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September 1964

MASTER

G-1 CORE ASSEMBLY VIBRATION TEST

(Title Unclassified)

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Astronuclear Laboratory
Westinghouse Electric Corporation

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Astronuclear

WANL-TME-819

September 1964

MASTER

(59)

SUMMARY REPORT
G-1 CORE ASSEMBLY VIBRATION TESTS
(Title Unclassified)

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1. SUMMARY

A test and analytical program was initiated in 1963 to subject a reactor of the NRX-A design to an external mechanical vibratory and transient dynamic environment. The original designation for the program was E-1 "Reactor Assembly Mechanical Tests". In order to eliminate any misunderstanding with engine numbers, the terminology was later changed to G-1. The test specification is outlined in WANL-TNR-095 (Rev.) "NRX-A Component Test Program" and described more completely in WANL-TME-378 "NRX-A Reactor Vibration and Shock Test Plan". Results of the first series of tests, called the Core Assembly Vibration Series, are described in this report.

The G-1 Core Assembly configuration, designated as ND-20101, consists of the reactor core, lateral support system, inner reflector and core support plate.

In the case of the lateral and torsional tests, sinusoidal vibratory inputs were applied directly to the inner reflector. The dome end support ring was bolted to a heavy bed plate. In the axial vibration tests, sinusoidal vibratory inputs were applied at the periphery of the core support plate with the reactor suspended in the up-firing position.

The lateral and torsional tests were performed concurrently from September 19th through November 23, 1963. After a partial disassembly for inspection and rebuild to the axial configuration, the test specimen was vibrated axially during January 1964. Over 5,500,000 cycles were applied to the Core Assembly Tests. The main results of the test has been increased knowledge and confidence in the NRX-A design.

The primary objectives of the test have been met, in that:

1. The mechanical bundling ability of the lateral support system has been verified.

2. Areas of possible design difficulty have been uncovered and have resulted in further test investigation.
3. Resonance conditions and frequencies have been determined in the axial and lateral directions.

Some of the more important detailed conclusions are:

1. The frictional force for lateral motion between the inner reflector and core was found to be about 9,000 lb.
2. The core and inner reflector were locked together by the friction below 3 G lateral vibration input. The high friction forces prevent the core from returning to its central position when displaced by vibratory motion.
3. No abrupt motion took place at initiation of relative motion between the core and inner reflector.
4. The vibration test results agree very well with analog computer studies of models which consider the core as a solid mass.
5. Resonances in the lateral vibration tests were very high damped.
6. Indention of the seal segments, caused by abrasion of the plungers, resulted in the removal of the positioning holes used for Reactor Assembly.
7. Tie rod resonances were apparent. The value of damping was very low. However, no visible damage was caused by impacting of the tie rod against the fuel elements.
8. Shim rod springs suffered permanent deformation in the axial vibration tests in the case of the more heavily loaded shim rods.
9. Titanium bolt failures, which occurred during the axial vibration tests, have resulted in reactor bolt modifications by Reactor Mechanical Design.

10. The maximum tensile and compressive stresses in the filler strips were 120 psi during the torsional tests. Maximum fuel element stresses in the lateral test were below 25 psi for tension or compression.
11. Insignificant wear was detected on pyrocup, fuel elements, filler strips, antirotation pins and support blocks.

II. INTRODUCTION

The KIWI B-4A reactor design configuration suffered extensive damage during the hot and cold flow tests. Motion pictures of the cold flow test confirm that core vibration contributed to, or was responsible for, its failure in 1962. The NRX-A2 reactor incorporated features different from the KIWI B-4A. The core and lateral support system were designed to eliminate flow induced vibration in the core. Additional components have been modified to withstand environmental vibration and shock loads. Analysis of the system cannot be readily performed due to non-linear effects and the complex configuration. Only through full-scale testing, under actual boundary conditions, can structural dynamics of such systems be studied and evaluated.

This report presents the findings of the Core Assembly Test Series. The partial assembly was vibrated in the lateral, torsional and axial directions as shown in Figures 1, 2 and 3. A bar graph, Figure 4, shows the accumulated cycles for the tests.

III. PURPOSE

The Core Assembly Test Series was initiated to systematically study the NRX-A core and inner reflector assemblies under external vibratory loads.

