamic Properties of the GSA Package

To estimate the response of the GSA from the transportation design response spectra, dynamic properties (natural frequencies, mode shapes, mode participation factors) of the GSA container padding tie-down package were determined by using the finite element method.

The finite element model, which uses three dimensional beam and boundary elements, is presented in Figure 8. The properties of the beam elements were based on the sectional properties of the hexagonal channel because the fuel rods and grid spacers contributed little to the sectional properties of the assembly. For the vertical shipping mode, the foam rubber support at the base of the assembly was assumed to have a stiffness of 10^6 lb/in., laterally, and 10^6 in.-lb/rad, in rocking and torsion. The masses of the fuel rods and grid spacers were appropriately added to the mass of the channel, assuming that transverse accelerations of the fuel rods with respect to the channel are negligible. This assumption was confirmed later in dynamic tests. The modulus of elasticity and Poisson's ratio of the material were assumed 28×10^6 psi and 0.3 , respectively.

Similar to Eq. (1), the equation of motion of the package is given by

$$\underset{\sim}{M}\,\ddot{\underset{\sim}{u}} + \underset{\sim}{C}\,\dot{\underset{\sim}{u}} + \underset{\sim}{K}\,\dot{\underset{\sim}{u}} = -\underset{\sim}{M}\,\underset{\sim}{1}\,a(t) \qquad (3)$$

in which $\underline{M}$ = Mass matrix of the package
$\underline{C}$ = Damping matrix
$\underline{K}$ = Stiffness matrix
$\underline{u}$, $\underline{\dot{u}}$, $\underline{\ddot{u}}$ are respectively the vectors of displacements, velocities, accelerations of the GSA assembly relative to the trailer floor, and
a(t) is the acceleration of the trailer floor.

The natural frequencies ω_n's and mode shapes ϕ_n's are obtained from

$$\underline{K\phi n} = \omega_n^2 \underline{M}\, \underline{\phi}_n \tag{4}$$

The mode shapes satisfy orthoganality conditions

$$\underline{\phi}_m^T \underline{M}\, \underline{\phi}_n = \underline{o},\ m \neq n \text{ and } \underline{\phi}_n^T \underline{M}\, \underline{\phi}_n = M_n^*$$
$$\underline{\phi}_m^T \underline{K\phi n} = \underline{o},\ m \neq n \tag{5}$$

The nth mode participation factor γ_n is defined by

$$\gamma_n = \left(\underline{\phi}_n^T \underline{M}\, \underline{1}\right)/M_n^* \tag{6}$$

The first two natural frequencies were computed as 5.22 Hz and 51.2 Hz. Obviously these frequencies are well separated. Frequencies higher than the second were disregarded because their contribution to the response, judging from the modal participation factors and the response spectrum was negligible.

Examination of the mode shapes revealed that the first mode was primarily associated with the padding and tie-down properties, and the second mode was caused by lateral bending of the channel. The participation factor in the first mode was the highest.

The vertical shipping mode of the GSA package results in natural frequencies which are significantly smaller than the natural frequencies of the package in the horizontal shipping mode, because of the longer bearing support for the latter. As noted earlier, the maximum acceleration response is higher for systems with a fundamental frequency greater than 5 Hz. This implies that the maximum acceleration response in the horizontal shipping mode would be less than that in the vertical mode. For conservatism in design, however, the vertical shipping mode was assumed, thus allowing shipment of the GSA in either horizontal or vertical mode.

4.3 The GSA Package Response to Truck Transport

Based on the transportation design response spectra, (Figure 7), and the dynamic properties of the GSA package evaluated in the preceding section, maximum acceleration and stress responses were computed by

$$\max_t |\ddot{U}_n(t)| = \phi_n(x)\, \gamma_n\, S_a(\omega_n, \xi_n)$$
$$\max_t |U_n(t)| = \omega_n^2\, \phi_n(x)\, \gamma_n\, S_a(\omega_n, \xi_n) \tag{7}$$

in which ξ = Modal damping ratio in the nth mode (assumed = 0.01) and
$S_a(\omega_n, \xi_n)$ is the nth mode spectral value from Figure 6.

Knowing the modal maximum displacements, the finite element stresses were determined as usual.

