STRUCTURAL ANALYSIS OF Ka-BAND GIMBALED ANTENNAS FOR A COMMUNICATIONS SATELLITE SYSTEM

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ABSTRACT

This paper presents the FE modeling, structural analysis and test validation, of the Ka-band gimbaled antennas developed by COM DEV Ltd. for a satellite communications system. The modeling methods and dynamic stress analysis approaches are highlighted in order to achieve a compromise of the technical accuracy, computational cost and effectiveness requirements. The antenna launch-lock mechanisms with small clearance are modeled as equivalent linear springs (CELAS2's) by using the iteration technique. A linearized and locally remeshed assembly model is then effectively used to perform dynamic and stress analyses, by employing the MSC/NASTRAN programs. It is shown that the analysis results of the nonlinear gimbaled antennas, in terms of major modal frequencies, sine and random acceleration response, correlate the measured qualification test data very well.

Key words: Gimbaled antenna structures, nonlinear mechanism, equivalent linearisation, normal modes and frequency response analysis, component's stresses, and test validation.

1. INTRODUCTION

Two types of Ka-band two-axis gimbaled antennas have been developed by COM DEV Ltd., Canada, for a constellation of a low-earth-orbit (LEO) commercial communications satellite system. The crosslink moveable antennas (XMA) transmit messages between east/west adjacent satellites; while gateway moveable antennas (GMA) link the satellite to up to 4 ground based gateway stations. The antennas are driven by an elevation over azimuth gimbal system which employs two rotary actuators.

In order for the gimbaled antenna structures to survive the excessive launch loads, launch lock mechanisms are employed to restrain motion about the azimuth and elevation axes during the launch. Once the satellite is placed in the space orbit the launch locks will then be released to deploy the antennas. Paraffin actuator is selected for the launch lock application due to its simplicity, reliability and low cast for volume production. The launch lock mechanism has a small clearance between the pin and the bushing, for an ease and reliable release of the antenna in the space. The gimbaled antennas in locked position are also designed to meet stiffness requirement in order to be dynamically decoupled from the major modes of the space vehicle.

The two-axis gimbaled antennas are clearly nonlinear structures with a small gap at launch lock locations. For design and test qualification, the launch environments for the antennas are specified in terms of quasi-static, base excited sine vibration, base excited random vibration, and acoustic loadings.

It can be shown that the existing analytical techniques for solving nonlinear structure systems subjected to stochastic excitations all have certain limitations in application to the above mentioned nonlinear gimbaled structures [1,2]. For finite element analysis, nonlinear structural problems, with either material or geometric nonlinearity, can be solved for the static and the dynamic transient analyses, with properly selected iterative and incremental methods [3]. On the other hand,well established linear finite element tools are very powerful and cost effective in dealing with large and complex structural dynamic problems, such as, for example, MSC/NASTRAN program for normal modal analysis, frequency domain response and stochastic excitations [4,6].

In this paper, the FE modeling methods and the dynamic and stress analysis approaches are highlighted with respect to a compromise of the technical accuracy, computational cost and effectiveness requirements. The antenna launch-lock mechanisms with small clearance are modeled as equivalent linear springs (CELAS2's) by using the iteration technique. The MSC/NASTRAN program is then effectively used to perform dynamic analysis of the linearised antenna assembly models.

The analysis results of the nonlinear gimbaled antennas, in terms of major modal frequencies, sine and random acceleration response, are compared with those of the measured qualification test data, in order to validate the proposed equivalent linear models and the analysis results.

2. GIMBALED ANTENNA MODELS

One of the gimbaled antennas in stowed (locked) configuration is presented in Fig.1. Two launch locks are employed in the antenna to restrain the antenna motion about both azimuth and elevation axes. The launch locks provide additional load paths from the gimbaled antenna to the support frame and thus increase the stiffness of the antenna structure.