The test series was designed to permit application of vibratory forces directly to the core by driving the inner reflector in the lateral and torsional tests and the core support plate in the axial test. Since the outer reflector components were not used, it was possible to observe in detail the characteristics of the lateral support components and the core supporting structures and to

SUMMARY TABLE

Component	Acceleration		Significance	Frequency (CPS)	Page Referenced
	Input G	Drive Direction			
ND-20101 Config- uration	5	Lateral	No major resonance	1000 to 10	21-24
Core and Inner Reflector	3	Lateral	Breakaway	1000 to 10	23-24
Fixture and Inner Reflector	<5	Lateral	Core Isolation	> 200	24-28
Tie Rods	<5	Lateral & Torsional	Component Resonance	22	28-30
Core and Inner Reflector	<5	Torsional	No Breakaway Exhibited	500 to 10	30
Core	<5	Axial	Resonance	70	33
Dome End Support Ring & Hold Down Barrel	<5	Axial	Isolation	200	33
Core and Inner Reflector	<5	Axial	Isolation	> 200	33
Drive Fixture	<5	Axial	Resonance	500	39

FIGURE 1

Single Thruster - Lateral Test

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FIGURE 2

Dual Thruster - Lateral and Torsional Test

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FIGURE 3

Three Thrusters - Axial Test

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Cycles Run During Lateral, Torsional and Axial Tests

FIGURE 4

Total Cycles Accumulated During
G-1 Core Assembly Test Series

uncover potential problem areas. The tests were developmental in nature, serving as a prerequisite to the full assembly series in which all core boundary and support conditions are present.

IV. OBJECTIVE

The primary objective was to determine the ability of the reactor to withstand vibratory inputs. Detailed objectives of the Core Assembly Vibration Tests were as follows:

1. To observe in detail the behavior of the lateral support and core components under vibratory conditions.
2. To determine the bundling ability of the lateral support system and measure translational excursions due to the sinusoidal vibratory inputs.
3. To determine the vibratory stress levels in the fuel elements and filler strips for various levels of input vibration.

V. CONCLUSIONS

The core assembly test results have increased the knowledge and confidence in NRX-A design. The NRX-A1 cold flow test, performed in March of 1964, has shown that the flow induced vibration encountered by KIWI B-4A has been corrected. Therefore, the results of external vibratory inputs, as in the Core Assembly Series, are directly applicable.

The tests have accomplished their objectives in that they have:

1. Developed a confidence in the integrity of graphite components as designed in the NRX-A configuration.
2. Determined the mechanical bundling ability of the lateral support system.
3. Determined resonance conditions and frequencies of concern for the axial and lateral directions.
4. Determined areas of concern which require further detailed investigations.

VI. RECOMMENDATIONS

During the vibration tests performed on the ND-20101 configuration, the lateral support system was found to have a much higher friction force for lateral motion than was originally calculated in WANL-TME-358 and WANL-TME-365. This large value of friction force prevents the lateral support system from acting as a vibration isolation system for vibratory inputs below 3 G. The tests did indicate, however, that this isolation function may not be required due to the low core stresses measured. However, it is important to determine the optimum lateral support pressure necessary to mechanically bundle the core under vibratory inputs and still have low enough pressures so that inter-element friction does not prevent free axial growth of the core. It is recommended that analytical studies to determine an optimum mechanical bundling pressure and lateral support friction values be initiated. These would be coupled with the results of the core vibration experiments and the core axial friction and lockup experiments being run.

The problem of determining representative conditions of vibration for reactor tests requires greater effort than has been provided in the past year. Without accurate information on vibratory conditions on the reactor pressure vessel, test stand and on internal reactor parts, no progress can be made in determining reactor dynamic environments during flow and hot tests. It is, therefore, recommended that future reactor tests be instrumented more fully with respect to dynamic data; that the data system provide reasonable data accuracy up to at least 2,000 cps, and that a program be actively undertaken to arrive at test specifications from which new tests can be run to prove future designs. Engineering Mechanics and Reactor Design are now evaluating NRX-A1 and KIWI data towards determining reactor dynamic environments.

For future tests, it is recommended that the test reactor be subjected to higher vibration levels and for a longer time duration than used thus far. The design margin of strength, under dynamic conditions, will then be obtained.

Further, it is recommended that serious consideration be given to random vibration tests for future reactor tests, to more closely simulate the environments measured in the NRX-A1 cold flow and KIWI B4D hot test.

Since the tie rod is a critical member, work will be continued in determining vibratory conditions, fatigue life and damper design for the tie rod.

In order to make certain that the design margins in the critical area of bolted flange designs are sufficient, additional analysis and further fatigue tests work should be required.

Further investigation of the significance of graphite dust generation caused by graphite components is indicated.

VII. TEST SPECIMEN

The core assembly tests were performed utilizing a reactor simulating the NRX-A design in all important respects with the exception of the core band. The test reactor consisted of a core, inner reflector, NRX-A type lateral support system, support plate, dome end support and test fixtures. These tests did not utilize the outer reflector assembly, aluminum barrel or pressure vessel.

