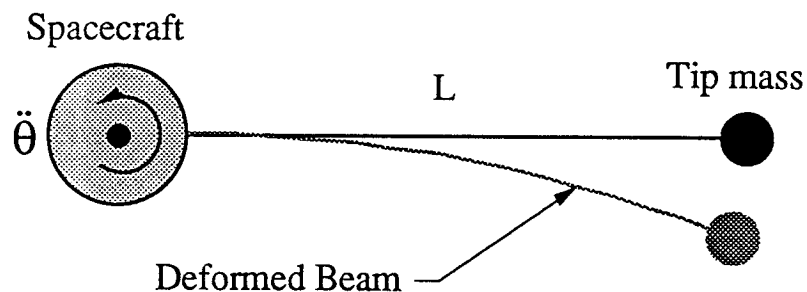*IN-18-CR*

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PRELIMINARY DESIGN METHOD FOR DEPLOYABLE SPACECRAFT BEAMS

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PRELIMINARY DESIGN METHOD FOR DEPLOYABLE SPACECRAFT BEAMS

by

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ABSTRACT
There is currently considerable interest in low-cost, lightweight, compactly packagable deployable elements for various future missions involving small spacecraft. These elements must also have a simple and reliable deployment scheme and possess zero or very small free-play. Although most small spacecraft do not experience large disturbances, very low stiffness appendages or free-play can couple with even small disturbances and lead to unacceptably large attitude errors which may involve the introduction of a flexible-body control system. A class of structures referred to as "rigidized structures" offer significant promise in providing deployable elements that will meet these needs for small spacecraft. The purpose of this paper is to introduce several rigidizable concepts and to develop a design methodology which permits a rational comparison of these elements to be made with alternate concepts.

INTRODUCTION
There is currently considerable interest in low-cost, lightweight, compactly packagable deployable structural elements for various future space missions involving small spacecraft[1]. In addition to these requirements, simplicity and reliability of deployment are of paramount concern. In many instances the concern over the cost and reliability of deployable components leads spacecraft designers to either not consider them at all, or to use existing deployable components with low stiffness or joint deadbands. In either case, spacecraft performance for such missions can be severely compromised due to the lack of well accepted, high performance deployable components.

Although most small spacecraft do not experience large disturbances, very low stiffness appendages can couple with even small disturbances and lead to unacceptably large attitude errors which may involve the introduction of a flexible modes control capability onboard which increases spacecraft cost. Thus, there exists a need in small spacecraft for stiff deployable components which are truly low cost and reliable.

The current commercially available SOA for deployable beam elements includes the unfurlable STEM, the continuous-coilable-longeron mast, the FASTMAST as used for the Tether satellite, and unfolding "Lazy-Tong" devices which deploy a few bays of panels such as on the SEASAT. The other approach used in deployment is to simply hinge panels or elements together with no supporting structure. None of these available deployable devices satisfies all of the desired requirements for the new generation of lighter, faster, and cheaper, small spacecraft.

1

A class of structures referred to as "rigidized structures[2]" offer significant promise in providing high performance structural components for the new small spacecraft. A large reflector based on inflatable and rigidized concepts is currently being built for an In-Step flight experiment[3] as a proof of concept for a microwave and VLBI antennas. Such reflectors have also been studied for optical interferometers[4] and solar concentrators[5]. The purpose of this paper will be to introduce several new "rigidizable" structural concepts, and to demonstrate their performance potential through systematic design studies and through comparisons with alternate concepts.

BEAM DESIGN METHODOLOGY

General Approach for Developing Beam Weight Equations

The primary purpose of this paper is to develop an approach to enable a rational comparison to be made of weight and diameter of different deployable beam concepts for small spacecraft. The four general beam concepts to be compared are as follows:

  a. Unfurlable BI-STEM[6]
  b. Coilable longeron[7]
  c. Space rigidizable organic matrix composite (inflatably deployed)[8]
  d. Unfurlable thin walled aluminum (inflatably deployed)[2,9,10]

For purposes of comparing the relative merits of the different beam concepts, the deployable beam is assumed to be cantilevered from a spacecraft and supporting a tip mass as shown in Sketch-a. The design methodology developed herein could readily be applied to other applications such as supporting a distributed solar array. Although there are numerous factors which contribute to the concept selection and design of deployable beams for spacecraft, attention in this paper is focused on two primary design drivers. These are 1), a lowest beam natural frequency constraint, and 2), a cantilever root angular acceleration constraint $\ddot{\theta}$ as shown in Sketch a.
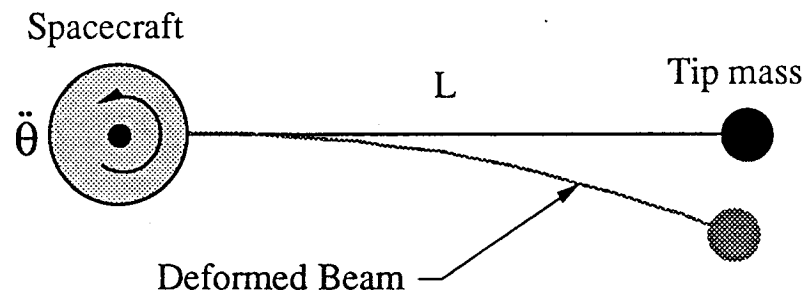
Sketch a.- Schematic of Spacecraft with Deployable Beam

The constraint on frequency is commonly imposed upon spacecraft components to deal with flexible control issues, while the constraint associated with a root angular acceleration imposes a loading that the beam must be able to withstand without failure. A general description of the four beams considered in this paper and how they are modeled are given in the next sections.

a. Unfurlable STEM.- This class of structures involves materials which are thin enough to be rolled up for packaging without yielding, and subsequently unfurled into the deployed state. The classic example of this type of structure is the STEM and BISTEM[6]. The STEM structure is a thin metallic sheet which is coiled into a compact cylindrical roll and deployed on-orbit. The design is such that no involved bending strains exceed yield. These structural elements unfurl into a long tubular shape, thus forming a deployed beam. The major shortcomings of these elements is that a slit is required along the length to accommodate low strain packaging, resulting in very low beam torsional stiffness and that the deployment mechanism is quite heavy. The BISTEM beam is composed of two interwoven STEMS which provides additional torsional stiffness from the resulting friction between the overlapping elements. Under some loading conditions a slippage can occur between the overlapping elements resulting in unwanted deformation or dynamic perturbations. In the present paper the STEM is treated as a simple steel tube with a wall thickness of 0.005" (5 mils).

b. Coilable longeron.- The coilable longeron beam is a highly used deployable beam and is well described in reference 7. The weight and performance equations for this beam are taken from ref. 7. The popularity of this beam arises from its high reliability and wide experience base. Its shortcomings are that is requires a relatively heavy canister and is limited in size to about 20" in diameter due to high straining in the stowed condition.