At early stage of the development of the gimbaled antennas, the antenna was first modeled as a linear finite element structure model by neglecting the small clearance between the pin and the bushing of the launch locks. The initial low level vibration survey test of the engineering model (EM) indicated that the resonant frequencies of the antenna model were shifting with respect to vibration excitation levels, and the stiffness of the antenna assembly was lower than that predicated by using the simple linear model. It is then identified that the gimbaled antenna structure is very sensitive to the small clearance of the launch lock. The influence of the clearance on the antenna stiffness and the dynamic response can not be neglected.

In order to utilize the powerful linear dynamic analysis tool for the stiffness driven antenna structure design, the gimbaled antennas with a small gap at launch lock locations are modeled as an equivalent linearized models. Section 3 will present the linearization method used to obtain the equivalent linear springs which approximate the dynamic characteristics of the gap elements.

The normal modes analysis (SOL103) is then employed for the stiffness driven structural design. For dynamic stress analysis of the components, it usually requires large a mount of work involving the worst case analysis of the interface loads for each components concerned, the processing and transforming dynamic loads to component models, and the component model runs. In this project, a different approach was taken for dynamic stress analysis. A locally re-meshed antenna assembly model is constructed based on the strain energy intensity information from the normal modes analysis. The procedure and criteria will be dressed in section 4.

A basic finite element model of the gimbaled antenna is shown in Fig.2.

3. MODEL LINEARIZATION

In order to obtain an equivalent linear stiffness coefficient for a gap element at the launch lock location, a generalised discrete harmonic linearization technique is employed [5]. Consider a single DOF vibration system, the equation of motion of the system is given by:

$$m\,\ddot{x}(t) + f(x, \dot{x}, t) = p(t) \qquad (1)$$

where m is the mass, x is the displacement, t the time, f represents the nonlinear force generated by nonlinear elements of the system, and p the excitation load. The linearized stiffness coefficient is computed by equating the value of the energy function of the gap spring to that of a linear spring, through an iterative process. The nonlinear force then can be approximated as:

$$f(x, \dot{x}, t) = K_{eq}(N_k, X)\,x(t) + C_{eq}(N_c, X)\,\dot{x}(t) \qquad (2)$$

The resulted equivalent linear spring, as well as damper, is a function of the excitation level, and the characteristics of the nonlinearity, such as gap size, and stiffness of non-gap portion.

For the case of a launch lock mechanism with a gap in the -Y XMA antenna structure, the computed equivalent spring stiffness is shown in Fig.3. Where the gap size is 0.002 in, the linear spring value for the launch lock with non-gap is 3.94E5 lb/in, the excitation level is qualification level random vibration. The resulted equivalent linear dynamic spring is computed as 19000 lb/in.

The antenna launch lock mechanisms with small clearance are modeled as equivalent linear springs by using CELAS2 elements in the MSC/NASTRAN finite element model. The MSC/NASTRAN program is then effectively used to perform dynamic analysis of the linearised antenna structure model. The normal modal analysis (SOL103) is first employed for the stiffness driven configuration and basic dimension design of the antenna structure. The frequency response analysis (SOL111) is then used for dynamic response and component stress evaluation, and detailed dimension design.

4. MODEL LOCAL REMESH

For dynamic stress analysis of the components, it usually requires large a mount of work which involves the worst case identification in terms of interface loads for each components concerned, the dynamic load processing and transforming for each component model, and the component model runs for stress results [10]. For this project, a different approach was taken for dynamic stress analysis. A locally re-meshed antenna assembly model is constructed with a reasonable size of elements, in terms of a compromise of stress analysis accuracy and available computational space. This approach is to eliminate the work associated with the interface loads transformation and the separated component's level stress analyses.

It is known that from the strain energy intensity information obtained via the normal modes runs (SOL103), the high stress areas of each component in the antenna assembly model can be identified. Each sub-model of critical components in the assembly can be, therefore, locally re-meshed to achieve stress analysis accuracy. The criterion for determining the resolution of the remesh is based on the stress levels deference in adjacent elements. For this project a 25% difference in stresses between the adjacent elements was used. The locally re-meshed antenna assembly model is thus constructed with a reasonable size and number of elements, with respect to a compromise between the stress analysis accuracy and the limited CPU resource. The elements for stress recovery of the dynamic response run are also identified and selected during this local remesh process for each of the components.

