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by

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SUMMARY

Dynamic response of a candidate flexible beam for a space experiment on control of flexible structures is investigated. Studies of natural frequencies reveal a beam length in which torsion and bending frequencies virtually coincide. Eccentric tip mass causes small shifts in natural frequencies but introduces coupled torsional/bending mode shapes. Transient response studies indicate significant effects on tip responses of low damping and first bending mode excitation at higher frequencies. Steady state responses suggest displacement and acceleration measurements could be made up to 5-12 Hz for the actuator forces/torques assumed.

INTRODUCTION

As spacecraft increase in size and mission control requirements become more demanding, there is a need to investigate control of flexible structures. NASA has initiated a research program (ref. 1) which involves ground and flight tests of a large flexible beam. In figure 1, the deployment and test sequence is indicated for a flexible beam carried to orbit by the shuttle orbiter. The structure considered is a deployable graphite truss with demanding precision requirements representative of large flexible space antenna masts.

The purpose of the present paper is to present results of preliminary dynamic response studies of a flexible beam suitable for the flight experiment. The fundamental frequencies of the candidate beam are determined. Transient and steady state vibration responses are investigated to determine feasible ranges of measurement for the flight experiment.

SYMBOLS

A	equivalent area of lattice beam
a_x, a_y	accelerations in x and y directions
E	Young's modulus of lattice beam longeron
G	shear modulus for twisting of lattice beam
I_p	mass moment of inertia per unit of beam length about z axis

I_{xx}	equivalent area moment of inertia of lattice beam about x axis
I_{yy}	equivalent area moment of inertia of lattice beam about y axis
$\bar{I}_{xx}$	mass moment of inertia of tip mass about x axis
$\bar{I}_{yy}$	mass moment of inertia of tip mass about y axis
$\bar{I}_{zz}$	mass moment of inertia of tip mass about z (longitudinal) axis
J	equivalent torsional constant of lattice beam about z axis
L	length of beam
T_y, T_z	applied cyclic torques about y and z axes
P_x	applied cyclic load in x direction
u,v,w	displacements in x, y, and z directions
x,y,z	beam coordinates (see fig. 2)
$\tilde{x},\tilde{y}$	location of eccentric concentrated mass at tip of beam
β	frequency equation coefficient (eqs. (1) and (2))
ρ	mass density of beam material

ANALYSIS APPROACH

The candidate mast-beam properties are presented in figure 2. The stiffness properties are based on equivalent beam properties for double-bay single-laced latticed beams developed in reference 2 with a slight modification to account for unequal area longerons. Analysis of beams several bays long with averaged stiffness properties have been shown to give good accuracy when compared to finite element models of latticed truss beams.

Dynamic response studies were performed using two finite element analyses: the Engineering Analysis Language (EAL, ref. 3) and MSC/pal (ref. 4) a finite-element analysis for a micro-computer. In both cases, dynamic analyses were conducted on cantilevered beams with 24 elements in EAL and 9 elements for MSC/pal. The effects of eccentrically mounted tip masses on the beam were investigated by adding an additional very stiff beam element with a concentrated tip mass oriented in the x-y plane (fig. 2). In a preliminary study, it was found that the lowest torsion and second bending frequencies were virtually identical when the beam was 44 m in length. Since the in-orbit experiment involves partial and full deployment of the beam (fig. 1), beam lengths of 44 m and 60.5 m were investigated. In the experiment, the beam is to be excited (and damped) by actuators mounted on the beam. For this study, a single tip-mounted actuator was investigated with actuator forces represented by cyclic forces or torques applied at the beam tip.

ANALYSIS RESULTS

Natural Frequency

The calculated natural frequencies are presented in table I and plotted in figure 3. Comparison of MSC/pal and EAL results indicate excellent correlation for non-eccentric bending results indicating sufficient number of nodes for convergence. Only good correlation was obtained for eccentric bending/torsion results and only fair correlation for non-eccentric torsion frequencies.

The differences in torsion results was traced to the incorrect use by EAL of $I_p = \rho J$ instead of the value $\rho(I_{xx} + I_{yy})$. For the built up lattice beam of the present paper this latter expression, which is correct for a uniform beam, properly accounts for the mass of the longerons but neglects the mass of the battens and diagonals according to the continuum theory of reference 2. Since J was an order of magnitude smaller than I_{xx} and I_{yy}, differences in results from the two programs occur for any mode involving significant torsion.

If the beam has a distributed mass moment of inertia I_p per unit of beam length L, the torsional frequencies f are given by (ref. 5)

$$f = \frac{1}{2\pi} \frac{\beta}{L} \sqrt{\frac{GJ}{I_p}} \tag{1}$$

where β is found from solutions of the transcendental equation

$$\beta \tan \beta = \frac{I_p L}{\bar{I}_{zz}} \tag{2}$$

in which GJ is the torsional stiffness of the beam and $\bar{I}_{zz}$ is the mass moment of inertia of the tip about the longitudinal (z) axis of the beam.

For the simple case without eccentricity, MSC/pal torsional frequencies were found to correlate within 1-4 percent with solutions of equation (1), while the torsion frequencies from EAL are in error because of the incorrect value of I_p.

In figure 3, the MSC/pal frequencies with no eccentric mass are plotted for the 60.5 m and 44 m beams. For plotting purposes, the lowest pair of bending frequencies was grouped as mode 1, with the next highest pair as mode 2, etc. The lowest torsion mode was plotted as mode 2 and higher modes as mode 3, etc. For the 60.5 m case, results presented in figure 3 indicate that torsion frequencies are generally higher than pairs of bending frequencies; only the lowest torsion mode is within 30 percent of a bending frequency.

As illustrated in figure 3, however, torsional frequencies for the 44 m beam are within 15 percent of bending frequencies over the frequency range plotted and the lowest torsion mode is almost coincident with the second pair of bending modes. The near coincidence of bending and torsion frequencies is an ideal situation for a challenging dynamics and control experiment, since the identification and control of responses of the real structure is complicated by the response of neighboring frequencies.

A second method of obtaining complex response is by inertially coupling bending and torsion responses through use of an eccentric (offset) mass at the tip of the beam. Frequency results for this case are summarized in table I. Results show relatively small shifts in frequencies from the non-eccentric case; however, mode shapes are changed significantly in that bending is in two directions and torsion is present in all modes. Representative mode shapes for the 44 m beam with eccentric mass are presented in table II. Note for the third, fourth, and fifth frequencies presented, the mode shape involves noticeable u and v displacements for all frequencies. For the third frequency, torsion is coupled with bending deformations as evidenced by larger z rotations and significant values of u and v. Eccentric beam modes with significant torsional rotation were found from animated simulations of modes and are indicated in table I.