This type of core configuration made it possible to apply the vibration excitation directly to the core components which were believed to be the most susceptible components of the reactor to vibration damage. Stroboscopic observation and high speed photographs of lateral support components and the nozzle end of core were obtained.

The unimpregnated inner reflector was modified by adding 290 one-inch diameter observation holes. These holes were drilled radially at selected locations around the periphery for vibration probes and visual observations of the seal segments and filler strips. An additional 24 instrumentation holes for displacement transducers and longitudinal wire grooves running the length of the barrel completed the modifications.

The core was not banded and consisted of uncoated and unfueled components. Fourteen filler strips and 27 fuel elements were modified to receive the instrumentation. Spiral liners were not used in the central fuel elements of the instrumented clusters. Liner tubes were used in 17 clusters of sector one. Tie rods were made of stainless steel, rather than Inconel, with brazed on heads and threaded ends.

The dome end support ring was a prototype unit with the exception of a thinner wall thickness than the latest design. No flow holes were drilled in the flanges. A stiffening ring fixture was used to simulate the effect of the outer reflector on the dome end support.

All tests in this series were designed to use a flat ring in lieu of the conical support ring to expose the tie rod top hardware assembly between the support plate and core for visual observation. With this configuration, the axial springs were also exposed for visual observation during the tests.

The axial springs used during the lateral and torsional tests were shortened so that the actual preloaded position of the inner reflector would be simulated. The inner reflector, not being preloaded against the axial springs, had a tendency to unseat from its normal operating position. Hold down cables were installed between the lower drive beam and the top flange of the dome end support at $\theta = 160, 180$ and 200 degrees. This modification was successful in maintaining contact between inner reflector and the axial springs. For the axial test, prototype springs were used and a preload was applied to the inner reflector by a lightweight aluminum barrel and nozzle ring fixture.

The configuration used for the lateral and torsional tests, was designated as ND-20101, Build 1, and defined by Drawing No. 709J984 and List of Parts No. 709J702. The assembly for the axial tests, was designated as ND-20101, Build 2, and defined by Drawing No. 710J096 and List of Parts No. 710J276.

The major subassemblies for each build are tabulated below:

ND 20101 BUILD 1

Assembly	709J984
Cylinder (Inner Reflector)	709J909 G01
Core Sub-Assembly	709J986 G01
Support Assembly, Dome End	709J508 H01

ND 20101 BUILD 2

Assembly	710J096
All sub-assemblies and items are the same as ND-20101 Build 1 with the additional -	
Barrel and Nozzle Ring	NT 700106

VIII TEST EQUIPMENT

1. Fixtures

Three methods were used for coupling the reactor to the electrodynamic shakers. Vibration fixtures for the first method consisted of three circumferential bands, located at the nozzle, midpoint and dome ends of the inner reflector. A longitudinal beam joined the three bands at $\theta = 0$ degrees, providing a distribution of force into the inner reflector. One 10,000 lb vibration exciter was attached to the midpoint of the longitudinal beam. The direction of force passed through the approximate center of gravity of the core assembly. A second fixture was designed because the load was not well distributed to the end bands on the inner reflector.

The second method utilized a drive beam with two circumferential bands spaced 16 inches apart. The arrangement accommodated one 10,000 lb exciter with the direction of force through the approximate center of gravity of the core. Figure 1 shows the test setup. For the lateral test series, this configuration proved satisfactory. However, additional excitation force was necessary to cause relative motion between the core and inner reflector due to a higher than calculated friction.

The third method applied vibration excitation to the reactor with two 10,000 lb exciters driving the reactor tangentially at $\theta = 90$ degrees and $\theta = 270$ degrees (see Figure 5). The vibratory loads were transmitted to the reactor by two longitudinal drive beams at $\theta = 90$ degrees and $\theta = 270$ degrees.

Torsional tests of the reactor were conducted using the same two 10,000 lb exciters and fixtures as shown in Figure 5. The exciters

FIGURE 5

Lateral and Torsional Setup Drawing

were electrically driven 180 degrees out-of-phase in order to apply pure torsional vibratory loading on the inner reflector of the reactor. For the lateral and torsional tests, the reactor was clamped to a bedplate by means of a split ring attached to the dome end support.

In the axial test series, vibratory loads were applied using three 10,000 lb exciters driving the periphery of the core support plate through a conical drive fixture. During this series of tests, the ND-201 reactor was suspended by cables through four low frequency isolation mounts attached to the support structure. Since natural frequency of the mounts was about 2.5 cps, the cable support system did not affect the vibration characteristics of the reactor. Figure 6 shows the general axial test setup.