A typical locally remeshed component, a slotted waveguide array radiating antenna panel, is shown in Fig.4.

Both dynamic response of the antenna assembly and stresses of the critical components were evaluated based on the same linearised and locally re-meshed antenna structural model. The dynamic response runs employ the modal frequency response method (SOL111) restarted with the database generated from normal modes run (SOL103). The loading environments include base excited sine vibration and random vibration, and acoustic excitations. The component stress analyses were accomplished along with the antenna assembly model runs, which saved a lot of the work, effort, time, and CPU resource as required for the conventional analysis approach.

5. RESULTS CORRELATION

The gimbaled antenna design are qualified through both analysis and testing programs. The linearised finite element models are validated through results correlation with the data from the qualification tests of the antenna assembly. Both normal modes and frequency response results are used to compare with those of the qualification vibration tests, conducted at COM DEV Ltd..

A correlation of the major vibration modes between the analysis and testing results is summarised in Tables 1 to 3, for three XMA antennas with different configurations, launch lock clearances and excitation levels, respectively. It can be observed that the analysis results for the first major mode correlated the measured test data very closely and within a relative error of 1%. As the dynamic mode number increases the relative errors also increases. All the relative error of the major modes data are, however, less than 15% and within an acceptable engineering analysis tolerance.

Table 1: +Y XMA #1 Major Frequencies and Modes
(Clearance $\delta = 2.0 \times 10^{-3}$, Acceptance random)

Test Frequency (Hz)	Analysis Frequency (Hz)	Error %	Mode No.	Analysis Effective Modal Mass (%)			Mode
				X	Y	Z	
92	91.9	0.1	1	15.4	5.4	< 1.0	Major X Mode
109	97.3	10.7	2	5.2	35.4	1.9	Major Y Mode
381	353.5	7.2	8	< 1.0	5.0	< 1.0	Flex W/G op.2
415	466.7	12.4	10	< 1.0	< 1.0	17.1	Major Z Mode

Table 2: -Y XMA #2 Major Frequencies and Modes
(Clearance $\delta = 2.0 \times 10^{-3}$, Acceptance random)

Test Frequency (Hz)	Analysis Frequency (Hz)	Error %	Mode No.	Analysis Effective Modal Mass (%)			Mode
				X	Y	Z	
89	88.4	0.7	1	20.5	3.8	< 1.0	Major X Mode
108	104.7	3.0	2	< 1.0	34.0	< 1.0	Major Y Mode
380	346.8	8.7	8	< 1.0	3.8	< 1.0	Flex W/G op.2
402	438.9	9.1	9	< 1.0	< 1.0	11.6	Major Z Mode

Table 3: -Y XMA #3 Major Frequencies and Modes
(Clearance $\delta = 2.4\ x10^{-3}$, Qual level vibration)

Test Frequency (Hz)	**Analysis Frequency (Hz)**	**Error %**	**Mode No.**	**Analysis Effective Modal Mass (%)**			**Mode**
				X	**Y**	**Z**	
82	81.2	0.9	1	20.4	2.0	< 1.0	Major X Mode
100	104.7	4.7	2	< 1.0	36.0	1.9	Major Y Mode
291	296.2	1.8	7	< 1.0	3.4	< 1.0	Flex W/G op.1
385	423.2	9.9	9	< 1.0	< 1.0	10.0	Major Z Mode

Table 4 presents a summary of the dynamic response results as compared with those of the linearised finite element model, due to a random vibration excitation. For the dynamic response analysis, a damping value Q of 20 was assumed. As can be seen that the equivalent damping values for real antenna assembly are different with respect to the excitation axis. The average damping value of the antenna structure is Q of 16. The simulated maximum response results are, therefore, on the safe side.

