METHODS OF EVALUATION OF INFLATABLE STRUCTURES FOR SPACE APPLICATIONS

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DISCUSSION

General

There are many combinations of shapes, materials, fabrication methods, environmental effects, and functional requirements involved in the design of an inflatable structure for use in space. The problem always arises as to which combination of shape, material, and method of fabrication is best suited to function properly in the environment resulting from a particular mission. This paper offers a convenient method for evaluating the merits of an inflatable structure and for comparing one approach against its alternates within the environment of its mission by introducing a figure of merit, F, which is a measure of the structure's efficiency.

Figure of Merit

Because when designing an inflatable structure there are many possible arrangements that can be conceived, it would be most convenient to have some yardstick by which to compare the structural efficiency of any two designs. Such a yardstick or guideline is the figure of merit, F, defined as

$$F = \frac{pV}{W}, \qquad (1)$$

where

p = internal pressure in pounds per square inch,

V = internal volume in cubic inches, and

W = structural weight in pounds.

If the figure of merit is known for a particular type of vehicle constructed of a particular type of material, the structural weight for other vehicles of the same type and made of the same material can be approximated from Equation 1. Part of this paper, therefore, is devoted to determining the structural weights of some typical inflatable space structures constructed of various materials in order to determine some typical figures of merit.

The basic assumptions on which the structural weight is determined are given below.

1. Vehicles operate between 100- and 500-naut mi altitude
2. Materials are pliable and expandable
3. Vehicles are designed primarily by internal pressure
4. Weight is the basis for comparison
5. Structural weight consists of:
   a. Primary structural material (including seams and reinforcements)
   b. Sealant and elastomer
   c. Pressurizing gas (when retained)
   d. Pressurization system
   e. Partitions required for safety
6. Meteoroid and radiation protection material are not included in structural weight
7. Shapes considered are surfaces of revolution or combinations

The total structural weight, of course, can be broken down into the following constituents or parameters.

1. Material properties (including effect of environment)
2. Shape of vehicle
3. Method of construction
4. Internal pressure
5. Safety factor
6. Volume enclosed by vehicle
7. Weight of elastomer
8. Weight of sealant
9. Seams
10. Cutout reinforcement
11. Joint reinforcement
12. Pressurization system
13. Partitions
14. Dynamic loads

The types of structures considered in this paper are confined to pliable expandable structures for low earth-orbit applications. In general, the structures are pressure vessels in the form of surfaces of revolution. The basic vehicle shapes considered are: (1) sphere, (2) spheroid, (3) cylinder, and (4) torus. Also, three methods of construction are considered: (1) isotensoid, (2) uniform sheet, and (3) optimum sheet.

Isotensoid construction implies a filament-wound structure in which the continuous filament is everywhere loaded by the internal pressure to an identical stress level.

Uniform-sheet construction employs a sheet of constant thickness. The thickness of the sheet is determined by the largest of all the principal stresses in the pressurized structure.

Optimum-sheet construction is achieved if the pressure vessel is constructed of sheet material whose thickness varies everywhere in such a manner that at each point the maximum principal stress is constant.

To determine the effect of the many parameters on the weight of the structure an initial weight is established that represents the weight of

the structure as it is prior to consideration of all the external effects, such as environment, seams, shape, and method of construction. For comparison, a final weight is determined taking into account the effect of all the parameters. In addition, the degree to which each parameter affects the weight is also established.

This initial weight is hereafter designated as the isotensoid weight (W_I), which can be determined[1, 2] from the following equation:

$$W_I = \frac{C_0 Vpn}{k_0}, \tag{2}$$

where

C_0 = coefficient for isotensoid structures = 3.0,

V = volume of structure,

p = internal pressure,

n = safety factor, and

k_0 = strength-weight ratio of structural filament under creep load at room temperature.

The isotensoid weight does not include the weight of the binder usually associated with fabric construction but is the weight of the structural material alone. Also, for all isotensoid structures, regardless of shape, the coefficient C_0 is equal to 3.0. As a result, the structural weight is proportional to the volume of the structure. In determining the isotensoid weight, the strength-weight ratio of the material is based on room temperature creep strength. The final weight or total weight (W_{TOT}), as it is hereafter called, is divided into two categories: basic structural weight (W_B) and secondary structural weight (W_S); or,

$$W_{TOT} = W_B + W_S . \tag{3}$$

The basic structural weight is nothing more than the isotensoid weight modified to reflect the effect of methods of construction (other than filament winding) on the structural weight as well as the effect of the fabrication process and environment on the material, which in turn affects the structural weight. This relationship can be written:

$$W_B = W_I \frac{k_0}{k_1} \frac{C_1}{C_0} , \tag{4}$$

or

$$W_B = \frac{C_0 Vpn}{k_0} \frac{k_0}{k_1} \frac{C_1}{C_0} = \frac{C_1 Vpn}{k_1} , \tag{5}$$

where

k_1 = strength-weight ratio of material based on effects of fabrication process and environment, and

C_1 = coefficient based on shape and method of construction.

