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ASTRO

RESEARCH
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CARPINTERIA, CALIFORNIA

EFFICIENT STRUCTURES
FOR GEOSYNCHRONOUS SPACECRAFT
SOLAR ARRAYS

PHASE I, II, and III FINAL REPORT

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SECTION 1

INTRODUCTION

The purpose of this program is to evaluate structural concepts for deploying and supporting lightweight solar-array blankets for geosynchronous electrical power. The following discussion is based on material contained in reference 1.

Communication by geosynchronous satellites may require power systems in excess of 20 kW, with efficiencies from 60 W/kg to over 400 W/kg. The state-of-the-art of the present hardware is considered to be the Solar Electric Propulsion System (SEPS) array, which has an array effectiveness of 66 W/kg.

The SEPS blanket surface has a mass density of 0.90 kg/m^2 and a power density of 100 W/m^2. The blanket mass is 113 kg and the structure mass is 76 kg. With the advent of the NASA-OAST ultrathin (50 μm) silicon solar cell, significantly higher solar-array blanket specific power became possible. (A solar-array blanket is the celled area of the array but without any supporting or deploying structure.) Blanket specific powers ranging from 200 to over 700 W/kg appear achievable as the cell, cover, and interconnect technologies mature. Three somewhat arbitrary blanket designs are specified in the JPL Statement of Work (Table I), representing three stages of blanket development. (For blanket design details see reference 2.) To derive the maximum benefit from high-performance blankets, it is necessary to develop new structural concepts, especially since existing structural designs are not entirely compatible with these high-technology blankets. The resulting blanket and structure combination achieves the high specific power levels necessary to support the projected requirements of the growing number of launch mass constrained geosynchronous missions.

TABLE I. DESIGN PROGRESSION COMPARISONS FOR 126-m^2 ARRAYS*.

PARAMETER	SEPS BASELINE	DESIGN 1	DESIGN 2	DESIGN 3
Array blanket	0.90 kg/m^2	0.64 kg/m^2	0.42 kg/m^2	0.24 kg/m^2
	100 W/m^2	135 W/m^2	189 W/m^2	189 W/m^2
Array power	12.5 kW	17 kW	23.9 kW	23.9 kW
Blanket mass	113 kg	81 kg	53 kg	30 kg
Structure mass	76 kg	($\leq$ 54 kg)	($\leq$ 35 kg)	($\leq$ 21 kg)
$\dfrac{\text{Blanket mass}}{\text{Blanket mass+structure mass}}$	0.6	($\geq$ 0.6)	($\geq$ 0.6)	($\geq$ 0.6)
$\dfrac{\text{Array power}}{\text{Array mass}}$	66 W/kg	($\geq$ 125 W/kg)	($\geq$ 270 W/kg)	($\geq$ 470 W/kg)
Length to width ratio	7.9	TBD	TBD	TBD
Natural frequency	0.05 Hz	TBD	TBD	TBD

*Values in parentheses indicate Statement of Work specified values.

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One of the primary concerns of these new low-mass structural concepts is structural stiffness. Structural stiffness, as indicated by high natural design frequency, is important so as to avoid interaction with the attitude control system and to provide strength during transient events such as eclipses. SEPS has a 0.05-Hz natural frequency. While adequate for thicker solar cell arrays, a higher natural frequency is desirable for a thin cell array.

Natural frequencies are increased by deepening the structure, by using materials of higher modulus to density ratio, and by lowering the array aspect ratio. Blanket stiffness is obtained by tensioning the blanket, by making periodic attachments to the support structure, and by using lateral stiffening beams.

Since the solar blanket itself is an expensive and fragile component, its protection is of primary concern. To minimize blanket handling, the blanket should be assembled to the structure late in fabrication. Launch loads on the blanket must be fully reacted by a containment structure. Deployment must be slow and controlled so that dynamic loadings on the blanket are minimized.

The following sections present a list of symbols used in this report.

1.1 SYMBOLS

A	cross-sectional area (m^2)
a	blanket width (m)
a/ℓ	composite honeycomb parameter
b	beam width, blanket batten width (m)
β	batten-diagonal angle (degrees)
D	beam diameter, flexural rigidity (m, N-m)
d	member diameter (m)
Δ	deflection (m)
ΔT	temperature difference (K)

E	modulus of elasticity (N/m^2)
ϵ	strain, efficiency
f	beam cantilever frequency (Hz)
GJ	torsional rigidity $(N\text{-}m^2)$
h	beam depth, core thickness (m)
I	area moment (m^4)
K	vibration constant
k	multiplying factor
L	beam length (m)
ℓ	bay length (m)
M	mass, moment (kg, N-m)
m	surface density (kg/m^2)
m'	linear density (kg/m)
μ	friction coefficient
N	total mass/longeron mass
n	number of blanket segments, number of standoffs
ν	Poisson's ratio
P	load (N)
q	pressure (N/m^2)
r	radius (m)
s	stack height (m)
ρ	density (kg/m^2)
σ	stress (N/m^2)
T	blanket or guide cable tension (N)
t	thickness (m)
U	panel unsupported length (m)
v	deployment velocity (m/s)

1.2 SUBSCRIPTS

AM	Astromast
a	adhesive, actuator
att	attachment
b	battens, bending
beam	beam

bl	blanket
c	honeycomb core
cont	container
d	diagonals; diameter
depl mech	deployment mechanism
ESS	Extendible Support Structure
Eu	Euler
f	frequency; facing
fb	face battens
fd	face diagonals
g	fiberglass
gc	guide cables
h	harness
j	joints
ℓ	longerons; length
m	mass
mem	members
p	load
pan	panels
s	strut
SB	Astro STACBEAM
so	standoffs
sys	system
t	tape
tors	torsion
tot	total
tu	tubes
sb	side battens
sd	side diagonals

1.3 SUPERSCRIPTS

m	empirical numbers

SECTION 2
GROUND RULES

Three structural concepts will be evaluated for efficiency in deployment and support of solar-array blankets. In order that the merits of each are considered in an unbiased manner, certain ground rules are established.

2.1 ACCORDIAN-PACKAGE BLANKET

The SEPS array employs a flexible blanket which is folded in accordian fashion. This is a lightweight approach which has a great amount of investment behind it.

2.2 NONSTRUCTURAL BLANKET

The solar-array blanket does not contribute to the stiffness of the structure in cantilever vibration. (Its tension-induced stiffness is a factor, however, in torsional vibrations.) Therefore, changes in blanket design have little effect on the structural behavior.

2.3 INCOMPATIBLE DEFLFCTIONS

Thermally induced changes in length are different for the blanket and for the structure and must be compensated for in the design.

2.4 ROTARY ATTACHMENT

Array angular accelerations relative to the spacecraft are buffered through a rotary attachment.

2.5 WINGLIKE CONFIGURATION

The solar blanket has a high aspect ratio, so that the view from the spacecraft is blocked only slightly.

2.6   SMALL RELATIVE MASS

The mass of the blanket and its structure is considered to be small in comparison so the total spacecraft mass. Therefore, the cantilever vibration frequency approximates the lowest frequency mode.

2.7   THERMALLY ISOLATED BLANKET

Each concept is designed so that no radiating structure is close to the blanket.

2.8   NONSPECIFIC BLANKET

The structure places no specific demand on blanket cell pattern.

SECTION 3
GENERAL DESCRIPTION

The general concept of a system for deploying and supporting a solar-cell blanket is shown in Figure 1. The system consists of several components, as follows.

3.1 BLANKET

The solar-cell blanket is of the flexible type with various surface densities defined by Designs 1, 2, and 3 (see Table I). It is divided into segments by alternating folds so that it packages accordian-fashion. The area of the blanket is 126 m^2 with a platform of 4 by 31.5 m.

3.2 BLANKET CONTAINMENT STRUCTURE

The blanket is protected and restrained during storage and launch by the containment structure which consists of a pair of panels, above and below the the packaged blanket (see Figu: 2). For deployment, the upper panel is released and pivots away from the blanket.

3.3 SUPPORT STRUCTURE (BEAM)

The blanket is pulled out and supported by a deployable beam which packages next to the packaged blanket. The beam lattice material is assumed to be graphite/epoxy composite in either tube or rod form; this material has extremely high specific strength (σ/ρ). The beam lattice configuration is adjusted so that the cantilever frequency is approximately 0.15 Hz.

Three concepts of support structures are to be investigated. The ESS (Extendible Support Structure) and the Astromast were developed by Astro Research Corporation (Astro) had have successfully flown in space; the STACBEAM is under development at Astro.

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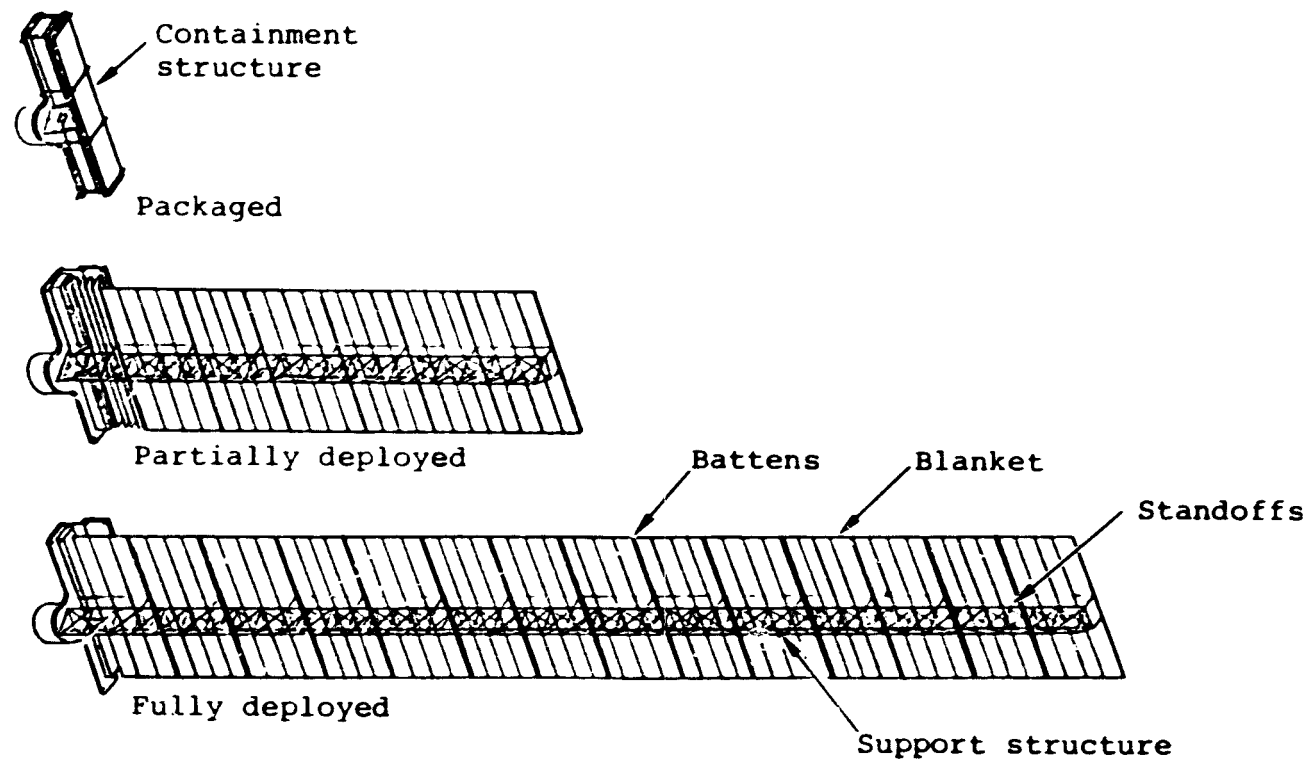
Figure 1.   General system concept for deploying and supporting
            a solar-cell blanket.

Packaged assembly

Upper panel

Section of facing sheet
showing lattice of uni-
directional graphite
filament tapes
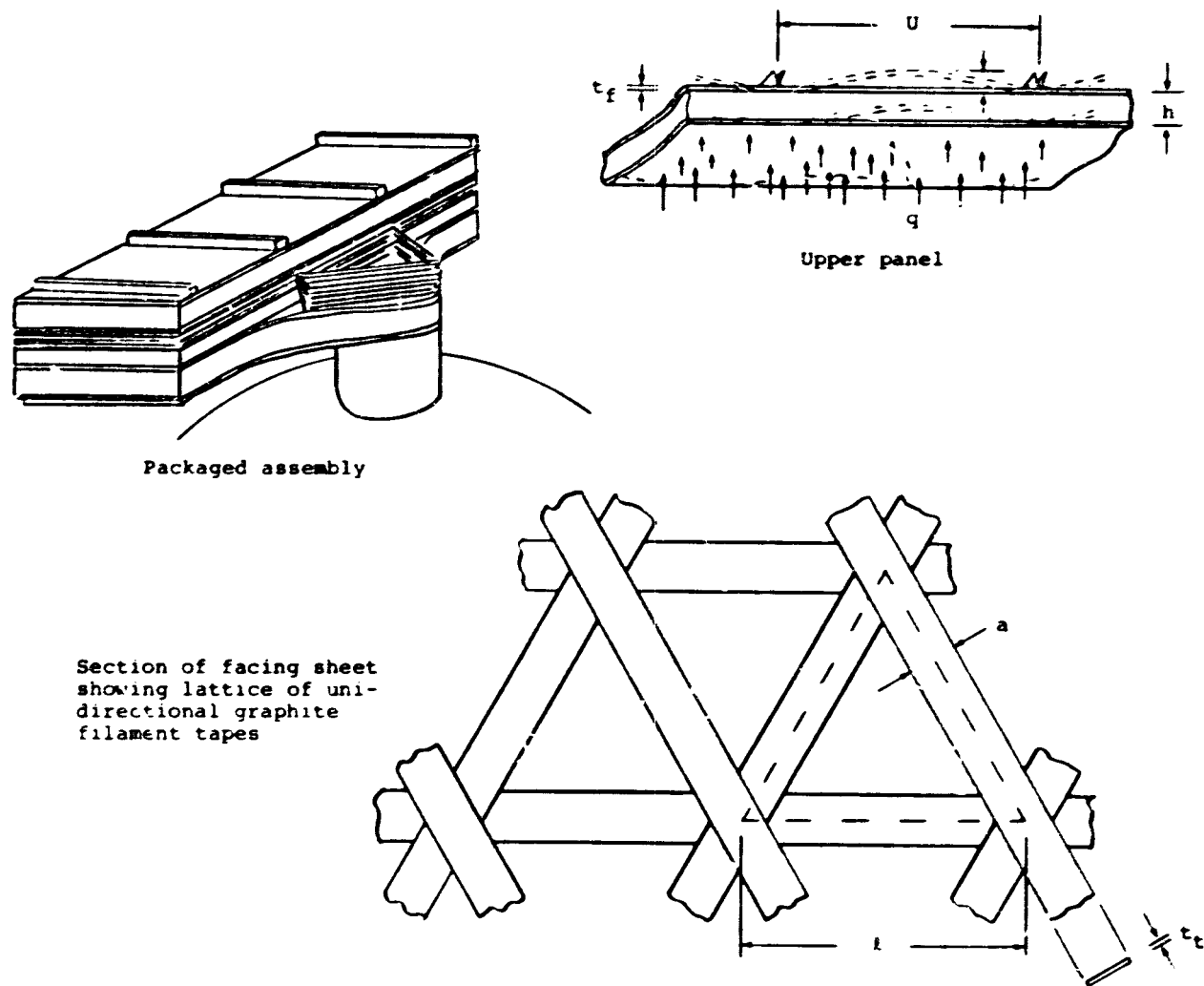
Figure 2.   Blanket containment structure.

3.4 BLANKET STIFFENING BATTENS

Lateral stiffness is developed in the blanket by attached deployable battens. These are flat when the blanket is in its packaged condition. Upon deployment, they become beams of triangular cross section (see Detail A, Figure 3).

3.5 STANDOFFS

The blanket is attached to the structure indirectly by standoffs. The standoffs hold the blanket away from the structure, thus minimizing radiative heating of the blanket. ˀ addition, the standoffs make relative deflections possible between the blanket and structure by pivoting along the wing axis.
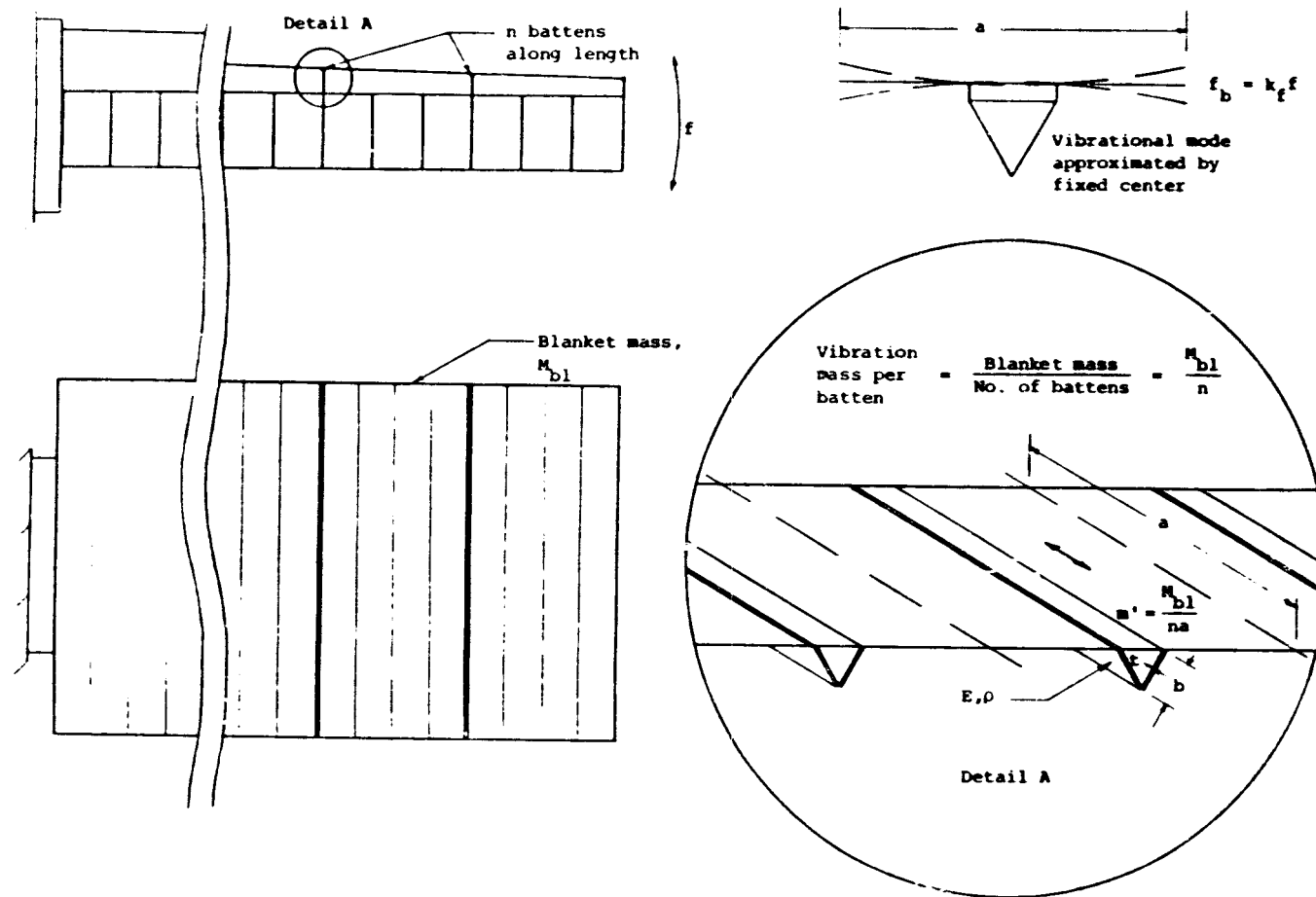
Figure 3.   Blanket stiffening battens.

SECTION 4
SYSTEM MASS

The system mass is the sum of the masses of the blanket and its container and harness, beam, deployment mechanism, actuator, guide cables, blanket stiffening battens, and standoffs, as follows:

$$M_{sys} = M_{bl} + M_{cont} + M_{harn} + M_{beam} + M_{depl\ mech}$$

$$+ M_a + M_{gc} + M_b + M_{so} \tag{1}$$

The first three terms depend only on blanket design, and the remaining terms are affected by beam structural configuration. These terms are addressed sequentially in the following sections.

4.1 BLANKET

The mass of the blanket is a design variable; values to be addressed are given in Table I. In the scope of this investigation, the blanket is of specific size: 126 m^2 in area with a 7.9 aspect ratio. However, its surface density is variable and values are used which reflect Designs 1, 2, and 3 of Table I with blanket surface densities of 0.643, 0.421, and 0.238 kg/m^2, respectively. These surface densities correspond to blanket masses of 81, 53, and 30 kg, respectively.