A random response at the antenna tip of the linearised antenna model XMA #2 is shown in Fig.5. The corresponding random PSD plot sensed at the same location of the antenna assembly is presented in Fig.6. It can be seen that the results correlated very well at low and medium range frequency range, and shown a large difference at higher frequency modes. This is due to the fact that the linearization is only a first order approximation. But the total rms values are correlated well within an accepted tolerance for engineering analysis.

Table 4: XMA #2 Random Vibration Response

Excitation Axis	Test Response G_{rms}	Random Vibration Analysis		Equivalent Damping "Q"
		Damping "Q"	Response, G_{rms}	
X Axis	8.32	20.0	10.9	15.3
Y Axis	7.08	20.0	12.8	11.2
Z Axis	13.96	20.0	13.01	21.5

6. CONCLUSIONS

The gimbaled nonlinear antenna, developed by COM DEV Ltd., is successfully modeled as an equivalent linearized structural model. The launch lock mechanisms with small clearance are modeled as equivalent linear springs (CELAS2's) by using the iteration technique. The MSC/NASTRAN program is then effectively employed to perform the dynamic analysis of the linearized antenna model. A locally remeshed antenna assembly model is constructed based on the strain energy intensity information. The dynamic stress analysis for components is accomplished along with antenna assembly model runs which eliminated the work associated with interface loads transformation and separated component's level stress analyses. It has been shown that the presented linearization method and local remesh approach have achieved a good compromise in terms of analysis accuracy, computational cost and effectiveness.

The qualification test data also show that the analysis results of the nonlinear gimbaled antennas, in terms of major modal frequencies, sine and random acceleration response, correlated the measured qualification test data very well. The linearised antenna finite element models are, therefore, validated and the FE based antenna design is qualified.

7. ACKNOWLEDGMENTS

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Fig.1 A Gimbaled Antenna In Stowed Configuration

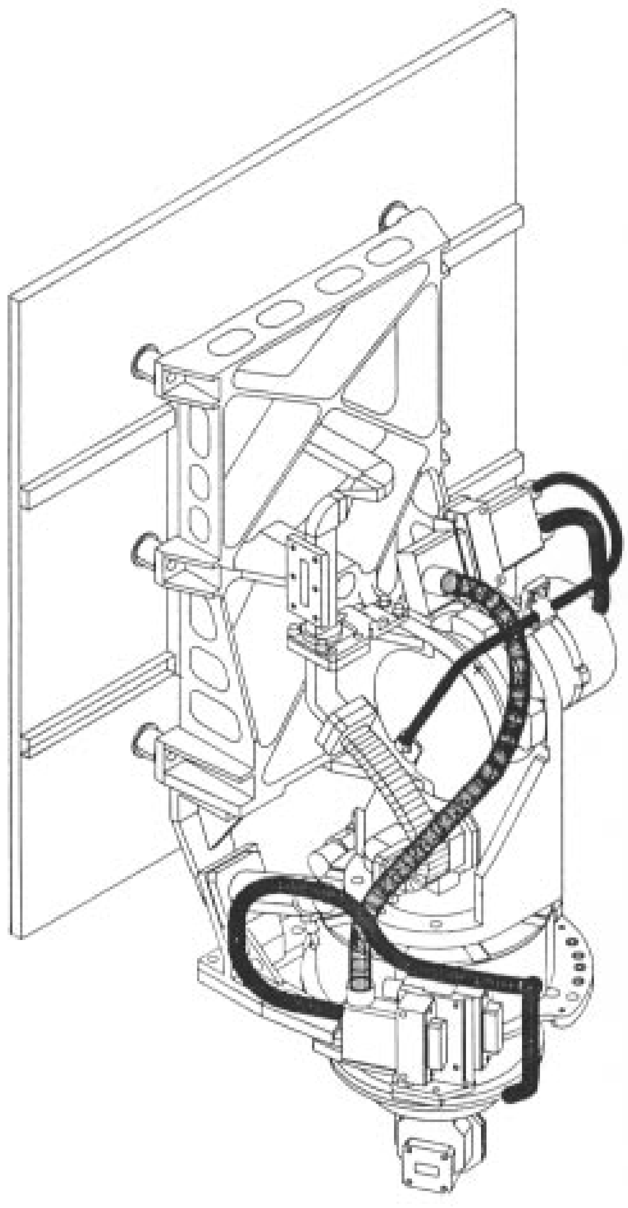
Fig.2 A Finite Element Model of Gimbaled Antenna