The factor k_0/k_1 actually is a nondimensional number that reflects the importance of material strength as it is influenced by the manufacturing process and environment in determining structural weight. Likewise, C_1/C_0 is a measure of the importance of shape and method of construction on the weight. W_B, like W_I, defines only the weight of the structural material required to resist the internal pressure and does not include such things as the binder or seams, included in the secondary structural weight.

The secondary structural weight is defined as the sum of the weights of those items, other than the basic structural weight, that comprise the structural wall of the space vehicle or that are necessary to ensure the structural integrity of the wall. The items considered here as comprising the secondary structural weight are as follows:

W_E = weight of elastomer or binder

W_D = weight of material added to resist dynamic loads

W_{SM} = weight of material and cement added for seams

W_C = weight of reinforcing material around cutouts

W_J = weight of material added for reinforcement of joint at intersection of two structures

W_P = weight of partitions required for safety

W_S = weight of sealant required to seal seams, joints, or walls

W_{PS} = weight of pressurization system, if pressure must be maintained for structural integrity

The equation defining the secondary structural weight can be written as follows:

$$W_S = W_E + W_D + W_{SM} + W_C + W_J + W_P + W_S + W_{PS} . \tag{6}$$

These factors, in general, have no simple mathematical relationship to the total weight or to one another, except that they might be some function of the vehicle volume or area. These relationships were established empirically from test data presently available or by calculation where possible. The equation for the total weight then can be written:

$$W_{TOT} = \frac{C_1 Vpn}{k_1} + W_E + W_D + W_{SM} + W_C + W_J + W_P + W_S + W_{PS} . \tag{7}$$

Having discussed the method for determining the structural weight, let us re-examine the figure of merit. First rewriting Equation 1,

$$F = \frac{pV}{W_{TOT}} .$$

From Equations 4 and 5 can be written:

$$W_B = W_I \frac{k_0}{k_1} \frac{C_1}{C_0} = \frac{C_1 Vpn}{k_1} . \tag{8}$$

But from Equation 3,

$$W_{TOT} = W_B + W_S ,$$

and

$$W_{TOT} = W_B\left(1 + \frac{W_S}{W_B}\right) = \frac{C_1 Vpn}{k_1}\left(1 + \frac{W_S}{W_B}\right). \quad (9)$$

Therefore,

$$F = \frac{pV}{W_{TOT}} = \frac{pV}{C_1 \frac{Vpn}{k_1} \quad 1 + \frac{W_S}{W_B}}, \quad (10)$$

and

$$F = \frac{lb\frac{in.^3}{in.^2}}{lb} = \text{inches} . \quad (11)$$

It is noted from Equation 11 that the figure of merit has the dimension of length (inches), the same as the strength-weight ratio, k_1. It is in fact equal to k_1 divided by dimensionless factors. The absolute minimum value of the denominator is 1.5 (where $C_1 = 1.5$, $n = 1.0$, and $W_S/W_B = 0$); thus, the figure of merit is at most equal to two-thirds k_1 with one-half k_1 representing a more practical upper limit.

The method for determining W_{TOT} and C_1 has been touched on but lightly here. There is a more detailed discussion in Appendix A. The structural weights in Table I were taken from Figures A-21 through A-25 of Appendix A.

Range of Values of "F" to be Expected in Practice

If the figure of merit is to be useful to the designer, he must be aware of the range of values that can be expected in practice for various types

TABLE I - FIGURE OF MERIT FOR VARIOUS SPACE VEHICLES

Vehicle	Shape	Method of construction	W_{TOT} (lb)	p (psi)	V (cu ft)	F (in.)
Passive communication satellite (unmanned)	Spherical	Wire grid and film	1,922	0.000266	33.5×10^6	8,000
	Lenticular	Wire grid and film	1,096	*	*	*
Solar concentrator or antenna (unmanned)	Paraboloidal sphere	Mylar film	426	0.027	523,000	58,400
	Paraboloidal ellipsoid	Mylar film and foam	354	0.027	128,000	16,900
Large toroidal space station (manned)	Torus	Filament-wound Dacron-neoprene	6,014	7	37,700	75,800
	Torus	Square-woven Dacron-neoprene	8,889	7	37,700	51,400
	Torus	Uniform-thickness Mylar film	11,382	7	37,700	40,100
	Torus	Uniform-thickness hypothetical film	5,611	7	37,700	81,400
	Torus	Filament-wound sapphire-whiskers-neoprene	2,522	7	37,700	180,800
Large dumbbell space station (manned)	Dumbbell	Filament-wound Dacron-neoprene	2,553	7	14,320	68,000
	Dumbbell	Square-woven Dacron-neoprene	3,648	7	14,320	47,400
	Dumbbell	Uniform-thickness Mylar film	4,866	7	14,320	35,600
	Dumbbell	Uniform-thickness hypothetical film	2,308	7	14,320	75,000
Small cylindrical space station (manned)	Cylinder	Filament-wound Dacron-neoprene	321	7	1,123	40,200
	Cylinder	Square-woven Dacron-neoprene	519	7	1,123	26,180
	Cylinder	Uniform-thickness Mylar film	684	7	1,123	19,860
	Cylinder	Uniform-thickness hypothetical film	332	7	1,123	40,900