4.2 BLANKET CONTAINMENT STRUCTURE

The packaged blanket is held in position by a pair of panels as shown in Figure 2. The design pressure on the blanket is based on that used on SEPS (1.5 psi). On the rationale that this pressure provides restraint forces which overcome blanket inertia during launch, this pressure is made proportional to blanket mass.

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$$q = \left(\frac{M_{bl}}{113 \text{ kg}}\right) (10{,}300 \text{ N/m}^2) \qquad (2)$$

Each panel consists of a honeycomb structure which consists of a core of aluminum honeycomb material with facing sheets. For the condition shown in Figure 2, the panel deflection is

$$\Delta = \frac{K_b q U^4}{D} \qquad (3)$$

where

K_b = bending deflection constant = 1/384 for fixed ends

U = unsupported length, consistent with K_b (Appendix A) = 1.064 m

D = flexural rigidity = $Eh^2 t_f / 2(1 - \nu^2)$

E = modulus of facing material = 1.3×10^{11} N/m^2

h = core thickness

t_f = facing thickness

ν = Poisson's ratio = 0.326

Then, the required core thickness is given by

$$h^2 = \frac{q U^4}{215 \Delta E t_f} \qquad (4)$$

The facing in this case consists of a lattice of unidirectional graphite filament tapes shown in Figure 2. The stiffness and mass of a panel consisting of this type of facing sheet are given in reference 3 which investigates this concept in detail. The stiffness is obtained by using an effective facing thickness, whereby the lattice is considered to be smeared over the entire facing sheet so that

$$t_f = 2.3 \frac{a}{\ell} t_t \tag{5}$$

where a/ℓ is the lattice parameter and t_t is the tape thickness. The panel mass per unit area is

$$m = 2\sqrt{3} \frac{a}{\ell}(2\rho_t t_t + \rho_c h + 2\rho_a t_a) - 3\left(\frac{a}{\ell}\right)^2 (\rho_c h + 2\rho_a t_a) \tag{6}$$

where

$$
\begin{aligned}
a/\ell &= \text{lattice parameter} = 0.3 \\
\rho_t &= \text{tape density} = 1520 \text{ kg/m}^3 \\
t_t &= \text{tape thickness} = 0.14 \text{ mm} \\
\rho_c &= \text{core density} = 48 \text{ kg/m}^3 \\
\rho_a &= \text{adhesive density} = 5770 \text{ kg/m}^3 \\
t_a &= \text{adhesive thickness} = 0.0254 \text{ mm}
\end{aligned}
$$

The above quantities were extracted from information presented in reference 2. The allowable panel deflection Δ is, determined from foam padding, force-deflection characteristics.

The SEPS pad (ref. 4, Figure 3-13) had the following crush pattern:

1) The initial 1.5 cm of crush produces little resisting force;

2) The next 0.5 cm builds elastically to 3500 P_a (N/m^2);

3) The next 0.5 cm crushes plastically, remaining at 3500 Pa; and

4) The final 0.6 cm builds elastically to over 10,000 Pa.

Designing to this fourth region, whereby the pad is crushed beyond 2.5 cm, the pressure load on the SEPS blanket varies over its surface in the range of 9000 to 12,000 Pa if the panel deflection is

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$$\Delta = 0.3 \text{ cm}$$

Thus, the required core thickness in meters can be expressed as a function of blanket mass in kilograms:

$$h = 0.00309 \sqrt{M_{bl}} \qquad (7)$$

Panel masses to contain the various blankets are shown in the following table using Eqs. (6) and (7), assuming a pair of panels contain a folded blanket having length and width of 4.0 meters and 0.4 meters.

DESIGN	M_{bl} (kg)	h (cm)	M_{pan} (kg)
1	81	2.78	5.42
2	53	2.25	4.79
3	30	1.69	4.14

Additional mass is required for support beams which contribute a mass of approximately 1.0 kg (see Appendix A). For attachment to the spacecraft, these masses are doubled to account for miscellaneous coupling hardware. Thus, the containment masses for Designs 1, 2, and 3 are 13.9, 12.1, and 10.1 kg, respectively.

4.3 HARNESS

The mass of the electrical harness is dependent on the blanket mass and is based on the SEPS harness mass (6 kg). It is assumed that technological improvements associated with increasing blanket voltage will also decrease the harness mass. The table below lists the values used in this analysis.

BLANKET MASS (kg)	HARNESS MASS (kg)
81	6
53	4
30	3

4.4 BEAM

The mass of the beam is determined as a function of its cantilever frequency. This is examined in detail in each separate concept section.

4.5 DEPLOYMENT MECHANISM

The mass of the deployment mechanism is examined separately with each concept.

4.6 ACTUATOR

The mass of the deployment actuator is assumed proportional to the power requirement to deploy the array which is determined by blanket tension. The actuator motor is required to move the structure outward against the force of the blanket tension at a predetermined deployment velocity. The mass is

$$M_a = k_a \frac{T\mu v}{\epsilon}$$

where

k_a = motor weight coefficient = 0.2 kg/W input power
μ = beam deployment friction coefficient = 2.0
v = beam deployment velocity = 0.055 m/s
ϵ = motor efficiency = 0.5

Thus, for blanket tension T expressed in Newtons, the actuator mass in kilograms is

$$M_a = 0.044 \ T$$

4.7 BLANKET GUIDE CABLES

The two steel guide cables ($\rho = 7700$ kg/m^3) are designed to be at a working stress of $\sigma = 210$ MPa (30,000 psi) when reacting the longerons at their Euler buckling limit. That is

$$T_{gc} = P_{Eu}$$

$$\sigma A_{gc} = \pi^2 EI/\ell^2$$

$$A_{gc} = \frac{\pi^3 Ed^4}{64\ell^2\sigma}$$

The guide cable mass per unit length is

$$m'_{gc} = 2\rho A_{gc}$$

$$= \frac{\pi^3 Ed^4}{32\ \ell^2} \left(\frac{\rho}{\sigma}\right)_{gc}$$

$$= 4.0 \times 10^6\ d^4/\ell^2 \tag{8}$$

This quantity, evaluated for nominal values of member diameter, wall thickness, and length, is three orders of magnitude less than the blanket mass-per-unit length. It is thus considered insignificant in terms of affecting beam frequency, and is evaluated only as it affects the total system mass.

4.8 BLANKET STIFFENING BATTENS

The method of periodic blanket stiffening is shown in Figure 3. The batten vibration frequency is

$$f_b = \frac{K}{2\pi} \sqrt{\frac{EI}{m'a^4}} \qquad (9)$$

where K depends on fixity:

$$
\begin{aligned}
K &= 3.52 \text{ for fixed end (cantilever)} \\
&= 9.87 \text{ for pinned ends} \\
&= 14.08 \text{ for fixed center} \\
&= 22.4 \text{ for free-free}
\end{aligned}
$$

Because the beam width is relatively small compared to the blanket width, the fixed-center solution must closely approximate the vibration mode, and its K value is used in Eq. (9).

The area moment of inertia of the equivalent section is

$$I = \frac{1}{8} tb^3$$

Assuming n stiffening battens at the standoff positions, the mass-per-unit length across the blanket, neglecting the batten mass itself, is

$$m' = \frac{M_{bl}/n}{a}$$

The batten frequency f_b is greater than the beam frequency f by a factor k_f:

$$f_b = \frac{K}{2\pi} \sqrt{\frac{Etb^3 n}{8 M_{bl} a^3}} = k_f f$$

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Thus, the necessary batten width is

$$b = 1.168 \left(\frac{k_f^2 \ell^2 M_{bl}}{Etn}\right)^{1/3} a \tag{10}$$

and the total batten mass is

$$M_b = 2 \, b t a \rho n \tag{11}$$

This quantity is approximately two orders of magnitude less than the blanket mass-per-unit length, and is disregarded as is the guide cable mass in the frequency calculations.

4.9   BLANKET STANDOFFS

The blanket standoffs satisfy two requirements:

1.   To hold the solar-cell blanket away from the supporting structure to minimize local radiative heating of the blanket, and

2.   To allow for attachment in the packaged condition.

It has been determined (see Appendix B) that a five-member diameter is sufficient standoff distance for the first requirement. For the second requirement, a standoff distance at the base of the assembly equal to the packaged stack height s is required. The n pairs of standoffs of average length $(5d + s)/2$, made of longeron material (area A_ℓ, density ρ) contribute a mass-per-unit length of

$$m_{so}' = \frac{n(5d + s)}{L} \rho A_\ell \tag{12}$$

This quantity, as is the case for the guide cables and stiffening battens, is insignificant compared to the payload mass-per-unit length (three orders of magnitude less). This quantity is disregarded in the frequency calculations.

SECTION 5

GENERAL DESIGN APPROACH

The following approach to beam design is applicable for beams in which member length and diameter are independent, which is the case for the ESS and STACBEAM structures, but not for the Astro-mast (see Figure 4). The Astromast design approach is described in Section 7. For the ESS and the STACBEAM, the relationship of beam mass to frequency is determined as follows. For certain blanket mass and length, and for various values of bay length, begin with an estimate of tube diameter and wall thickness (or just rod diameter). Determine the cantilever vibration frequency. Now impose the requirement that the blanket vibration frequency be three times the cantilever frequency, thus determining the required blanket tension (assuming some blanket attachment spacing). Lastly, compare the longeron buckling safety factor, or maximum member deflection due to reacting the blanket tension, to a predetermined number. In general, a difference is found and the calculation is repeated using different member sizes until agreement is obtained.

5.1 FREQUENCY DETERMINATIONS

5.1.1 Beam Cantilever Frequency

The vibration frequency of the cantilever beam is given by

$$ f = \frac{3.52}{2\pi} \sqrt{\frac{(EI)_{beam}}{ML^3}} \tag{13} $$

where

$(EI)_{beam}$ = beam bending stiffness where E = material modulus and I = beam equivalent area moment of inertia of beam

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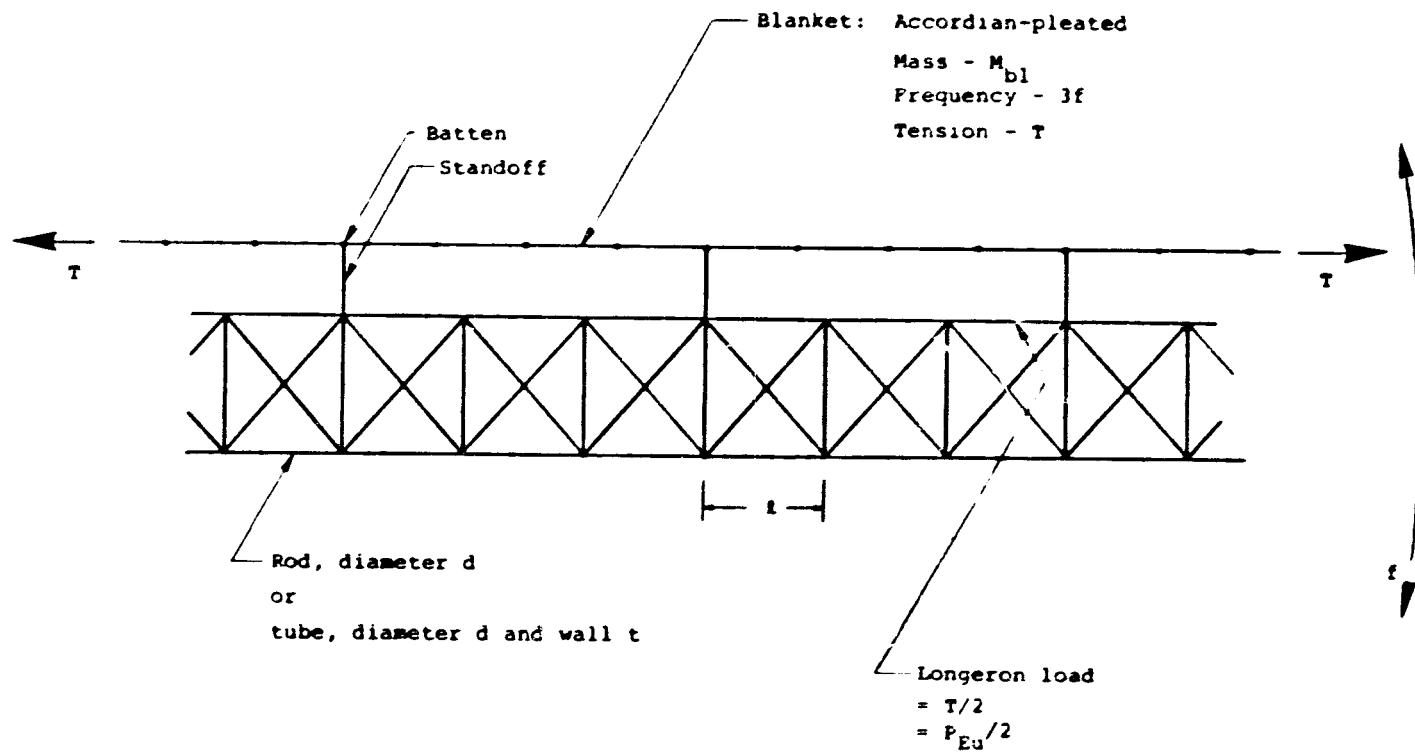
Figure 4. General design approach.

M = beam mass (including payload)

L = beam length

5.1.2 Blanket Frequency

The vibration frequency of a tensioned blanket is given by

$$f_{bl} = \frac{1}{2\ell_{att}} \sqrt{\frac{T}{m'_{bl}}} \tag{14}$$

where

ℓ_{att} = L/n = blanket attachment length (n = number of blanket segments)

T = blanket tension

m'_{bl} = M_{bl}/L = blanket mass-per-unit length

It is required that the blanket frequency be higher than the cantilever frequency by a factor k_f, or

$$f_{bl} = k_f f \tag{15}$$

Then

$$T = \left(\frac{2k_f f}{n}\right)^2 M_{bl} L \tag{16}$$

5.2 MEMBER SIZE DETERMINATION

Two methods of sizing members are presented here. The first method results in a relatively light structure by considering the longeron Euler buckling safety factor. The second method results in a heavier structure by requiring that all member cross sections

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be equal to that of the member with the highest bending moment. The first method is the one used in subsequent analyses, with the assumption that local strengthening is done where required.

5.2.1 Buckling

The required longeron size is determined by specifying its Euler buckling capacity. The blanket tension T is reacted by the two longerons adjacent to the blanket so that the longeron load is

$$P = \frac{T}{2} \tag{17}$$

The longeron Euler capacity is given by

$$P_{Eu} = \frac{\pi^2 (EI)_{mem}}{\ell^2} \tag{18}$$

where

$(EI)_{mem}$ = member bending stiffness where E = material modulus and I = member area moment of inertia as follows:

$$I = \frac{\pi}{4} r^4 = \frac{\pi}{64} d^4 \qquad \text{for solid rod}$$

$$I = \pi r^3 t = \frac{\pi}{8} d^3 t \qquad \text{for thin-wall tube}$$

r, d = section radius, diameter

t = wall thickness

ℓ = member length

It is required that the Euler buckling capacity P_{Eu} be higher than the longeron load P by a factor of k_p, or

$$P_{Eu} = k_p P \tag{19}$$

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Combining Eqs. (16), (17), (18), and (19) yields, for solid rod members,

$$f = \left(\frac{\pi^{3/2}}{4} \sqrt{\frac{E}{k_\rho M_{bl} L}} \frac{n}{k_f \ell} \right) \frac{d^2}{2\sqrt{2}} \qquad (20)$$

or, for thin-wall tube members,

$$f = \left(\frac{\pi^{3/2}}{4} \sqrt{\frac{E}{k_\rho M_{bl} L}} \frac{n}{k_f \ell} \right) \sqrt{d^3 t} \qquad (21)$$

Equations (20) and (21) say that the cantilever frequency is $1/k_f$ times the blanket frequency when the blanket tension is at a level which loads the longerons to $1/k_p$ times their buckling capacity. This is compared to Eq. (13), which determines the beam frequency directly from its stiffness and mass. Member size d is that which yields the same result for Eqs. (13) and (20) or (13) and (21).

5.2.2 Bending

The amount of bow in the greatest deflecting member is considered when using this method of member sizing. This member is one of two at the end of the beam which reacts the blanket tension at a standoff distance of five member diameters, so that

$$M = \frac{5}{2} Td \qquad (22)$$

The amount of deflection is (see ref. 5, Table 3, Item 3e)

$$\Delta = 0.0642 \frac{M \ell_b^2}{EI_b} \qquad (23)$$

where

$$\ell_b = k_\ell \ell = \text{length of bending member}$$

$$I_b = \text{bending member section moment}$$

$$= \frac{\pi}{64} d^4 \qquad \text{for solid rod}$$

$$= \frac{\pi}{8} d^3 t \qquad \text{for thin-wall tube}$$

The deflection is limited to

$$\Delta = \kappa_d d \qquad\qquad (24)$$

Combining Eqs. (16), (22), (23), and (24) yields, for solid rod members,

$$f = \frac{1}{6.41}\left(\sqrt{\frac{\pi k_d E}{M_{bl} L}} \frac{n}{k_f k_\ell \ell}\right) d^2 \qquad\qquad (25)$$

or, for thin-wall tube members,

$$f = \frac{1}{2.27}\left(\sqrt{\frac{\pi k_d E}{M_{bl} L}} \frac{n}{k_f k_\ell \ell}\right) \sqrt{d^3 t} \qquad\qquad (26)$$

Equations (25) and (26) say that the cantilever frequency is $1/k_f$ times the blanket frequency, when the blanket tension is at a level which deflects supporting members k_d times their diameter. This is compared to Eq. (13) which determined the beam frequency directly from its stiffness and mass. Member size d is that which yields the same result for Eqs. (13) and (25) or (13) and (26).

This approach results in a conservatively heavy beam because each member is sized to meet the bending moment of the two end members.

SECTION 6
EXTENDIBLE SUPPORT STRUCTURE CONCEPT

The Extendible Support Structure (ESS), originally developed by Astro Research Corporation to support the synthetic aperture radar antenna, was flown on the Seasat spacecraft. The ESS concept for deployment and support of a solar array is shown in Figures 5 and 6. The beam is of width b, depth h, and baylength ℓ, and is composed of tubes of diameter d and wall thickness t. For the Seasat application, rigid solar panels were an integral part of the structure in both the packaged and deployed state. For this case, the solar array and the support structure are essentially separate, except for periodic attachments.

The method of sizing the structure is outlined in Section 5. The ESS is, in general, a structure with rather long tubular members. A conservative design is obtained using the bending method of Eqs. (13) and (26). A lighter design with a smaller load margin is obtained using the Euler buckling method of Eqs. (13) and (21).

6.1 BEAM PROPERTIES

Equation (13) contains terms which pertain to intrinsic beam properties of mass and stiffness.