c. Space rigidizable organic matrix composite (inflatably deployed).- This concept is basically a simple tubular beam fabricated from a fiber fabric impregnated with a matrix that is rigidized[8] after pressure deployment in space. This concept is still in the development stage, however, it offers the promise of a very simple, lowcost, compactly packagable beam. In the present paper the tube is considered to be fabricated from a bi-directional KEVLAR fabric impregnated with a rigidizable matrix. The effective properties assumed for the beam material are: $E = 4x10^6$ psi, thickness = 0.011", and a weight density of 0.05 lb/in^3.

d. Unfurlable thin-walled aluminum (inflatably deployed).- This concept is basically a thin-walled aluminum tubular beam presuure deployed in space. For this approach the tubes are made from thin (~3 mil) low-yield-stress aluminum sandwiched between two thin layers of reinforced Kapton film for structural strength and initial inflatant containment[2,9]. The deployment is obtained by pressurization of the tube to a cylindrical shape. After deployment the pressure is increased to yield the thin aluminum into its final wrinkle-free state. The tube is then de-pressurized and remains in a cylindrical shape providing a high performance structural member. Although this concept is still in the development stage a full scale deployable solar array has been built and ground demonstrated[9]. This concept has the potential for being an extremely simple, low-cost, and reliable deployment system. The primary shortcoming of this concept is the low level of development that has occured in exploring different hybrid wall concepts. In the present paper the beam is simply considered to be fabricated from 0.003" thick aluminumwith a modulus of $10x10^6$ psi.

Weight of a Thin Walled Tubular Beam Subjected to Frequency and Root Moment Constraints

The weight of a thin walled tubular beam as shown in Sketch b can be written as:

$$W_{T.B.} = \rho AL = \rho(2\pi Rt)L \qquad (T1)$$

where ρ is the weight density of the beam material, and R and t are radius and thickness of the tubular beam respectively.
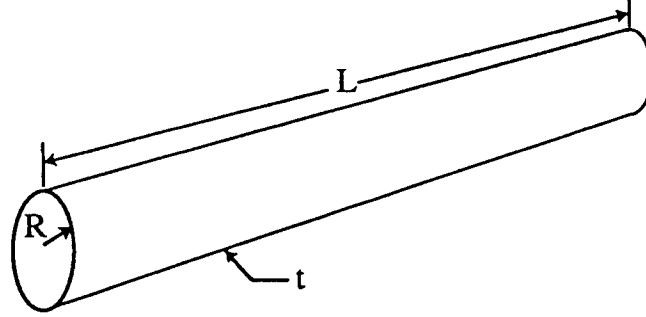
Sketch b. Schematic of tubular beam

There are two unknowns, R and t, in equation T1 that must be determined to assure that the beam performs as required. The two constraints, frequency and cantilever root angular acceleration provide the two equations for determining these two unknowns.

Frequency Constraint.- For a cantilever beam with a tip mass, m_{tip}, large enough that the mass of the beam can be neglected, the first natural bending frequency f is given by beam theory as:

$$f = \frac{1}{2\pi}\sqrt{\frac{3EI}{L^3 m_{tip}}} \qquad (T2)$$

where E is the extensional modulus of the tube material. If the tube is made of an orthotropic material, E is the extensional modulus in the long direction of the beam. The moment of inertia of a thin walled tubular beam is approximated by:

$$I = \pi R^3 t \qquad (T3)$$

Substituting for I from equation T3 into equation T2 yields the following equation governing R and t:

$$R^3 t = \frac{L\bar{f}}{3E\pi} \qquad (T4)$$

where $\qquad \bar{f} = L^2 m_{tip}(2\pi f)^2$

4

Beam Root Moment Constraint.- Using the rotational dynamic equilibrium equation, the moment M at the root of a cantilever beam with a large tip mass subject to a root acceleration $\ddot{\theta}$ is:

$$M = \alpha\ddot{\overline{\theta}} \tag{T5}$$

where $\quad\quad\quad \ddot{\overline{\theta}} = L^2 m_{tip}\ddot{\theta}$

and α is the dynamic overshoot factor due to a suddenly applied root acceleration. In this paper the dynamic overshoot factor α is conservatively taken as 2 for numerical comparisons.

The root stress σ in the cylinder due to this moment is:

$$\sigma = \frac{MR}{I} = \frac{MR}{\pi R^3 t} \tag{T6}$$

The failure mode in the tubular beam is assumed to be local wall buckling of the cylinder which is given by:

$$\sigma_{local} = C\frac{Et}{R} \tag{T7}$$

This is a generalization of the wall buckling equation for an isotropic cylinder which has a theoretical value of $C = 0.6$ In the present paper, the constant C is determined for the particular orthotropic wall construction being considered.

Combining equations T5, T6, and T7, a second equation relating R and t is obtained as:

$$Rt^2 = \frac{\alpha\ddot{\overline{\theta}}}{\pi CE} \tag{T8}$$

Closed Form Solution for Tubular Beam Weight.- A single equation governing the weight of a tubular beam can be found by substituting the expressions for t and R from equations T4 and T8 into the weight equation T1 to obtain:

$$W_{T.B.} = \frac{2(3)^{4/5}\pi^{2/5}}{3}(\alpha)^{2/5}\left(\frac{\rho}{E^{3/5}C^{2/5}}\right)L^{6/5}\left(\overline{f}\right)^{1/5}\left(\ddot{\overline{\theta}}\right)^{2/5} \tag{T9}$$

where the tube radius is given by

$$R = \left(\left(\frac{L\overline{f}}{3\pi E}\right)^2\frac{\pi CE}{\alpha\ddot{\overline{\theta}}}\right)^{1/5} \tag{T10}$$

and the tube thickness is given by

5

$$t = \frac{L\bar{f}}{3\pi ER^3} \qquad \text{(T11)}$$

Equation T9 relates the beam weight to the loading constraints $\bar{f}$ and $\bar{\bar{\theta}}$, and is useful for investigating how the beam weight varies as a function of the constraint values as well as the beam length, the overshoot parameter, and the material parameters. In this equation the radius r and the thickness t vary as a function of $\bar{f}$ and $\bar{\bar{\theta}}$. For the different tubular beam concepts considered in the present paper, there are practical constraints on the thicknesses due to the compact packaging constraints and these are dealt with in the next section.

Beam weight considering thickness contraints. - For each of the tubular beam concepts considered in the present paper there is a limitation on the tube wall thickness. This limitation is imposed in order to compactly package the material without excessive damage. To account for this thickness constraint, weight equations are derived in this section for both frequency and root moment constraints considering the thickness to be a constant.

The radius for a constant thickness t_c and for the frequency constraint is obtained directly from equartion T4 as

$$R = \left(\frac{L\bar{f}}{3E\pi t_c} \right)^{1/3} \qquad \text{(T12)}$$

This expression for R is then substituted into the weight equation T1 to obtain

$$W_f = \frac{2\pi^{2/3}}{3^{1/3}} \frac{\rho t_c^{2/3}}{E^{1/3}} L^{4/3} \bar{f}^{1/3} \qquad \text{(T13)}$$

The radius for a constant thickness t_c and for the root moment constraint is obtained directly from equation T8 as

$$R = \frac{\alpha\bar{\bar{\theta}}}{\pi C E t_c^2} \qquad \text{(T14)}$$

This expression for R is then substituted into the weight equation T1 to obtain

$$W_{rm} = \frac{2\rho L \alpha\bar{\bar{\theta}}}{C E t_c} \qquad \text{(T15)}$$

For a given design condition the greater of the two weights as given by equations T13 and T15 must be chosen. It should be noted that the weight given by equation T13 increases for increasing thickness t_c while the weight given by equation T15 decreases for increasing thickness t_c.