FORCED RESPONSE RESULTS

To determine feasible ranges of measurement for the flight experiment, the beams were subjected to sinusoidal excitations at the tip of the beam (station 10 for MSC/pal, station 25 for EAL). Both transient and steady state responses were obtained under a lateral force of 30 N or a cyclic torque of 10 N-m. In all cases solutions contained at least 9 modes, with modal damping of 1/2 percent. In modeling the input force or torque for MSC/pal the input sine function was represented by 560 points (a program limit) with 8 points per sine wave. For EAL, sine waves were modeled with 24 points per sine wave. Similar results were obtained from both solutions.

Transient Response

Typical results for transient response are presented in figures 4, 5, and 6. In figure 4, transient results for the 44 m beam with non-eccentric mass under a 30 N lateral force are presented. Deflections of node 6 (the maximum nodal deflection in the 44 m beam) and node 10 (the beam tip) based on MSC/pal calculations are presented. The maximum displacement of the beam under 21 seconds of applied load is 1.52 cm. Maximum tip deflection is about 0.55 cm. As expected, the forcing frequency excites the second beam bending mode. However, the vibration of the beam tip also generates noticable response in the first bending mode that is evident in both plots.

Vibration in the first bending mode is also apparent for response of the 60.5 m beam presented in figure 5. In this case, the maximum deflection of the beam driven at the second bending mode is 5.52 cm and the tip deflection is 2.14 cm at 1.74 Hz. Studies of transient response indicated that as the frequency is increased, it becomes more difficult to excite higher frequency beam modes.

Because of low damping in the structure, transient response of the beam tip will have some significant implications on how the beam orbital tests will be conducted. In figure 6, the transient response of the beam is shown over a much longer time period than in figure 5. Note that total response peaks in about 60 seconds but it takes at least 10 minutes before steady state response is reached at the beam tip. Note also that the peak transient response is greater than the steady state response. The interaction of inertial masses with beam vibrations can limit the performance of tip actuators (see for example refs. 6 and 7), so that the configuration investigated herein would appear to offer a significant research challenge in control of flexible bodies.

Steady State Responses

Investigation of limiting (steady state) responses was conducted using EAL. Results for steady state displacement amplitudes (u,v) and accelerations (a_x,a_y) are presented in table III for cyclic applied forces of 30 N and cyclic applied torques of 10 N-m. For brevity, responses are shown at the beam tip and mid-beam. In some cases, larger responses could be found at other stations for particular modes.

Limitations of orbital measurement systems are expected to be about 2 mm displacement using optical systems and 50 micro-g' using sensitive accelerometers. Data obtained from table III and other transient response studies suggests that deflection measurements can be made up to 5 Hz for the 60.5 and 44 m beams. Acceleration measurements appear feasible over the entire frequency range (up to 12 Hz). If the coupling between the second torsion mode and third pair of bending modes is of interest in the experiment, larger forces or torques will be needed for the actuators.

CONCLUSIONS

A preliminary investigation of the dynamic response of a lattice beam has been conducted. The beam is a candidate for a flight experiment to investigate control of flexible structures. The study included analysis of vibration modes and frequencies, transient and steady state response, and the effect of eccentric mass on dynamic response.

Studies of vibration frequencies indicated a beam length could be found in which the lower torsion and bending modes are virtually coincident. Investigation of the effects of eccentric tip mass showed only small changes on natural frequencies. However, mode shapes were found to have noticeable coupling between torsional and bending deformations. Two methods of analysis were compared and found to be in good agreement except where the effects on torsion frequencies due to distributed beam inertia had been neglected.

Transient response studies showed significant excitations of the lowest bending modes when attempting to excite higher frequency beam modes. Achievement of steady state response for the beam tip required minutes of excitation at the tip. This behavior suggests that great care will have to be taken in the design of tip controllers for the beam and may require additional sensors for control. Steady state response calculations show that acceleration and displacement measurements are feasible up to 5-12 Hz. Based on the limited dynamic response results presented herein, the beam investigated appears to have sufficiently complex dynamic behavior to provide a challenging test bed for research on control of flexible structures.

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TABLE I - BEAM NATURAL FREQUENCIES

(a) 60.5 m beam

No Eccentric Mass			With Eccentric Mass	
MSC/pal	Mode Shape	EAL	MSC/pal	EAL
0.182	1st Bend	0.182	0.182	0.181
0.197	1st Bend	0.197	0.197	0.197
1.742	2nd Bend	1.743	1.737	1.738
1.887	2nd Bend	1.888	1.875	1.878
2.539	1st Tors	3.058	2.585*	3.101
5.410	3rd Bend	5.410	5.414	5.418
5.859	3rd Bend	5.858	5.873	5.884
9.393	2nd Tors	---	9.421*	---
11.139	4th Bend	11.124	11.133	11.114
12.062	4th Bend	12.044	12.081	12.060
17.458	1st Axial	17.463	17.275	17.195
17.639	3rd Tors	---	17.658*	---
18.934	5th Bend	18.862	19.044	18.968
20.498	5th Bend	20.416	20.564	20.479
26.656	4th Tors	---	26.673*	---

*Torsion Dominated Mode

(b) 44 m beam

No Eccentric Mass			With Eccentric Mass	
MSC/pal	Mode Shape	EAL	MSC/pal	EAL
0.304	1st Bend	0.301	0.303	0.302
0.328	1st Bend	0.327	0.329	0.327
3.123	1st Tors	3.59	3.037*	3.183
3.241	2nd Bend	3.24	3.294	3.396
3.510	2nd Bend	3.51	3.598	3.823
10.139	3rd Bend	10.13	10.128	10.121
10.979	3rd Bend	10.96	10.986	10.993
12.593	2nd Tors		12.653*	---
20.868	4th Bend		20.336	20.250
21.437	1st Axial		21.674	21.587
22.591	4th Bend		22.812	22.864
24.053	3rd Tors		24.094*	---

*Torsion Dominated Mode

TABLE II - MODE SHAPES FOR 44 M BEAM WITH ECCENTRIC MASS

MODE NO. 1 AT 3.03181E-01 CPS (1.90494E+00 RAD/SEC)