IX RESULTS

1. Inspection

Two major inspections of the reactor were made during the Core Assembly Vibration Series. One was performed at the conclusion of the lateral and torsional test series. The other was made after the axial test series.

During the lateral, torsional and axial tests, the screws that fastened the leaf springs to the inner reflector were periodically checked to determine if they had become loose during the run. They remained properly torqued throughout the duration of the core assembly test series. All other bolts, screws and locking devices also remained properly torqued throughout the test series.

The wearing of the seal segments under the plungers was observed during the core assembly test series. (Figure 7.)

FIGURE 6

Axial Test Setup Drawing

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FIGURE 7

Typical Seal Segment Wear

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The majority of the wear occurred during lateral vibration at levels from 3 G to 5 G. During the lateral vibration test, graphite dust was observed at the seal plungers. (Figure 8.) However, the amount of accumulated dust indicated that it was probably due to residual graphite particles left in the hole of the inner reflector and on the surface of the plungers after machining which worked its way out when relative motion between the plunger and inner reflector occurred.

The other area of concern in the lateral test was the slight chipping of the support blocks at the core periphery near the fuel element interface. This chipping was believed to be associated with the assembly and disassembly handling.

Inspection of the reactor after the lateral test revealed that 28 of the 48 titanium bolts used to attach the core support ring had either fractured or yielded. The location of these bolts at $\theta = 0$ degrees and 180 degrees indicated that a rocking mode about an axis perpendicular to the drive direction may have caused the failure. Furthermore, the torques on the remaining bolts measured less than the value required at assembly. An examination disclosed that a sharp corner on the simulated support ring interfered with the corner fillet on the core support plate. The removal of this interference was accomplished before the axial test series. The axial tests were run with no failure occurring to the bolts in this bolt circle.

After the axial tests, graphite dust was observed in the vicinity of the pyrocups and at the surface of the fuel elements (Figure 9). Insignificant wear was also detected at the seal plungers, pyrocups, fuel element, filler strips, antirotation pins in the filler strips and

FIGURE 8

Dust Accumulation at Seal Plungers

FIGURE 9

Graphite Dust Accumulation on Support Blocks

inner reflector seal segment grooves. Inspection revealed that these and adjacent components were still within drawing tolerances.

At the conclusion of the axial test series, 19 clusters were completely disassembled and inspected. All cluster plates exhibited a peening from the shim spring retainers. Two of the four shim springs had a permanent set due to excessive loading from the heavier shim rods (Figure 10). In some cases, the springs supporting the lighter shim rods also yielded. The ends of the shim rods adjacent to the springs were rounded, but had not chipped. Since no chips were found and the expected acceleration levels for the NRX-A1 reactor test were below 5 G, these observations were of no real concern.

2. Lateral Test Quantitative Results -

In the lateral test configuration, sweep and dwell tests were performed in the frequency range of 10 - 1000 cps. The maximum input level, as measured on the inner reflector, was 5 G. No major component resonances were noted.

Input levels referred to in the body of this report, and discussed in Appendix B and E, are the magnitude of the control accelerometer after filtering. By control is meant servo control provided by the vibration exciter on one accelerometer identified as the control point. The effect of a filtered and unfiltered waveform is important in evaluation of the data. Unfiltered data has higher harmonics caused by transients, distortion and non-linearities in the system, superimposed on the fundamental at certain of the test frequencies. The level of the fundamental component (first harmonic in the Fourier analysis) will have a somewhat lower value than the peak value of the unfiltered data at these frequencies. The total acceleration is recorded, although

FIGURE 10

Cluster Hardware Inspected at
Completion of Core Assembly Test Series

control is on the fundamental. A plot of peak values of the input acceleration vs. frequency will therefore vary somewhat from a constant level over the frequency range. Values of single peak magnitude vs. frequency are plotted in all X-Y plots presented in this report, since this is the characteristic operation of the plotter system used.

It is necessary to know the waveshape in order to fully understand the behavior of the non-linear system. The ND-20101 reactor is damped by coulomb friction, which produces characteristic acceleration traces which can be determined analytically. Knowing this response, additional system non-linearities caused by rattling and individual component resonances can be understood and separated from the data.

An input level of at least 3 G is necessary to cause relative motion between the core and inner reflector. This condition is generally referred to as breakaway. Damping forces were found to be of the coulomb friction or dry sliding type. The magnitude of this damping force can be calculated as:

$$\begin{aligned}\text{Friction Force} &= (\text{Weight of Core}) \times (\text{breakaway G level}) \\ &= (3000 \text{ lb}) \times (3) \\ &= 9000 \text{ lb}\end{aligned}$$

This high value was not originally expected, but the latest calculations utilizing all the friction forces existing on plungers and seals, with the latest "V" groove seal design, confirm this higher value.