*Figure of merit not applicable.

of pressure vessels. If a spherical pressure vessel ($C_1 = 1.5$) built from a high-strength steel is considered, (tensile yield strength = 220,000 psi) with a factor of safety over working pressure of 1.25, and with very minimum secondary weights ($W_S/W_B \rightarrow 0$), then:

$$F = \frac{220,000/0.286}{1.5 \times 1.25 \times (1+0)}$$

$$= 410,000 \text{ in.}$$

Therefore, a value of F of 400,000 in. represents an efficient pressure vessel and can be used as an upper limit for comparison. Values of F down to about 4,000 in. can be expected in pressure vessels of inefficient shape, built from low-strength materials by fabrication processes that result in variable quality, and burdened with many secondary weight items.

Values of F for various types of inflatable space structures are given in Table I. They vary from 8000 to 180,800 in. F = 8000 in. for passive communication satellites, which quite often are designed by stiffness rather than by strength. The value of 180,800 in. is due largely to the use of sapphire whiskers as a structural material to calculate the weight of the structure. This is an extreme case because the strength of the material is beyond the present state of the art, and therefore the figure of merit is much higher than is practical at this time. A more practical upper value would be about 100,000 in. for a sphere.

In general the values of F for inflatable structures are considerably less than those for rigid structures because of the higher safety factor presently being used for nonrigid materials. The safety factor for rigid materials usually ranges from 1.25 to 2.00, while for nonrigid materials the range is from 2.00 to 4.00. In comparing the values of the figure of merit cited above one would expect the structure using the sapphire whisker material to have a higher figure of merit than one constructed of high-strength steel, but actually it is less (180,800 in. compared to 410,000 in.). This condition is due to the difference in safety factor (3.00 compared to 1.25) and the difference in shape factor (2.00 compared to 1.5).

The use of the figure of merit is not confined to any particular type of structure but can be useful for evaluating any type of pressure vessel, whether it be a flexible inflatable structure or a rigid pressurized structure. Table II presents data on the figure of merit for pressure vessels other than inflatable space structures, including various sizes of airship envelopes, an example of a flexible pressurized structure. The values of F range from 13,100 to 26,100 in. An airship structure is not designed purely by the internal pressure but must resist bending due to aerodynamic and landing loads. As a result, the figure of merit is lower than if the structure were designed by pressure alone. Also, the ZPG-3W fabric thickness was varied approximately in accordance with stress. The fabrics for the other airships were, in general, of uniform thickness.

Two types of rigid pressure vessels are presented in Table II: (1) a solid propellant booster, and (2) differently shaped pressure bottles made of various materials. The solid-propellant, filament-wound booster case proved to be a highly efficient structure (F = 371,000 in.) primarily because of a low safety factor (1.25) based on burst pressure, which was justified because of an intensive test program before delivery.

Spherical pressure bottles also proved to be very efficient structures, despite a safety factor of 2.0 based on burst pressure, primarily because of the use of high strength-weight materials and the efficiency of the spherical shape. The toroidal pressure bottle was less efficient because of the use of a lower strength-weight ratio material and the less efficient toroidal shape. The cylindrical pressure bottle proved the least efficient because of the use of low strength-weight ratio material, the less efficient shape, and the very inefficient design of the ends.

Solid-Propellant Booster Tank Design

An important factor in the performance of rocket vehicles is the ratio of hardware weight to usable propellant weight. The weight of the propellant container may be a significant part of the hardware weight, particularly for solid-propellant rocket motors in which the propellant container also acts as a combustion chamber and must resist high combustion pressures. Knowing the figure of merit for a particular type of booster, either by calculation or by experience, it is a simple matter to calculate α, the ratio of hardware weight to usable propellant weight.