Fig.3 Launch Lock Spring Stiffness Models

Fig.4 A Local Remeshed Component, Antenna Plate Model

Fig.5 An Analytical Random Response of XMA #2 Antenna Model

Fig.6 A Measured Random Response of XMA #2 Antenna Assembly



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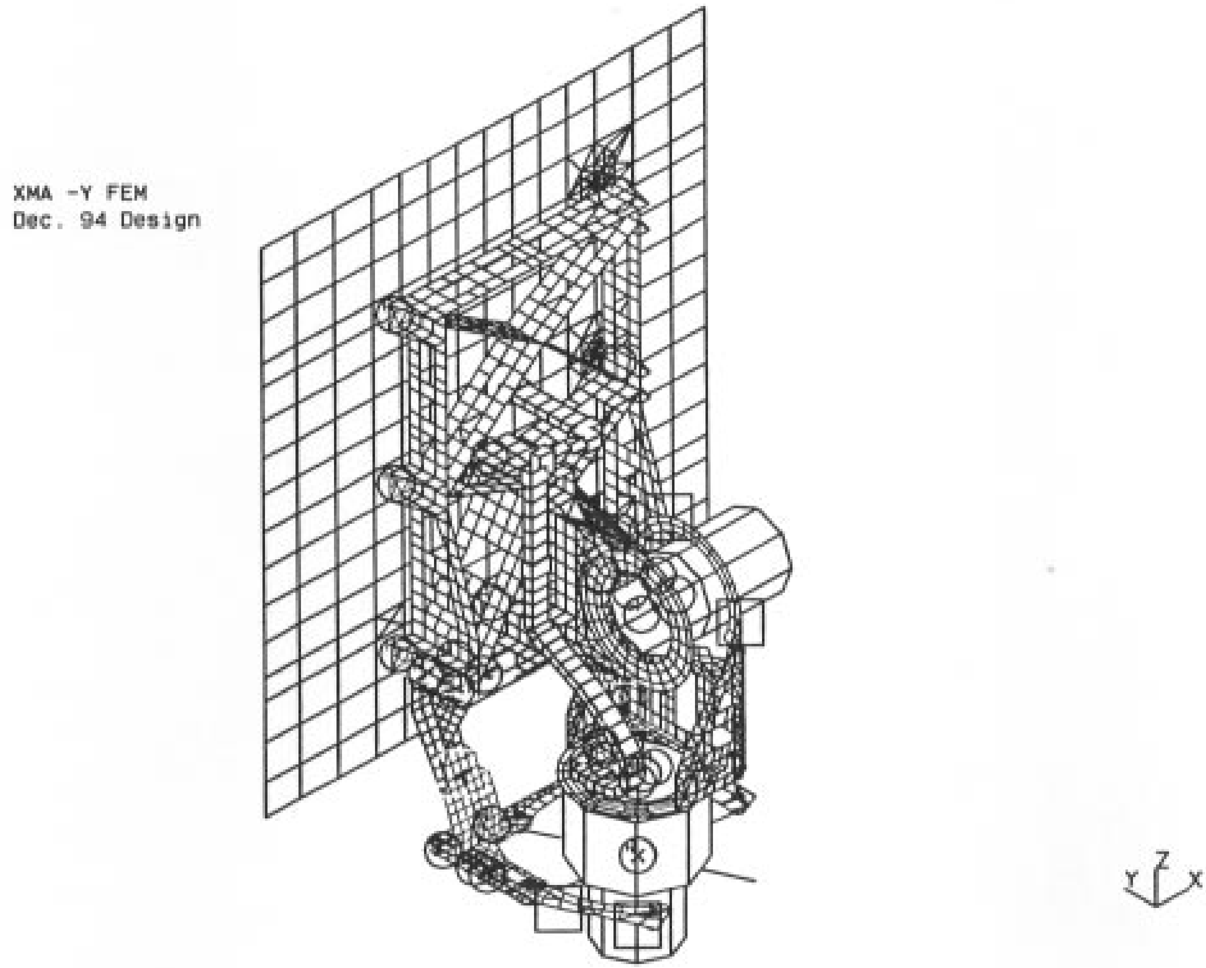