Stroboscopic and high speed motion picture observations indicate that the seal segment plunger combination followed the core

at all input conditions. Relative motion or slipping usually occurred between the seal segments and filler strips. In some instances, slipping occurred between the plungers and seal segments. Typical wave-shape samples are shown in Figure 11.

Typical inner reflector accelerations filtered and unfiltered are shown in Figure 12. Comparisons of waveshapes with computer models are as a single mass. The system response to a sinusoidal force input is therefore linear and the measured accelerations are sinusoidal. The relative displacement transducers show no response. When the input level is increased above 3 G, friction is broken and the system behaves in a non-linear manner. The acceleration traces become typical for a system damped by coulomb friction as can be seen by comparison with a lumped mass analog computer representation. The peak value of the first harmonic (filtered wave) of the non-linear acceleration is approximately 70% of the unfiltered wave. The relative motion between the core and inner reflector exhibits stops which are also typical for this system. A more detailed discussion of coulomb damping and its effect on a vibration system can be found in WANL-TME-389 "Analog Computer - B-5 Tests" and in Appendix E.

The high friction damping prevented any major system resonance buildup. This can be seen in Figure 13, where a plot of input impedance is shown. This plot indicates no resonances at lower frequencies. At frequencies higher than 200 cps, fixture and inner reflector flexibility isolate the core. The impedance plot also shows that below 100 cps, at the input levels of this test, the effective mass of the core and inner reflector remains constant 4,000 lb. A plot of core

FIGURE II

Typical Relative Displacement Between Core
 and Inner Reflector - 30 cps Dwell Series

FIGURE 12

Typical Control Accelerometer Output
 30 cps Dwell Series

FIGURE 13

Input Impedance for the Lateral Test with 5 G Input, $Z = \frac{\text{Input Force, lb.}}{\text{Input Velocity, in./sec.}}$

acceleration as measured on the filler strips and support block is shown in Figure 14. In the lateral test series, a 250 cps harmonic was observed on the support block acceleration trace.

No significant stress was measured in the fuel elements or filler strips. The stresses which were measured never exceeded 25 psi for all input conditions. No trends in the stress patterns which would indicate fuel element bending or separation are apparent.

The bundling forces on the core were sufficient to prevent any significant cluster separation or bending as evidenced by the low core component stress at all input levels and by the displacement triads on the cluster support blocks. Some support block rattling was noticed at lower frequencies. This occurrence can be attributed to some rocking motion of the core causing a loss of tie rod preload.

The lateral force excitation was applied through drive bands wrapped about the inner reflector as shown in Figure 5. The highest measured stress on the inner reflector of 200 psi occurred under these bands.

The tie rods were the only core components which exhibited a definite resonance. Vibration persists at a frequency of 22 cps as indicated from strain gage data, for more than a minute after the exciters are shut down. This resonance is in agreement with independent tie rod testing data which has determined that a preload force of 40 lb will produce a 25 cps first mode of vibration. The log decrement was calculated to be 0.00568. Prior to shutdown, peaks were noted on the tie rod stress traces indicating that the tie rods were hitting

A. Typical Support Block Acceleration - Lateral Series

B. Typical Filler Strip Acceleration - Lateral Series

FIGURE 14

Typical Radial Acceleration in Direction of
Drive During Lateral Test Series

against the sides of the fuel elements. No fuel element damage has been found as a result of this impact. The elements in which these instrumented tie rods are located are without liner tubes. The spiral liner tubes would further restrain motion. Typical tie rod strains are shown in Figure 15.

3. Torsional Quantitative Results

In the torsional test configuration, sweep and dwell tests in the frequency range of 10 - 500 cps were performed.

The highest level vibration input measured tangentially on the inner reflector was 5 G. At this input level, friction was not broken between the core and inner reflector. This can be seen from Figure 16 showing a comparison of inner reflector and core accelerations. The core acceleration was measured tangentially on a peripheral support block. The accelerations remain equal in magnitude with the same waveshape at all input levels.

The tie rod motions were similar to those reported in the previous section covering the lateral test as shown in Figure 17.

Maximum tensile and compressive stresses in the fuel element was found to be 25 psi. Maximum stresses in the filler strips, occurring at low frequency, were found to be 120 psi for tension and compression.

4. Axial Test Quantitative Results

In the axial test configuration sweep and dwell tests were performed in the frequency range of 10 - 2000 cps. The maximum input (filtered) as measured on the core support plate was 5 G.