If

$$\alpha_p = \frac{\text{weight of propellant container, } W}{\text{weight of propellant, } W_p}, \quad (12)$$

and

$$W_p = V\beta\rho_p, \quad (13)$$

where

β = volumetric loading efficiency and

ρ_p = propellant density,

then,

$$W = \frac{pV}{F}, \quad (14)$$

and

$$W_p = V\beta\rho_p. \quad (15)$$

Therefore,

$$\alpha = \frac{p}{F\beta\rho_p}. \quad (16)$$

For example,

$p = 1000$ psi,

$F = 200,000$ in.,

$\beta = 0.90$,

$\rho_p = 0.06$, and

$$\alpha = \frac{1000}{200,000 \times 0.90 \times 0.06}$$

$$= 0.0925.$$

Versatility of Figure of Merit

The figure of merit can be defined in various ways for specific purposes. In one case it can be

TABLE II - FIGURE OF MERIT FOR VARIOUS FLEXIBLE AND RIGID STRUCTURES

Structure	Shape	Material	W_{TOT} (lb)	P (psi)*	V (cu ft)	F (in.)
ZPG-3W airship (flexible)	Surface of revolution	Dacron-neoprene	12,456+	0.1084	1,490,000	22,400
	Surface of revolution	Cotton-neoprene	15,967+	0.1084	1,465,000	17,200
ZPG-2W airship (flexible)	Surface of revolution	Cotton-neoprene	10,290	0.0858	975,000	14,100
ZSG-4 airship (flexible)	Surface of revolution	Cotton-neoprene	5,776	0.0361	527,000	10,000
Solid-propellant booster (rigid)	Cylinder	Fiberglass (filament-wound)	880.0†	945	200	371,000
Pressure bottle (rigid)	Sphere	Titanium	87.5	3000	4.63	274,500
		Alloy steel	8.20	3000	0.29	183,000
		Fiberglass	6.25	3000	0.17	141,000
	Torus	Steel (HT 190,000)	8.3	3000	0.116	72,500
	Cylinder	Steel (HT 125,000)	5.0§	3000	0.026	26,900

*Operating pressure.

+Thickness of fabric varied approximately according to stress.

†Weight does not include insulation. With insulation included F = 225,000 in.

§Cylinder had extremely heavy ends and large connecting fitting.

based on the short-time bursting strength in essentially a laboratory environment. In other cases it may be necessary to base the figure on long-time performance in a hostile environment, in which case a number of factors, such as factors of safety, loss of material due to erosion, and deterioration of material properties would need to be included in establishing a meaningful figure of merit.

The point to be emphasized is that pressure vessels constructed in a particular way for a particular type of service will have a weight proportional to the design pressure times the volume. Therefore, if the constant of proportionality, F, (the figure of merit) for this class of pressure vessels is known, the weight required for a new application can be determined quickly. Also, the excellence of a particular design solution can be evaluated by comparing its figure of merit with a standard known to be representative of efficient designs for this same general class of applications.

SUMMARY

This paper provides some insight into the most significant parameters that constitute the weight of various inflatable space structures and offers a means of estimating their structural weight. Structural weights for typical inflatable space structures of different shapes and constructed of different materials were calculated for the purpose of evaluating the structural efficiency of each configuration.

The structural efficiency of each configuration was evaluated by the use of the figure of merit, which is a measure of the structural efficiency of the structure. Although the figure of merit was applied primarily to inflatable structures in this paper, its usefulness is not confined to these types of structures. It can serve as a yardstick for measuring the structural efficiency of any type of pressure vessel or pressurized structure provided that it is designed primarily by the internal pressure. Significant bending loads or compression loads distort the figure of merit and make the structure appear less efficient than it really is.

The figure of merit can range from about 400,000 in. for an extremely efficient structure to 4000 in. for a very inefficient structure. Knowing this range, one can quickly establish the relative efficiency of any pressurized structure or make a quick comparison of alternate designs.

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APPENDIX A - DETERMINATION OF TOTAL STRUCTURAL WEIGHT

General

In order to establish some typical figures of merit for various types of inflatable space structures, the structural weights, internal pressures, and internal volumes of some representative inflatable space structures must be determined for comparison. The types of structures for which these values are determined in this paper are:

1. Passive communications satellites
2. Inflatable solar concentrators
3. Large toroidal space stations
4. Dumbbell-type space stations
5. Small cylindrical space stations

Each of the weight parameters that constitute the total structural weight of an inflatable space structure is discussed in some detail in the following sections, along with a summary of structural weights of each of the above vehicles constructed of various types of materials.[A-1] These values are also given in Table I.

Shape and Method of Construction

From Equation 5 the basic structural weight of a pressure vessel in the shape of a surface of revolution can be written:

$$W_B = \frac{C_1 Vpn}{k_1}, \qquad (A\text{-}1)$$

where

V = volume,

p = pressure,

n = safety factor,

k_1 = strength-weight ratio, and

C_1 = constant that depends on shape and method of construction, except isotensoid, in which case $C_1 = C_0 = 3$.