To obtain the maximum value of the total response, the modal maxima were combined by the square root of the sum of squares (SRSS) method as follows

$$\max_{t} \mid \ddot{U}(x) \mid = \sum_{n=1}^{N} \left(\mid \max_{t} \ddot{U}_n(t) \mid \right)^2 \qquad (8)$$

As noted earlier, the natural frequencies of the assembly are well separated; therefore, the SRSS method of combining the modal maxima was justified in estimating the maximum response. For conservatism, the trailer bed acceleration response spectra (Figure 6) in the vertical direction were assumed applicable also to the lateral and transverse directions. Maximum responses of the GSA were:

Maximum acceleration	1.18G
Maximum shear force - lb	136
Maximum moment - in.-lb	6862

Apparently, the maximum acceleration responses of the GSA package in either the vertical or horizontal shipping mode are well within the maximum allowable accelerations of 6G's vertically and 2G's horizontally. Furthermore analytical results indicate that the maximum dynamic stresses during truck transport would be small enough to cause little fretting and wear.

5. Laboratory Tests

Dynamic analyses of the GSA presented in the preceding section were based on the assumption that the GSA would be shipped in either the vertical or horizontal position. However, because of substantial practical incentive to ship the GSA in the horizontal mode, static displacements of the spacer tabs relative to the channel recesses under the weight of the fuel bundle must be small so that the static weight displacement, combined with dynamic displacements, do not result in disengagement of the tabs. This was verified as follows in static and dynamic (sinusoidal sweep and shock) tests on the assembly supported in the horizontal position.

5.1 Static Tests

As noted earlier, the purpose of these tests was to verify that static displacements of the spacer tabs, relative to the recesses under the dead weight loading of the fuel bundle, are well within the minimum desirable spacer engagement limits (Figure 3).

A bench test was first performed, using a short hexagonal channel and rod bundle assembly, to determine the deflection of the rods and grid-spacer tabs relative to the channel. See Figure 9. Slots were machined in the channel section to provide access for measurement of rod and tab displacement and to further load the rod bundle between the grid spacers. This test provided the correlation that the tab displacement is approximately 30 to 50% of the rod displacement. The sag in the weighted dummy rods varied between 1 and 3 mils with the channel and rod bundle assembly in the horizontal position. The results from this test indicated that a sag of approximately 3 mils could be expected from the rods with the assembly in the horizontal position. The resulting deflection of the tabs would be less than 2 mils.

Next, the GSA was tested by lowering from the vertical to the horizontal position (Figure 10), thereby loading the grid spacers from no load to the full dead load of the dummy rods. X-rays of the test assembly in the vertical position showed that all six grid spacers, under study in the weighted or fueled section, were properly located with tabs engaged in the channel wall recesses. Measurements were made of the rod displacement before, during, and after rotation of the assembly from the vertical position to the horizontal position and back to the vertical. Micrometer readings of rod displacement were taken through pressure tap openings in the channel and proximity sensors were used to monitor rod displacement during rotation of the assembly. Rotation of the assembly to the horizontal position produced a measurable rod movement of between 0 and 4 mils. X-rays, taken with the assembly in the horizontal position, showed similar displacement of the rods along the channel. All tabs appeared to remain engaged in the channel wall. Rotation of the assembly back to the vertical position demonstrated that the rods returned to their initial position with no perceptible motion of the grid spacers in the channel. This test was repeated with the rods returning again to their initial vertical positions. This amount of movement is insignificant, relative to the nominal 31 mils of grid-spacer tab engagement in the channel recesses (Figure 3). A substantial margin of tab engagement is still provided with the assembly in the horizontal position.

5.2 Dynamic Tests

These tests involved dynamic loading of the assembly in the horizontal position (Figure 11). Because of the extremely small vibrational amplitudes predicted from dynamic analyses, two vibration measurement methods were used to measure rod displacements. The first method consisted of internally mounted fuel rod accelerometers, and the second method involved proximity sensors positioned at edge and corner rods so that the two methods could provide comparable vibration data. The proximity sensors were mounted on the channel at the same axial and radial locations as the accelerometers contained in the respective edge and corner rods. Figure 4 shows the axial location of the accelerometers and proximity sensors with respect to the grid spacers and the radial positions of the sensors in the bundle. During the dynamic tests, only select sensors were used to measure rod displacements.

The grid-spaced hydraulic test assembly was dynamically tested on a large shaker table at the GE test facility. The test assembly was instrumented with accelerometers and proximity sensors to monitor the dummy fuel rod displacement during the test. The test consisted of the following phases:

1. Axial direction frequency sweeps (1-30 Hz) at acceleration levels up to 2.5G's.
2. Transverse direction frequency sweeps at acceleration levels up to 2.0G's.
3. Dual direction frequency sweeps at acceleration levels up to 2.5G's axial and 2.0G's transverse.
4. Shock tests at accelerations up to 6.0G's axial and 2.0G's transverse.