Major system resonances were noted at 70 and 160 cps. Resonance is defined here as a relative maximum in a transducer

FIGURE 15
TYPICAL TIE ROD STRAIN OUTPUTS RECORDED AFTER
THRUSTER SHUTDOWN

FIGURE 16

Waveshape Comparison of Inner Reflector
and Core Acceleration Traces
Torsional Test at 50 cps

response with the input level held constant. Resonances as defined here will occur near the natural frequency of individual components within the system. The input impedance will show a relative maximum at these frequencies.

A plot showing input impedance for a 4 G sweep is shown in Figure 18. Sweep tests at other input levels exhibit the same maximums. Plots of core and inner reflector acceleration are shown in Figure 19.

Above 200 cps, the core and inner reflector are isolated. At 160 cps the dome end support ring - hold down barrel combination, moving together as a mass, go into resonance. The components are pictured in Figure 20. The spring restraint causing this resonance is a result of bending across the simulated support ring and the nozzle preload ring. No full reactor resonance condition can be associated at this time with this occurrence as full assembly boundary conditions are not duplicated.

At 70 cps the dome end support ring - hold down barrel combination moves with the input. The core mass restrained by the tie rod stiffness goes into resonance as shown in Figure 21. The tie rod strain (Figure 17) also shows a peak at this frequency. The inner boundary conditions with the inner reflector preloaded against the pressure vessel nozzle ring, thus restraining its motion, this resonance should be more highly damped by coulomb friction.

In the frequency range of 90 - 150 cps and 10 - 50 cps, the core and inner reflector move together, locked by coulomb friction.

(a) TYPICAL TIE ROD STRAIN~TORSIONAL SERIES

(b) TYPICAL TIE ROD STRAIN~AXIAL SERIES

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FIGURE 17

Typical Tie Rod Strains

(a) TYPICAL IMPEDANCE PLOT - AXIAL SERIES

(b) TYPICAL THRUSTER DRIVE FORCE - AXIAL SERIES

(c) TYPICAL CONTROL ACCELEROMETER UNFILTERED OUTPUT - AXIAL SERIES

597305B

FIGURE 18

Typical Drive Characteristics

(a) TYPICAL SUPPORT BLOCK ACCELERATION~AXIAL SERIES

(b) TYPICAL INNER REFLECTOR ACCELERATION~AXIAL SERIES

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FIGURE 19

Typical Core and Inner Reflector Accelerations

FIGURE 20

Area of Simulation Where Resonance Occurred

FIGURE 21

Core Resonance Curve
(Filtered Values)

As the frequency approaches 10 cps, the response of the entire system approaches the input. A resonance exists at 500 cps. This resonance has been attributed to the drive cone.

During the sweep test series, two silver plated titanium bolts failed. These bolts attached the simulated support ring to the outer reflector support ring. Instrumented bolts were installed in order to measure induced stress levels.

An alternating stress superimposed on the preload stress was measured in the bolts. With an input of 4 G, 7,500 psi single peak alternating axial stress was measured at 160 cps. The stress at 85 cps was 5,000 psi. In addition to the resonance points, a 4,000 psi stress was also measured through the frequency range of 10 - 50 cps. Maximum bending stress in the bolts is of the same order of magnitude at the axial stress.

Theoretically, the bolt stress is equal to the initial tension in the bolt produced by tightening. The applied external vibratory load has no effect until this load is exceeded. The inertia loading across the bolt flange as calculated from the known component accelerations, did not exceed the preload force. The excess bolt stress is therefore caused by some combination of bending and leverage effect across the flange.

After disassembly for inspection, a static test was performed using both the actual and simulated core support ring. The simulated ring is more flexible than the actual core support cone. This flexibility produces a more flexible connection and consequently more bending across the bolt circle.

Some response peaks and higher harmonics in addition to those discussed, can be observed from the data. These peaks or resonances could be significant or could be occurring as a result of fixtures and simulated parts.

The tie rod and fuel element strain plots, Figures 17 and 22 show resonant responses at 1600 cps. A fifth harmonic is present on the core support acceleration at 160 cps (Figure 23). All resonances noted during the core assembly series will be investigated in the Reactor Assembly Vibration Series which will be conducted under exact boundary conditions.

X. APPENDICES

1. Appendix A - Terminology

The following definitions are consistent with the terminology used during the G-1 Core Assembly Series 1 through 34.

Series - Because of the limited number of channels available in the data acquisition system, it was necessary to select a few transducers at any one time. Each set of transducers were considered as a Series.