Figure A-1 shows the effect of shape and method of construction on structural weight. If the surfaces of revolution shown all have the same volume, the same internal pressure, the same safety factor, and all are made of a material having the same strength-weight ratio, then the structural weight of each is proportional to the coefficient C_1.

The following conclusions can be drawn from a study of a plot of C_1 versus the ratio a/b, based on the above assumptions:

1. The isotensoid method is the heaviest method of construction for all shapes ($C_0 = 3.0$), and it would not be competitive except that it uses very high strength-weight ratio materials that more than overcome deficiencies in its method of construction and make it the lightest method for moderately high-stress applications.
2. The uniform-sheet method is the lightest practical method of construction, but it cannot compete with isotensoid construction because it is restricted to materials having a low strength-weight ratio. It can be competitive, however, in low-stress applications.

The lightest shape is the sphere ($C_1 = 1.5$). Other shapes appear to be competitive, but they merely approach the sphere.

The torus is one of the heavier structures; however, as its diameter increases, the unit weight decreases and it actually approaches its optimum weight.

Although $C_0 = 3.0$ for all isotensoid shapes that are ideally wound, only three shapes can be wound ideally: the torus, the cylinder, and the geodesic ovaloid.

Effect of Environment on Material Strength

This study shows that the space environment has less effect on the structural material than originally anticipated. This is brought about by the fact that unmanned vehicles are generally pressurized only for deployment and the structure is essentially not loaded during its long exposure to the space environment. In the case of manned vehicles, the internal pressure produces high loads in the structure for the duration of the flight, but the effect of the space environment on the structure is greatly reduced by the protection afforded by the meteoroid barrier.

Therefore, it is assumed for purposes of determining the strength-weight ratio of the material that the only environmental effects are time and temperature. To establish a realistic strength-weight ratio it is desirable to determine the strength of the material based on any combination of time and temperature. Material strength can be determined by using the Larson-Miller parameter, which has been used extensively for metals. Figure A-2 is a plot of the Larson-Miller parameter, $\theta = T(C + \log t_h)$, versus the strength-weight ratio for Dacron. It was established from test data that the value of C in the Larson-Miller parameter for Dacron is equal to 21.4 when the time (t_h) is in hours and the temperature (T) is in degrees Rankine. From this

curve the strength-weight ratio of Dacron, subjected to any combination of time and temperature, can be determined.

For the purpose of drawing the Mylar line in Figure A-2, it was assumed that because Dacron and Mylar are chemically similar, the Larson-Miller parameter would be the same for each material. It was further assumed that the Mylar line would be parallel to the Dacron line but displaced from it an amount equal to the difference in their quick-break strengths. The Mylar line has not been verified by tests.

Safety Factor

All polymeric materials exhibit creep to some extent. Structures made of these materials are always subjected to different stresses at different times; therefore, a load history must be known in order to design the structure. However, there is no proved theory to evaluate the effect of the given load-time history on the strength of the structure. Furthermore, a load-time history implies fatigue as well as creep, but the fatigue damage theory is almost always presented as range of stress versus cycles of variation; that is, time is omitted completely. The effect of creep and fatigue when both occur is generally considered by the designer empirically. In this paper the effects of environment and time under load are accounted for in the strength-weight ratio so that the factor of safety must allow for variability of loads and material properties.

In order to specify a structural factor of safety in the design of aerospace vehicles, it is first necessary to define a probability of failure that can be tolerated. Then both the load and material property spectra must be subjected to a statistical analysis to determine what distribution functions best fit them. A factor of safety can be deduced from the specified probability of failure and the parameters of the distribution function. Actually, a factor of safety need not be computed because the specified probability of failure and the statistical parameters determine an allowable stress. The factor of safety is a convenient and familiar way of presenting the allowable stresses.

For this paper the log-normal distribution is used (see Figure A-3), where the lower curve is representative of (1) standard aerospace materials and (2) comparable but higher variability in loads. The upper curve represents the case in which the variability in material and loads is high (for example, concrete made under poor control conditions). The coefficient of variation for material properties is given as σ_R/R_0; the coefficient of variation for loading is given as σ_W/W_0.

Structural reliability is only one part of the reliability of a mission and hence must be higher than that of the mission as a whole. Over-all mission reliability is not likely to be specified much less than 0.9 nor much greater than 0.99. If the structural reliability is assumed to be three orders of magnitude better than the mission reliability, that is, 0.9999 to 0.99999, then the probability of structural failure would be reduced to 10^{-4} to 10^{-5}. The lowest factor of safety would be 1.3 and the highest about 4.1 (Figure A-3), depending on the variability of loads and material properties and the allowable probability of failure. These values suggest rounded-off factors of safety of 1.5 to 4.5. The weight penalty for poor material and poor loading information is about three to one.

Elastomer and Sealant

Conventional structural fabrics consist of woven or nonwoven structural fibers or yarns coated with an elastomer. The elastomer transfers load between fibers at discontinuities and also acts as a gas retention barrier.