6

Weight of a Coilable Longeron Beam Subjected to Frequency and Root Moment Constraints

The weight $W_{C.L.}$ of a coilable longeron beam as shown in Sketch c, is taken from reference 7 as:

$$W_{C.L.} = 3.4\left(3\rho A_{\ell} L\right) \tag{C1}$$

where A_{ℓ} is the area of each longeron which is positioned at a radius R from the beam's centroid. For the coilable longeron beam, the two unknowns to be determined from the frequency and root rotational acceleration constraints are A_{ℓ} and R.

Cross-Section
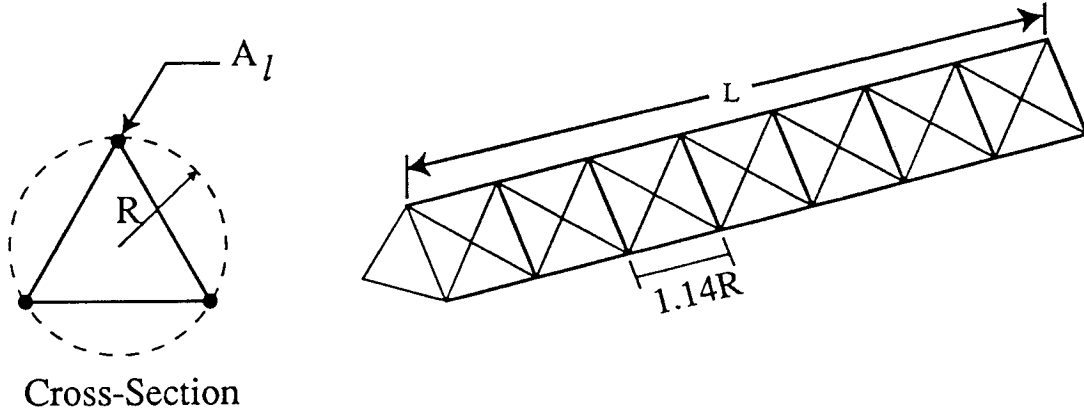
Sketch c. Schematic of coilable longeron beam

The quantities inside the parentheses of eq. C1 are the weight of the three longerons and the factor 3.4 is an empirical constant which accounts for the beam's battens, diagonals and joints. This empirical constant is taken from reference 7 where it was determined by curve fitting data from several coilable longerons beams which had been built.

Frequency Constraint.- The frequency equation for a cantilevered coilable longeron beam with a large tip mass is taken to be the same as that for the tubular beam and is given by equation T2. The bending moment of inertia of a three longeron beam about an axis passing through its centroid is given by:

$$I = 1.5 A_{\ell} R^2 \tag{C2}$$

It should be noted that the moment of inertia of a three longeron beam is independent of the angle of the axis which passes through the beam's centroid. In other words the beam behaves elastically similar to a cylindrical beam with the same radius R, and the same amount of material at that radius.

Substituting the expression for I from eq. C2 into the frequency equation T2, results in one equation governing R and A_{ℓ} as:

$$R^2 A_{\ell} = \frac{L\bar{f}}{3(1.5)E} \tag{C3}$$

7

Root Moment Constraint.- The root moment is taken to be the same as that for the tubular beam and is given by eq. T5. For the coilable longeron beam, failure is assumed to be buckling of the root longeron due to compression from the root moment. Longeron buckling is taken as the simple support Euler load as:

$$P_{Euler} = \frac{\pi^2 EI_\ell}{(1.14R)^2} \tag{C4}$$

where the beam bay length is given in reference (Crawfordf) as 1.14R, and the moment of inertia I_ℓ of a longeron is:

$$I_\ell = \frac{A_\ell^2}{4\pi} \tag{C5}$$

where

$$A_\ell = \frac{\pi d^2}{4}$$

Combining equations C4, C5, T5, and T6 yields a second equation governing R and A_ℓ as:

$$\frac{R}{A_\ell^2} = \frac{1.5\pi E}{4\alpha(1.14)^2 \ddot{\theta}} \tag{C6}$$

Equation C6 represents a second equation which governs the coilable longeron radius R and the longeron area A_ℓ. In addition to equations C3 and C6, an additional constraint[7] must be imposed upon the longeron diameter to accomodate packaging as discussed in the next section.

Longeron Packaging Constraint.- For elastic packaging[7], the longeron diameter, d, must be limited as follows:

$$\frac{d}{2R} = \varepsilon = \frac{\sqrt{\frac{4A_\ell}{\pi}}}{2R} \tag{C7}$$

where ε is the longeron allowable strain. In reference 7 this strain value was taken as 0.0133 for fiberglass and is the same value used in the present paper. It should also be pointed out there is a factor of 2 error in equation 12 of reference 7. The left hand side of eq. 12 should read d/2R as in eq. C7 rather than d/R.

Coilable Longeron Beam Weight. - Equations C3, C6, and C7 represent three equations for the two unknowns R and A_ℓ, thus the design is overspecified and must be separated into three possible design cases to determine which two of the three conditions govern. The three possible cases are: (1) impose the frequency constraint eq. C3 and the root

moment constraint eq. C6, (2) impose the root moment constraint eq. C6 and the stowage constraint C7, and (3) impose the frequency constraint eq. C3 and the stowage constraint eq. C7. Case 3 is never critical, thus, the higher of the two weights resulting from cases (1) and (2) must be taken as the coilable longeron weight and are given as follows:

(1) Frequency/root-moment constrained, coilable longeron beam weight

From equations C3 and C6, the following two expressions are obtained for A_ℓ and R:

$$A_\ell = \left(\frac{L\bar{f}}{3(1.5)E} \left(\frac{4\alpha\bar{\bar{\theta}}(1.14)^2}{1.5\pi E} \right)^2 \right)^{1/5} \quad \text{(C8)}$$

and

$$R = \frac{1.5\pi E}{4\alpha(1.14)^2 \bar{\bar{\theta}}} A_\ell^2 \quad \text{(C9)}$$

A single equation governing the weight of the coilable longeron beam can be found by substituting these expressions for R and A_ℓ into C1 to obtain:

$$W_{C.L.} = 3.4(3) \left(\frac{1}{3(1.5)} \left(\frac{4(1.14)^2}{1.5\pi} \right)^2 \right)^{1/5} (\alpha)^{2/5} \left(\frac{\rho}{E^{3/5}} \right) L^{6/5} (\bar{f})^{1/5} (\bar{\bar{\theta}})^{2/5} \quad \text{(C10)}$$