Test - Sweep and dwell tests were performed with the shakers positioned for driving the reactor in the lateral, torsional and axial directions.

Runs - During the dwell runs, a constant frequency was held while the acceleration was varied in incremental steps. The variables were reversed in the sweep runs.

(a) TYPICAL FUEL ELEMENT COMPRESSION STRAIN~AXIAL SERIES

(b) TYPICAL FUEL ELEMENT BENDING STRAIN~AXIAL SERIES

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FIGURE 22

Typical Fuel Element Strains

FIGURE 23

Unfiltered Acceleration Waveshape Showing the Presence
 of a Fifth Harmonic at the Center of the Support Plate
 During 160 cps Dwell

The accelerations were held constant while the frequency was varied automatically. A cycling oscillator was used to program the log frequency at a constant scanning rate. The time duration for the sweep runs were approximately 7-1/2 minutes. Dwell time varied depending on motions observed. In general, changing incremental acceleration levels took from 5 to 10 seconds. Increments were held for 15 to 45 seconds.

Lateral excitation was applied through the frequency spectrum from 5 cps to 500 cps and to 5 G acceleration levels. For the axial test, the frequency range was increased to 2,000 cps.

2. Appendix B - Thruster Control

The controlling accelerometer was initially mounted on the drive beam during the lateral and torsional tests. In order to minimize the effect of the drive fixture, the accelerometer was moved to the inner reflector for the final group of tests. All data cited in this report refers to the accelerometer on the inner reflector. For the axial test, the accelerometer was attached to the center of the support plate. The signal from the accelerometer was fed through an impedance matching acceleration amplifier and into the servo amplifier and monitor chassis. These signals control the gain of the control amplifier so that the acceleration level or displacement remains constant, regardless of the frequency. The control amplifier responds to average level, rather than peak or RMS.

The input control accelerometer was filtered using a 20 cps bandwidth tracking filter. The resulting fundamental sine wave was fed into

a B&K controller which reads an average value. This average value was converted to a peak reading which was the input filtered level. The implications of filtering are discussed briefly in Section IX, second paragraph, and in Appendix E.

3. Appendix C - Transducer

Detailed descriptions of transducers used during the Core Assembly Series were reported in WANL-TME-377. These transducers monitored strains, relative displacement and accelerations. Data was recorded at 233 out of a possible 365 transducer location. A limited number of strain gages sampled indicated very low stresses in filler strips, fuel elements and inner reflector. Therefore, numerous strain gage locations were considered redundant and were not evaluated.

As the program developed, the most useful numerical data was gained from the accelerometers mounted on the support plate, inner reflector, support blocks, drive beams and by means of a hand probe accelerometer. The significant strain gage outputs were monitored on the drive rods and cluster tie rods.

4. Appendix D - Data Handling

Raw data was initially transcribed on oscillograph, pen and tape recorders. Outputs from significant transducers were recorded on the pen recorder and oscillograph for viewing as the test progressed. The data stored on tape was replayed on oscillograph and through an X-Y plotter at the completion of the test series. The X-Y plotter was used to develop impedance and amplitude curves. The X axis drive was controlled by the logarithmic value of frequency. The Y axis represented either drive impedance or a peak sensor output.

A manual reduction of dwell data was used. Plots were made for each sensor which show the peak values reached during the dwell tests. The curves indicate linearity and the amount of attenuation or amplification. A portion of the oscillograph was sampled for each point plotted. Variations in waveshapes, due to noise and harmonics, were applied.

5. Appendix E - Analog Computer Studies

Lumped mass models representing the core assembly configuration have been run on the analog computer. These studies have been used to predict and understand test results.

The equations of motions are:

$$M_1 \ddot{X}_1 + K_1(X_1 - X_2) \pm F \cos \omega t$$

$$M_2 \ddot{X}_2 + K_1(X_2 - X_1) + K_t X_2 \mp P = 0$$

where	K_1	- lateral support spring	=	44.6×10^3 lb/in.
	P	- damping force	=	8200 lb
	K_t	- tie rod in bending	=	4×10^3 lb/in.
	M_1	- inner reflector weight	=	$1.9 \text{ lb-sec.}^2/\text{in} = 750 \text{ lb}$
	M_2	- core weight	=	$8 \text{ lb-sec.}^2/\text{in.} = 3100 \text{ lb}$

Damping was found experimentally to be approximately 9000 lb.

Plots of core acceleration $\ddot{X}_2$ and relative motion $(X_2 - X_1)$ are shown in Figures 24 and 25. The response is non-linear, typical of a system damped by coulomb friction. Abrupt changes occur in the acceleration traces when the friction force changes sign. Stop motion is present for inputs to at least 7 G. The core response is highly damped exhibiting no resonances at the input levels shown. The core acceleration approaches P/M_2 (2.6 G) at higher frequencies.