It has been found experimentally that woven cloths will perform properly as a structural material when the amount of elastomer just fills the cloth interstices. Thus, the elastomer weight has been defined as that amount of elastomeric binder that just fills the interstices of a set of structural fibers. This assumption has not been checked experimentally for filament-wound or cord-type fabrics, but based on available data it is justifiable.

As a general rule, the weight of the elastomer applied to a woven cloth is approximately equal to the weight of the cloth (see Figure A-4). A cord-type and filament-wound Dacron-neoprene fabric was found analytically to require 40 percent less elastomer weight than a woven cloth when the fabric was designed close to the maximum strength per ply (Figure A-4).

The major use of structural fabrics will be in inflatable structures that are highly loaded or manned. These applications will require maintenance of high-pressure atmospheres for long periods of time. The elastomeric coatings presently available and applied so they just fill the cloth interstices will not be able to contain these atmospheres for long durations. Additional material will be required to reduce leakage; this added material is referred to as a sealant.

The atmosphere of a manned space structure usually consists of various parts of oxygen, nitrogen, carbon dioxide, and water vapor. It is possible to add more elastomer to the fabric to contain these gases or vapors for the required period, but the result is a heavy inflexible material. Another method of improving the impermeability of the fabric is to construct a composite material using a very thin layer of polymeric film (Saran) that is lightweight and highly impervious to the atmospheric components. This very thin layer of film sealant would in all instances be more effective than adding elastomeric material to reduce atmospheric losses.

Calculations indicate that a sealant of one-mil Saran will be sufficient to keep the loss of vehicle atmosphere to less than 10 percent of the weight of the vehicle's atmosphere for a one-year flight. For the purpose of determining sealant weight, therefore, a weight of 8.85×10^{-3} psf of external surface will be used for all vehicles requiring a sealant.

Weight of Seams

The arrangement of seams in an inflatable structure depends on (1) the shape of the vehicle, (2) the dimensions of the available material, (3) the

required accuracy of shape, and (4) the permissible angle between principal stress and principal strength directions.

This latter criterion is of particular concern when the construction material is woven wire or textile filaments, since for shapes of revolution other than the cylinder the direction of the load-carrying filaments will coincide with the principal stress direction at only one point on the flat pattern.

For symmetrical longitudinal seaming of bodies of revolution, the seam weight (W_{SML}) can be expressed as a percentage of the surface weight (W_F) by three nondimensional parameters as shown below:

$$\frac{W_{SML}}{W_F} = J_L \frac{b}{c} j , \qquad \text{(A-2)}$$

where

J_L = a shape factor,

b = the seam width,

c = the maximum seam spacing, and

$j = 1 + \gamma_c/\gamma_F$.

In this expression, γ_c is the cement weight per unit area (also includes taping, sewing, etc.) and γ_F is the weight per unit area of the surface material and includes the basic weight, elastomer weight, and the weight increase due to dynamic loads if applicable. The seam width also includes the effect of dynamic loads on seam strength requirements. The expression is plotted in Figure A-5 for several common shapes. The curves of Figure A-6 are also applicable to hoop seams in a cylinder.

The hoop seam weight (W_{SMH}) can be approximated by a similar set of parameters. That is,

$$\frac{W_{SMH}}{W_F} = J_H \frac{b}{r} j , \qquad \text{(A-3)}$$

where j and b have the same definition as for longitudinal seams but refer, of course, to the hoop seam construction. The shape factor (J_H) depends on the shape and the number of hoop seams and is plotted in Figure A-7 for several shapes of revolution. The physical significance of the parameter r varies with the shape (see Figure A-8). For the sphere, r refers to the radius of the sphere; for the cone, r is the radius of the base.

The total seam weight as a percentage of the surface is the summation of Equations A-2 and A-3.

Joints

The intersection curve of two surfaces of revolution is a circle if the two surfaces have a common axis of revolution. Other intersection curves are skew curves that, in general, require compression stresses for equilibrium or plane curves that require a curtain for stability.

For the circular intersection the weight of the cord required for stability is included in the basic weight equation. A transition sphere can be used for this type of intersection. The penalty due to the extra volume at the intersection is usually small.

At an intersection additional seams are required to transfer load from one surface to another and to the cord. A certain amount of latitude is available in the location of an intersection so that each intersection requires the addition of only one seam. An approximation of the effect of intersections on seam weight (W_{SM}) can be expressed as

$$\frac{W_{SM}}{W_{SM_0}} = 1 + \frac{N}{n} , \qquad \text{(A-4)}$$

where

W_{SM_0} = seam weight with no intersections considered,

N = number of intersections, and

n = number of seams with no intersections considered.

Since n generally is much larger than N, the increase in seam weight is very small.