6.1.1 Stiffness

The stiffness of the ESS is expressed as (see ref. 6)

$$EI = \frac{2}{3} h^2 EA$$

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Legend:

ℓ baylength

b beam width

h beam depth

T blanket tension
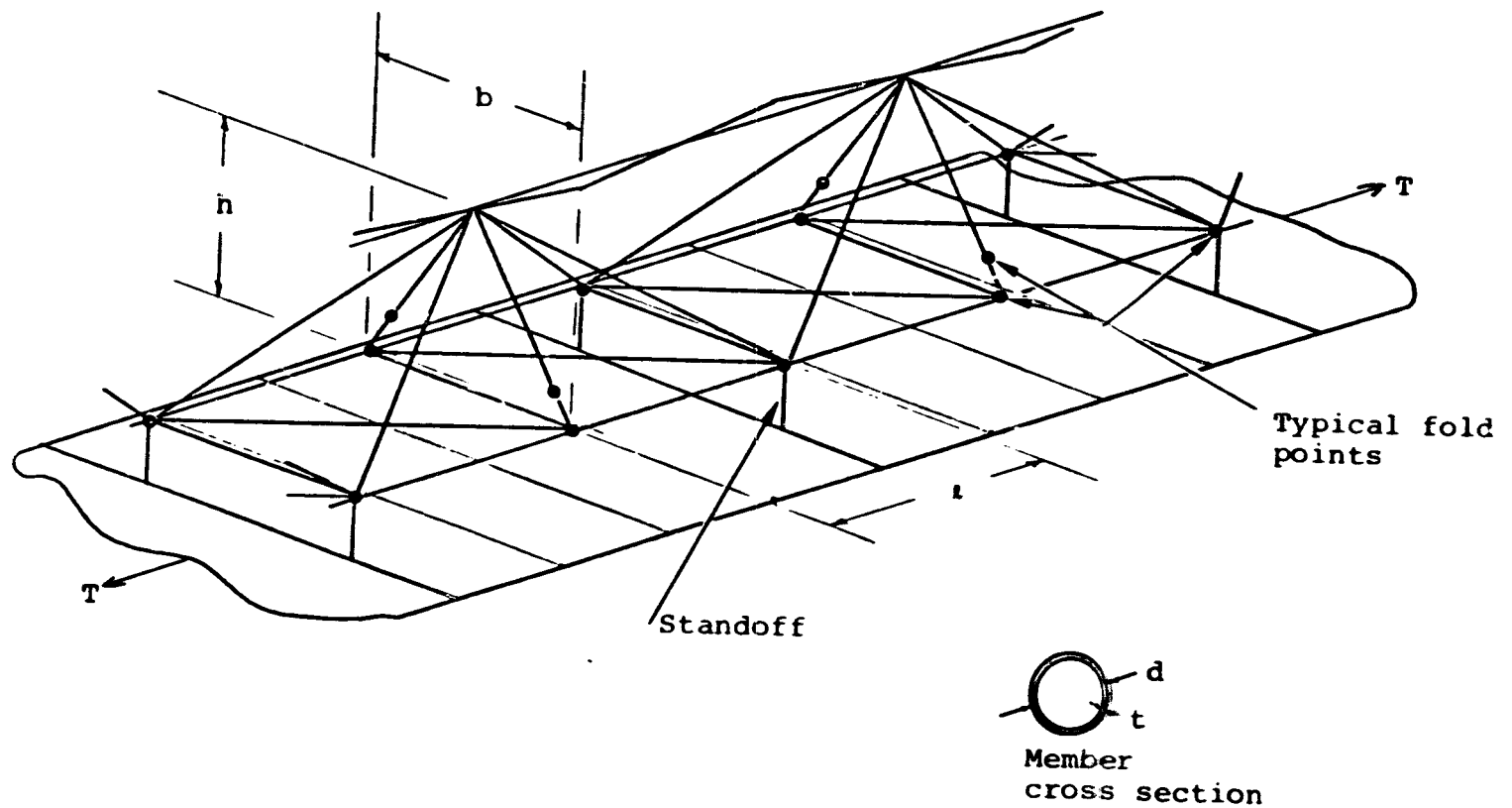
Figure 5. ESS concept.
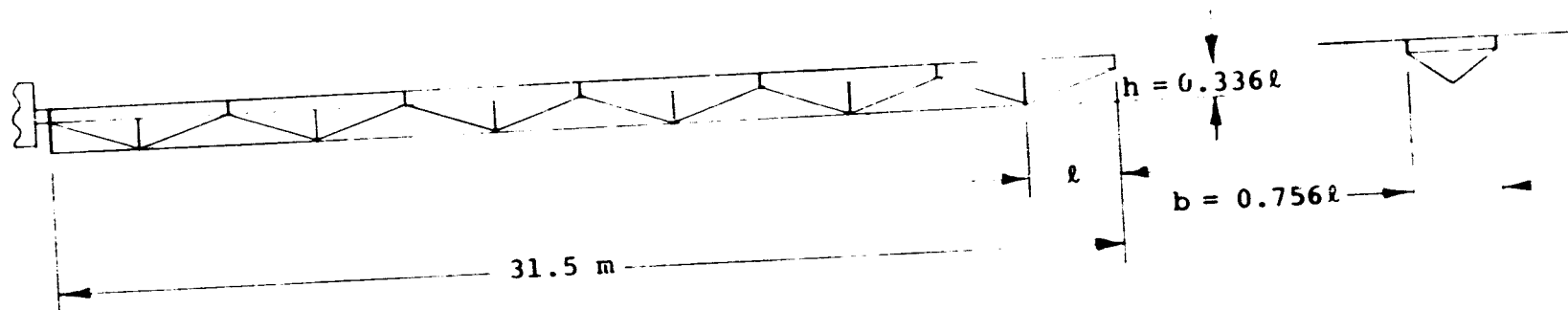
Material:

    Graphite/epoxy tubing: $E = 110$ GPa, $\rho = 1520$ kg/m^3

    Wall thickness $t_w \geq 0.25$ mm

    Diameter $d \geq 0.25$ cm

Structure equivalents:

$$EI = 0.2364 \; Edt_w\ell^2 \; Nm^2$$

$$GJ = 0.03261 \; Edt_w\ell^2 \; Nm^2$$

$$\frac{Mass}{Length} = 27.5 \; k\rho dt_w \; kg/m \quad (k \doteq 2)$$

Figure 6. ESS configuration.

where

    h = beam depth

    A = πdt = member area

An arbitrary design constraint is applied, whereby the ratio of beam depth h to bay length ℓ is equal to the ratio used for Seasat, or

$$h = 0.336 \, \ell$$

Then

$$EI = 0.2354 \, E\ell^2 dt \tag{27}$$

6.1.2 Mass

The cantilevered mass of the ESS includes the masses of its tubular members and joints and the blanket payload itself, or

$$M = M_{tu} + M_j + M_{bl} \tag{28}$$

6.1.2.1 <u>Mass of Tubes</u> - The tubular mass is determined by multiplying the linear density by the beam length, or

$$M_{tu} = m'_{tu} \, L \tag{29}$$

The beam consists of longerons, face and side diagonals, and face and side battens, so that

$$m'_{tu} = m'_{\ell} + m'_{fd} + m'_{sd} + m'_{fb} + m'_{sb}$$

$$= 4\rho A + \sqrt{1 + \left(\frac{b}{\ell}\right)^2}\,\rho A + 2\sqrt{1 + \left(\frac{b}{2\ell}\right)^2 + \left(\frac{h}{\ell}\right)^2}\,\rho A$$

$$+ \frac{b}{\ell}\,\rho A + \frac{1}{2}\sqrt{\left(\frac{b}{\ell}\right)^2 + \left(\frac{2h}{\ell}\right)^2}\,\rho A$$

where ℓ, b, and h correspond to the lengths shown in Figure 6, ρ is tube material bulk density, and A is the tube cross-sectional area. Thus,

$$m'_{tu} = 8.75\ \rho A \tag{30}$$

6.1.2.2 **Mass of Joints** - The joint mass is expressed as a function of tube mass by

$$k \equiv \frac{M_j}{M_{tu}} + 1$$

The joint factor k, multiplied by the tube mass, gives the beam mass and is estimated by investigating the Seasat ESS assembly. The joint mass in the ESS for Seasat was 8 kg, over a length of 10.7 m, so that

$$m'_{j\ Seasat} = 0.75\ \text{kg/m}$$

The tube diameter was 0.0127 m with a 0.00075-m wall thickness. In accordance with Eq. (30), the graphite tube mass per length is determined to be 0.40 kg/m. Therefore, for Seasat,

$$k-1 = 0.75/0.40 = 1.88$$

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and

$$k = 2.88.$$

This joint factor, obtained for an assembly having compara-
tively high wall thickness, should be greater for joints connect-
ing thinner wall tubing. Efforts directed toward decreasing the
Seasat joint weight (see Figure 7) have succeeded in lowering the
factor to approximately 1.9. Projecting from these results, the
following joint factors are assumed.

TUBE WALL THICKNESS t (mm)	JOINT FACTOR k
0.75	1.9
0.50	2.2
0.25	2.5

6.2 SIZE-FREQUENCY RELATIONS

Table II is a listing of an HP34-C calculator program which
correlates payload mass and frequency with ESS mass and frequency.
The latter three terms are determined by member length and cross-
sectional properties as outlined in Section 5. Figures 8, 9, and
10 indicate tube size, beam mass, and frequency trends of the ESS
system configuration for Designs 1, 2, and 3, respectively, sized
by the Euler buckling method of Eqs. (13) and (21).

6.3 DEPLOYMENT MECHANISM

The ESS deployment mechanism consists of latching clamps
which hold the packaged assembly rigidly together. Upon release
of these clamps, the assembly is free to expand and does so in a
controlled fashion by motorized extension of the rear scissors
longeron. There are three clamps, one for each stack of hinges,
which make a direct load path to the spacecraft. Such an assembly
is expected to have a mass approximately equal to the joint mass.

Object on right: Joint J2-1 used on Seasat
                   ESS structure. Mass: 92 gm.

Object on left: Joint J2-1 lightened as follows:
                   Web reduced from 0.30 to 0.08
                   inch; hole diameter increased
                   from 0.25 to 0.30 inch.
                   Mass: 46 gm. Reduction: 50
                   percent.

Figure 7. Joint mass reduction.

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TABLE II. ESS MASS-FREQUENCY.

REGISTER		VALUE		FLAG	SET	CLEAR
I	f			0	Buckling	Bending
0	t	0.00075,		1		
		$-50,-25$		2		
1	ℓ			3		
2	d					
3	M_{beam}			**ROUTINE**		
4	M_{bl}	81,53,30		A	Buckling	
5	L	31.5		B	Bending	
6	ρ	1520		0	Continue	
7	E	1.1×10^{11}		1	Solve for d	
8	ℓ_{tot}/L	8.75		2	Continue	
9	K_{EI}	0.2364		3	Buckle calc.	
.0	K_ℓ	0.560		4	Continue	
.1	k_ℓ	1.1206		5	Store t, M_{bl}	
.2	k_p	2.0				

ENTRY: Enter with ℓ
      A: Buckling
      B: Bending

1	LA	Buckling:	20	RTN
	SFO	set flag		L1
	GTO			S2
	LB	Bending:		PSE
	CFO	clear flag		↑
	LO			x
	S1			RO
	R2	⎫		x
	↑	⎬ guess d		R6
10	↑			x
	+	⎭	30	R8
	Solve 1	solve for d		x
	GT2			3
	R/S			RO
	L2			2
	R1	ℓ		EEX
	R3	M_{beam}		3
	RI	f		x
	R2	EXIT: d		-
				x

(with "area" bracketing the ↑, x, RO, x group and "joint factor k" bracketing the 3, RO, 2, EEX, 3, x group)

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TABLE II. (concluded).

4J	R5		
	x		$\div$
	S3		π
	R9		x
	R7		4
	x	90	$\div$
	R1		F?0
	x^2		GT3
	x		2
	R2		$\div$ $\sqrt{1/k_d}$
50	x		R.1
	R0		$\div$
	x		GT4 f_{bend}
	R3		L3
	R4		R.2
	+	100	$\sqrt{}$
	$\div$		$\div$
	R5		R.0
	3		$\div$
	y^x		L4 f_{buckle}
60	$\div$		$-$
	$\sqrt{}$		EEX
	R.0		3
	x		x
	SI	f	INT
	R2	110	x $\neq$ 0
	3		LST x
	y^x		RTN
	R0		L5
	x		S4
70	R7		$\downarrow$
	$\div$		S0 new t, M_{bl}
	R4		RTN
	$\div$		
	R5		
	$\div$		
	$\sqrt{}$		
	R5		
	R1		
	$\div$		
80	2		
	$\div$		
	x		
	R1		
	$\div$		
	3	κ_f	

Blanket: 81 kg; 0.643 kg/m^2
Tube wall thickness = 0.75 mm
Graphite/epoxy
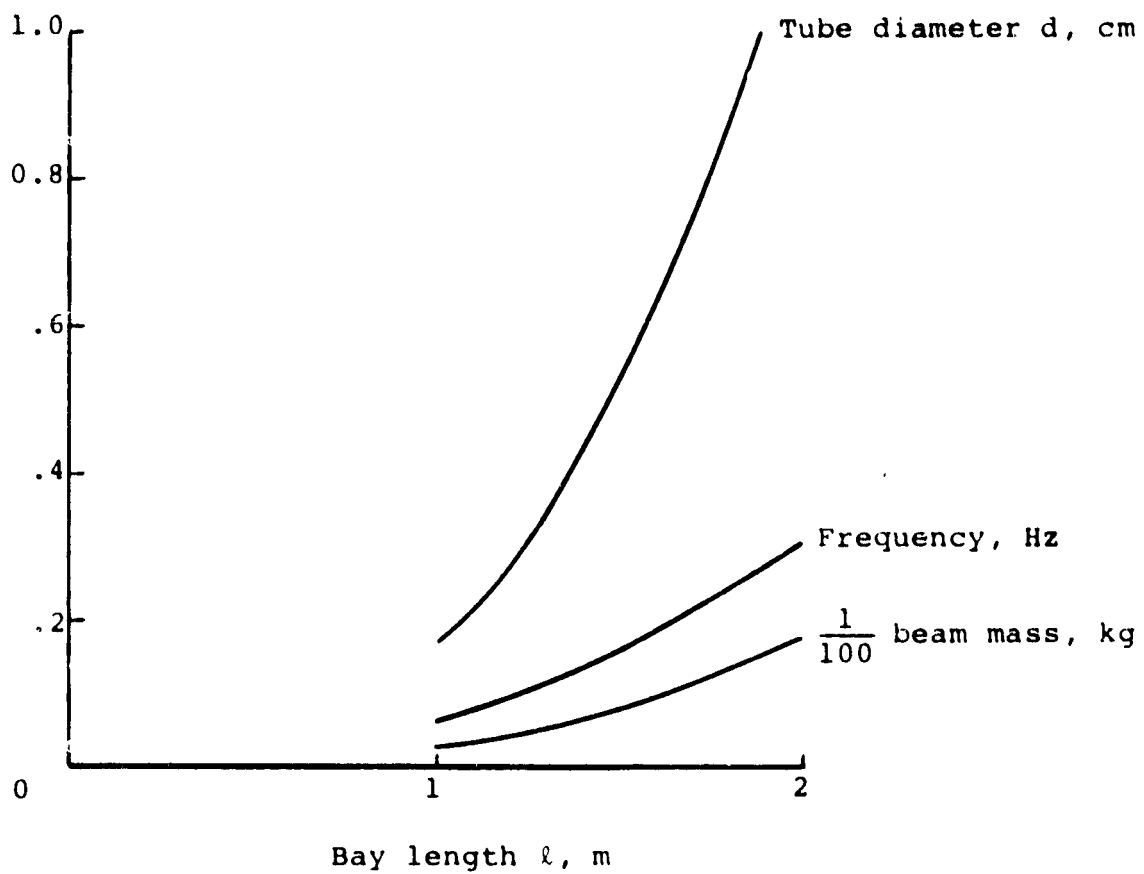
Figure 8.   ESS configuration: Design 1.

Blanket: 53 kg; 0.421 kg/m^2

Tube wall thickness: 0.50 mm

Graphite/epoxy
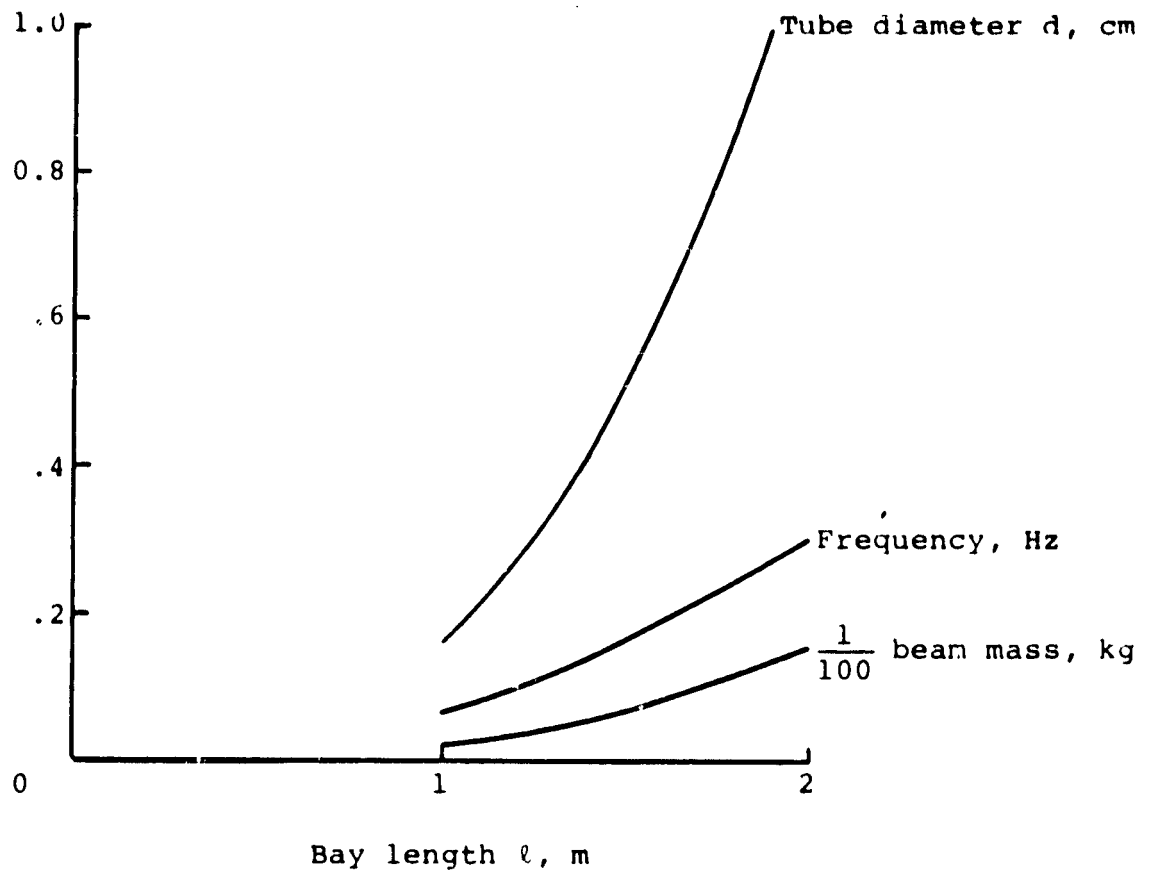
Figure 9.   ESS configuration: Design 2.

Blanket: 30 kg; 0.238 kg/m^2

Tube wall thickness = 0.25 mm

Graphite/epoxy
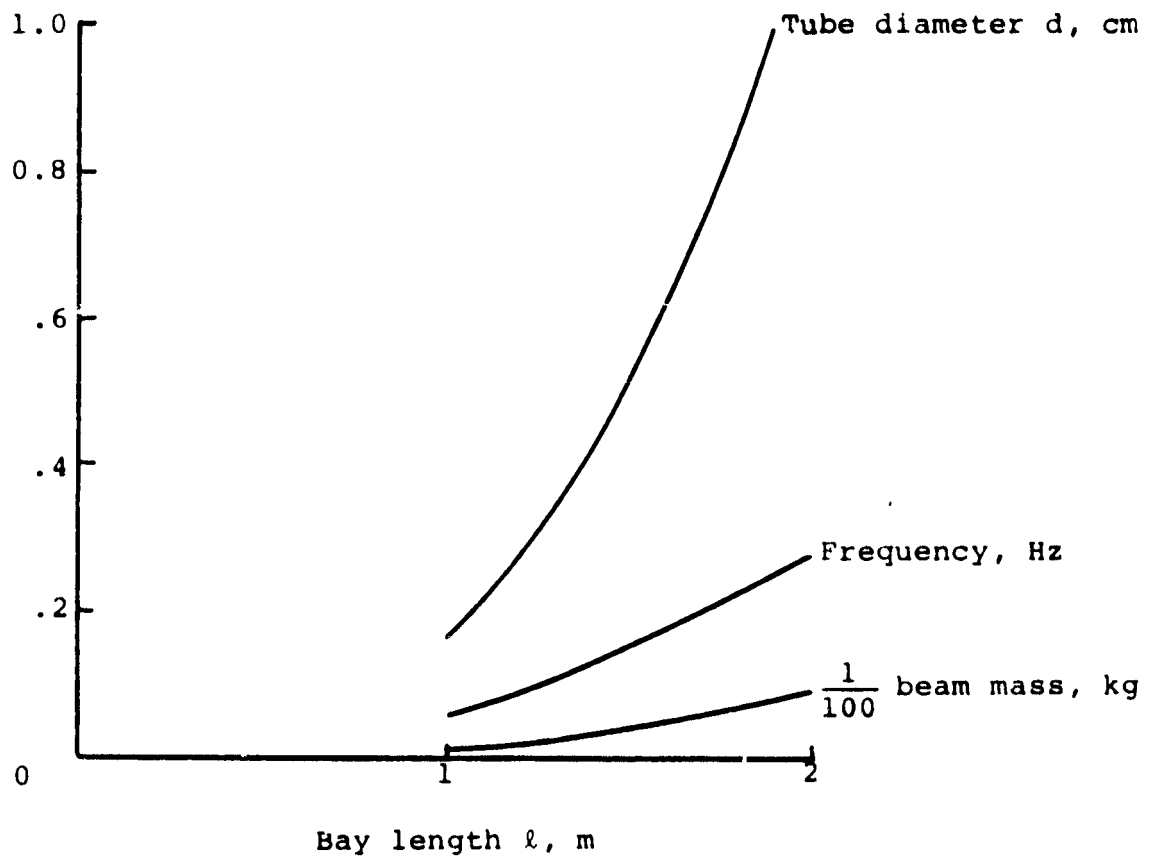
Figure 10. ESS configuration: Design 3.