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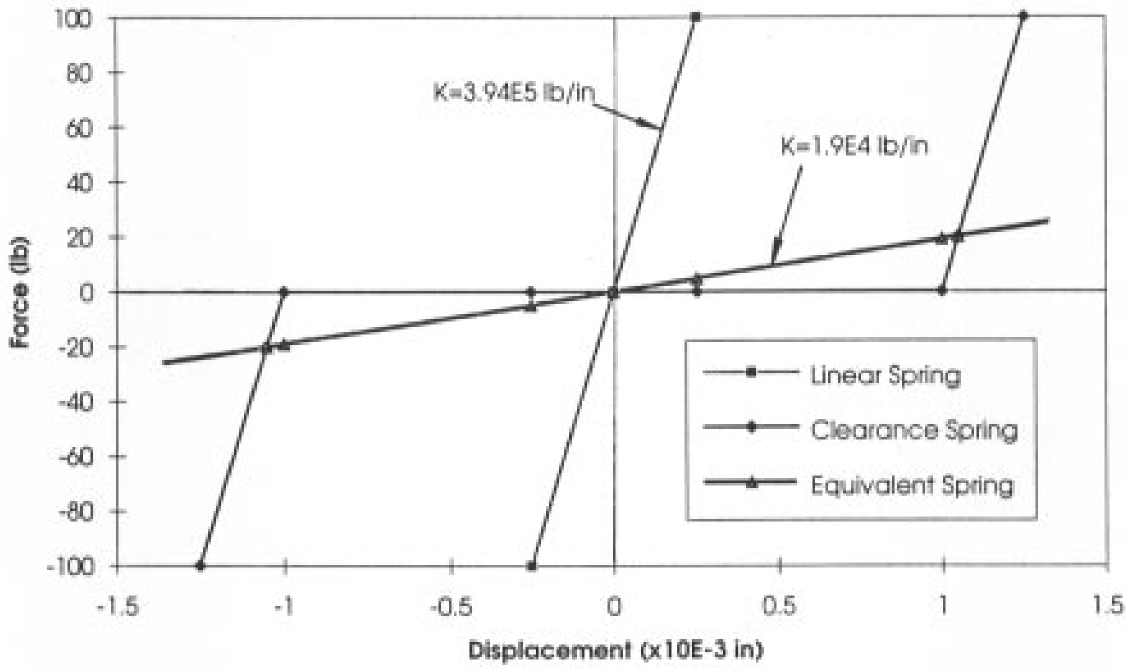