Cutouts

Additional weight at cutouts is due to (1) additional material in the form of a cable or rigid ring required for stabilization, (2) seam weight, and (3) the differential weight (omitted in this discussion) between the material removed and that put in its place. The intersection curve of a cutout can be a plane curve, in which case it can be circular, elliptical, or some other closed curve. The stress distribution around closed curves is simple only for circular cutouts that can be stabilized by cables. Other plane closed curves could be stabilized by curtains, but these would destroy the usefulness of the cutout. Therefore, a rigid ring is required for stabilization, or the material closing the cutout must be able to stabilize the opening. In many cases the cutouts are roughly rectangular because the edges follow meridian lines and latitude lines. In surfaces of revolution under internal pressure there is no shear along those lines.

The isotensoid weight accounts for the fact that more material is required in regions of higher stress than in regions of lower stress. The weight penalty due to a cutout can be estimated from the weight of the isotensoid material removed to make room for the cutout. An almost exact estimate of the weight of the cutout area removed is given by:

$$W_C = (\sigma_1 + \sigma_2)\, n \frac{A_C}{k} , \qquad \text{(A-5)}$$

where

σ_1 and σ_2 = principal stresses (assumed constant) acting on cutout area,

A_C = cutout area,

n = factor of safety, and

k = strength-weight ratio.

For the sphere and cylinder the weight per

unit area is constant, and for the important torus shapes it is approximately constant. Therefore, the weight of the cutout area is assumed to be proportional to the area removed. To allow for the discrepancies in shapes other than spherical or cylindrical and to account for the structural inefficiency of cutout reinforcement, it is proposed to double the weight of the cutout area to arrive at the reinforcement weight. Simply, 5-percent cutout area with 10-percent added results in a 5-percent net penalty for reinforcement weight.

As mentioned before, the weight of the material that closes the cutout must be added to this reinforcement weight. The material will depend on the purpose of the cutout.

Figure A-9 indicates the weight penalty due to the reinforcement required for a circular cutout in a pressurized sphere based on the isotensoid weight equation and the factor of two given above.

Partitions

In the event that a meteoroid would penetrate the meteoroid bumper and puncture the structural wall, a warning system would be required that also would activate an emergency air supply to replace the air lost through the puncture and to maintain the internal pressure. If emergency air is not available, the rate at which internal pressure is lost would depend on the area of the puncture, the volume of the station exposed to the puncture, and the pressure. If it is assumed that the hole would provide an ideal orifice and that expansion of the air would be isothermal, the rate of pressure change would be given by:[A-2]

$$\frac{dp}{dt} = 0.578 \frac{pa}{V_0} \sqrt{g\gamma RT} \qquad \text{(A-6)}$$

where

- p = internal pressure at any time,
- t = time in hours,
- a = total area of holes produced by meteoroid impacts,
- g = acceleration of gravity at earth's surface,
- γ = ratio of specific heats (taken as 1.4),
- R = gas constant, and
- T = temperature inside structure.

Integrating this expression gives:

$$\frac{p}{p_0} = \exp\left(-0.578\, at \frac{\sqrt{g\gamma RT}}{V_0}\right) \qquad \text{(A-7)}$$

where

- p_0 = initial pressure at time zero, and
- V_0 = initial volume of air inside the structure.

Also, assuming that the temperature of the air in the station would be 78 F, Figure A-10 shows the time required for the pressure to drop to one-third of the initial pressure in cylindrical modules with hemispherical ends. For larger volumes, a longer time would be available to the crew to locate and repair the puncture. This implies that open partitions would provide greater safety from this standpoint than closed partitions since the entire station volume is exposed to the puncture. However, closed partitions would prevent a pressure drop throughout the station, and if each section is provided with a puncture warning system, the area of search would be narrowed immediately to one section.

If an emergency air supply is activated at the instant of puncture and internal pressure is held constant, the additional station weight of the air and the air container must be considered. The weight of air required to replace that lost through punctures of various diameters, using the same assumptions as above and also a station pressure of seven pounds per square inch (see Figure A-11), for the same cylindrical module with hemispherical ends. This air possibly could be taken from the same source that would replace air lost through permeation. A puncture, then, would reduce the mission life of the vehicle.

If the puncture would be of such a size that very little time would be available to evacuate the section or if the air supply would be inadequate, permanently closed structural partitions would save the station. Since most stations would be combinations of cylinders and spheres, the most efficient partition at any cross section would be a hemisphere whose diameter is that of the cross section. This same relationship is shown in Figure A-12 for cylinders with hemispherical ends as a function of the number of partitions (up to four) and the module aspect ratio assuming a uniform sheet construction.