NODE	W	U	V	Z ROT	X ROT	Y ROT
1	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00
2	-4.1016E-08	1.1203E-02	5.7655E-06	-2.8981E-04	-2.3661E-06	4.4877E-03
3	-1.2099E-07	4.2948E-02	2.3601E-05	-5.7899E-04	-4.9914E-06	8.4038E-03
4	-1.9992E-07	9.2445E-02	5.4267E-05	-8.6682E-04	-7.5110E-06	1.1751E-02
5	-2.9085E-07	1.5693E-01	9.6601E-05	-1.1532E-03	-9.7202E-06	1.4535E-02
6	-3.8305E-07	2.3366E-01	1.4782E-04	-1.4390E-03	-1.1134E-05	1.6766E-02
7	-4.6225E-07	3.1998E-01	2.0469E-04	-1.7257E-03	-1.1997E-05	1.8458E-02
8	-5.3029E-07	4.1329E-01	2.6397E-04	-2.0142E-03	-1.2191E-05	1.9632E-02
9	-5.5074E-07	5.1113E-01	3.2374E-04	-2.3040E-03	-1.2224E-05	2.0312E-02
10	-5.6540E-07	6.1115E-01	3.8356E-04	-2.5944E-03	-1.2275E-05	2.0531E-02

MODE NO. 2 AT 3.29193E-01 CPS (2.06838E+00 RAD/SEC)

NODE	W	U	V	Z ROT	X ROT	Y ROT
1	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00
2	9.3000E-08	4.3094E-06	1.1197E-02	4.3449E-04	-4.4856E-03	1.7685E-06
3	1.6852E-07	1.7239E-05	4.2930E-02	8.7101E-04	-8.4014E-03	3.5190E-06
4	2.4652E-07	3.9090E-05	9.2418E-02	1.3099E-03	-1.1749E-02	5.5159E-06
5	3.2933E-07	7.2968E-05	1.5689E-01	1.7508E-03	-1.4535E-02	8.6040E-06
6	4.1124E-07	1.2560E-04	2.3363E-01	2.1933E-03	-1.6767E-02	1.3066E-05
7	4.8928E-07	2.0003E-04	3.1996E-01	2.6391E-03	-1.8460E-02	1.7171E-05
8	5.5913E-07	2.9085E-04	4.1328E-01	3.0874E-03	-1.9634E-02	1.9702E-05
9	6.2982E-07	3.9042E-04	5.1113E-01	3.5379E-03	-2.0315E-02	2.0897E-05
10	6.8597E-07	4.9413E-04	6.1116E-01	3.9885E-03	-2.0534E-02	2.1411E-05

MODE NO. 3 AT 3.03661E+00 CPS (1.90796E+01 RAD/SEC)

NODE	W	U	V	Z ROT	X ROT	Y ROT
1	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00
2	4.0396E-06	2.3792E-02	-1.3505E-02	5.3696E-02	5.0880E-03	8.9164E-03
3	8.0683E-06	7.9297E-02	-4.5525E-02	1.0694E-01	7.5889E-03	1.3004E-02
4	1.2090E-05	1.4365E-01	-8.3771E-02	1.5927E-01	7.6816E-03	1.2632E-02
5	1.6112E-05	1.9682E-01	-1.1734E-01	2.1026E-01	5.7603E-03	8.5979E-03
6	2.0135E-05	2.2372E-01	-1.3780E-01	2.5947E-01	2.4386E-03	2.1206E-03
7	2.4172E-05	2.1602E-01	-1.4017E-01	3.0648E-01	-1.5053E-03	-5.2944E-03
8	2.8198E-05	1.7300E-01	-1.2346E-01	3.5089E-01	-5.2231E-03	-1.2071E-02
9	3.2209E-05	1.0133E-01	-9.0784E-02	3.9233E-01	-7.9102E-03	-1.6816E-02
10	3.6191E-05	1.3523E-02	-4.8869E-02	4.3045E-01	-8.9130E-03	-1.8547E-02

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TABLE II - MODE SHAPES FOR 44 M BEAM WITH ECCENTRIC MASS - Concluded

MODE NO. 4 AT 3.29415E+00 CPS (2.06978E+01 RAD/SEC)

NODE	W	U	V	Z ROT	X ROT	Y ROT
1	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00
2	2.1555E-05	4.9593E-02	1.7404E-02	-2.8365E-02	-6.5212E-03	1.8485E-02
3	4.3107E-05	1.6337E-01	5.7985E-02	-5.6448E-02	-9.5042E-03	2.6331E-02
4	6.4648E-05	2.9111E-01	1.0499E-01	-8.3969E-02	-9.2196E-03	2.4432E-02
5	8.6191E-05	3.8944E-01	1.4374E-01	-1.1065E-01	-6.2529E-03	1.4702E-02
6	1.0771E-04	4.2675E-01	1.6320E-01	-1.3624E-01	-1.4978E-03	1.0000E-05
7	1.2920E-04	3.8706E-01	1.5727E-01	-1.6046E-01	3.9424E-03	-1.6204E-02
8	1.5067E-04	2.7145E-01	1.2542E-01	-1.8309E-01	8.9160E-03	-3.0510E-02
9	1.7211E-04	9.6448E-02	7.2523E-02	-2.0389E-01	1.2408E-02	-4.0129E-02
10	1.9353E-04	-1.1067E-01	7.6989E-03	-2.2266E-01	1.3707E-02	-4.3512E-02

MODE NO. 5 AT 3.59796E+00 CPS (2.26067E+01 RAD/SEC)

NODE	W	U	V	Z ROT	X ROT	Y ROT
1	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00	.0000E+00
2	1.1858E-05	-1.0075E-02	5.1317E-02	3.2319E-02	-1.9118E-02	-3.7337E-03
3	2.3705E-05	-3.2773E-02	1.6885E-01	6.4254E-02	-2.7167E-02	-5.1816E-03
4	3.5541E-05	-5.7351E-02	3.0038E-01	9.5427E-02	-2.5088E-02	-4.5583E-03
5	4.7370E-05	-7.4692E-02	4.0086E-01	1.2547E-01	-1.4890E-02	-2.3161E-03
6	5.9188E-05	-7.8435E-02	4.3761E-01	1.5402E-01	4.1973E-04	8.8091E-04
7	7.0999E-05	-6.5780E-02	3.9429E-01	1.8074E-01	1.7247E-02	4.2628E-03
8	8.2804E-05	-3.7646E-02	2.7235E-01	2.0531E-01	3.2025E-02	7.1047E-03
9	9.4571E-05	1.9425E-03	8.9226E-02	2.2743E-01	4.1896E-02	8.8867E-03
10	1.0630E-04	4.7203E-02	-1.2668E-01	2.4686E-01	4.5312E-02	9.4291E-03