6.4 SYSTEM MASS

The mass of the ESS system, using Eq. (1), is itemized for Design 2 at a 0.2-Hz cantilever frequency in Table III. It appears that the design goal (blanket mass/system mass $\geq$ 0.6) is attainable at this frequency. The ESS system mass for Designs 1, 2, and 3 is given as a function of frequency in Figures 11, 12, and 13. These curves serve as a means of comparison of the various blanket support schemes and as an indication of the mass penalty involved in attaining higher frequencies.

6.5 SYSTEM DESCRIPTION

The system for deployment and support of a solar-array blanket using the ESS is shown in Figures 14 and 15. The blanket is 4 m wide by 31.5 m long, with 80 accordian-pleat segments. It is attached to the beam on standoffs at a spacing of two baylengths ℓ.

Deployment of the ESS system is in a continuous motion, whereby each part of the blanket and structure move at an essentially constant velocity. The time required for deployment, in which the tip speed is 5 cm/s, is 10.5 min.

The beam shown in Figure 14 is sized to support the 53-kg Design 2 blanket at a cantilever frequency of 0.2 Hz. The dimensions are presented in Table III.

The package volume of the stowed 0.2-Hz structure is approximately 0.28 m^3, consisting of a rather flat box (1.7 m long by 1.25 m wide by 0.13 m high).

TABLE III. BLANKET DESIGN 2: POINT DESIGNS FOR
ESS, ASTROMAST, AND STACBEAM
(frequency = 0.2 Hz).

DESIGN COMPONENTS	ESS	ASTROMAST	STACBEAM
$M_{blanket}$, M_{bl}	53 kg	53 kg	53 kg
$M_{container}$, M_{cont}	12.1 kg	12.1 kg	12.1 kg
$M_{harness}$, M_h	4.0 kg	4.0 kg	4.0 kg
Blanket Tension	26 N	47 N	24N
Strut Diameter (wall)	0.68 cm (0.50 mm)	0.35 cm	0.31 cm
Bay Length (diameter)	1.65 m	(0.70 m)	0.45 m
M_{beam}	9.0 kg	4.1 kg	10.0 kg
$M_{depl\ mech}$	4.5 kg	38.8 kg	6.4 kg
$M_{actuator}$, M_a	2.6 kg	4.7 kg	2.4 kg
$M_{guide\ cables}$, M_{gc}	0.5 kg	0.5 kg	0.5 kg
$M_{battens}$, M_b	0.9 kg	0.7 kg	0.9 kg
$M_{standoffs}$, M_{so}	0.5 kg	0.5 kg	0.5 kg
System Mass	87.1 kg	118.4 kg	89.8 kg

$M_{bl} = 81$ kg; 0.643 kg/m^2

Astromast (AM)

Astro STACBEAM (SB)

Extendible Support Structure (ESS)
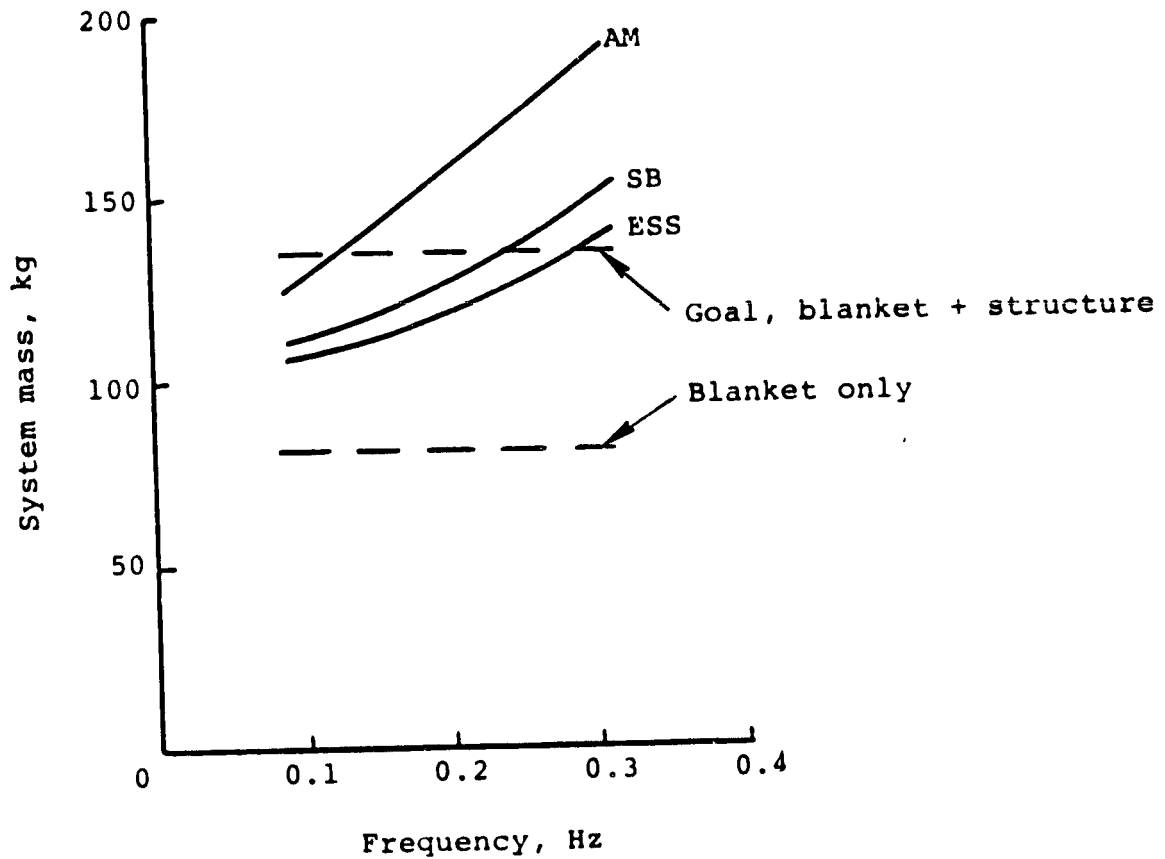
All graphite/epoxy

Figure 11. Astromast, STACBEAM, and ESS system mass versus frequency for Design 1 blanket.

$M_{bl} = 53$ kg; 0.421 kg/m^2

Astromast (AM)

Astro STACBEAM (SB)

Extendible Support Structure (ESS)
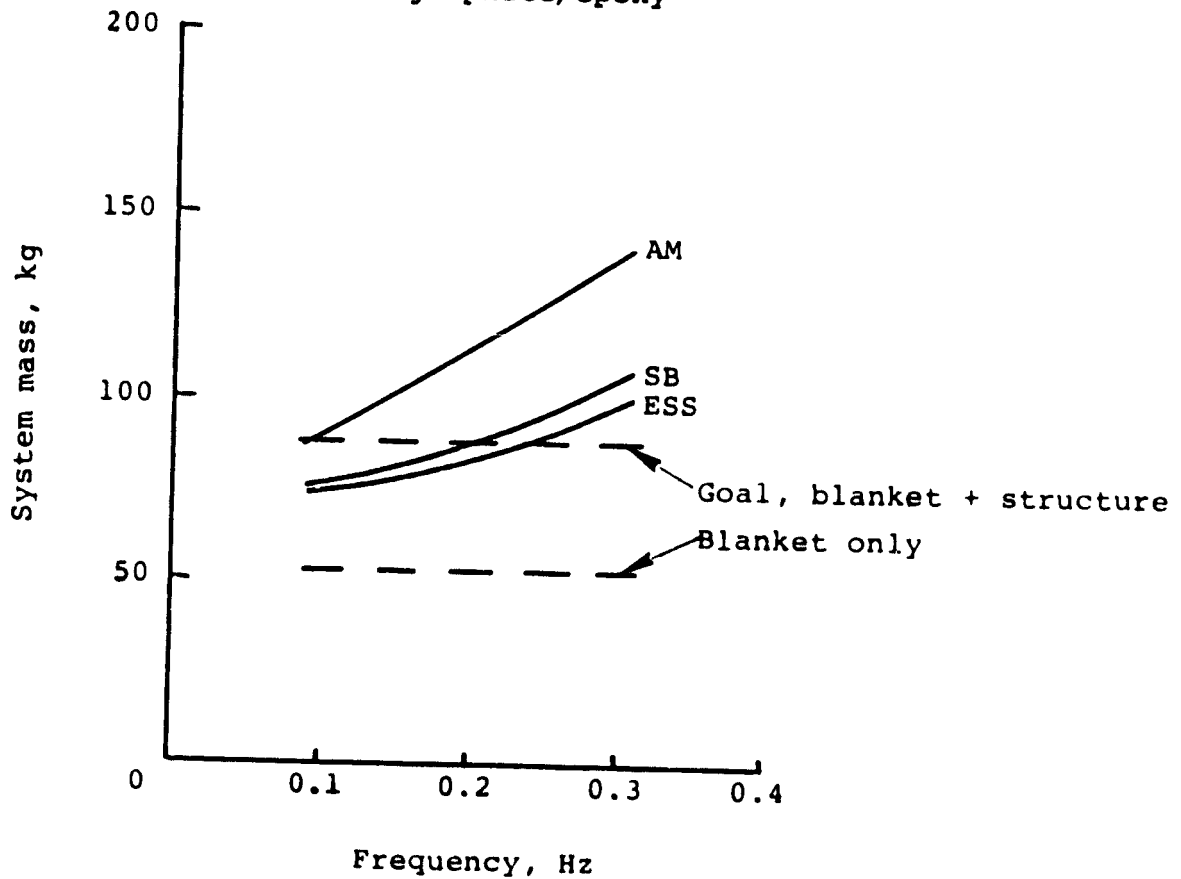
All graphite/epoxy

Figure 12. Astromast, STACBEAM, and ESS mass versus frequency for Design 2 blanket.
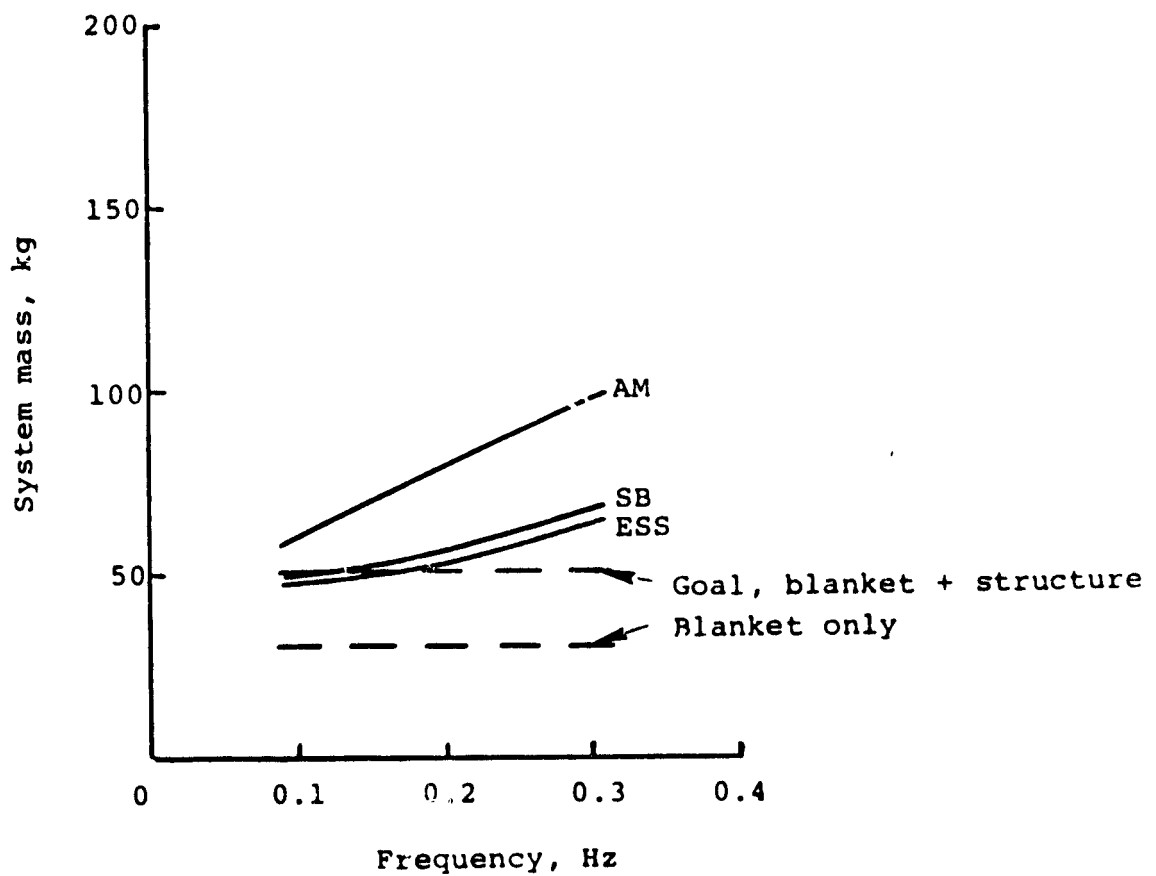
Figure 13. Astromast, STACBEAM, and ESS mass versus
frequency for Design 3 blanket.

Graphite/epoxy tubing:

d = 0.68 cm

t = 0.50 mm
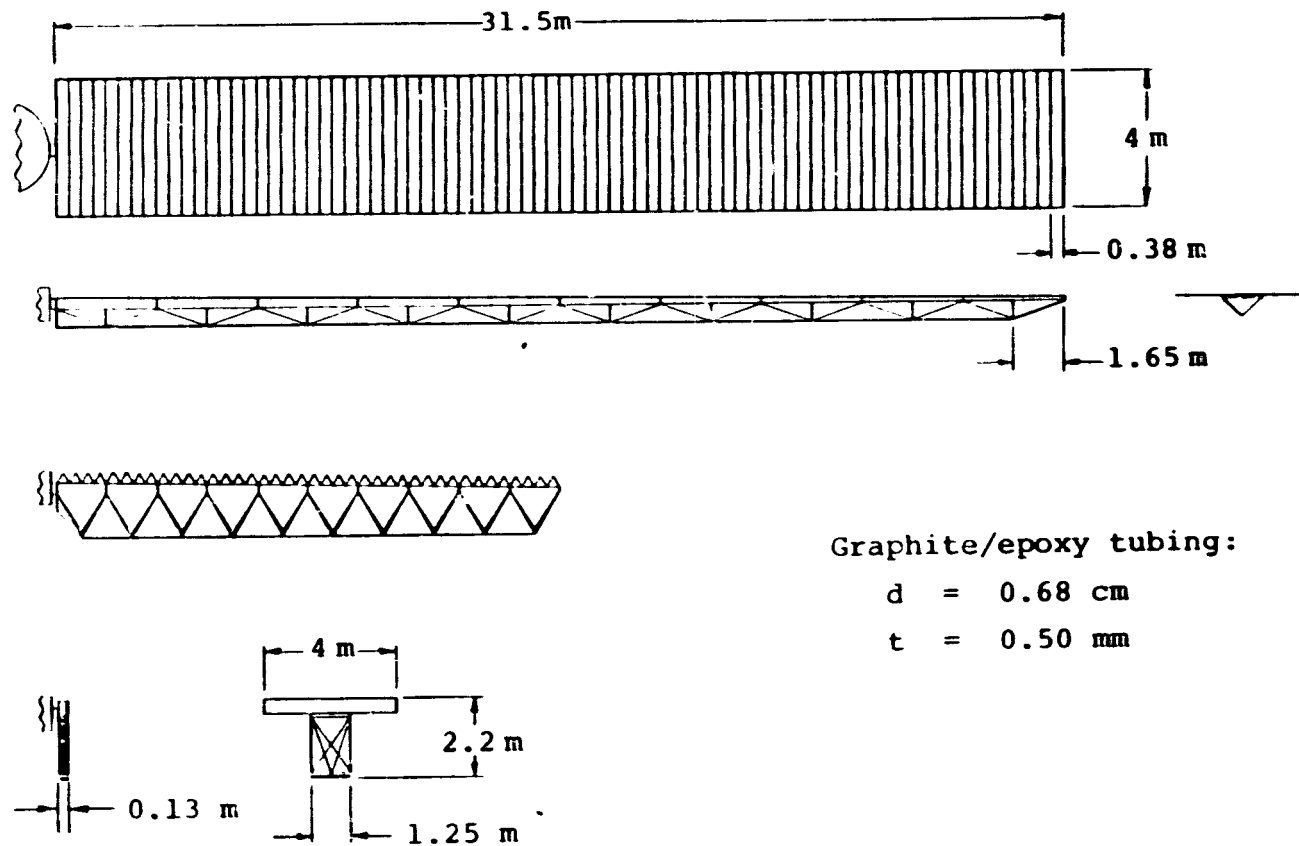
Figure 14. FSS system overall geometry.

See Detail A

Deployable
batten

Standoff

Detail A

0.13 m
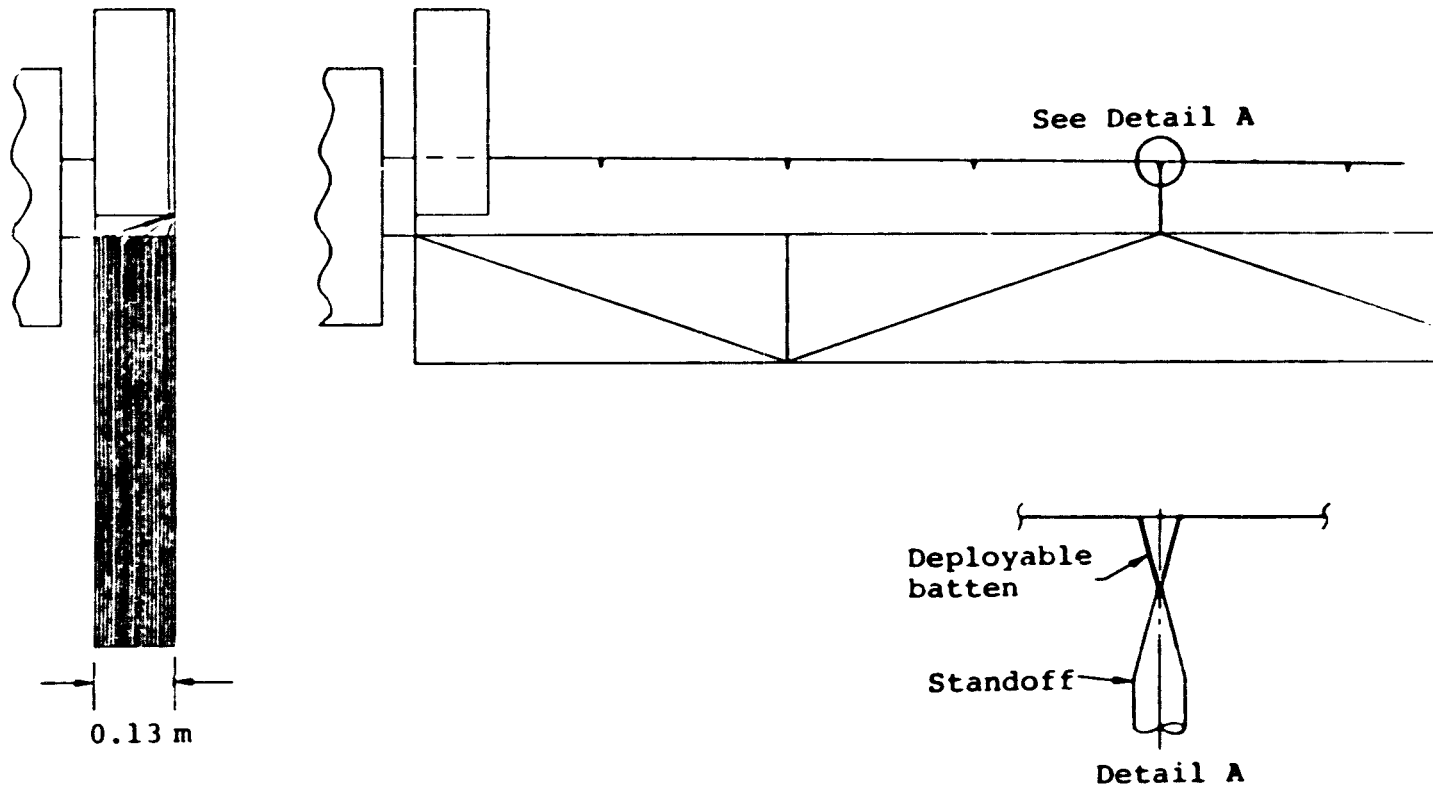
Figure 15. ESS system detail.

SECTION 7
ASTROMAST CONCEPT

The Astromast concept for deployment and support of a solar array is shown in Figure 16. The beam is of diameter D and consists of battens, diagonals, and continuous longerons (referred to as subscripts b, d, and ℓ, respectively). Battens are spaced at a distance ℓ, which is given the term baylength. In this concept, the blanket is necessarily detached from the structure in the packaged condition and is dynamically connected to the structure as it leaves the canister. This mode of attachment is required because the packaged portion of the Astromast rotates during deployment.