(2) Root moment/stowage constrained, coilable longeron beam weight

From eqs. C6 and C7 an expression for the radius is

$$R = \left(\frac{4 \times 1.14^2 \alpha\bar{\bar{\theta}}}{1.5\pi^3 E\varepsilon^4} \right)^{1/3} \quad \text{(C11)}$$

and the longeron area is found from equation C7. Substituting these into equation C1 yields the following equation for the coilable longeron beam weight as:

$$W_{C.L.} = \frac{3.4(12)}{\pi} \left(\frac{1.14^2}{2 \times 1.5} \right)^{2/3} \frac{\rho L \alpha^{2/3} \bar{\bar{\theta}}^{2/3}}{E^{2/3} \varepsilon^{2/3}} \quad \text{(C12)}$$

BEAM WEIGHT RESULTS AND DISCUSSION

A general comparison of the various deployable beams concepts constructed of different materials is difficult to make over a wide range of the frequency and loading design parameters $\bar{f}$ and $\bar{\bar{\theta}}$. The reason for this is that the different beams are governed by practical constraints such as limitations on material thickness which in turn are a function

of the level of the design parameters. In order to obtain insight into the relative weight and stowage efficiency of the deployable beam concepts considered in the present paper, a specific set of design requirements are selected which are considered to be representative of a range of typical spacecraft conditions. The first requirement considered is a root moment constraint imposed by an angular acceleration of the spacecraft. In the present paper this root moment is taken to be the same as that provided by a 0.03 g equivalent lateral static loading. This requirement is satisfied by equating the moment due to a 0.03g lateral load to the moment caused by a root angular acceleration as given by equation T5 and is written as:

$$M_{tip}(0.03g)L = L^2 M_{tip}\ddot{\theta} \qquad (R1)$$

Solving for $\ddot{\theta}$ yields:

$$\ddot{\theta} = \frac{0.03g}{L} \qquad (R2)$$

This equation shows that for this particular design condition, the allowable spacecraft rotational acceleration is simply a function of 1/L. The other design requirement considered is that of a lowest natural bending frequency constraint. Since the value of this constraint is typically not well defined, it is varied in this study to determine its impact on structural weight and beam diameter.

The other spacecraft input needed to make a design study is the tip mass on the beam. To obtain representative mass values for small spacecraft, the inflatable solar array of ref. 9 was used as an example. This solar array was about 11.5' (3.5) meters long and weighed about 7 pounds. Since there are two beams that support this weight, half of the weight is assigned to each beam or $m_{tip} = 3.5lb/386in/sec^2 \sim .01 \ lb \cdot sec^2/in$. For the current design study this mass is assumed to vary as a function of the square of the beam length to simulate area masses such as that associated with solar arrays.

To obtain insight into the relative weights and diameters of the deployable beams considered in the present paper, four beam lengths (3.5, 7, 14, and 28 meters) were investigated. For each length the beam weights and beam diameters are plotted as a function of natural frequency. The natural frequency was varied from .02 to 1 Hz to cover the range of interest for most spacecraft. The properties used for each of the four beams are presented in the following table.

	E, psi	ρ, lb/in^3	t_c, in
Aluminum	10×10^6	.1	.003
Rigidizable	4×10^6	.05	.011
Steel STEM	30×10^6	.3	.005
Coilable Longeron	7.5×10^6	.07	-----

For all calculations in this study the dynamic overshoot parameter α was taken as 1, and the local wall buckling constant C from eq. T7 was taken as 0.2.

L=3.5 meters. - For this beam length the angular acceleration is found from equation R2 to be approximately 0.04 rad/sec^2 and the tip mass is 0.01 $lb \cdot sec^2/in$ (3.86 lbs). The weights for the four deployable beams considered in this paper are shown in figure 1, and the associated beam diameters are shown in figure 2. In figure 1 it can be seen that the 3 mil aluminum beam is the lightest over the low frequency range, while the coilable longeron beam is lighter for frequencies greater than 0.35 Hz. For this relatively short length the three tubular beams are governed primarily by the frequency constraint, while the coilable longeron beam is governed by the root moment/stowage constraint at low frequencies and by root moment/frequency for the higher frequncies. At very low values of frequency (less than 0.1 Hz) the aluminum beam is governed by the root moment strength constraint.

For this short beam length the beam weights are quite low for all four concepts, however, as seen in figure 2 there is quite a difference in beam diameters. For the three tubular beams the required diameter is 3 inches or less while the required diameter for the coilable longeron beam ranges from 4 to 7 inches. The beam diameter will greatly influence the stowage volume and deployment weight required for the deployable beams. In the present paper the only weight considered for the deployable beams is that required for structural performance. For all of the beams there will be a system weight required to accomplish deployment. For the STEM and the coilable longeron beams there is a mechanical deployment canister required that is typically several times the weight of the beam structure[7]. For the inflatable beams there is the weight associated with the pressurization system that must be considered. The details of these auxiliary weights are beyond the scope of the present paper, however, these weights will be a strong function of beam diameter.

L=7 meters. - For this beam length the angular acceleration is 0.02 rad/sec^2 while the tip mass is .04 $lb \cdot sec^2/in$ (15.44 lbs). These beam weights are shown in figure 3 while the corresponding beam diameters are shown in figure 4. The aluminum beam is the lightest over much of the frequency range, however, the strength cutoff for the aluminum beam is now up to 0.3 Hz. The strength constrained weight of a tubular beam is inversely proportional to the assumed thickness t_c as shown by equation T15. Thus, if the aluminum thickness could be doubled to 6 mils, the weight of the aluminum beam in the low frequency range would be reduced by a factor of two. The tubular beam diameters are seen from figure 4 to be about one half of the coilable longeron beam diameters. In fact the coilable longeron diameter of almost 20 inches is at the size limit of practicality for these beams.

L=14 meters. - For this beam length the angular acceleration is 0.01 rad/sec^2 while the tip mass is .16 $lb \cdot sec^2/in$ (61.76 lbs). These beam weights are shown in figure 5 while the corresponding beam diameters are shown in figure 6. For this length the rigidizable material beam now is the lowest over a large portion of the frequency range. For these longer lengths it will probably be necessary to limit the design frequency to 0.4 Hz or less to keep the beam weights practical. The corresponding diameters for these beams are shown in figure 6. The coilable longeron beam diameter is out of the practical design range for frequencies greater than 0.4 Hz while the aluminum beam is probably

impractically large over the entire range. Even the rigidizable begins to assume impractical diameters for frequencies above 0.4 Hz.

L=28 meters. - For this beam length the angular acceleration is 0.005 rad/sec^2 as given by equation R2. However, for this length the beam tip mass was not increased over that for the 14 m beam. If the tip mass were increased as a function of a square of the length this would result in a tip weight of about 240 lbs. This was not considered to be practical for most applications so the tip weight was kept at 61.76 lbs. It should also be pointed out that this 28 m length is the same as the IAE support boom length[3]. In fact this length and tip mass are representative of design conditions for large inflatable reflector applications. The beam weights for this length are shown in figure 7 and the corresponding diameters are shown in figure 8. In figure 7 it can be seen that the rigidizable beam is lightest over most of the practical frequency range. Because of the rapid increase in beam weight and diameter with frequency, the design frequency for such structures would probably have to be restricted to 0.2 Hz or less.