TABLE III - STEADY STATE RESPONSE
(a) 44 m beam

Steady State Response, 44m beam with no eccentricity

Load		P_x = 30		M_y = 10	
Freq	Loc	u	a_x	u	a_x
Hz		mm	g	mm	g
.30	tip	3977	1.471	44.53	.016
	mid	1079	.399	12.09	.004
.33	tip	229	.099	2.56	.001
	mid	62.36	.027	.72	.000
3.24	tip	3.63	.153	.62	.026
	mid	17.5	.739	2.98	.126
3.51	tip	.51	.025	.04	.002
	mid	.94	.046	.18	.009
*3.59	tip	.44	.023	.03	.001
	mid	.69	.036	.13	.007
10.13	tip	.14	.058	.06	.026
	mid	.04	.297	.34	.141
10.96	tip	.04	.020	.00	.002
	mid	.03	.015	.02	.008

Steady State Response, 44m beam with eccentricity

Load		P_x = 30				M_z = 10			
Freq	Loc	u	v	a_x	a_y	u	v	a_x	a_y
Hz		mm	mm	g	g	mm	mm	g	g
.30	tip	3975	14.40	1.454	.005	5.69	.44	.002	.000
	mid	1079	3.91	.395	.001	1.54	.12	.001	.000
.33	tip	229	17.17	.099	.007	.39	7.52	.000	.003
	mid	62.53	4.66	.027	.002	.10	2.04	.000	.001
3.18	tip	1.28	.46	.052	.019	1.75	.65	.071	.027
	mid	9.49	2.15	.387	.087	12.9	3.04	.526	.124
3.40	tip	1.94	.72	.090	.033	2.53	.94	.118	.044
	mid	4.71	9.68	.218	.449	6.13	12.8	.284	.592
3.82	tip	2.44	3.88	.143	.228	4.83	7.77	.284	.457
	mid	2.78	6.90	.164	.406	5.51	13.8	.324	.812
10.12	tip	.16	.01	.066	.005	.03	.01	.014	.004
	mid	.77	.05	.317	.021	.16	.02	.065	.008
10.99	tip	.05	.03	.022	.013	.01	.04	.006	.021
	mid	.03	.12	.016	.058	.01	.19	.006	.094

* Torsion mode

TABLE III - STEADY STATE RESPONSE - Concluded
(b) 60.5 m beam

Steady State Response, 60.48m beam with no eccentricity

Load		P_x = 30		M_y = 10	
Freq	Loc	u	a_x	u	a_x
Hz		mm	g	mm	g
.18	tip	10321	1.377	83.7	.011
	mid	2815	.375	22.8	.003
.20	tip	594	.093	4.80	.001
	mid	163	.025	1.36	.000
1.74	tip	15.7	.191	1.52	.018
	mid	57.7	.706	5.62	.069
1.89	tip	1.86	.027	.09	.001
	mid	3.10	.044	.33	.005
*3.06	tip	.43	.016	.01	.000
	mid	.22	.008	.04	.001
5.41	tip	.64	.076	.16	.019
	mid	2.44	.287	.64	.075
5.86	tip	.15	.021	.01	.002
	mid	.11	.015	.03	.004
11.12	tip	.08	.041	.04	.019
	mid	.45	.222	.23	.116
12.04	tip	.15	.019	.00	.002
	mid	.11	.017	.02	.009

Steady State Respons 60.48m beam with eccentric'ty

Load		P_x = 30				M_z = 1			
Freq	Loc	u	v	a_x	a_y	u	v	a_x	a_y
Hz		mm	mm	g	g	mm	mm	g	g
.18	tip	10319	17.22	1.363	.002	7.27	.56	.001	.000
	mid	2814	4.68	.372	.001	1.98	.15	.000	.000
.20	tip	595.3	20.48	.092	.003	.49	9.66	.000	.001
	mid	162.9	5.58	.025	.001	.13	2.63	.000	.000
1.74	tip	14.2	.22	.172	.003	1.46	.04	.018	.001
	mid	54.9	1.88	.667	.023	5.65	.50	.069	.006
1.88	tip	1.76	1.21	.025	.017	.30	1.86	.004	.026
	mid	3.04	4.85	.043	.069	.50	7.51	.007	.106
3.10	tip	3.34	4.51	.129	.174	8.25	11.2	.319	.434
	mid	1.67	2.37	.065	.092	4.12	5.89	.159	.228
5.42	tip	.76	.07	.090	.008	.19	.04	.229	.005
	mid	2.65	.18	.312	.021	.67	.08	.079	.010
5.88	tip	.17	.13	.023	.019	.06	.25	.008	.035
	mid	.11	.43	.016	.059	.04	.79	.006	.110
11.11	tip	.09	.01	.046	.004	.02	.01	.009	.004
	mid	.47	.02	.234	.008	.09	.01	.046	.004
12.06	tip	.04	.01	.021	.008	.01	.02	.005	.014
	mid	.03	.07	.017	.004	.01	.11	.005	.067