The method of sizing the structure is not the iterative method outlined in Section 5. Rather, it is a straightforward calculation of frequency versus beam size. This is because in order to package the Astromast by coiling the longerons, the longeron diameter is a strict proportion of boom diameter, or

$$d_\ell = \epsilon D \qquad (31)$$

This proportionality fixes the boom stiffness and mass as functions only of boom diameter.

7.1 BEAM PROPERTIES

7.1.1 Stiffness

The bending stiffness of the Astromast is a function of beam diameter D and longeron cross-sectional area A_ℓ, or

Figure 16. Astromast concept.

$$EI = \frac{3}{8} D^2 (EA)_\ell$$

$$= \frac{3}{8} D^2 E\pi \frac{\epsilon^2 D^2}{4}$$

$$= \frac{3\pi}{32} \epsilon^2 E D^4 \qquad\qquad (32)$$

7.1.2 Mass

The cantilevered mass of the Astromast is

$$M_{beam} = 3k_m A_\ell \rho Lk + M_{bl}$$

7.1.2.1 Mass of Members - The factor k_m relates beam mass to longeron mass as follows:

$$k_m = \frac{M_\ell + M_b + M_d}{M_\ell}$$

$$= 1 + \frac{b}{\ell} \frac{A_b}{A_\ell} + 2\frac{d}{\ell} \frac{A_d}{A_\ell}$$

$$= 1 + (1.39)(0.64) + (2)(1.71)(0.09)$$

$$= 2.20 \qquad\qquad \text{in standard Astromast}$$

Then

$$M_{beam} = 1.65 \pi\epsilon^2 \rho D^2 Lk + M_{bl} \qquad\qquad (33)$$

Combining Eqs. (32) and (33) with Eq. (13) yields the cantilever frequency as a function of the beam diameter.

The required blanket attachment spacing is then determined according to blanket frequency and longeron buckling considerations. Using Eqs. (16), (17), (18), and (19) yields

$$\ell_{att} = 0.7875 \frac{\varepsilon^2}{k_f f} \sqrt{\frac{EL}{k_p M_{bl}}} \, D$$

7.1.2.2 __Mass of Joints__ - The joint factor $k = 1 + KD^m$ is the ratio of mast weight, including joints, to the weight of the struts alone and has been evaluated in terms of mast diameter for a fiberglass Astromast. The joint mass of a fiberglass Astromast per unit length is

$$m'_{jg} = K_g D^m N \rho_g \varepsilon_g^2 D^2$$

where

$$K_g, m = \text{empirically derived numbers}$$

$$N = \frac{\text{total strut mass}}{\text{longeron mass}}$$

The joint mass per bay is

$$m_{jg} = m'_{jg} \frac{\ell}{D} D$$

$$= K_g N \frac{\ell}{D} \rho_g \varepsilon_g^2 D^{3+m}$$

In terms of joint mass, to use a different strut material of allowable strain ϵ is analogous to reducing the diameter of the fiberglass Astromast to $(\epsilon/\epsilon_g)D$. Thus, the ratio of joint mass to strut mass per bay is

$$\frac{m_j}{m_s} = \frac{K_g N \frac{\ell}{D} \rho_g \epsilon_g^2 \left(\frac{\epsilon}{\epsilon_g} D\right)^{3+m}}{N \rho \epsilon^2 D^2 \frac{\ell}{D} D}$$

$$= K_g \frac{\rho_g}{\rho} \left(\frac{\epsilon}{\epsilon_g}\right)^{1+m} D^m$$

The joint mass per unit length is

$$m'_j = \frac{m_j}{m_s} \frac{m_s}{\ell}$$

$$= K_g \frac{\rho_g}{\rho} \left(\frac{\epsilon}{\epsilon_g}\right)^{1+m} D^m N \rho \epsilon^2 D^2$$

$$= K D^m N \rho \epsilon^2 D^2$$

where

$$K = K_g \frac{\rho_g}{\rho} \left(\frac{\epsilon}{\epsilon_g}\right)^{1+m}$$

The empirically derived quantities for the fiberglass Astromast ($\rho_g = 2280$ kg/m^3 and $\epsilon_g = 0.015$) are

$$K_g = 0.3842$$

and

$$m = -0.48$$

These quantities, converted for a graphite Astromast (ρ = 1520 kg/m^3 and ι = 0.005), are

$$K = 0.3255$$

and

$$m = -0.48$$

7.2 SIZE-FREQUENCY RELATIONS

Table IV is a listing of an HP34-C calculator program which calculates beam frequency as a function of beam diameter and blanket mass. This program also calculates the maximum allowable blanket attachment spacing. Figure 17 illustrates the beam mass and frequency trends of the Astromast system.

7.3 DEPLOYMFNT MECHANISM

The Astromast is deployed by an active mechanism: a rotating nut, which has a diameter equal to the mast diameter, lifts the mast through a transition section during deployment. Required components include a rotating plate for the packaged portion of the mast, a cylindrical can, and the rotating nut. The mass of this assembly has been determined to be

$$M_{depl\ mech} = 4.3\ D + 73\ D^2$$

expressed in kilograms, for a 31.5-m-long Astromast where diameter D is expressed in meters. This equation was taken from reference 7 with the mass scaled by the ratio of graphite density to fiber-glass density.

TABLE IV. ASTROMAST MASS-FREQUENCY.

REGISTER		VALUE	FLAG	SET	CLEAR
I	f		0		
0	D		1		
1	M_{beam}		2		
2	E	1.245×10^{11}	3		
3	t	0.005			
4	$3\pi/32$	0.2945			
5	ρ	1520	ROUTINE		
6	L	31.5			
7	1.65π	5.18	A	Calc. f, M_{beam}, ζ_{att}	
8	M_{bl}	81,53,30	B	New M_{bl}	
9	$3.52/2\pi$	0.560			
.0	k_f	3			
.1	k_p	2			
.2	0.7875	0.7875			
.3	-0.48	-0.48			
.4	0.3255	0.3255			

ENTRY:　A:　Enter with D
　　　　　B:　New blanket mass

1	LA		20	R7	·
	SQ			x	
	x^2			R0	
	x^2			R.3	
	R2			y^x	
	x			R.4	
	R3			x	
	x^2			1	
	x			+	
				x	
10	R4				
	x	EI	30	S1	M_{beam}
	R0			R8	
	R3			+	
	x			:	
	x^2			R6	
	R5			y^x	
	x			Y	
	R6			:	
	x			,	
				R9	

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TABLE IV. (concluded).

```
40    x
      SI      f
      R0
      ← →
      ⋮
      R3
      x²
      x
      R.0
      ⋮

50    R2
      R6
      x
      R.1
      ⋮
      R8
      ⋮
      ‚
      x
      R.2

60    x
      R0            D
      ← →           ℓ
      R1           M att
                     beam
      RI      EXIT:  f
      RTN
      LB
      S8
      RTN
```

Blanket: 81, 53, 30 kg; 0.643, 0.421, 0.238 kg/m^2
Longeron diameter = 0.005 x mast diameter
Graphite/epoxy
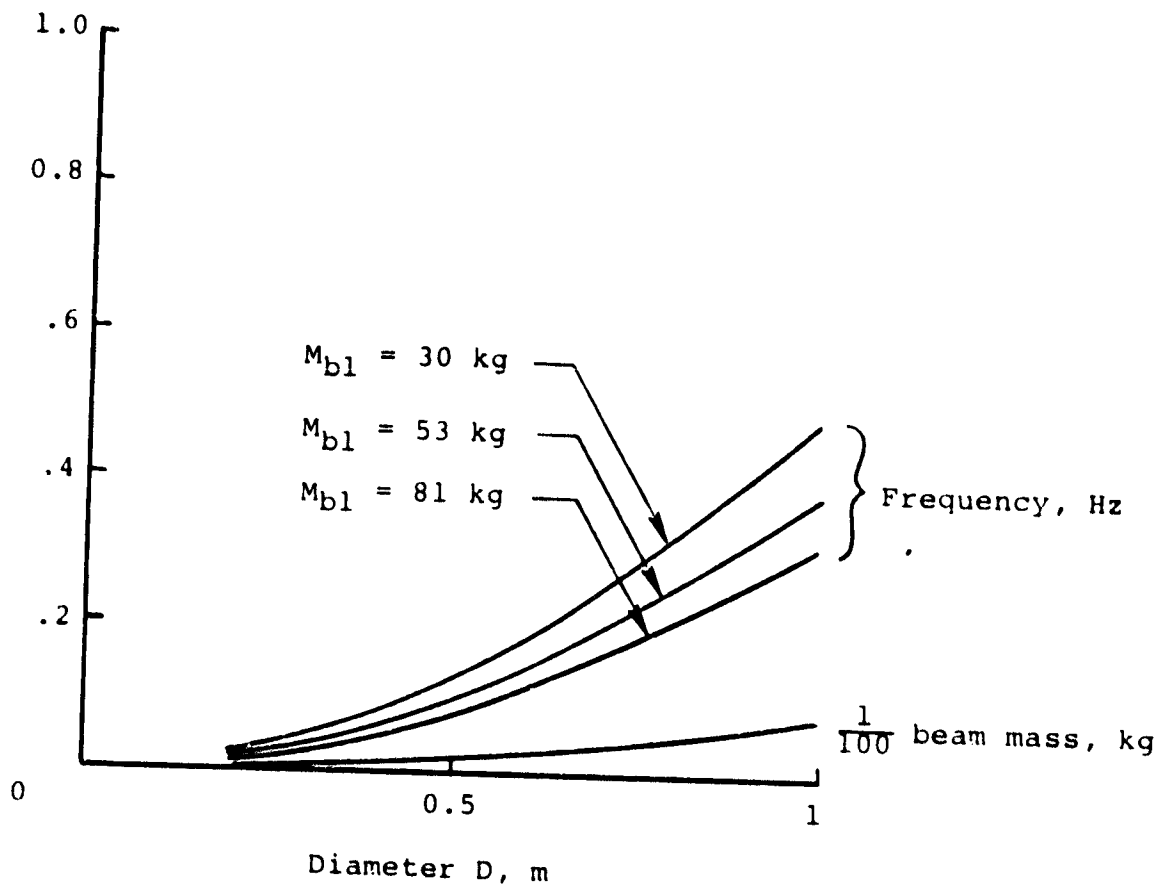
Figure 17. Astromast configuration for Designs 1, 2, and 3.

7.4 SYSTEM MASS

The mass of the Astromast system, using Eq. (1), is itemized for Design 2 at a 0.2-Hz cantilever frequency as shown in Table III. The design goal (blanket mass/system mass ≥ 0.6) is not met in this case. The canister weighs 38.8 kg and must be reduced to 8.7 kg in order to meet the design goal. The Astromast system mass is shown, as a function of frequency, for Designs 1, 2, and 3 in Figures 11, 12, and 13, respectively.

7.5 SYSTEM DESCRIPTION

The system for deployment and support of a solar-array blanket using an Astromast is shown in Figures 18 and 19. The blanket is 4 m wide by 31.5 m long, with 80 accordian-pleat segments. It is attached to the beam with standoffs at a sufficient number of points to maintain a high blanket frequency. Battens are attached to the blanket at the standoff locations to provide lateral stiffness.

The method of deployment is as follows: The Astromast deploys from an initial coiled condition by means of a standard canister. As the Astromast emerges from the canister, the blanket is attached to the mast by the method shown in Figure 19. A tip speed of 5 cm/s indicates a deployment time of 10.5 min.

The beam shown in Figure 18 is sized to support the 53-kg Design 2 blanket at a cantilever frequency of 0.2 Hz. The dimensions are presented in Table III.

The packaging volume of the stowed 0.2-kg structure is approximately 0.75 m^3, consisting of a cylinder 0.75 m in diameter and 1.7 m in length.
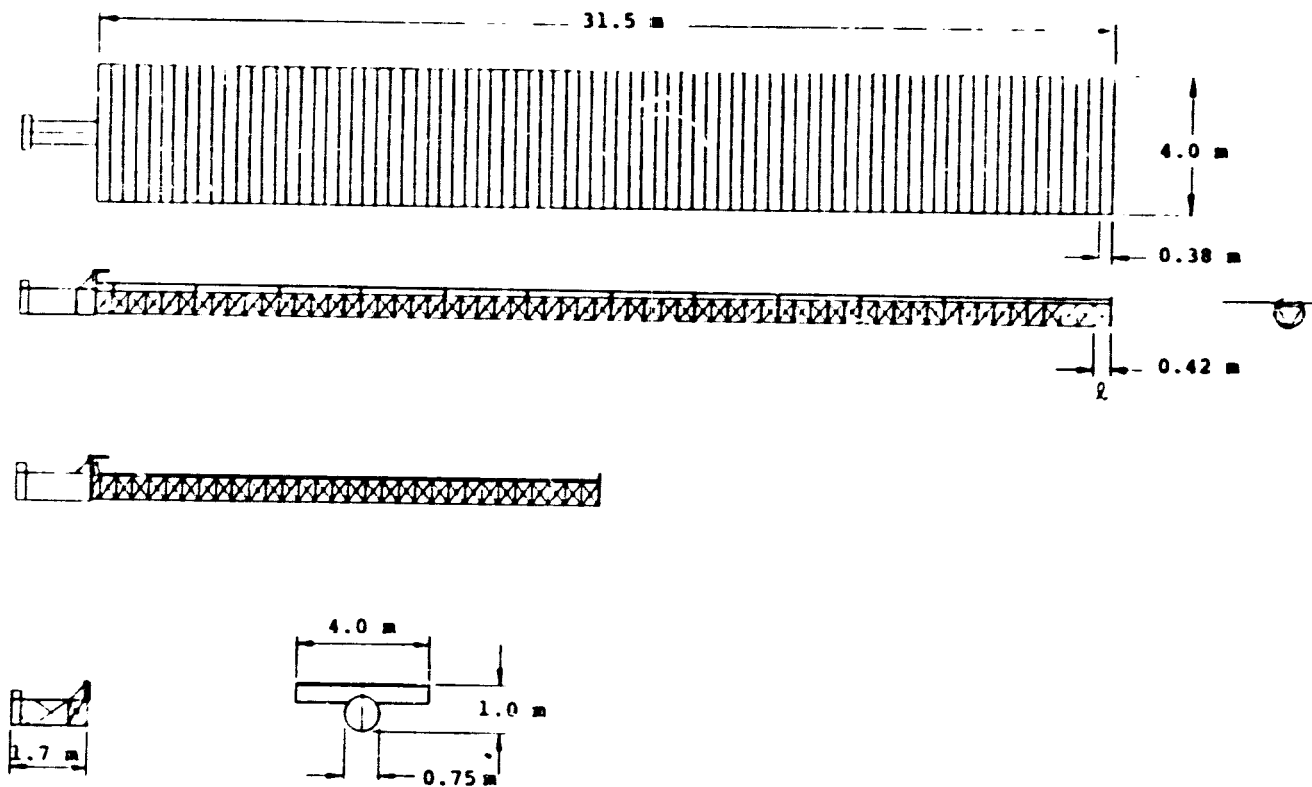
Figure 18.   Astromast system overall geometry.
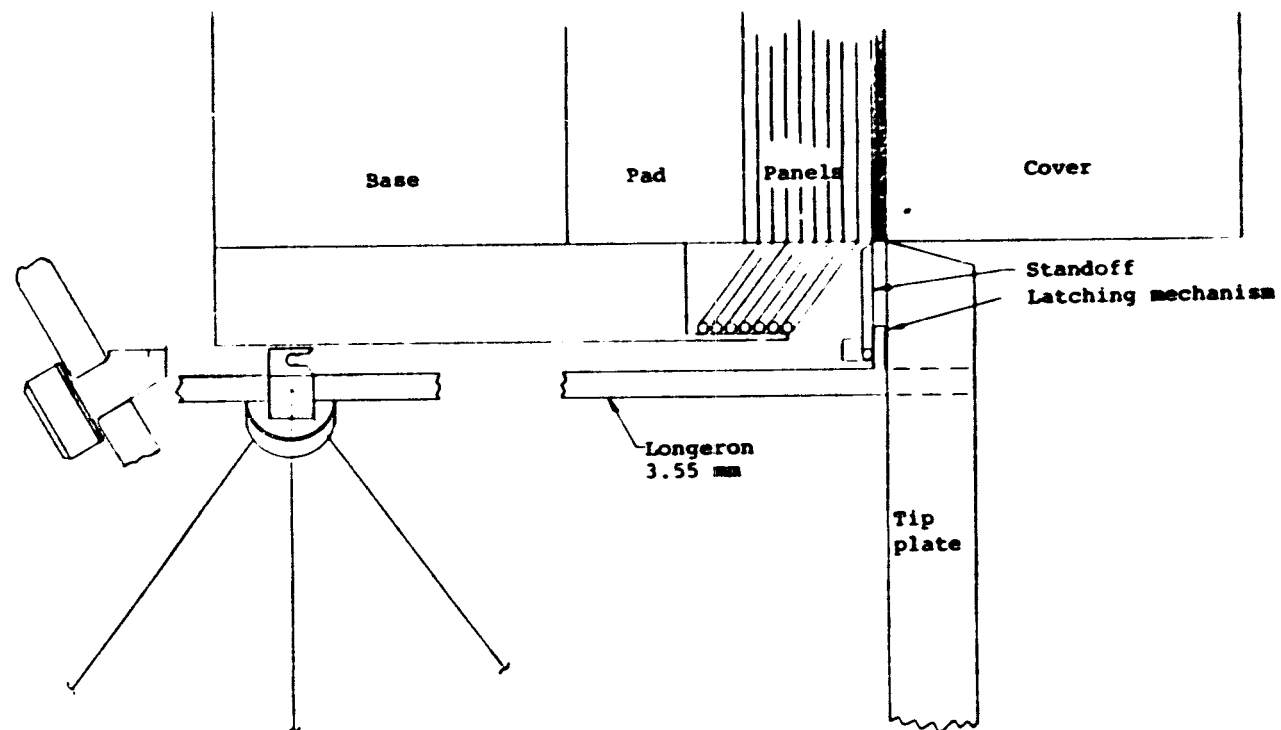
Figure 19.   Astromast stowed configuration detail.

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SECTION 8
STACBEAM CONCEPT

The Stacking Triangular Articulated Compact Beam (STACBEAM) concept for deployment and support of a solar-array blanket is shown in Figures 20 and 21. Special characteristics of the STACBEAM structure are as follows:

- It deploys sequentially, one bay at a time, by unfolding and locking hinges provided at the midpoint of each longeron and diagonal.
- The battens move linearly, without rotation, during this deployment.
- All hinges have a single degree of freedom.

The structure is sized using the iterative method outlined in Section 5. The beam is of width b, depth h and baylength ℓ, and is composed of solid rods. Equations (13) and (20) are thus used to size the beam by the Euler buckling method.

8.1 BEAM PROPERTIES

Equation (13) contains terms which pertain to intrinsic beam properties: stiffness and mass.

8.1.1 Stiffness

The expression for the stiffness of the STACBEAM is the same as for the ESS.

$$EI = \frac{2}{3} h^2 EA$$

where

h = beam depth

A = member area

Diagonals

$b = \sqrt{3}\ell$

ℓ

$h = \frac{3}{2}\ell$

Single degree of
freedom hinges

Battens

Longerons
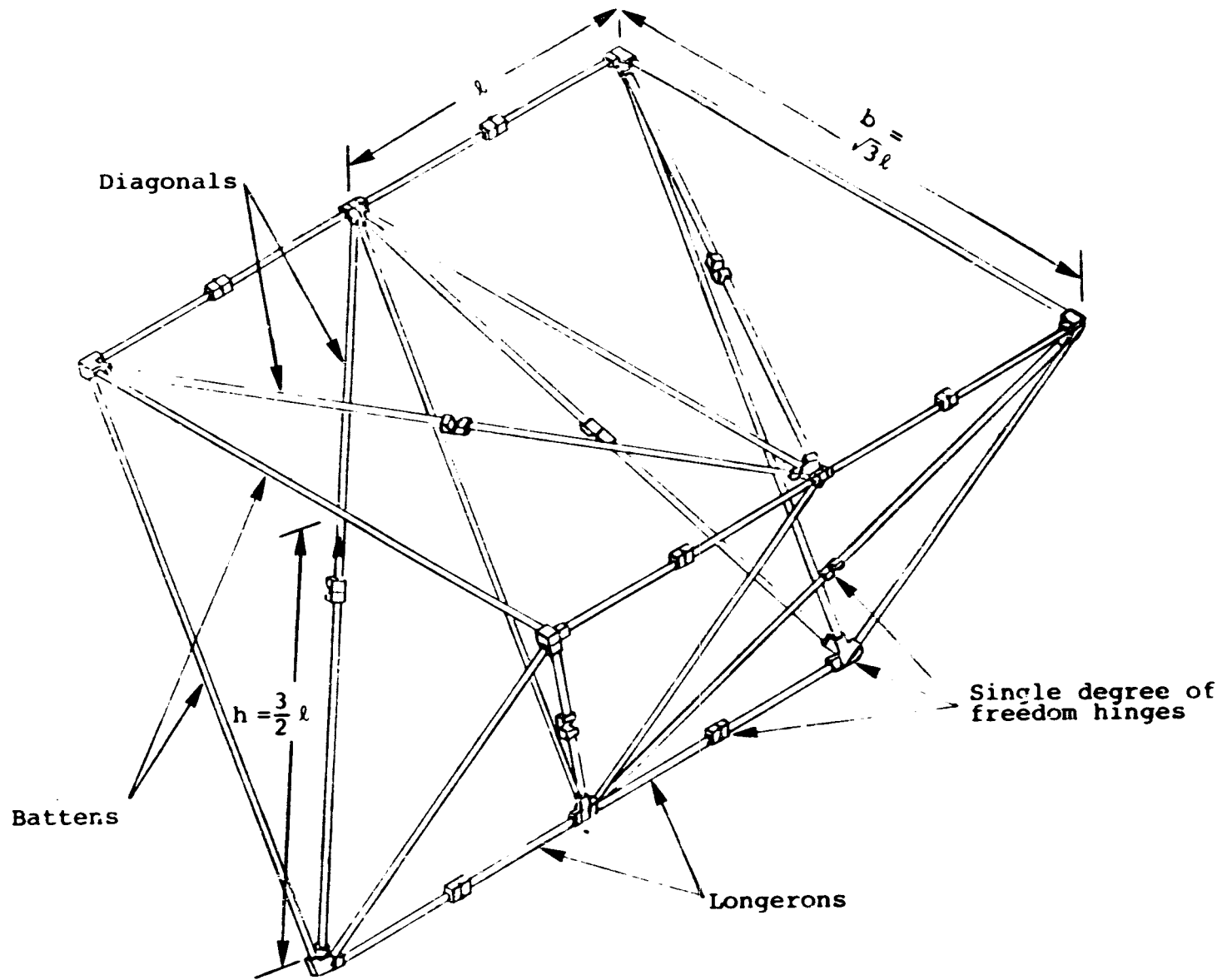
Figure 20. Schematic sketch of two bays of a STACBEAM column.