Weight as a function of length - In figure 9 the weights of the four beam concepts are plotted as a function of length for a fixed natural frequency of 0.2 Hz. This figure demonstrates that the rigidizable material beam is quite efficient for the longer lengths. In figure 10 the same weight curves are presented with the addition of a 0.006" thick aluminum tubular beam. As can be seen from the figure this thickness results in an aluminum beam with the same efficiency as the rigidizable material beam. Because of the relatively simple deployment process for the aluminum beam, a research effort should be conducted to determine if such a beam could be developed.

CONCLUDING REMARKS

The purpose of this investigation was to develop and demonstrate a design methodology for tubular, rigidizable, space beam structures. This methodology was applied to a new class of rigidizable beams to permit a rational comparison with alternate deployable concepts. Specifically the rigidizable beams were compared with the STEM and coilable longeron beams on a weight and diameter basis.

A series of closed-form equations were developed for the weight and diameter for each of the concepts for the condition of a long beam cantilevered from a spacecraft with a tip mass. The two design requirements considered were a lowest natural frequency constraint and a root moment constraint imposed by a spacecraft angular acceleration. Although it is difficult to draw completely general conclusions as to the relative efficiency of the different beam concepts, representative small spacecraft operational conditions were assumed to enable a comparison to be made.

The two primary rigidizable concepts investigated were a 0.011" thick KEVLAR fabric impregnated with a rigidizable matrix and a 0.003" thick aluminum tube which is rigidized by pressure yielding the material. Beam lengths ranging from 3.5 m to 28 m were investigated for a frequency range from 0.02 Hz to 1 Hz. The strength constraint imposed was that the beam be required to withstand a 0.03g lateral loading. The beams were assumed to have a tip mass that was larger than the mass of the beam. This tip mass

was inertially similar to the mass of a distributed solar array. Results from this study led to the following conclusions:

1) Because of the discrete practical thickness constraints imposed on the different tubular concepts the active design constraint, frequency or strength, is a function of beam length and required frequency. For the shorter lengths and higher frequency requirements the frequency constraint is active, while for longer lengths and lower frequency requirements the strength constraint is active.

2) The three tubular beams investigated, the KEVLAR rigidizable, the aluminum, and the steel STEM all have significantly smaller diameters than the coilable longeron beam.

3) For shorter length applications the 3 mil aluminum beam is the lightest and most compact for low natural frequency requirements.

4) For longer length applications the KEVLAR rigidizable beam is the lightest and most compact.

5) If thicker (~ 0.006") aluminum beam concepts could be developed, they would be very efficient over the entire range of parameters investigated. Since the rigidizable aluminum beam is so conceptually simple, it is recommended that alternate wall constructions be investigated to extend its range of application.

6) The closed-form weight and diameter equations developed herein enable a rational assessment to be made of the effect of material properties and thicknesses on deployable beam performance. For example, for frequency designed beams the weight is proportional to $1/E^{1/3}$ while for strength designed beams the weight is proportional to $1/E$. Such knowledge permits a quick assessment of the relative performance offered by alternate material systems.

7) The design methodology developed herein permits a rational assessment of the effect of spacecraft requirements (frequency and strength) on deployable beam weight and diameter. It is recommended that this methodology be used early in the design process to assist in establishing rational and reasonable spacecraft design requirements. Conducting a thorough sensitivity study of deployable beam weight and diameter to spacecraft design requirements early in the design process should lead to the most robust design at the lowest cost in terms of weight and stowage efficiency.

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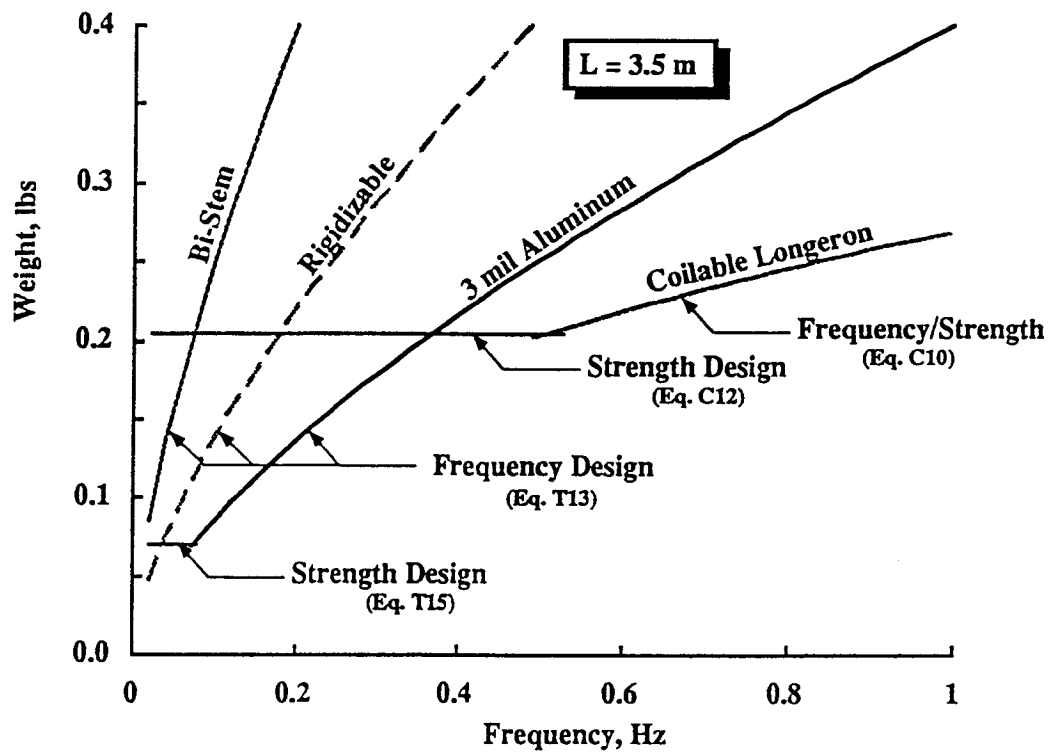
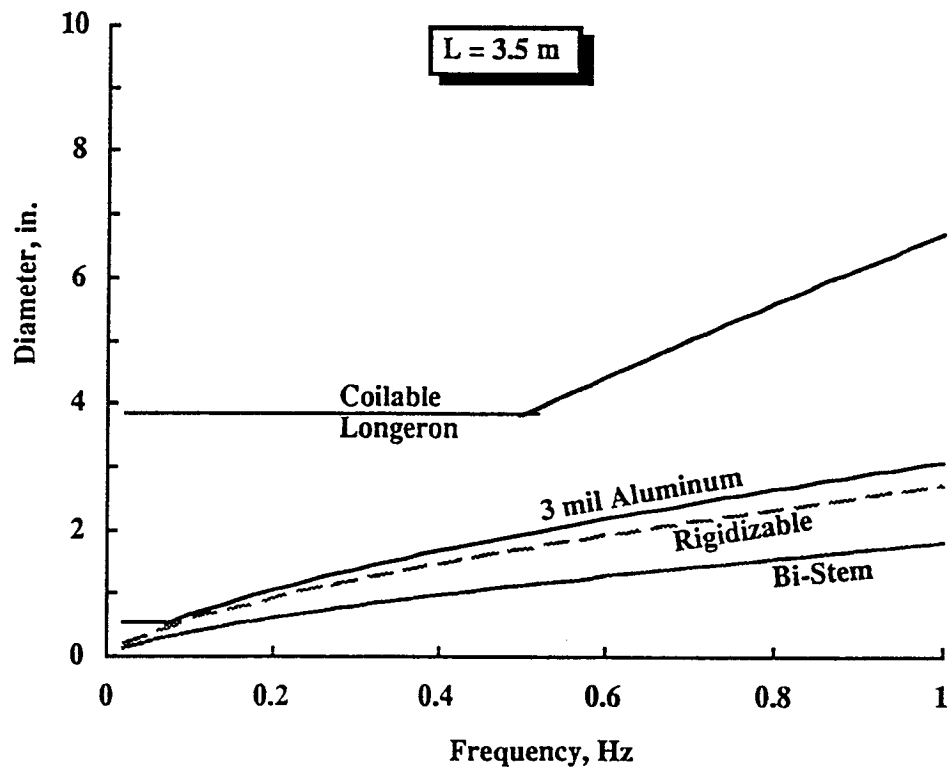Figure 1. - Deployable beam weight as a funtion of frequency for L = 3.5 m.