Coulomb damping force is constant in magnitude, not proportional to velocity as is viscous damping. As the magnitude of the input is increased, the damping ratio, which for this non-linear system can be defined as P/KX_1 , decreases. Thus, at higher input levels, dynamic resonances will become more apparent.

Vibration time response curves of the core assembly test are non-sinusoidal. One cause of this non-linearity is coulomb damping. Filtering the first harmonic from these non-linear waves using standard filtering techniques cannot always be used to determine response magnitudes (the first harmonic of a non-linear wave, as determined by Fourier analysis, does not necessarily have the same peak value as the wave). These non-sinusoidal curves must, therefore, be considered

FIGURE 24

Core Acceleration

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FIGURE 25

Relative Motion for Constant Acceleration Input

as the fundamental response of the system and must be filtered properly from the data. Distortions to these curves caused by rattling or local and fixture resonances can be more readily traced in this manner.

A narrow band tracking filter was driven by the analog computer in order to investigate this problem. A typical result is shown in Figure 26. It was found that the first harmonic of a typical lateral acceleration wave is approximately 70% of the actual peak value. Corresponding displacements have the same filtered as unfiltered peak values. Test acceleration outputs must, therefore, be filtered with lower harmonics included in the output if nonlinearities, other than those caused by coulomb friction, appear.

The need for this procedure can be seen from acceleration data from the lateral test with the outer leaf of each spring removed. Accelerometer outputs on the nozzle end of the inner reflector and on the 4C1 core support block measured in the direction of drive are shown in Figure 27. The input frequency is 23 cps. The core support block acceleration unfiltered, filtered and from the computer is shown. High cutoff frequency on the band pass filter was 130 cps (first 5 harmonics). Inner reflector acceleration is shown unfiltered along with the corresponding computer traces.

The model run for the axial test is shown below:

FIGURE 26

Typical Tracking Filter Results

FIGURE 27

Comparison of Test and Computer Curves

The equations of motion are:

$$M_0 \ddot{X}_0 + K_A(X_0 - X_1) + K_P(X_0 - X_2) + K_R(X_0 - X_3) = F \cos \omega t$$

$$M_1 \ddot{X}_1 + K_A(X_1 - X_0) \pm P = 0$$

$$M_2 \ddot{X}_2 + K_P(X_2 - X_0) + K_N(X_2 - X_3) \mp P = 0$$

$$M_3 \ddot{X}_3 + K_R(X_3 - X_0) + K_N(X_3 - X_2) = 0$$

where: K_A	- tie rod stiffness	- 1.44×10^6 lb/in.
K_P	- inner reflector preload spring	- 16.8×10^3 lb/in.
K_R	- simulated core support ring	- 0.9×10^6 lb/in.
K_N	- nozzle preload ring	- 0.4×10^6 lb/in.
M_0	- core support plate	- 1 lb-sec. ² /in.
M_1	- core	- 8 lb-sec. ² /in.
M_2	- inner reflector	- 1.94 lb-sec. ² /in.
M_3	- preload barrel + dome end support	- 1.34 lb-sec. ² /in.
P	- friction between filler strips and core	- 4000 lb

After teardown K_R was measured to be 1.6×10^6 lb/in. compared to the prototype support ring which has a stiffness of 20×10^6 lb/in. Both K_R and K_N remain linear until the inner reflector preload force across them is exceeded.

Acceleration response curves are shown in Figure 28 for a 2 G input. A core anti-resonance occurs at 70 cps due to the tie rod stiffness.

$$f_n = \frac{1}{2\pi} \sqrt{\frac{K_A}{M_1}} = \frac{1}{2\pi} \sqrt{\frac{1.44 \times 10^6}{8}} = 67.5 \text{ cps}$$

FIGURE 28

Acceleration Response Curves

The actual mass consists of both the core and inner reflector combined as friction is not broken at this input level.

An anti-resonance occurs near 160 cps due to the simulated core support ring stiffness acting on the preload barrel - dome end support mass.

$$\frac{1}{2\pi} \sqrt{\frac{K_R + K_N}{M_3}} \approx \frac{1}{2\pi} \sqrt{\frac{1.6 \times 10^6}{1.34}} \approx 173 \text{ cps}$$

The computer model shows resonances at the same frequencies as the actual system. The magnification factors are greater than those found in the test as no damping was placed across springs K_N , K_R and K_P while some damping will exist in the physical system. Friction is not broken at this input level and the system response is linear.

6. Appendix F - Bibliography

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