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a. Packaged

b. Partially deployed bay
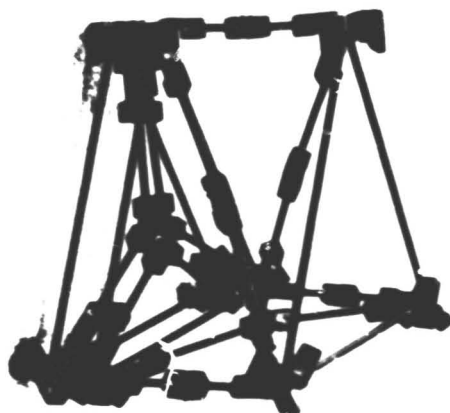
c. Fully deployed bay
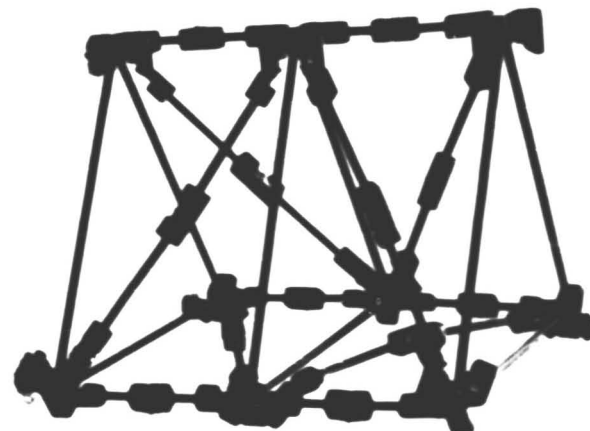
d. Two deployed bays

Figure 21. STACBEAM engineering model.

As shown in Figure 20, the beam depth h is related to bay-length ℓ by

$$h = \frac{3}{2} \ell$$

Then

$$EI = \frac{3}{2} \ell^2 EA \tag{34}$$

8.1.2 Mass

The cantilevered mass of the STACBEAM includes the masses of the members and joints, and the blanket payload itself.

$$M = M_{mem} + M_j + M_{bl} \tag{35}$$

8.1.2.1 Mass of Members - The member mass of their linear density times the beam length, or

$$M_{mem} = m'_{mem} L \tag{36}$$

This density is the sum given by the longerons, battens, and diagonals

$$m'_{mem} = m'_\ell + m'_b + m'_d$$

$$= 3\rho A_\ell + 3\sqrt{3} \rho A_b + 6\rho A_d$$

$$= 14.2 \ \rho A \qquad \text{for } A_\ell = A_b = A_d = A \tag{37}$$

8.1.2.2 Mass of Joints - The joint mass in the STACBEAM is assumed equal to the member mass. Then the joint factor

$$k \qquad \frac{M_1}{M_{mem}} + 1$$

is assigned the value $k = 2$.

8.2 SIZE-FREQUENCY RELATIONS

Table V is a listing of an HP-34C calculator program which correlates payload mass and frequency with STACBEAM mass and frequency; the latter three terms being determined by member length and cross-sectional properties as outlined in Section 5. Figures 22, 23, and 24 indicate member size, beam mass, and frequency trends of the STACBEAM system, using members consisting of solid rods.

8.3 DEPLOYMENT MECHANISM

The STACBEAM uses a reciprocating mechanism which deploys each bay separately, as shown in Figure 25. Each of the telescoping tube assemblies is sized to equal the stiffness of the STACBEAM as follows:

$$(EI)_t = (EI)_{SB}$$

$$\pi r_t^3 \, t_t E_t = \frac{3}{2} \, \ell_{SB}^2 \, \frac{\pi}{4} \, d_{SB}^2 \, E_{SB}$$

$$t_t = \frac{3}{8} \, \frac{(\ell d)_{SB}^2}{r_t^3} \, \frac{E_{SB}}{E_t}$$

The tube linear density is

$$m' = 2\pi (rt\rho)_t$$

$$= \frac{3\pi}{4} \left(\frac{\ell_{SB} d_{SB}}{r_t} \right)^2 \frac{E_{SB}}{E_t} \, \rho_t$$

TABLE V. STACBEAM MASS-FREQUENCY.

REGISTER		VALUE		FLAG	SET	CLEAR
I	f			0	Tube	Solid
0	t	0.00075,-50, -25		1		
1	ℓ			2		
2	d			3		
3	M_{beam}					
4	M_{bl}	81,53,30				
5	L	31.5				
6	ρ	1520				
7	E	1.1×10^{11}				
8	ℓ_{tot}/L	14.2				
9	k_j	2.0				
.0	k_p	2.0				
.1	k_f	3.0				
.2	n_f	10				

ROUTINE

A	Tube
B	Solid
0	Continue
1	Solve for d
2	Continue
3	Tube
4	π
5	Tube
6	Continue

ENTRY: Enter with ℓ
      A: Tube
      B: Solid

```
1      LA      Tube:              20      RTN
       SFO     set flag                   L1
       GTO                                S2
       LB      Solid:                     PSE
       CFO     clear flag                 F?0
       L0                                 GT3
       S1                                 x²
       R2    ⎫                            4
        ↑    ⎬ guess d                    ÷
10      ↑    ⎪                            GT4
       +     ⎭              30      L3  ⎫
       Solve 1  Solve for d              R0  ⎬ Tube
       GT2                               x   ⎭
       R/S                               L4
       L2                                π
       R1          ℓ                     x
       R3          M_beam                ↑
       RI          f                     ↑
       R2     EXIT:  d                   R8
                                         x
```

TABLE V. (concluded).

40	R6		80	:	
	x			R.2	
	R5			x	
	x			R1	
	R9			:	
	x			π	
	S3	M_{beam}		1	
	↔			.	
	1			5	
	.			y^x	
50	5		90	x	
	R1			4	
	x^2			:	
	x			P.2	
	R7			F?0	
	x			GT5	
	x	EI		x^2	} Solid
	R3			x	
	R4			2	
	+			:	
60	:		100	2	
	R5			┌	
	3			:	
	y^x			GT6	
	:			L5	} Tube
	┌			3	
	.			y^x	
	5			R0	
	6			x	
	x			┌	
70	SI	f	110	x	
	R7			L6	f_{buckle}
	R.0			−	
	:			EEX	
	R4			3	
	:			x	
	R5			INT	
	:			x ≠ 0	
	┌			LSTX	
	R.1			RTN	

Structure: Graphite/epoxy rods

Blanket: 81 kg; 0. ＿ kg/m^2

Figure 22. STACBEAM configuration: Design 1.

Structure: Graphite/epoxy rods

Blanket: 53 kg; 0.421 kg/m^2

Rod diameter d, cm

Frequency, Hz

$\frac{1}{100}$ beam mass, kg

Bay length ℓ, m
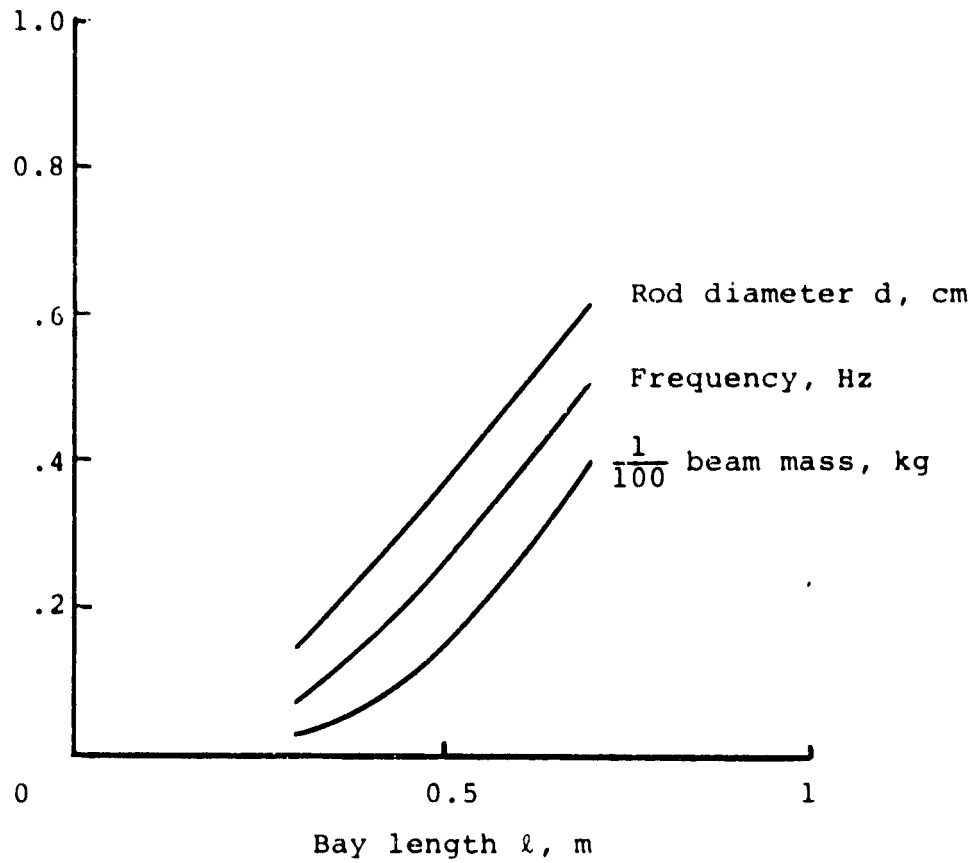
Figure 23. STACBEAM configuration: Design 2.
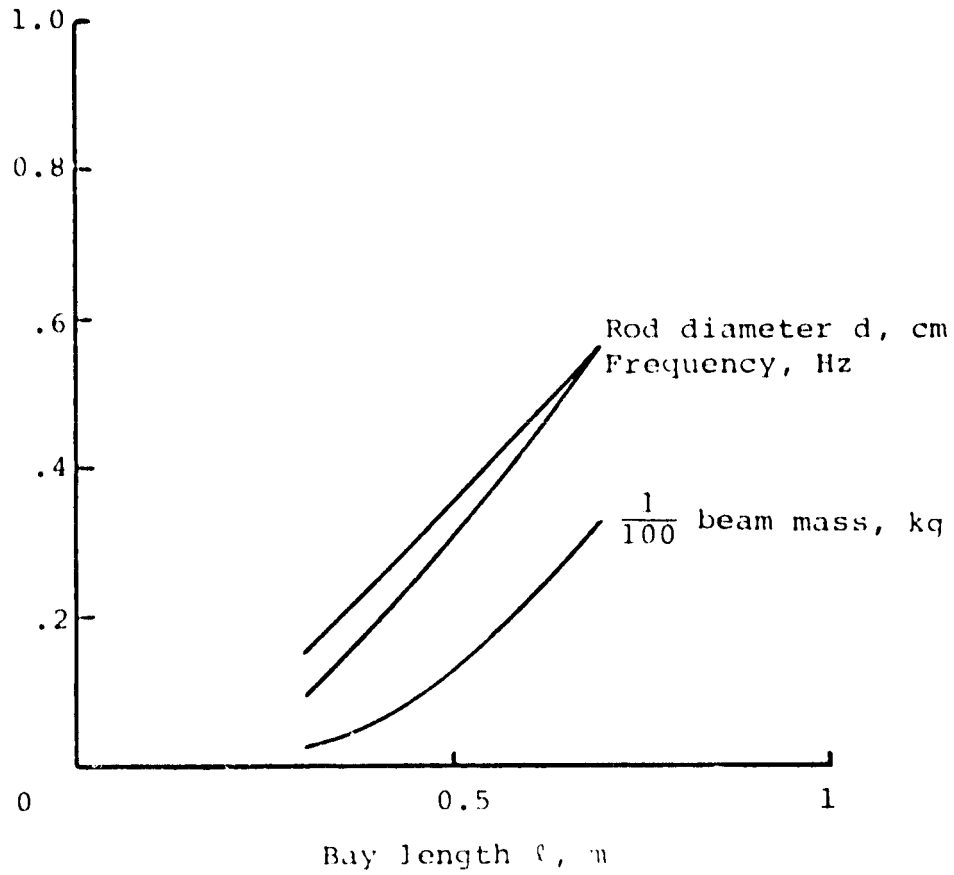
Figure 24. STACBEAM configuration: Design 3.

Stowed mast

Partial deployment
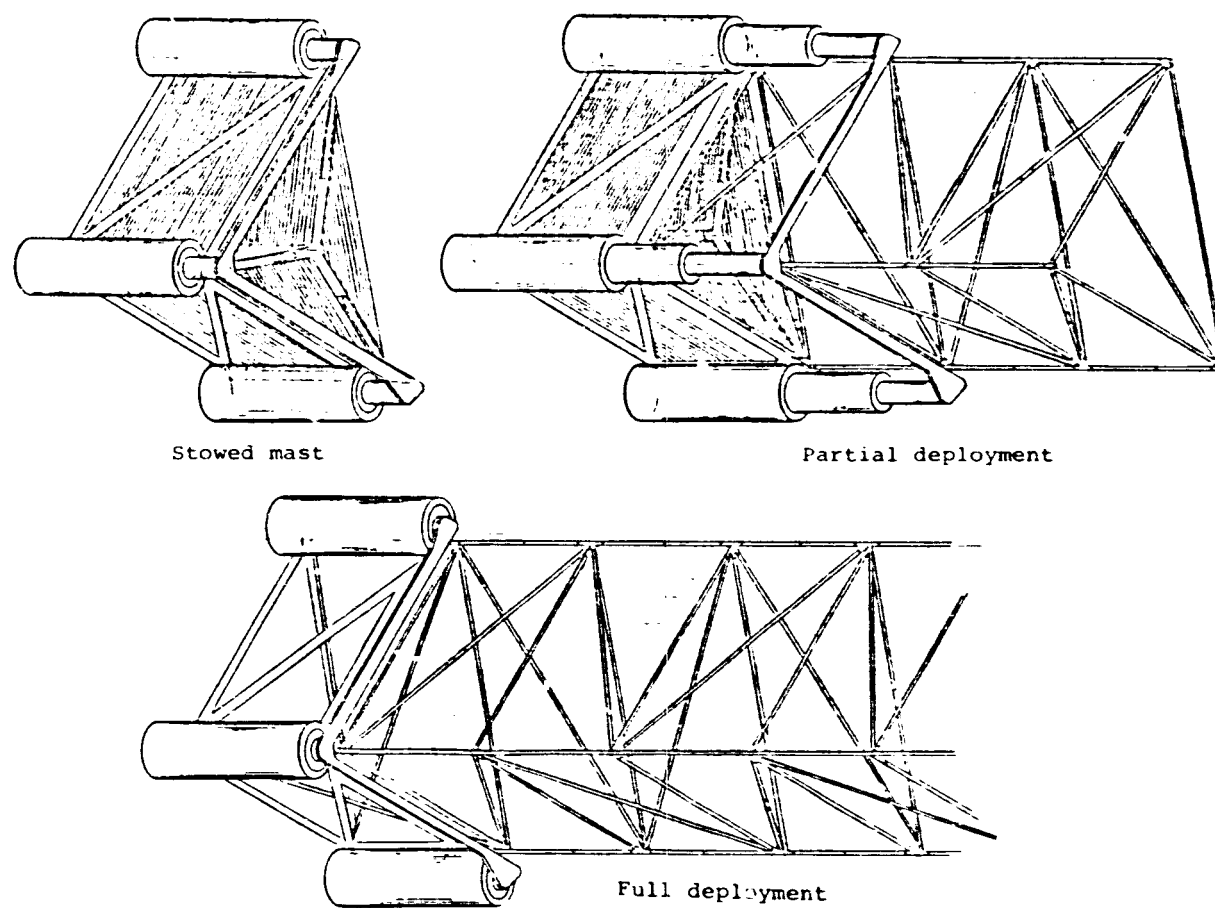
Full deployment

Figure 25.   STACBEAM system deployment mechanism.

The mass of the mechanism is the product of this linear density and the length of stiff tubing. Each of the three telescoping assemblies consists of three nested tubes of a length approximately the baylength so that the mechanism mass is

$$M_{depl\ mech} = 9\ m'\ell_{SB}$$

The tube radius is now arbitrarily set at one-fourth the beam baylength, yielding a deployment mechanism mass of

$$M_{depl\ mech} = 108\ \pi\left(\ell d^2 E\right)_{SB}\left(\frac{\rho}{E}\right)_t$$

8.4 SYSTEM MASS

The mass of the STACBEAM system, using Eq. (1) is itemized for Design 2 at 0.2-Hz cantilever frequency as shown in Table III. It appears that the design goal (blanket mass/system mass ≥ 0.6) is attainable at this frequency. The STACBEAM system mass for Designs 1, 2, and 3 is given as a function of frequency in Figures 11, 12, and 13.

8.5 SYSTEM DESCRIPTION

The system used for deployment and support of a solar-array blanket using a STACBEAM is shown in Figures 25, 26, 27, 28, and 29. The blanket is 4 m wide by 31.5 m long, with 80 accordian-pleat segments. It is attached to the beam on standoffs at ten equally spaced intervals. Battens are attached to the blanket at these locations.

The deployment mechanism erects the stacked assembly in a sequential manner, which ensures stiffness and integrity during deployment. Continuity of stiffness is established as shown in Figure 25. Telescoping erection tubes are used which equal the STACBEAM stiffness. During deployment, the beam consists of three distinct regions:

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Assembled view with corner of
batten frame held by starwheels
of keeper assembly and engaged
by lifter assembly.