* Torsion mode

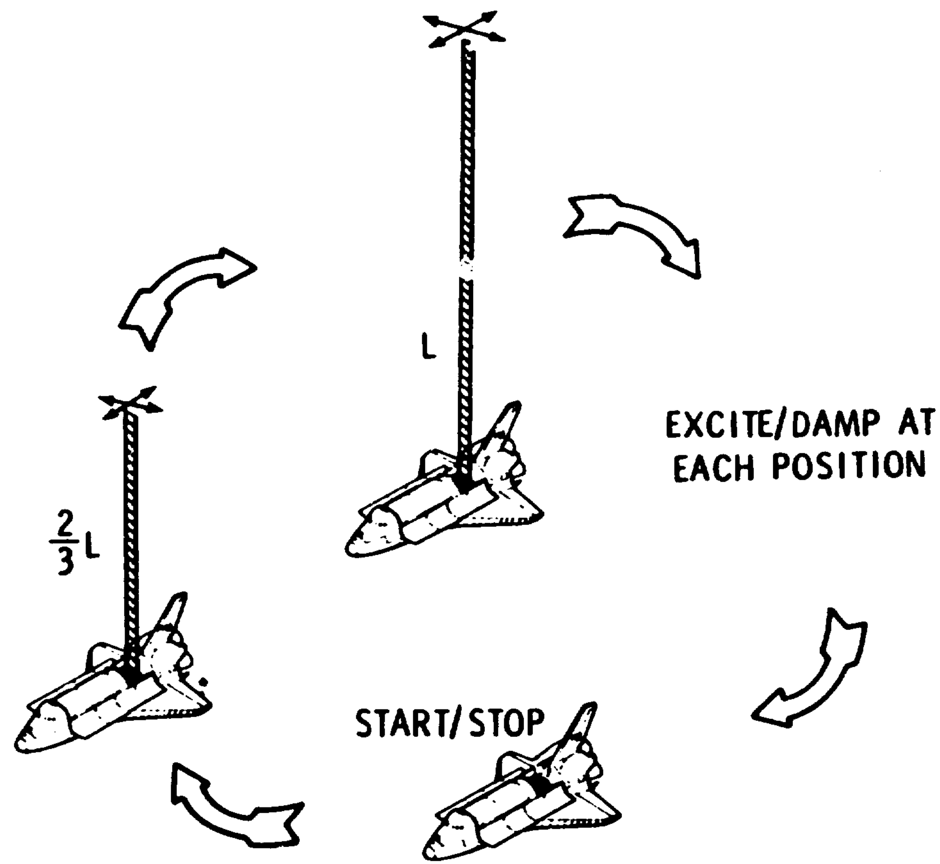
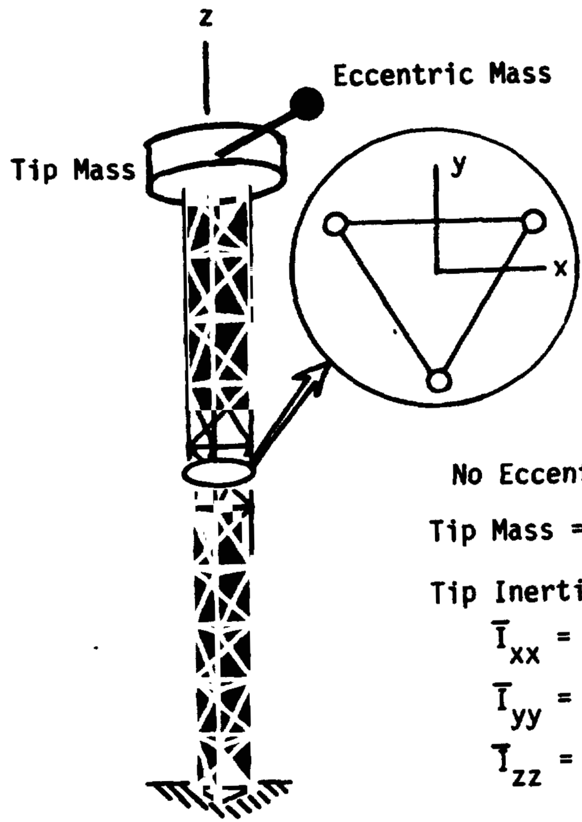


Figure 1.- Flight experiment with flexible deployable beam.



$E = 137.9$ GN/m^2

$G = 68.95$ GN/m^2

$\rho = 2758.7$ kg/m^3

$A = 1.305 \times 10^{-3}$ m^2

$J = 1.385 \times 10^{-5}$ m^4

$I_{xx} = 1.821 \times 10^{-4}$ m^4

$I_{yy} = 1.552 \times 10^{-4}$ m^4

No Eccentricity

Tip Mass = 170 kg

Tip Inertias:

$\overline{I}_{xx} = 21$ kg-m^2

$\overline{I}_{yy} = 21$ kg-m^2

$\overline{I}_{zz} = 42$ kg-m^2

With Eccentricity

Tip Mass = 100 kg

Tip Inertias:

$\overline{I}_{xx} = 12.35$ kg-m^2

$\overline{I}_{yy} = 12.35$ kg-m^2

$\overline{I}_{zz} = 24.71$ kg-m^2

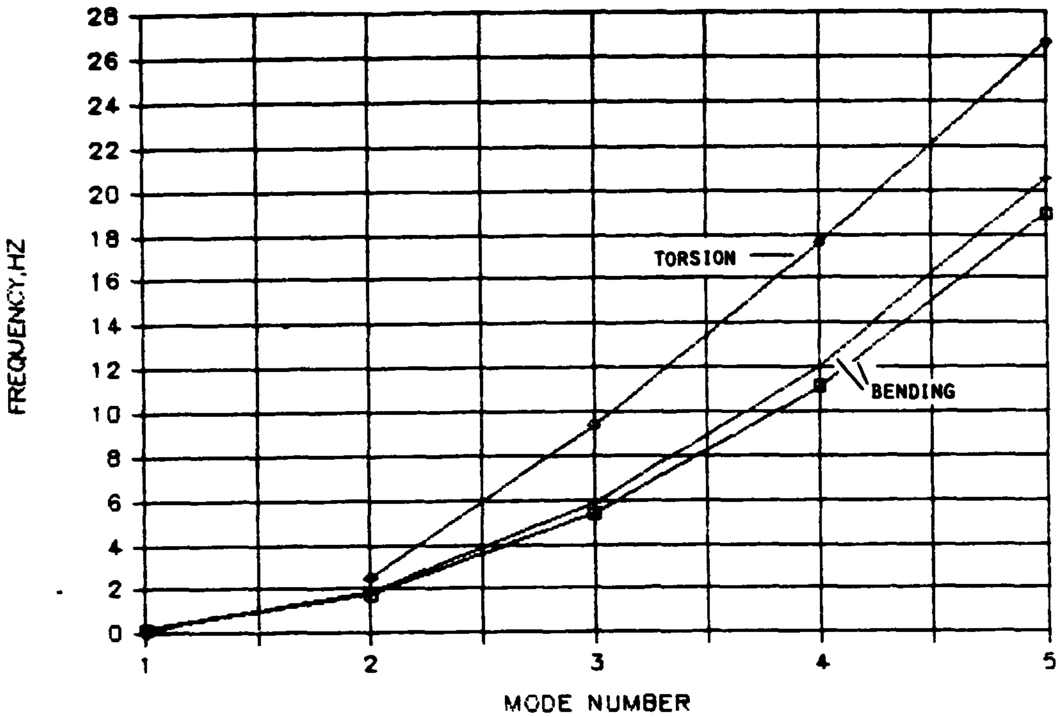
Eccentric Mass = 70 kg

Coordinates of Eccentric Mass

$\overline{x} = .48$ m $\overline{y} = .36$ m

Figure 2.- Mast-beam properties.