Samples of the sensor readouts during the dynamic sweep and shock tests are shown in Figures 12 and 13. Rod displacement was measured by depth micrometer readings of fuel rod position at five locations along the assembly before, between and after shock and vibration testing of the fuel assembly. The readings indicated no change in rod position as a result of the testing. Proximity sensor readings that were taken during vibration testing to measure the rod amplitude indicated rod movement of less than 4 mils. This amplitude occurred at an apparent natural vibration frequency of 20 Hz. X-rays of the channels after completion of the testing, with the test assembly vertical, showed that the grid tabs had remained engaged in the channel recesses. The X-rays were virtually identical with X-rays taken at the same axial locations after the initial static handling test and before the dynamic phase of testing.

Both static and dynamic test results indicated that staggered grid-spaced assemblies can be shipped and handled horizontally with no limitation on shock and vibration in excess of the current specified limits for fuel rods.

6. Conclusions

From the results of dynamic analyses based on the field test data and the static and dynamic tests summarized in the preceding sections, the following conclusions may be drawn:

1. The GSA design meets the criteria regarding maximum acceleration response limits defined in Reference 1.
2. The GSA may be shipped in either horizontal or vertical mode.
3. The details of packaging - the base padding and the tie down - should be carefully considered in transportation design to mitigate shock and vibration effects.
4. Dynamic analyses, based on transportation design response spectra, which are derived from tests under representative field conditions are useful in design qualification of the GSA.

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Editor's note: The figures 1, 2, 6, 7, 8, 9, 10, 11, 12, 13 are not reproducible.