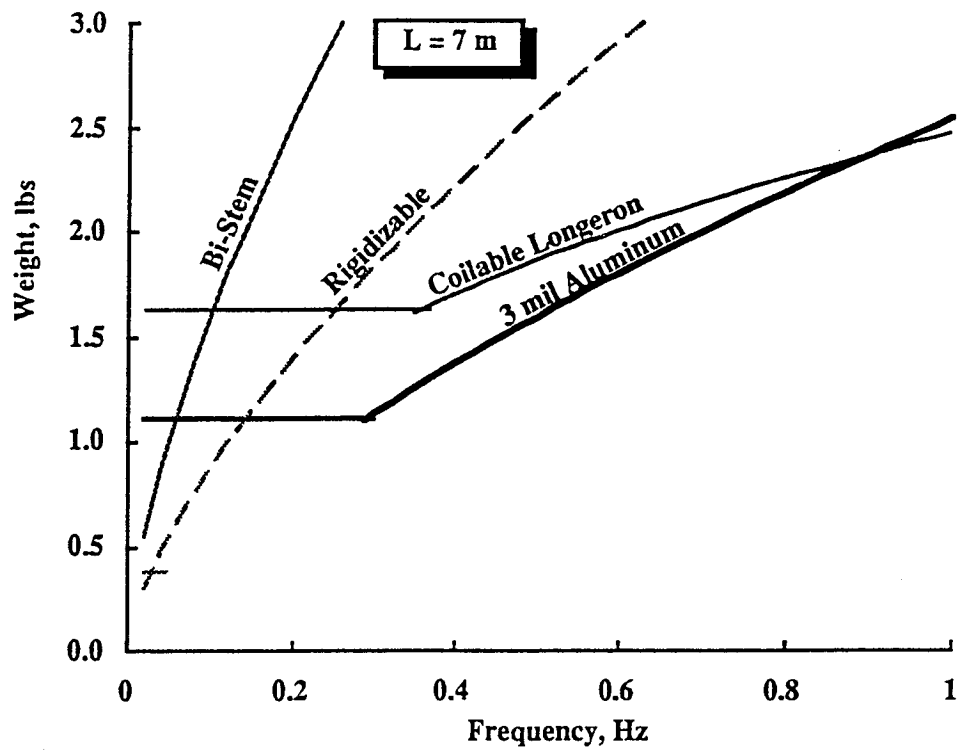Figure 2. - Deployable beam diameter as a funtion of frequency for L = 3.5 m.

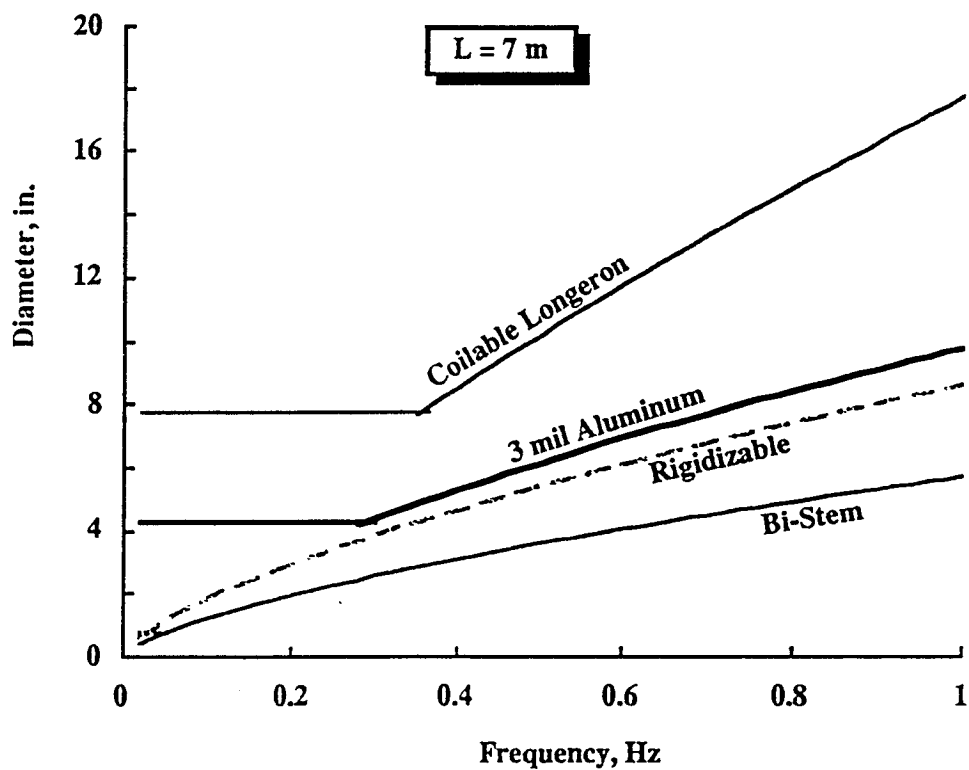Figure 3. - Deployable beam weight as a funtion of frequency for L = 7 m.

Figure 4 - Deployable beam diameter as a funtion of frequency for L = 7 m.