Keeper assembly with
gear linkage for
synchronized starwheel
rotation

Undeployed
STACBEAM

Lifter
assembly to
extend batten
frame
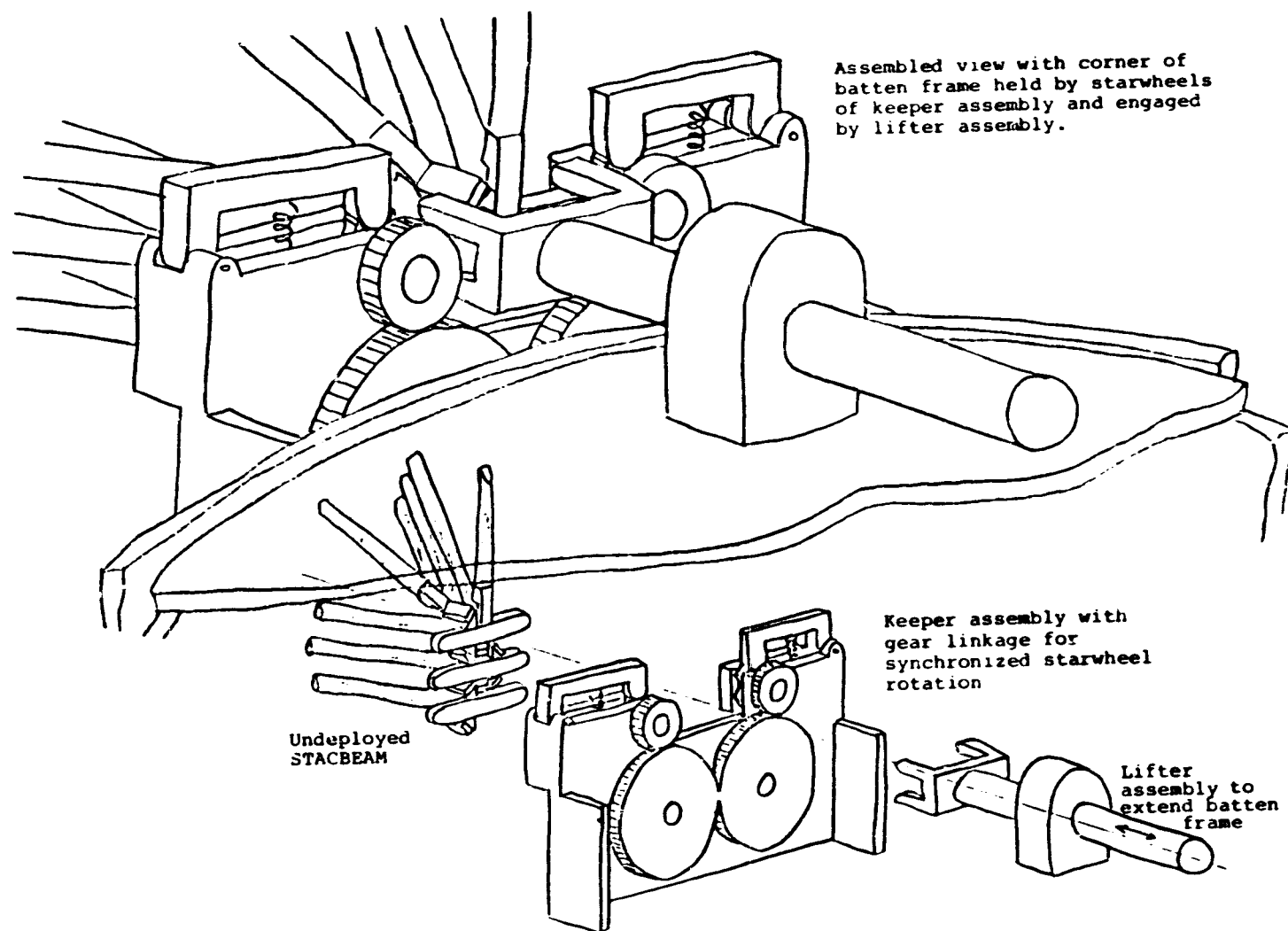
Figure 26.   Detail of batten deployment mechanism (assembled and
exploded views).
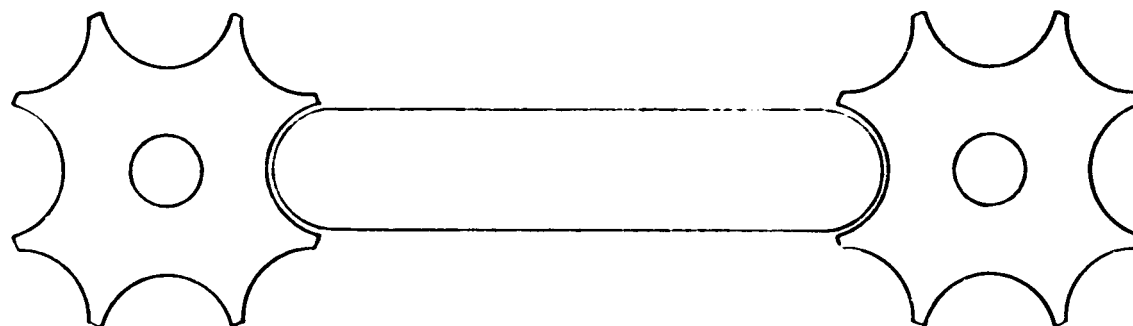
a. Detent position 1: Star wheels hold batten frame
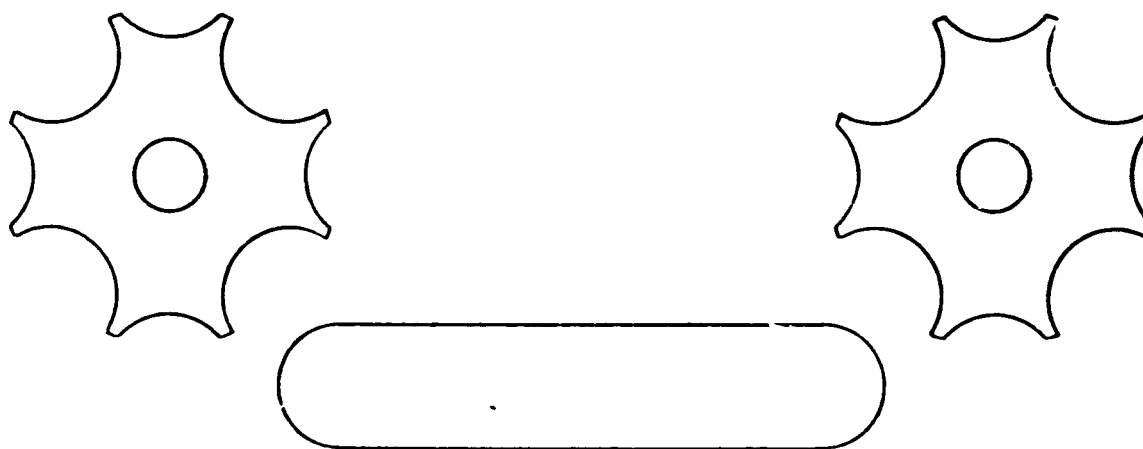
b. Detent position 2: Star wheels await next batten frame

Figure 27. Star wheel component of keeper assembly.

Deployable battens at each blanket hinge point

31.5 m

4.0 m

Full
deploy-
ment

Attachments between solar blanket
and STACBEAM column

Solar blanket   0.39 m

Accordian folds in solar
blanket

STACBEAM column

Partial
deploy-
ment

4.0 m

Packaged
config-
uration

0.43 m 0.75 m

1.06 m
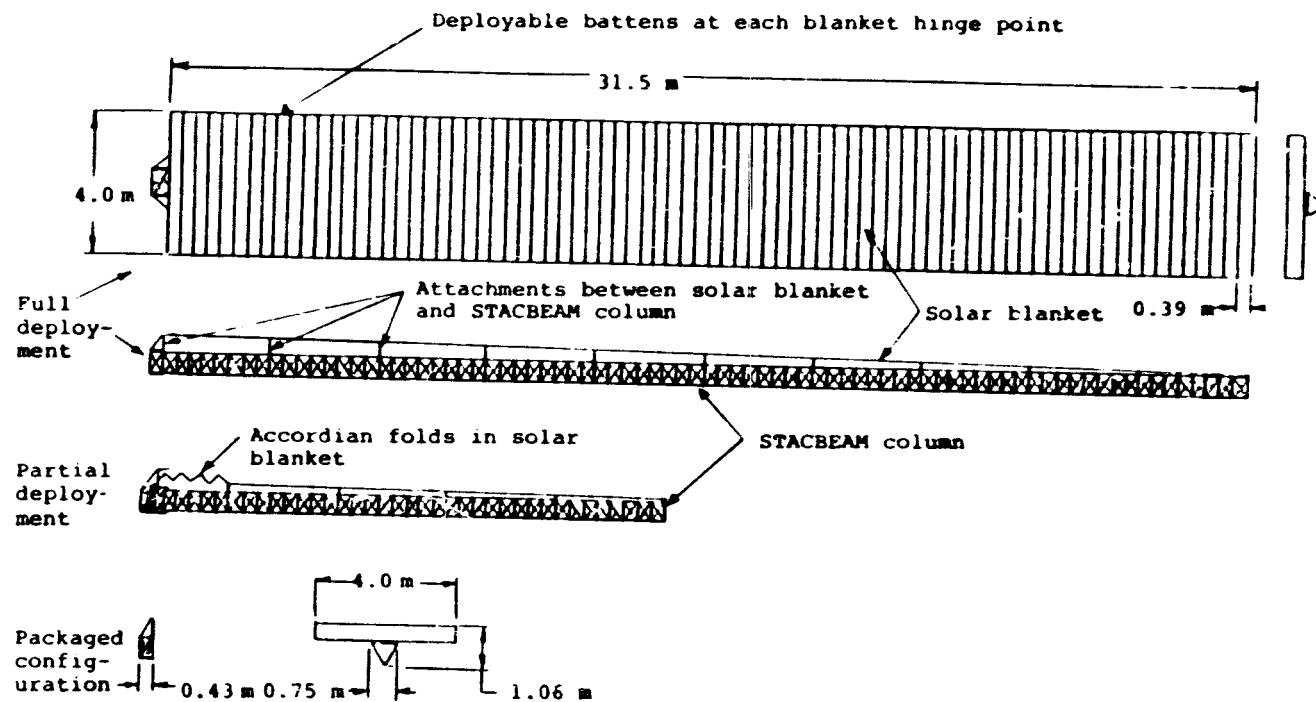
Figure 28.   STACBEAM solar blanket support structure.

Folded solar blanket

Solar blanket stowage bin

Foam pads

Honeycomb cover panels

Storage bin supports

Solar blanket folded behind cover panels

Storage bin support

Standoffs

Deployer jack screws

Deployer star wheel assembly

Folded longeron

Folded diagonal

Batten

Deployer jack screws

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Figure 29.   Packaged STACBEAM solar-array system.

- Region 1 is characterized by fully deployed bays, with all members extended, which form a rigid lattice structure.

- Region 2 is characterized by undeployed bays.

- Region 3 connects Regions 1 and 2, and consists of one partially erected bay.

The partially deployed bay of Region 3 is erected in a controlled fashion as follows. The outer end of this bay has its batten frame extended rigidly by three lifter assemblies, one at each corner, as shown in Figure 26. When full extension of this bay is reached, the three corners of the inner batten frame are brought into, and held rigidly by, the keeper assembly, Figure 26. The basic component of the keeper assembly is a pair of star wheels which engage the corner of the batten frame (see Figure 27). The rotation of each pair of star wheels is coupled so that the batten frame does not twist during engagement. The lifter 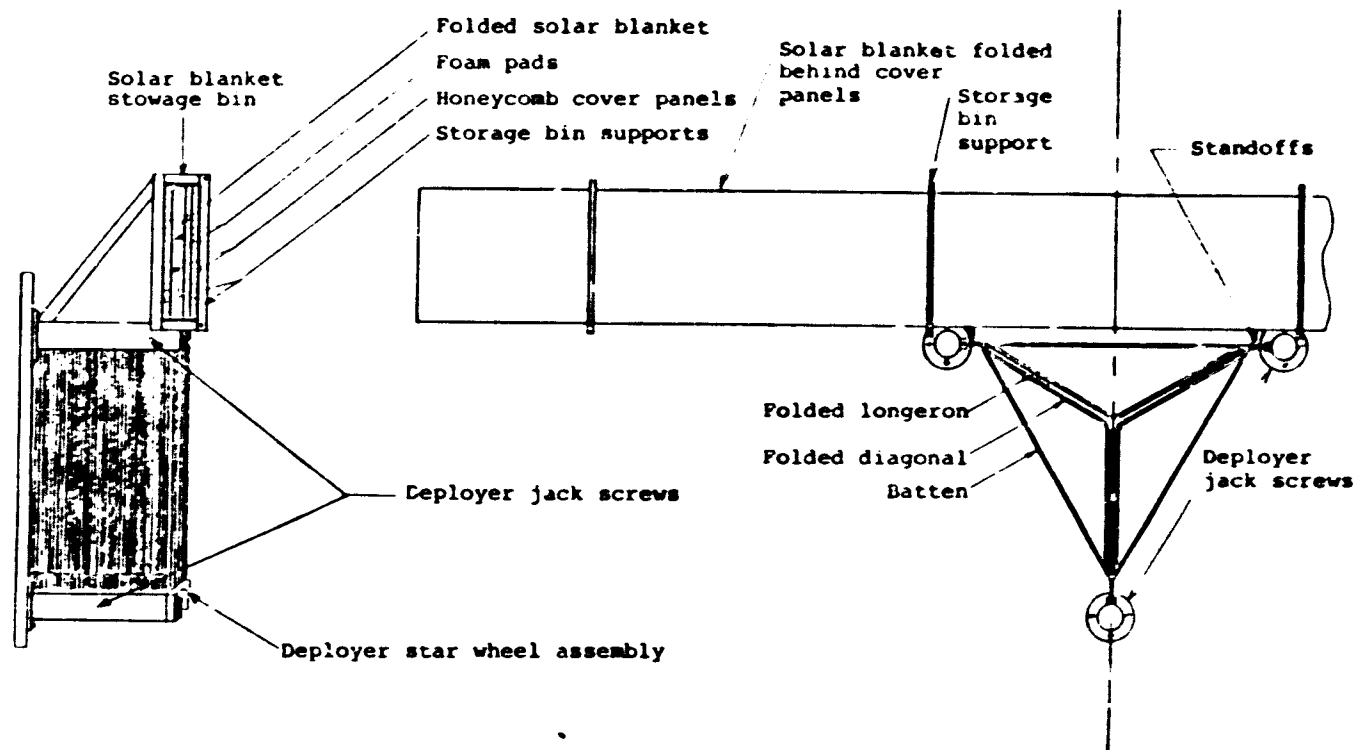assembly then disengages the outer batten frame, retracts, and engages the inner batten frame which is held by the keeper assembly.

The beam shown in Figure 28 is sized for Design 2 (53-kg blanket) at a frequency of 0.2 Hz. It is constructed of 0.31-cm-diameter solid members with a baylength of 0.45 m. This results in a beam mass of 10.2 kg. Additional details of the STACBEAM system are shown in Figure 29.

The reciprocal mode of deployment results in a longer deployment time compared to the other concepts. Assuming a sinusoidal reciprocation velocity with ±5 cm/s peaks, the average deployment velocity is 3.19 cm/s and the required deployment time is 33.0 min.

When stowed, the beam package resembles a triangular prism with 0.78-m sides and a height of 0.45 m. The packaging volume of this 0.2-Hz structure (excluding blanket package) is approximately 0.24 m^3.

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SECTION 9

COMPARISONS

The three structural concepts for blanket support are evaluated by using several bases of comparison: system mass, torsional stiffness, simplicity and reliability of deployment, package size, deployment force, deployment time, and growth adaptability.

9.1 SYSTEM MASS

Table III itemizes masses for Design 2 at 0.2 Hz and Figures 11, 12, and 13 indicate system mass trends, as a function of cantilever frequency, for each of the three structural concepts and each blanket design. Figure 11 shows that any of the three structural concepts is acceptable in terms of mass for Design 1 at frequencies of 0.1 Hz or less. For support of lighter blankets (Designs 2 and 3) the Astromast canister mass is too great a fraction of the blanket mass and causes the system mass to exceed the design goal.

9.2 TORSIONAL FREQUENCIES

The torsional frequency of the solar blanket arrays is affected not only by the beam stiffness but also by blanket tension which produces torques that oppose the direction of beam twist. The expression for beam torsional frequency is

$$f_{tors} = \frac{1}{2\pi} \left(\frac{\frac{Ta^2}{12L} + \frac{GJ}{L}}{\frac{M_{bl}a^2}{36} + I} \right)^{1/2}$$

for a blanket of mass M_{bl}, length L, width a, tension T, and a beam of torsional rigidity GJ and mass moment of inertia I about its longitudinal axis (see ref. 7).

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A beam of mass M with an equilateral triangular cross section of circumscribed radius r has a mass moment of

$$I = \frac{1}{2}\left(1 + \frac{M_\ell}{M}\right)Mr^2$$

where M_ℓ/M is the fraction of beam mass along the longerons.

9.2.1 ESS System

9.2.1.1 Torsional Rigidity - The torsional rigidity of the ESS (ref. 6) is

$$GJ = \frac{b^2 h^2}{\ell^2}\left\{1 + 2\frac{a^3}{\ell^3}\frac{(EA)_\ell}{(EA)_d} + \frac{d^3}{\ell^3}\frac{(EA)_\ell}{(EA)_d}\right.$$

$$\left. + \frac{1}{2}\frac{b^3}{\ell^3}\left[\frac{(EA)_\ell}{(EA)_{b_1}} + \frac{(EA)_\ell}{(EA)_{b_2}}\right]\right\}^{-1}(EA)_\ell$$

This function, evaluated for Seasat proportions ($a/\ell = 1.121$, $b/\ell = 0.756$, $d/\ell = 1.254$, $h/\ell = 0.336$, all EAs equal) is

$$GJ = 0.40 \, \ell^2 \, EA$$

9.2.1.2 Moment of Inertia - The fraction of beam mass, including joints, in the longerons of the ESS is

$$\frac{M_\ell}{M} = 0.73$$

The ESS radius, since its cross section is not an equilateral triangle, is approximated by

$$r_{ESS}^2 = \left(\frac{2}{3}h\right)^2 = 0.050 \, \ell^2$$

Then

$$I_{ESS} = \frac{1}{2} (1 + 0.73)M(0.050) \ell^2$$

$$= 0.043 \ M\ell^2$$

9.2.1.3 <u>Torsional Frequencies</u> - Using the above equations, tor-
sional frequencies for the ESS supporting either of the three
blanket designs, at cantilever frequencies in the range of 0.1 to
0.3 Hz, are consistently higher than the cantilever frequency by
a factor of 30 to 35.

9.2.2 <u>Astromast System</u>

9.2.2.1 <u>Torsional Rigidity</u> - The torsional rigidity of the Astro-
mast is

$$GJ = \frac{D^2}{8} \left(3 \sin \beta \cos^2 \beta \ EA_d \right)$$

where

β = batten-diagonal angle, ~35.7 degrees

$A_d = \dfrac{\pi (0.0015 \ D)^2}{4}$

$E = 1.245 \times 10^{11} \ N/m^2$

Then

$$GJ = 3.18 \times 10^4 \ D^4$$

9.2.2.2 <u>Moment of Inertia</u> - The fraction of beam mass in the longerons of the Astromast is

$$\frac{M_\ell}{M} \approx 0.61$$

Then

$$\tau = \frac{1}{2} (1 + 0.61) M \frac{D^2}{4}$$

$$= 0.20 MD^2$$

9.2.2.3 <u>Torsional Frequencies</u> - Using the above equations, torsional frequencies for the Astromast supporting either of the three blanket designs, at cantilever frequencies in the range of 0.1 to 0.3 Hz, are consistently higher than the cantilever frequency by a factor of 2 to 3.

9.2.3 <u>STACBEAM System</u>

9.2.3.1 <u>Torsional Rigidity</u> - The torsional rigidity of the STACBEAM is

$$GJ = \frac{b^2 h^2}{\ell^2} (EA)_\ell \left[1 + 3 \frac{a^3}{\ell^3} \frac{(EA)_\ell}{(EA)_a} + \frac{b^3}{\ell^3} \frac{(EA)_\ell}{(EA)_b} \right]^{-1}$$

This function, for $a/\ell = 2$, $b/\ell = \sqrt{3}$, $h/\ell = 3/2$, and all EAs equal is

$$GJ = 204 \ \ell^2 \ EA$$

9.2.3.2 **Moment of Inertia** - The fraction of beam mass, including joints, in the longerons of the STACBEAM is

$$\frac{M_\ell}{M} \approx 0.36$$

The STACBEAM radius is

$$r = \ell$$

Then

$$I = \frac{1}{2} (1 + 0.36) M\ell^2$$

$$= 0.68 M\ell^2$$

9.2.3.3 **Torsional Frequencies** - Using the above equations, torsional frequencies for the STACBEAM supporting either of the three blanket designs, at cantilever frequencies in the range of 0.1 to 0.3 Hz, are consistently higher than the cantilever frequency by a factor of 150 to 170.

9.3 DEPLOYMENT

Aspects of deployment, which are factors in evaluation, are time duration, reliability, force, and natural frequency during deployment. As discussed in Sections 6.5, 7.5, and 8.5, the ESS and the Astromast each take 10 min to deploy; the STACBEAM takes 33 min.

The relative reliabilities of deployment of each concept can be compared by considering their special characteristics.

- The ESS deploys by simultaneously opening all bays so that each member receives its deployment force by transmission through the assembly. The maximum number of bays which can be deployed successfully is not known.

- The Astromast deploys by uncoiling helically wound longerons. Therefore, attachment of the blanket to the mast must take place during deployment. This type of blanket attachment is a special development problem.

- The STACBEAM requires synchronized engagement and lifting of the three corners of each triangular batten frame. This can be accomplished electronically or mechanically, and is not considered to be a difficult development problem. The mass of this synchronization assembly is considered part of the actuator mass.

In order to properly tension the blanket, a high deployment force is required, particularly toward the end of deployment. Both the Astromast and the STACBEAM are capable of exerting a high deployment force, since the deployment mechanism operates directly on the fully erected portion of the mast. The ESS is not capable of delivering a high deployment force.