60.5 METER BEAM NATURAL FREQUENCIES



44 METER BEAM NATURAL FREQUENCIES

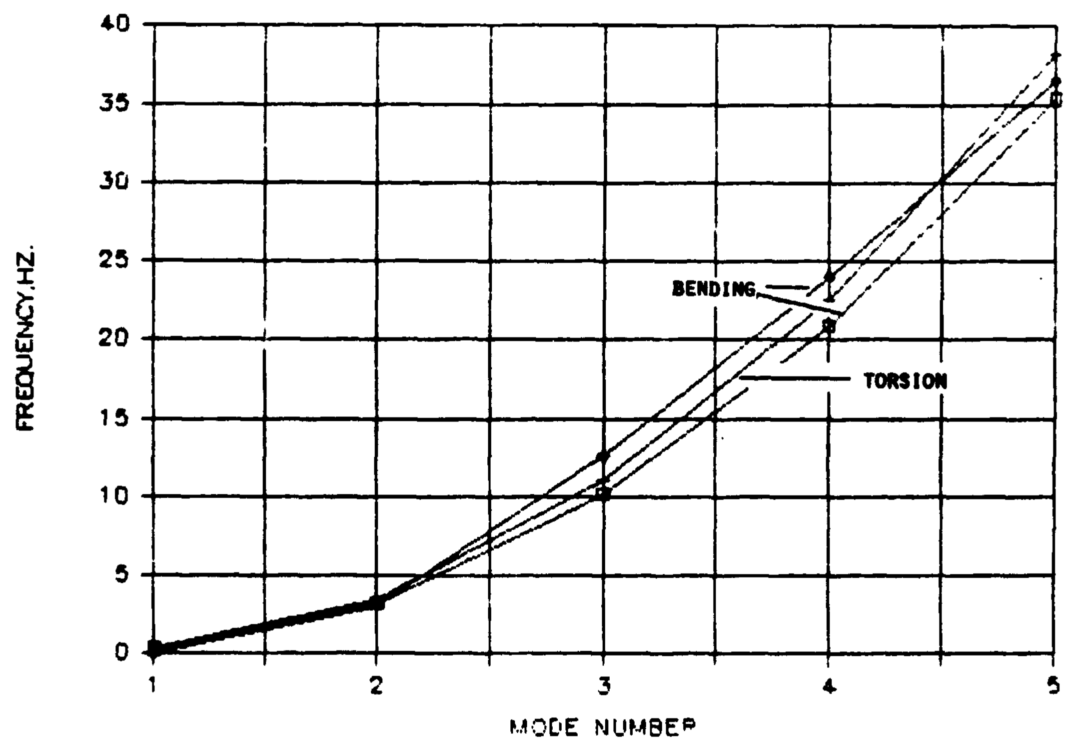
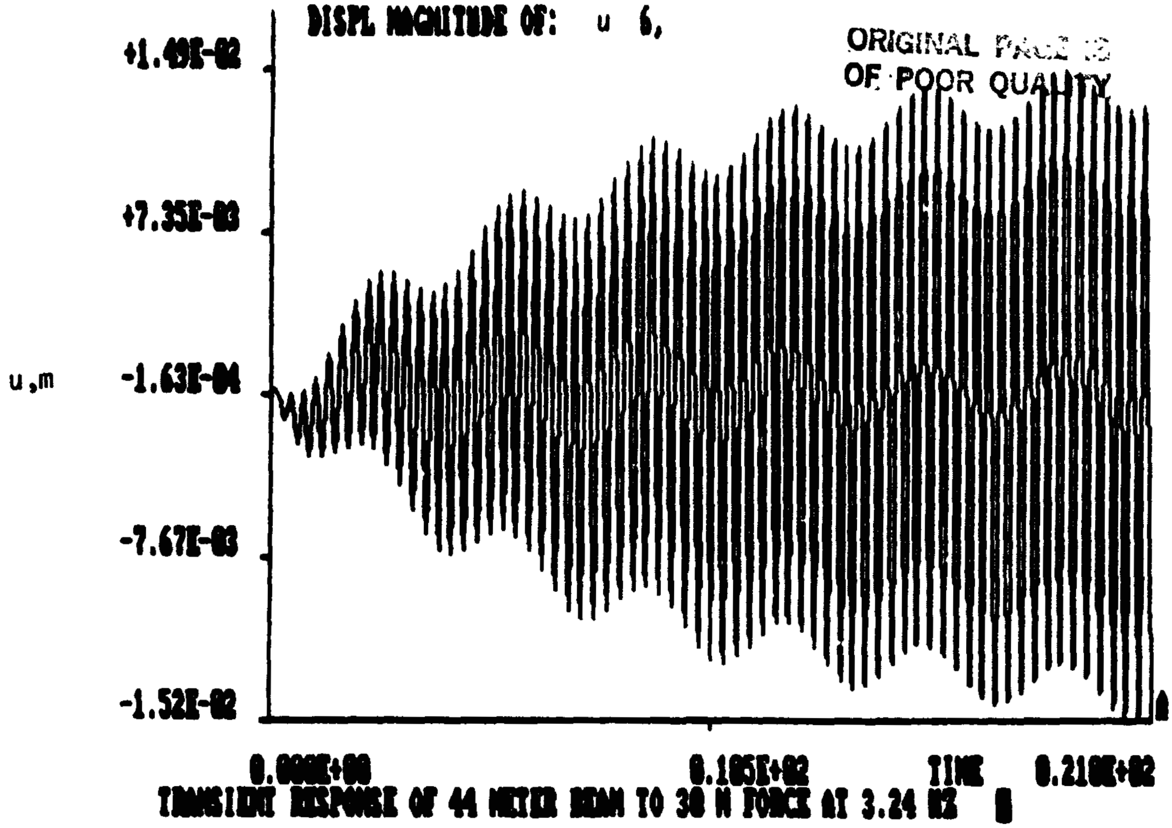


Figure 3.- Natural frequencies for 60.5 m and 44 m beams.