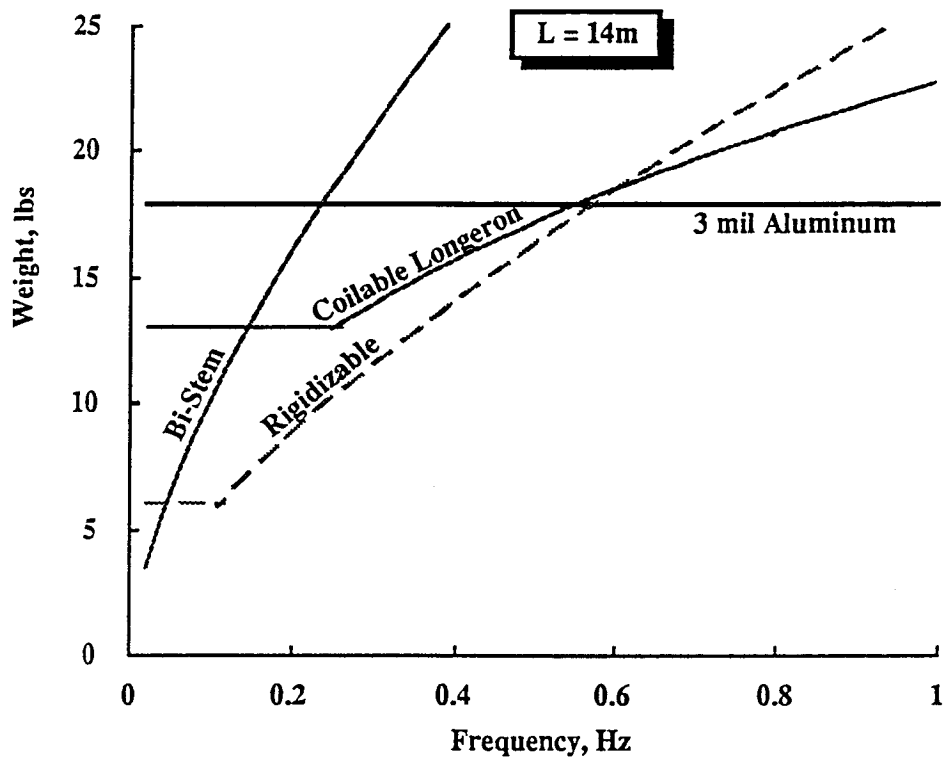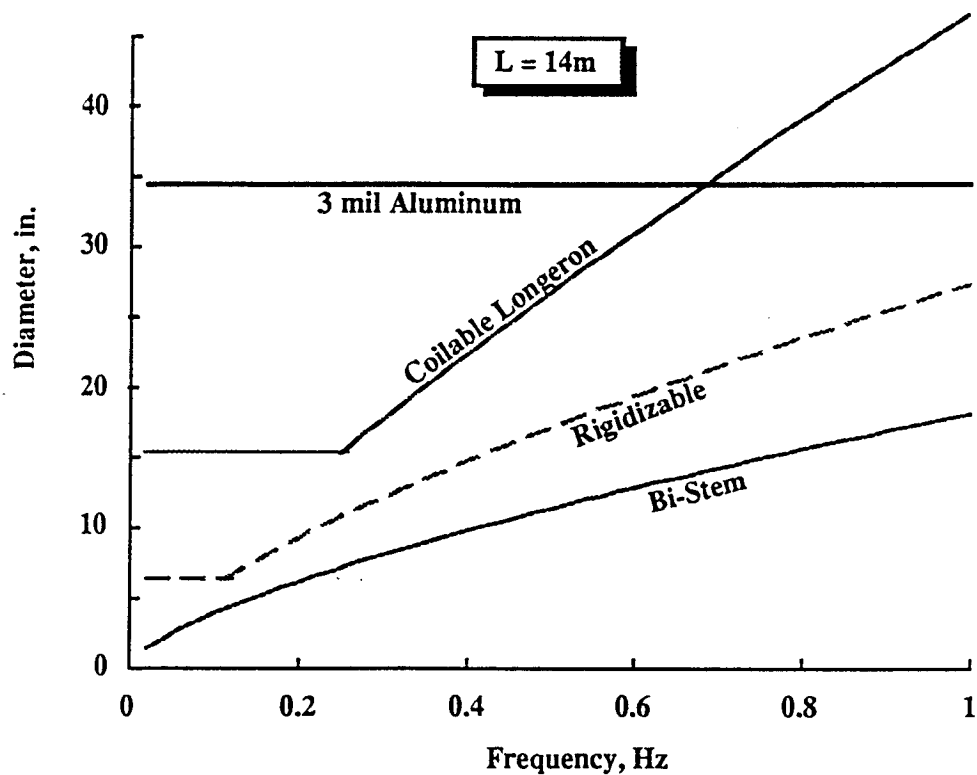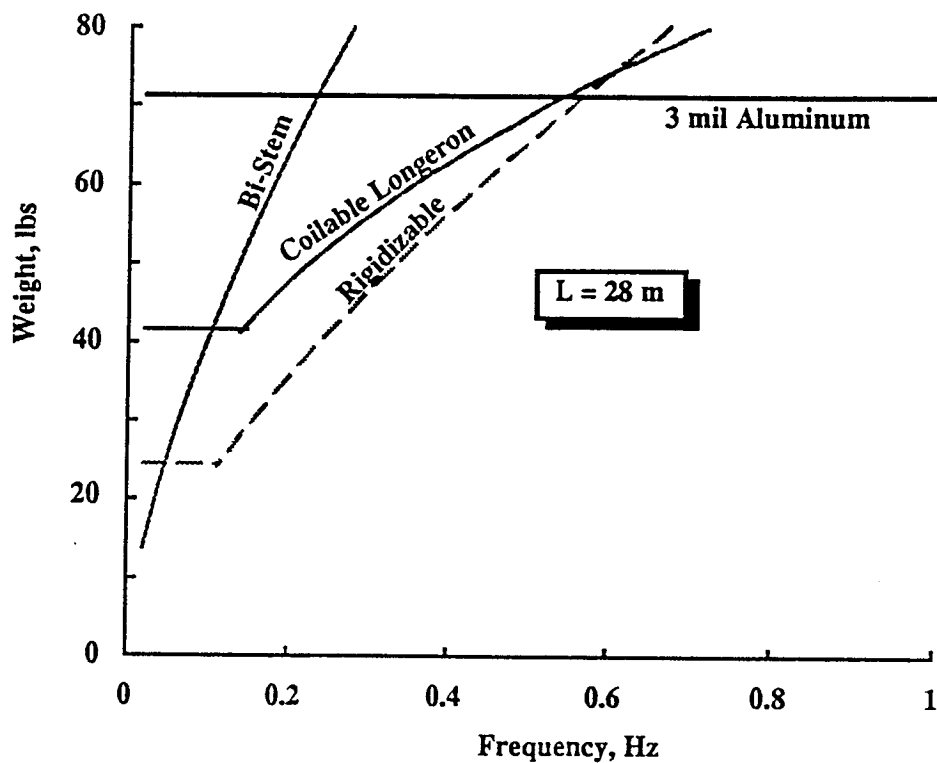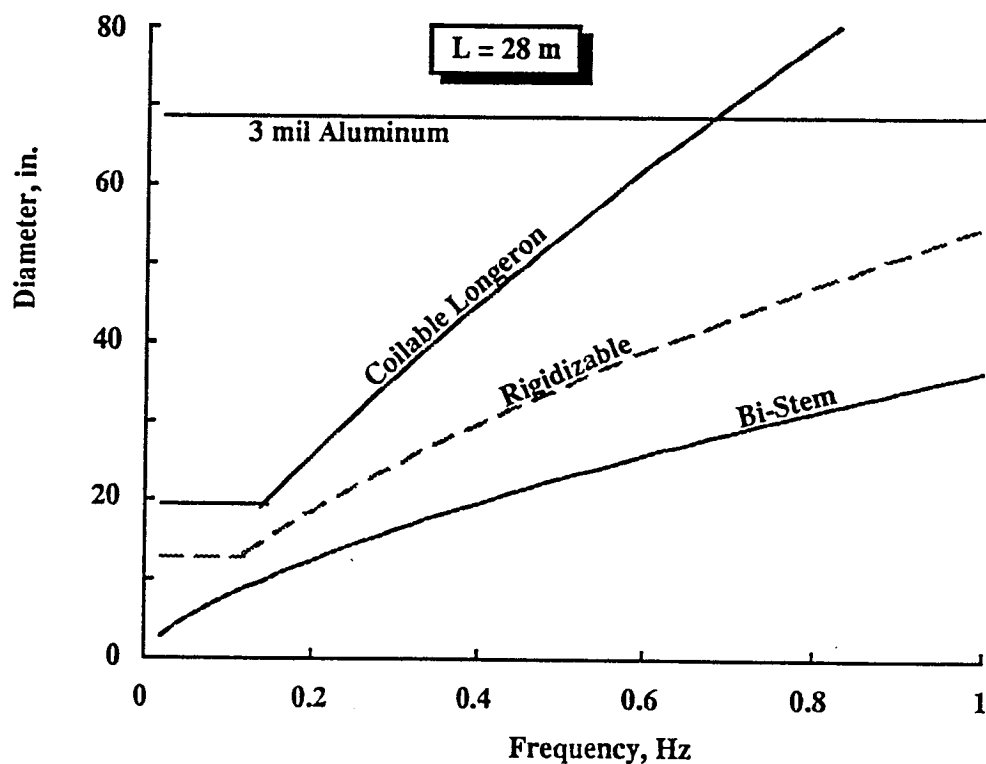Figure 5. - Deployable beam weight as a funtion of frequency for L = 14 m.