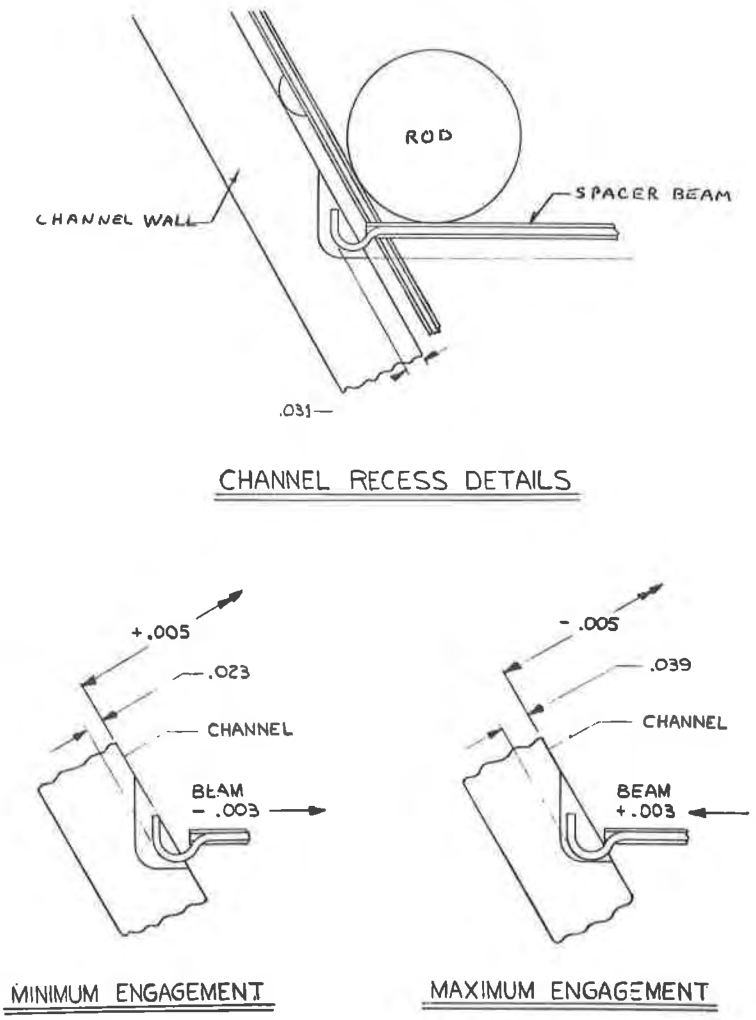


Figure 3 Details of Tab Engagement

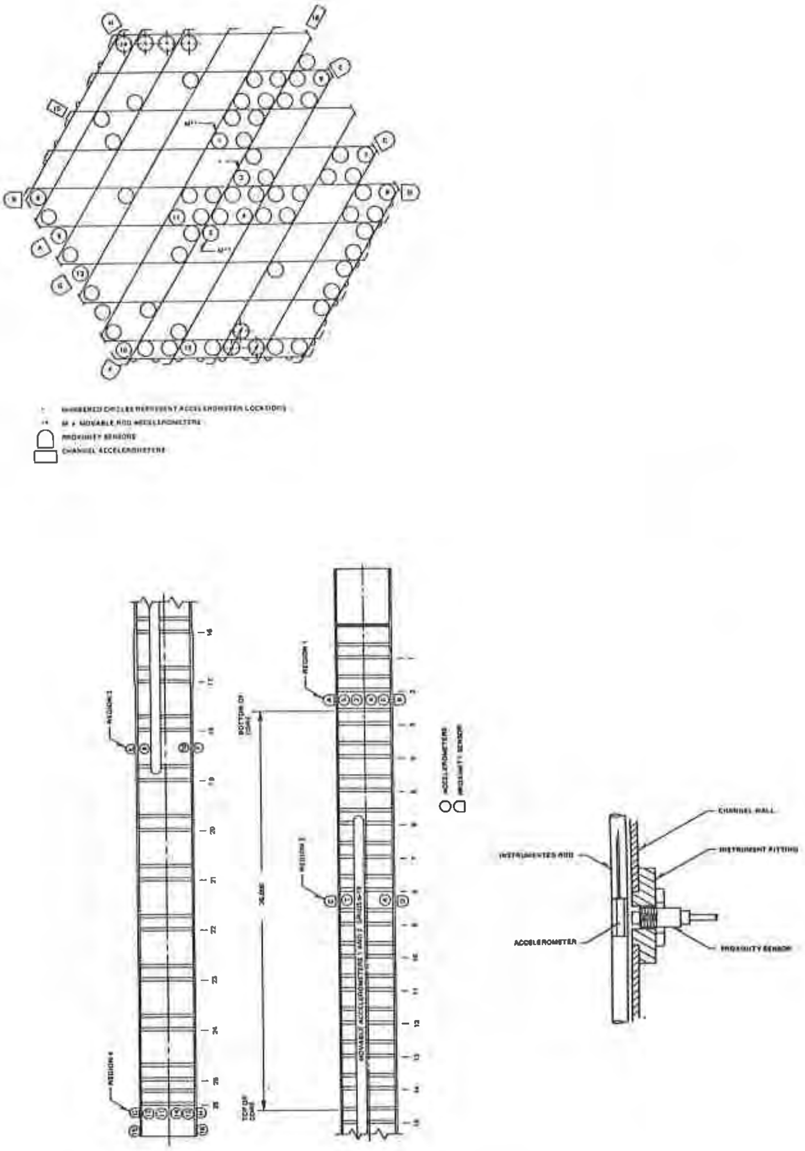
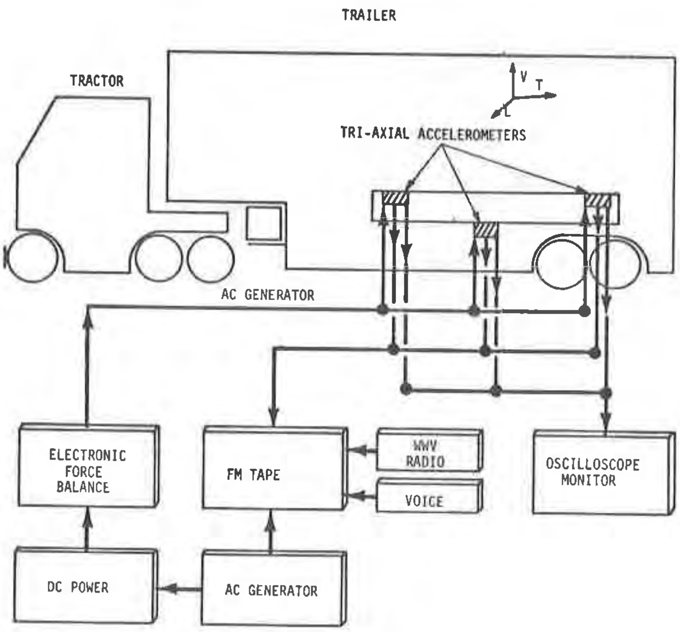


Figure 4 Arrangement of Rods and Spacers Including Instrumentation Used in Lab Tests



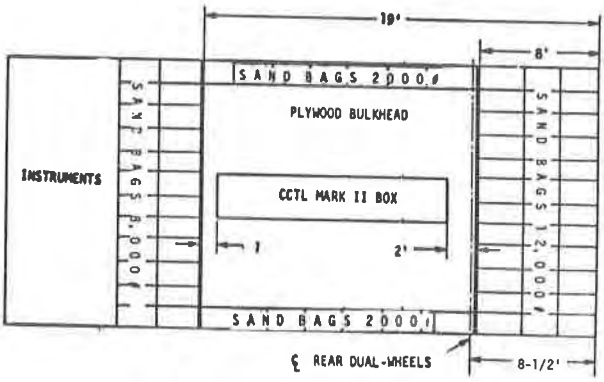


Figure 5 Instrumentation and Shipping Configuration of the Trailer Floor