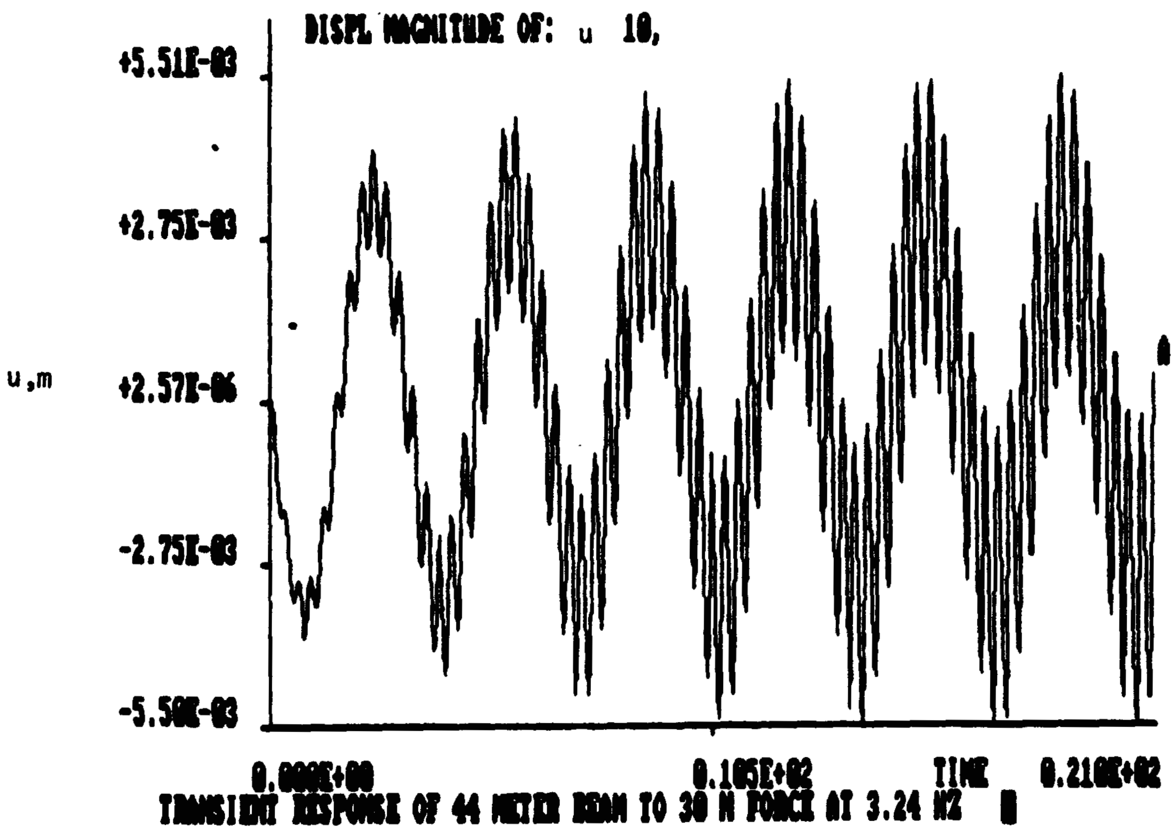


Figure 4.- Transient response of 44 m beam under cyclic tip force.

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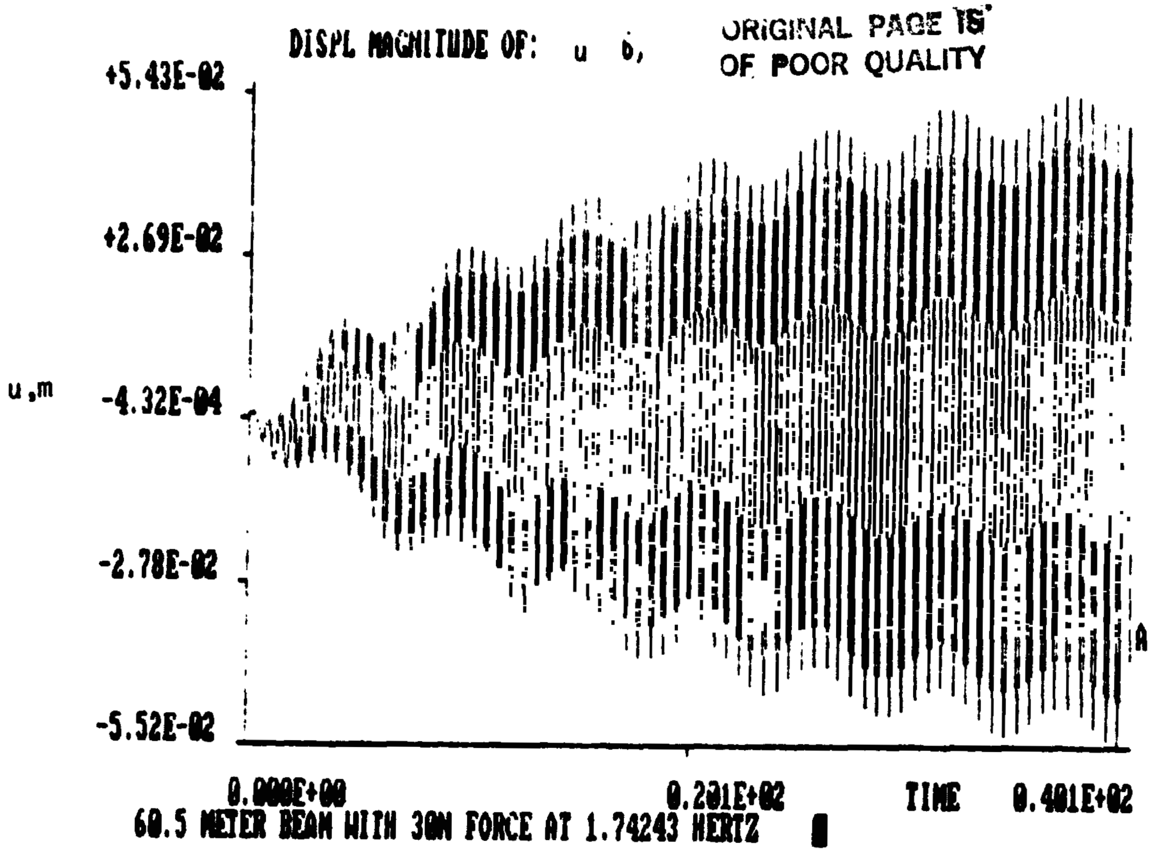