Fig.3 Launch Lock Spring Stiffness Models

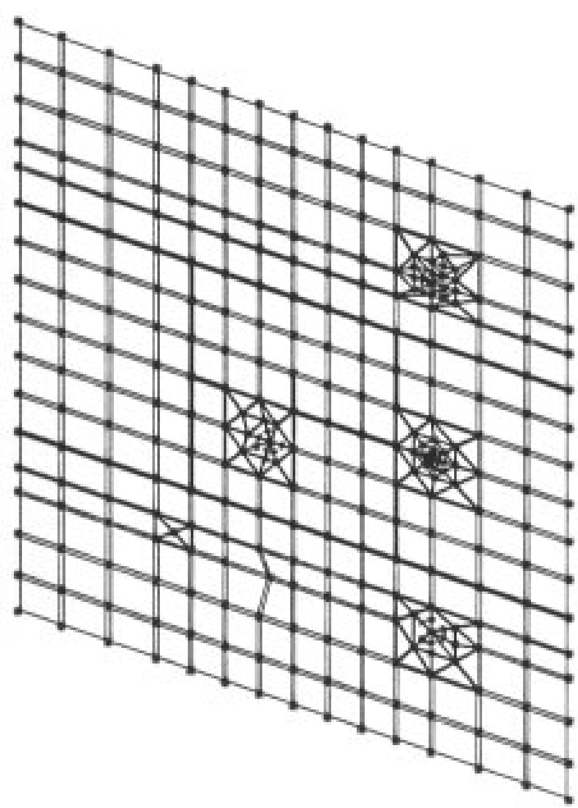

Fig.4 A Local Remeshed Component, Antenna Plate Model

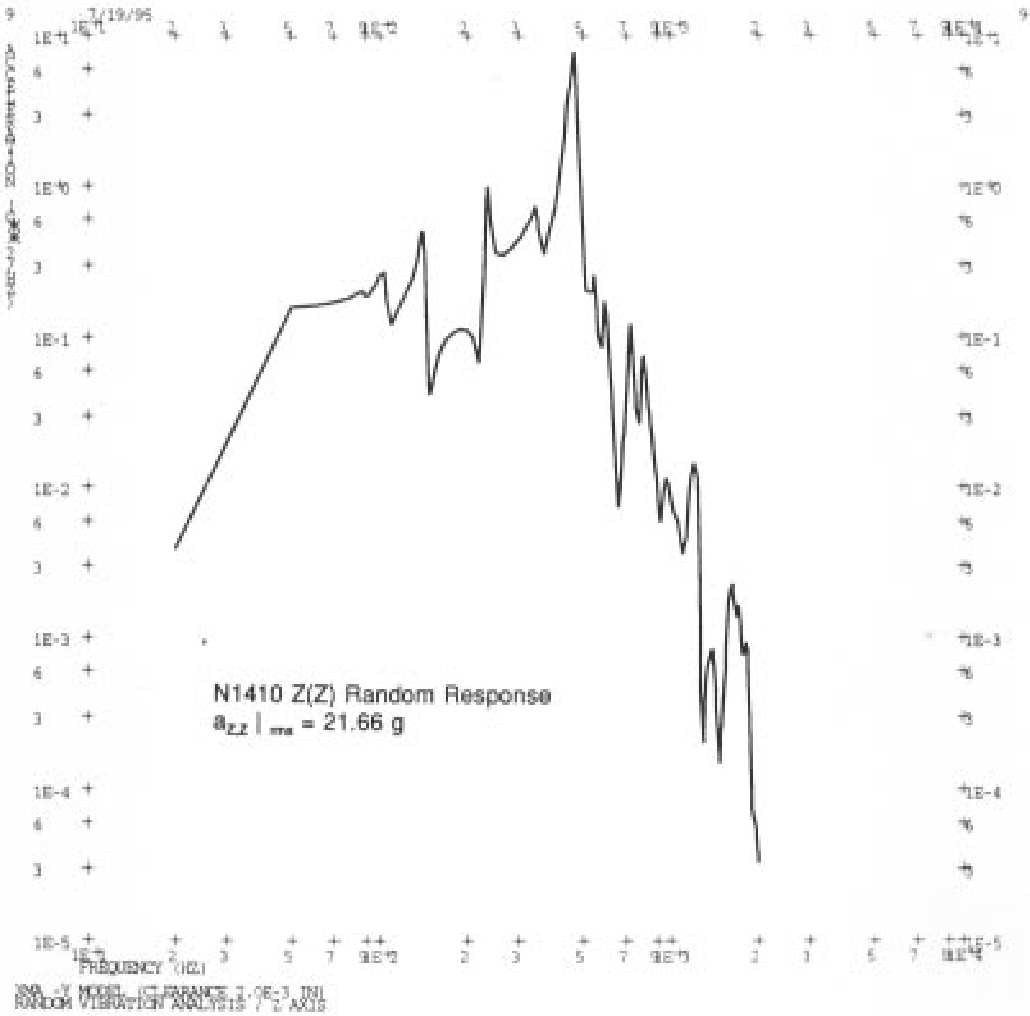