Figure 6. - Deployable beam diameter as a funtion of frequency for L = 14 m.

Figure 7. - Deployable beam weight as a funtion of frequency for L = 28m.

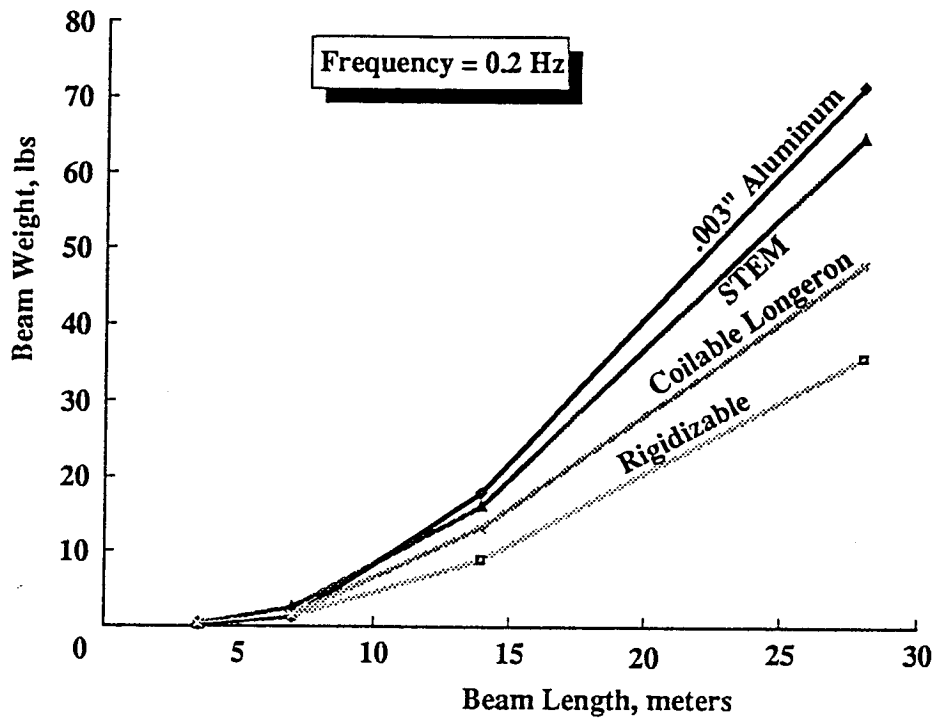Figure 8. - Deployable beam diameter as a funtion of frequency for L = 28m.

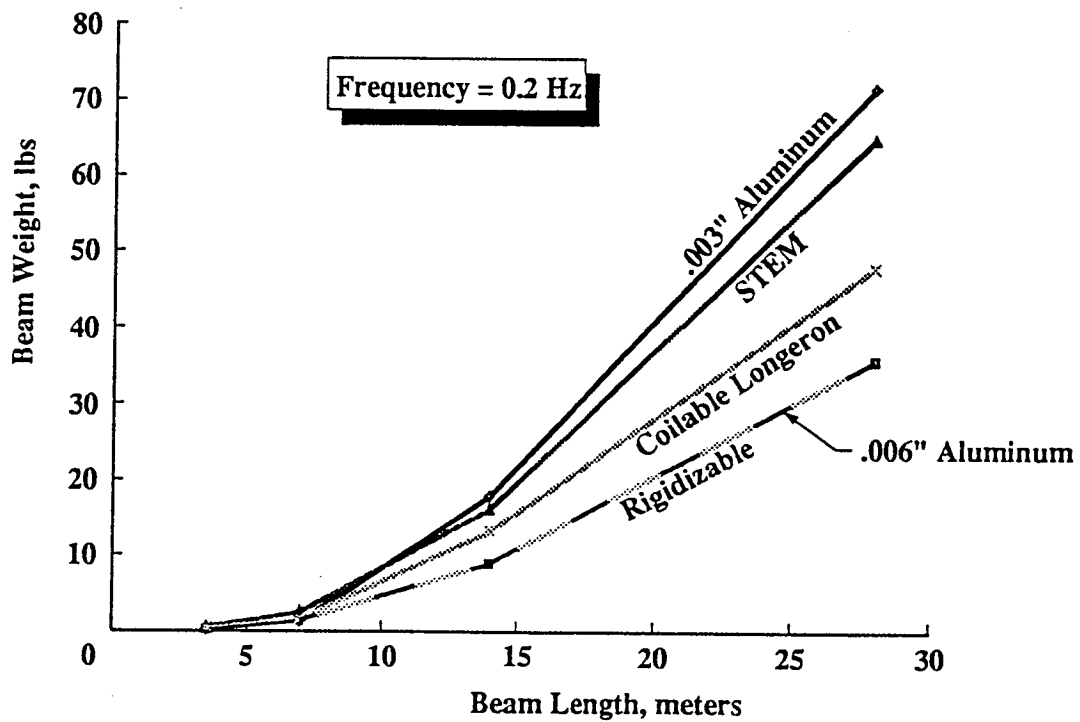Figure 9. - Beam weight as a function of length for f = 0.2 Hz.

Figure 10. - Beam weight as a function of length for f = 0.2 Hz with additional aluminum curve for .006" thickness wall.