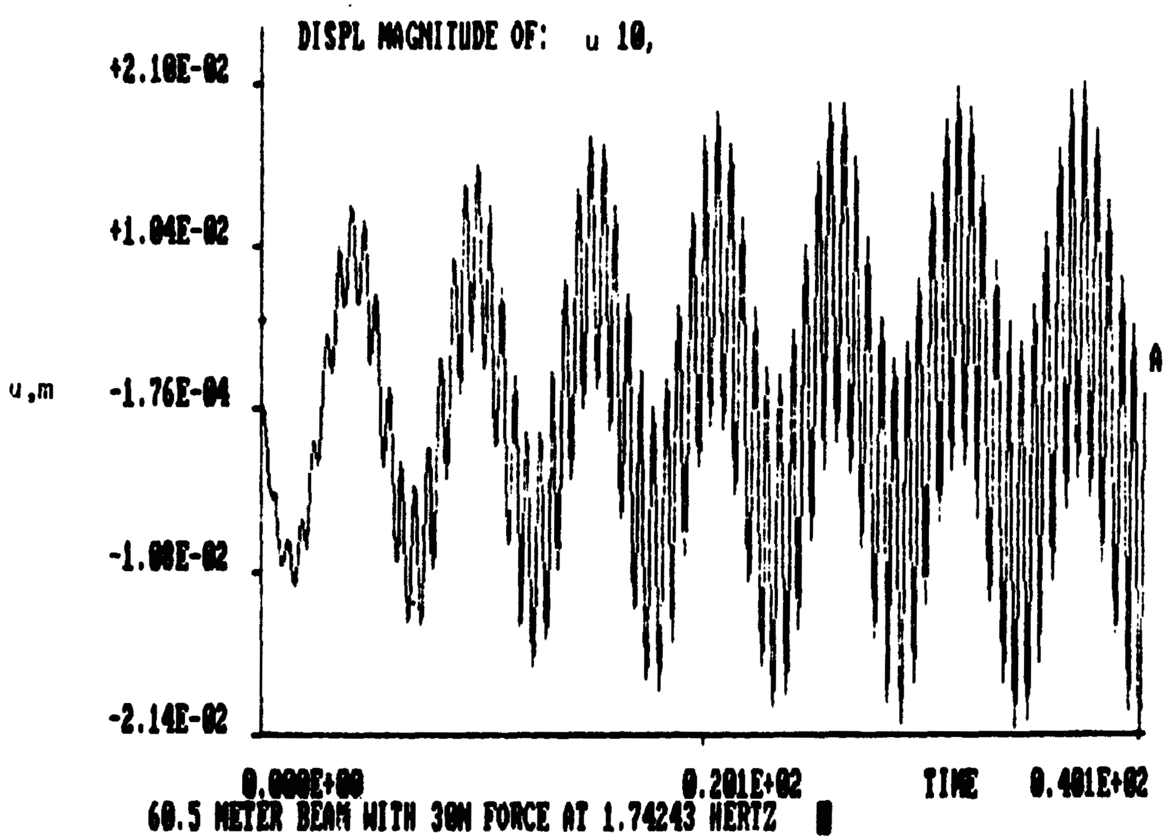

Figure 5.- Transient response of 60.5 m beam under cyclic tip force.

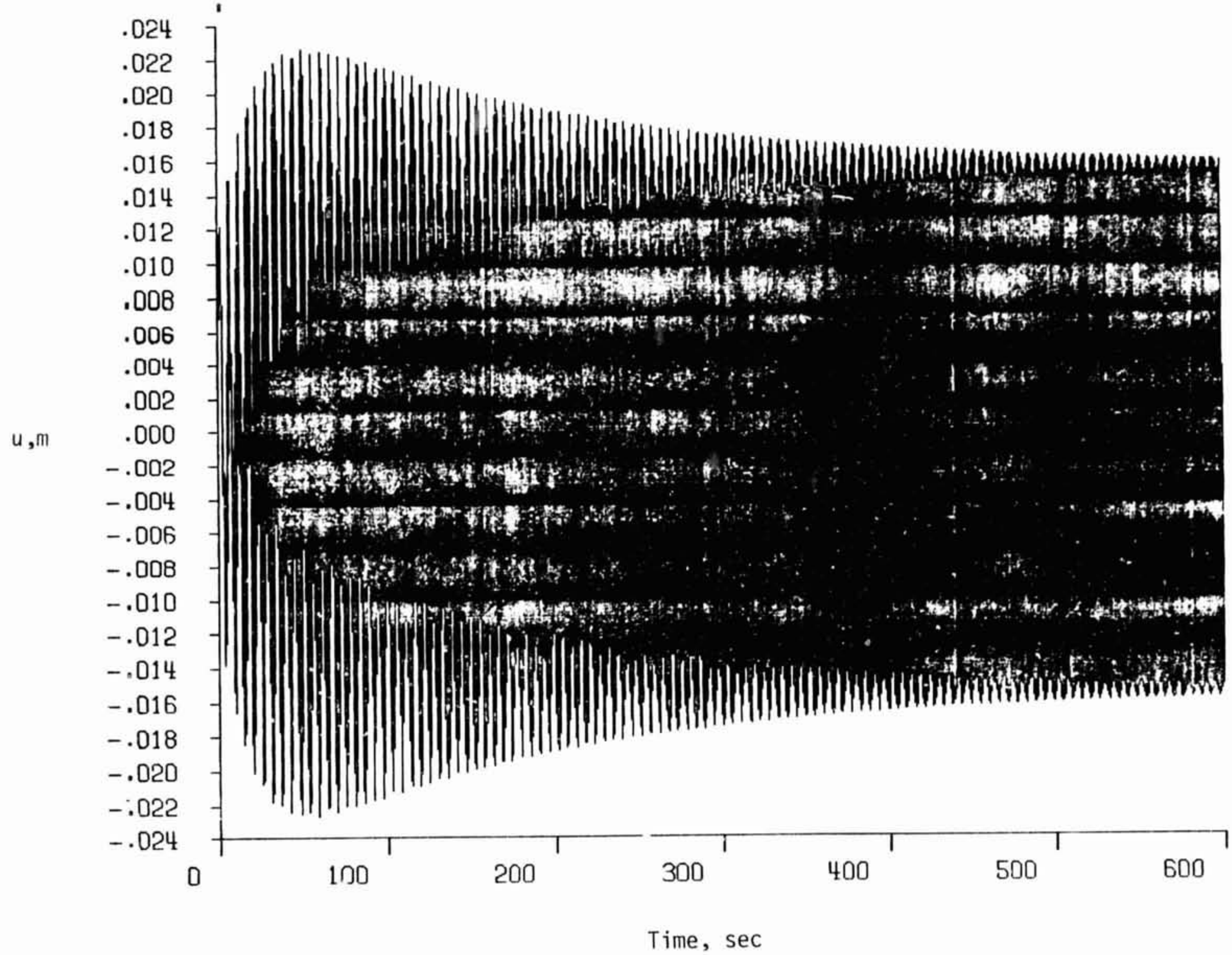

Figure 6.- Transient response of 60.5 m beam under 30 N force at 1.74 Hz.

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16. Abstract

Dynamic response of a candidate flexible beam for a space experiment on control of flexible structures is investigated. Studies of natural frequencies reveal a beam length in which torsion and bending frequencies virtually coincide. Eccentric tip mass causes small shifts in natural frequencies but introduces coupled torsional/bending mode shapes. Transient response studies indicate significant effects on tip responses of low damping and first bending mode excitation at higher frequencies. Steady state responses suggest displacement and acceleration measurements could be made up to 5-12 Hz for the actuator forces/torques assumed.

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