Since the full stiffness of the ESS is not developed until it is completely deployed, the system frequency is degraded during deployment. This problem is not encountered in the other systems. The blanket frequency is degraded during deployment in all cases, since full blanket tension is not developed until full deployment.

9.4 GROWTH ADAPTABILITY

As mentioned in Section 9.3, the maximum number of bays which can be deployed successfully is not known for the ESS. This limits its consideration for longer arrays.

Consideration of longer arrays would probably increase the attractiveness of the Astromast system, since the canister mass would not be as great a fraction of the total system mass. However, for shorter arrays, the Astromast is less attractive because the canister mass does not decrease as rapidly as other component masses.

The growth potential of the STACBEAM is excellent; deployment force is not a function of the total number of bays to be erected.

9.5 SYSTEM PACKAGING

The three concepts have three distinct package shapes. The
ESS package is a flat box, the Astromast package is a long cylinder,
and the STACBEAM package is a triangular prism.

SECTION 10
EFFECTS OF BLANKET SIZE AND SHAPE

It is desired to know how total system mass and package volume vary with blanket size and shape. These relationships are investigated using the STACBEAM structural concept with a blanket density of 0.421 kg/m^2 (Design 2). The baseline for comparison is the point solution for a cantilever frequency of 0.2 Hz, in which the blanket dimensions are 4 by 31.5 m. It is not reasonable that beams of all lengths vibrate at the 0.2-Hz reference frequency; in this case, the frequency is made to vary inversely with the beam length.

10.1 MASS

The system mass is the sum shown in Eq. (1) and the beam mass is determined using the iterative method outlined in Section 5. The mass of the blanket is the blanket density times its area. The container mass is determined using the method shown in Section .2, where the unsupported panel lengths are 0.532 and 1.064 m for the 2- and 4-m blanket widths. The 8-m blanket width is obtained using two 4-m assemblies. The harness mass is taken to be proportional to blanket area, at 0.03175 kg/m^2.

The deployment mechanism mass is determined as shown in Section 8.3. The actuator mass is 1, 2, and 4 kg for blanket widths of 2, 4, and 8 m. The guide cable mass is insignificant, as shown in Section 4.7.

The mass of the blanket-stiffening battens is determined as shown in Section 4.8. For the 8-m-wide blanket, a K factor of 3.52 is used because a cantilevered pair of 4-m-wide blankets is assumed. As shown in Section 4.9, the standoff mass is insignificant.

These masses are summed and divided by blanket area. The quantity obtained is the array unit mass and is plotted versus blanket area in Figure 30; itemized quantities are listed in Table VI.

10.2 VOLUME

The package volume is the sum of deployer volume and blanket container volume. Deployer volume is calculated as the volume of one deployed bay because the stack height is approximately equal to the bay length, and the equilateral triangle cross section is conserved from packaged to deployed condition (see Figure 25). This volume is

$$V_{depl\ mech} = \frac{3}{4} \sqrt{3}\ \ell^3$$

The blanket container volume is

$$V_{cont} = \left(2h + p + \frac{L}{w} t_{bl}\right) wa$$

where h, p, and t_{bl} are honeycomb panel, pad, and blanket thicknesses; and L, a, and w are blanket length, width, and fold spacing, respectively (see Figure 2).

Summing these volumes and dividing by blanket area yields the package unit volume which is plotted versus blanket area in Figure 31.

10.3 TRENDS

The optimum area for a given blanket width appears to be proportional to width; therefore, a preferred length of about 30 m is suggested. Regardless of the blanket area requirement,
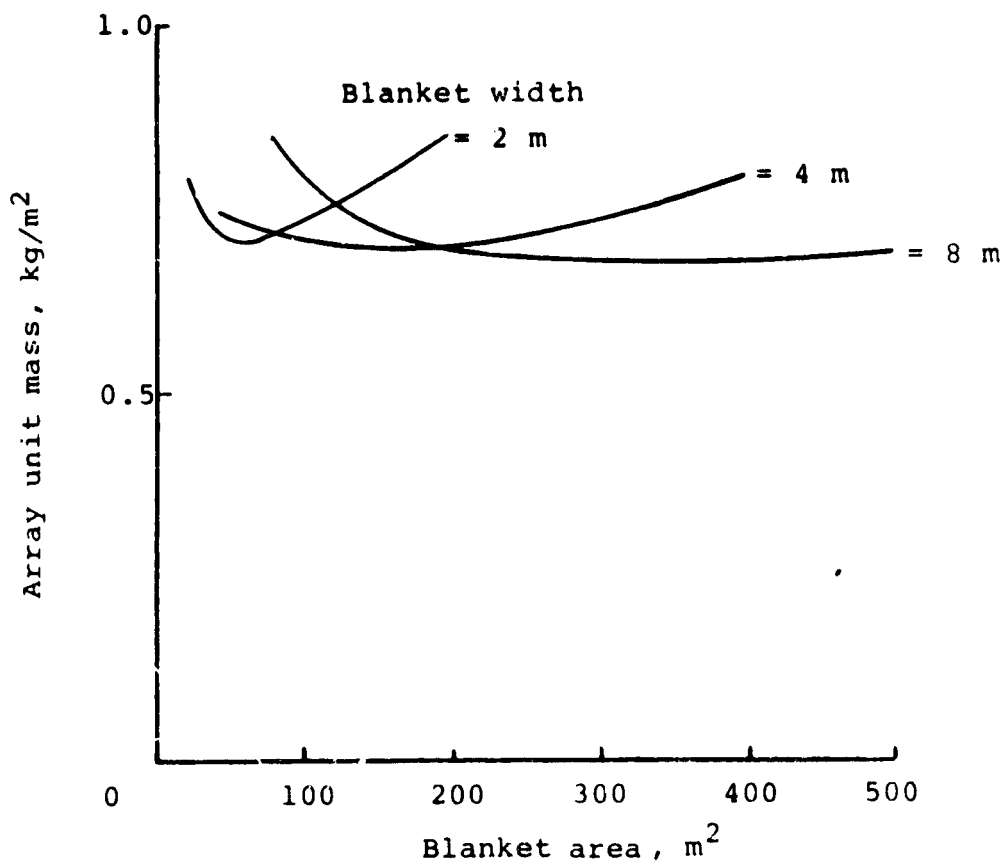
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Blanket: 0.421 kg/m^2

Figure 30. Array unit mass versus blanket area and width for STACBEAM system for Design 2.

TABLE VI. EFFECTS OF BLANKET SIZE AND SHAPE FOR STACBEAM STRUCTURE SUPPORTING BLANKET, DESIGN 2.

DESIGN WIDTH AND BLANKET TENSION	CALCULATED RESULTS																		
	L (m)	A (m²)	M_{bl} (kg)	f (Hz)	ℓ (m)	d (cm)	M_{beam} (kg)	V_{beam} (m³)	h (cm)	M_{cont} (kg)	V_{cont} (m³)	b (cm)	M_b (kg)	M_{dm} (kg)	M_h (kg)	ΣM (kg)	ΣV (m³)	$\Sigma M/A$ (kg/m²)	$\Sigma V/A$ (m²)
a = 2 m	10	20	8.41	0.630	0.182	0.165	0.93	0.008	0.346	3.3	0.025	0.89	0.269	1.8	1	17	0.033	0.850	0.00165
	20	40	16.83	.315	.295	.211	3.01	.034	.490	3.7	.032	.76	.214	4.7	2	31	.066	.775	.00165
T = 12 N	31.5	63	26.5	.200	.407	.247	6.53	.088	.614	4.1	.040	.61	.184	8.8	2	49	.128	.778	.00203
	40	80	33.65	.1575	.483	.270	5.95	.147	.692	4.4	.046	.56	.170	12.5	3	65	.193	.800	.00241
	60	120	50.48	.105	.649	.312	19.77	.355	.848	4.9	.058	.49	.148	22.4	4	103	.413	.858	.00344
	80	160	67.30	.07875	.801	.345	32.28	.568	.979	5.3	.071	.44	.135	33.8	5	145	.739	.906	.00462
	100	200	84.13	.063	.953	.381	49.13	1.125	1.095	5.7	.083	.41	.125	49.1	6	195	1.208	.975	.00604
a = 4 m	10	40	16.8	0.630	0.203	0.208	1.45	0.011	0.979	8.3	0.070	2.23	1.36	3.1	1	34	0.081	0.825	0.00203
	20	80	33.65	.315	.328	.264	4.72	.046	1.385	10.0	.093	1.77	1.03	8.1	3	63	.139	.778	.00174
T = 24 N	31.5	126	53	.200	.450	.310	10.25	.119	1.738	11.5	.116	1.52	.93	15.3	4	97	.235	.770	.00187
	40	160	67.3	.1575	.534	.336	15.27	.198	1.958	12.4	.132	1.41	.86	21.4	5	124	.330	.775	.00206
	60	240	100.95	.105	.717	.390	30.93	.479	2.40	14.2	.167	1.23	.75	38.7	8	196	.646	.817	.00269
	80	320	134.6	.07875	.879	.428	49.69	.882	2.77	15.7	.199	1.12	.68	57.1	10	270	1.081	.844	.00338
	100	400	168.25	.063	1.037	.473	75.75	1.449	3.10	17.1	.229	1.04	.63	82.3	13	359	1.678	.900	.00420
a = 8 m	10	80	33.65	0.630	0.227	0.261	2.31	0.015	0.979	16.6	0.140	5.63	6.84	5.4	3	72	0.155	0.900	0.00194
	20	160	67.3	.315	.365	.331	7.45	.063	1.385	20.0	.186	4.47	5.43	14.0	5	123	.249	.768	.00156
T = 48 N	31.5	252	106	.700	.500	.388	16.11	.163	1.738	23.0	.232	3.84	4.67	26.3	8	188	.395	.746	.00157
	40	320	134.6	.1575	.591	.420	23.96	.268	1.958	24.8	.264	3.55	4.31	36.4	10	238	.532	.744	.00166
	60	480	201.9	..05	.790	.486	48.11	.640	2.40	28.4	.334	3.10	3.77	65.2	15	366	.974	.762	.00203
	80	640	269.21	.07875	.970	.537	78.26	1.19	2.77	31.4	.398	2.81	3.42	97.7	20	504	1.588	.788	.00204
	100	800	336.51	.063	1.138	.590	118.12	1.92	3.10	34.2	.458	2.61	3.18	138.4	25	659	2.378	.824	.00297

L	blanket length	h	container panel thickness
A	blanket area	M_{cont}	container mass
M_{bl}	blanket mass	V_{cont}	container volume
f	array frequency	b	batten width
ℓ	bay length	M_b	batten mass
d	member diameter	M_{dm}	depl mech mass
M_{beam}	beam mass	M_h	harness mass
V_{beam}	beam package volume	ΣM	total mass

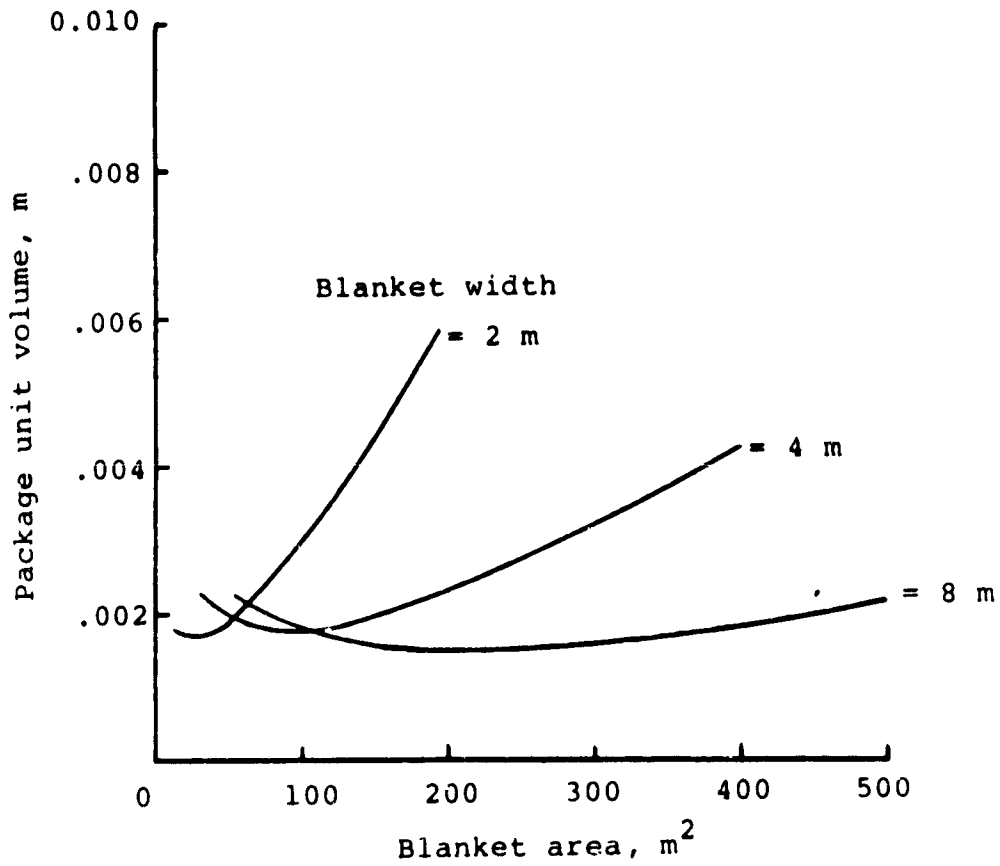|	
ΣV	total volume
$\Sigma M/A$	unit mass
$\Sigma V/A$	unit volume

Blanket: 0.421 kg/m^2

Figure 31. Package unit volume versus blanket area and width for STACBEAM system for Design 2.

a width can be chosen which will yield the minima of system mass per blanket area of about 0.75 kg/m^2 and package volume per blanket area of about 0.0015 m.

The 4-m width is a good choice for the 126 m^2 area. This should not be surprising, since a great deal of effort was expended in the SEPS program to select an efficient aspect ratio.

SECTION 11

CONCLUSIONS AND RECOMMENDATIONS

It is recommended that the STACBEAM solar-array system should be the object of further study and detailed evaluation. The STAC-BEAM system provides high stiffness at low mass, and with the use of a low mass deployment mechanism, full structural properties can be maintained throughout deployment. The stowed volume of the STACBEAM is acceptably small, and its linear deployment characteristic allows periodic attachments to be solar-array blanket to be established in the stowed configuration and maintained during deployment.

Properties of the STACBEAM structure which make it preferred are:

- It deploys sequentially, one bay at a time, by unfolding and locking the hinges located at the midpoint of each longeron and diagonal.
- The battens move linearly, without rotation, during this deployment.
- All hinges have a single degree of freedom.
- Structural elements are manufactured using graphite/epoxy composite rods.
- The deployment mechanism provides full structural stiffness during deployment.

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REFERENCES

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APPENDIX A

HONEYCOMB PANEL FIXITY

Consider a panel of total length L with a pressure load q, supported by intervals U as shown.
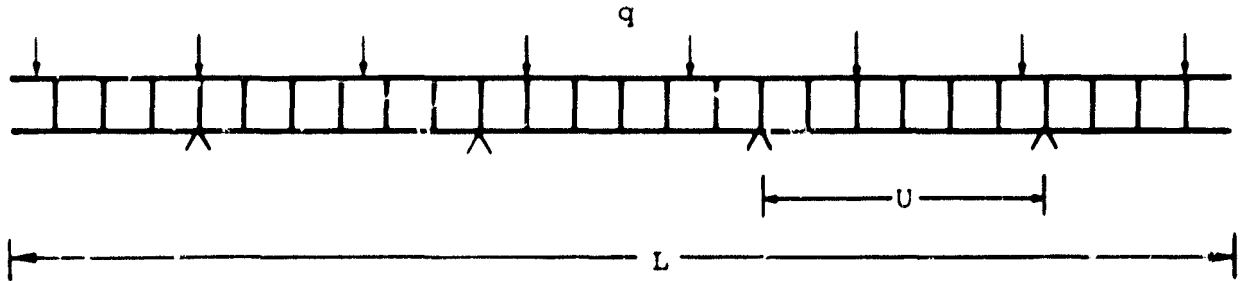
Panel deflection is expressed as (see Section 4.2)

$$\Delta = \frac{K_b q U^4}{D}.$$

where U is the unsupported length and K_b is a bending deflection constant depending on support fixity. Fixed-support conditions can be assumed if sufficient overhang is allowed at the ends, limiting end deflection so that it equals deflection between supports.

$$\Delta_{between} = \Delta_{end}$$

$$\frac{K_{between} q U^4}{D} = \frac{K_{end} \, q\left(\frac{L-3U}{2}\right)^4}{D}$$

$$K_{between} \, U^4 = K_{end}\left(\frac{L-3U}{2}\right)^4$$

If we use the following values,

$$K_{between} = \frac{1}{384}$$

$$K_{end} = \frac{1}{8}$$

$$L = 4 \text{ m}$$

the required support spacing is

$$U = 1.064 \text{ m}$$

Mass of Supports

The four pairs of supports thus located are each of length w, height b, and width c. Each supports a linear load of qU. Using Table 3, Item 2e of reference 5, and setting c/b = 1/5, the deflection is

$$\Delta = \frac{0.781 \ qU \ w^4}{Eb^4}$$

The support mass is

$$M_{supp} = 8 \ \rho w \ b^2/5$$

$$= 1.41 \ \sqrt{\frac{qU}{E\Delta}} \ \rho w^3$$

Evaluating for aluminum ($E = 7 \times 10^{10}$ N/m^2, $\rho = 2770$ kg/m^3), the supports (U = 1.064 m, w = 0.4 m, Δ = 0.003 m) have a mass of

$$M_{supp} = 0.0178 \ \sqrt{q}$$

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The three blanket designs require prepressures of q = 7383, 4831, and 2735 N/m^2 (see Section 4.2 of this report). Thus, the supports have masses of

$$M_{supp} = 1.53, 1.24, 0.93 \text{ kg}$$

APPENDIX B

THERMAL EFFECT OF STRUCTURE
ON BLANKET TEMPERATURE

The purpose of this section is to establish a criterion for the spacing between the blanket and the structure to ensure small enough thermal effect. The approximate analysis presented here is self-contained and therefore should be read without referring to the symbol list, for example, in the body of the paper.

Consider the blanket to be an infinite plane with the Sun shining normal to the front surface which has a solar-temperature absorptance of α and an emissivity at its temperature of ϵ_p. Let the infrared emissivity/absorptivity of the back surface be ϵ_1, and assume that the blanket has infinite conductivity through its thickness and zero conductivity in its own plane.

Model the obstructing structure behind the blanket as an infinitely long cylinder of diameter d, located a distance h between the blanket and the center of the cylinder. Let the emissivity/absorptivity of the cylinder be ϵ_2, and assume that its conductivity is infinite across its cross section.

The temperature of the cylinder T_2 is the same everywhere. The temperature of the blanket is constant in the direction parallel to the cylinder but varies perpendicular to it. The highest temperature T_1 will occur along the line directly in front of the cylinder. The thermal-balance equation at this point is

$$(\epsilon_0 + \epsilon_1)\sigma T_1^4 = \alpha\Phi + \epsilon_1 \sigma T_2^4 \frac{\theta}{2\pi}$$

where σ is the Stefan-Boltzman constant, Φ is the intensity of solar radiation, and θ is the angle subtended by the cylinder

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when viewed from the point on the blanket. The left-hand side represents the heat radiated from the blanket; the right-hand side represents approximately the heat received from the Sun and from the cylinder. Note that the approximation is conservative in that all the $(1 - \epsilon_2)$ reflected energy is included.

The thermal-balance equation for the cylinder is approximately

$$\epsilon_2 \sigma T_2^4 = \epsilon_2 \sigma T_1^4 \frac{\pi}{2\pi}$$

Note that the approximation is again conservative. Not only is all the $(1 - \epsilon_2)$ reflected energy included but also the energy coming from the blanket is assumed to be that for its highest temperature.

Denote the temperature of the blanket for no backing structure to be T_0. Then, combining equations yields

$$\left(\frac{T_1}{T_0}\right)^4 = \frac{\epsilon_0 + \epsilon_1}{\epsilon_0 + \epsilon_1 - \epsilon_1 \frac{\theta}{4\pi}}$$

Now,

$$\theta = 2 \sin^{-1} \frac{d}{2h}$$

Substituting and assuming $\epsilon_0 = \epsilon_1$ gives

$$\frac{\Delta T}{T_0} = \left(\frac{1}{1 - \frac{1}{4\pi} \sin^{-1} \frac{d}{2h}}\right)^{1/4} - 1$$

where ΔT is the temperature rise $T_1 - T_0$.

The variation is tabulated below:

$\dfrac{h}{d}$	$\dfrac{\Delta T}{T_0}$
1	0.010
3	.003
5	.002
7	.001

A reasonable value of T_0 is 300 K. Then the temperature rise for $h/d \geqslant 5$ should be less than 1 K.