Fig.5 An Analytical Random Response of XMA #2 Antenna Model

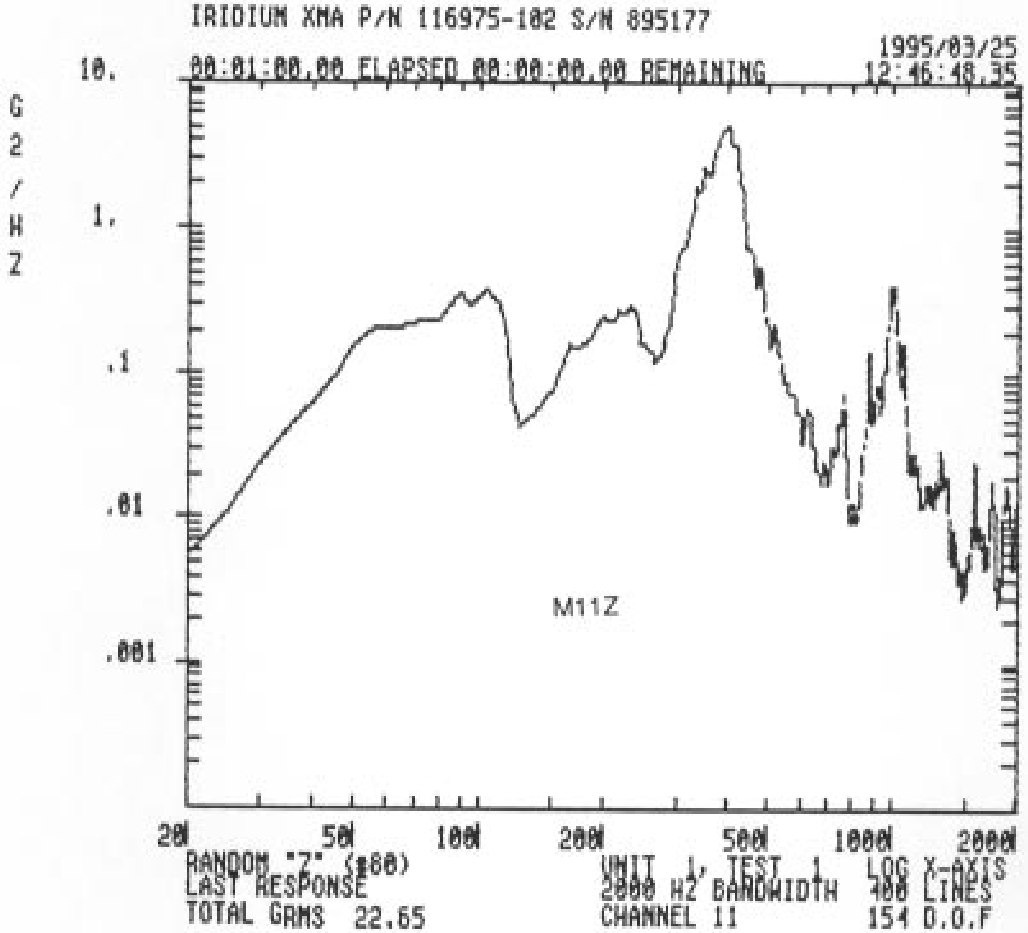

Fig.6 A Measured Random Response of XMA #2 Antenna Assembly