Dynamic Loads

Introduction. Space vehicles experience dynamic loads from such numerous sources as deployment (entrapped air), docking of other vehicles, rotation to produce artificial gravity, crew movements, and attitude control requirements. The response of the structure to a dynamic environment depends on the mass distribution throughout the vehicle and on the elastic characteristics of the structure, such as the bending, shearing, and axial stiffnesses. With the response to particular transient loading conditions determined by dynamic analysis and possibly model testing, control systems can be designed to prohibit large accelerations of the various masses and to ensure that the loading will not produce excessive deflections. For trapped air expansion, however, little if any control is possible.

Docking Loads. Loads experienced during a docking maneuver will be transient and will induce time-dependent accelerations throughout the structure. The analysis of this loading condition will be greatly simplified if the masses involved and the structural elastic characteristics can be represented by rigid bodies connected by springs and dashpots. In the least complex arrangement the station and capsule are assumed to be rigid masses with a single spring between them. The spring represents the docking mechanism and local deflections of both capsule and station. The g loads experienced by such an arrangement are shown in Figure A-13 for several closing velocities between the station and capsule.

For a three-mass, two-spring system (see

Figure A-14), a general solution for g loads becomes much more involved because of the many possible combinations of mass ratios, spring constant ratios, and closing velocities. Figure A-15 has been prepared for particular values of these ratios and for a capsule closing velocity of two feet per second.

Artificial Gravity Loads. Artificial gravity presents at least two areas for investigation with respect to loading. First, there is the condition of the constant g level during which the station rotates at a constant velocity. Second, it may be necessary to stop rotation for purposes of docking or to investigate the effects of prolonged zero g on crew comfort and performance. This latter condition as well as the resumption of rotation constitutes a transient loading during which torque is applied until the required angular velocity is obtained. As with docking, the response of the structure to this type of loading will depend on the dynamic characteristics of the system and on the time function of the applied torque.

Stresses due to artificial gravity, both the transient and steady state condition, usually will be in the direction of the minimum inflation pressure stress and will require very little additional weight for uniform-sheet materials. However, for the optimum type structure where material strength is proportional to the stress in the two principal directions, additional weight will be required.

Figures A-16 and A-17 show the percent of increase in weight required for this type of construction for the three-spoke torus and the dumbbell space stations at a constant angular velocity of four revolutions per minute. The curves shown for the torus are applicable only for the configuration shown because the structure is redundant and the relative stiffnesses between the spokes and the rim had to be assumed. The dumbbell configuration does not present this problem; therefore, the curves for this configuration are general.

For filament-wound structures the presence of shear requires multiple windings or bias plies, as in the case of docking loads, to prevent large deflections.

Deployment Loads. During packaging of a space vehicle, not all the air within the envelope of the vehicle can be removed. Remaining pockets of trapped air will expand within the container as the ambient pressure decreases during flight and will expand further after the container restraint is removed. The expansion may take place locally, forming a series of bubbles, or, if not restrained, the expansion will tend to accelerate the various masses of the station until the full geometry is realized. At full extension, the kinetic energy of the masses must be absorbed as strain energy in the structure.

The actual mechanism of trapped air deployment is unpredictable for several reasons. First of all, the actual volume of trapped air within the envelope will not be able to be measured accurately since part of the trapped air in the packaged vehicle will be within the folds but will be external with respect to the vehicle internal volume. Nor will the initial pressure of the trapped air be accurately known. It is possible that during launch heat will be added to the trapped air, in which case the enthalpy of the air at deployment will be greater than when packaged.

Figure A-18 shows the effect of trapped air deployment on the structural weight of an inflated sphere for an adiabatic expansion of air for V_1/V_0 equal to unity. A similar analysis has been made for the toroidal space station again using an adiabatic expansion for the trapped air. The results are shown in Figure A-19 for V_1/V_0 equal to unity.

Pressurization Systems

When an inflatable structure depends on its internal pressure for structural integrity, the weight of the pressurization system should be included in the structural weight.

For high-pressure structures, the weight of the pressurization system can contribute a considerable amount to the structural weight. It can be shown from the equation $W = CVpn/k$ that for a bottled gas system the empty weight of the bottles can actually approach the weight of the vehicle structure.

The equation $W = CVpn/k$ defines the structural weight not only of an inflated vehicle, but also of a pressure bottle. If both the vehicle and the bottle are constructed by the same method ($C_1 = C_2$), of the same material ($k_1 = k_2$), and with the same safety factor ($n_1 = n_2$), then the weights of the structure of the vehicle and of the bottles are proportional to their respective volumes multiplied by their respective internal pressures ($W_1 \propto p_1V_1$ and $W_2 \propto p_2V_2$). However, $p_1V_1 = p_2V_2$, and therefore, $W_1 = W_2$.

Some of the more feasible methods of pressurization are shown in Figure A-20. Pressurization system weight is plotted against inflation energy (volume times absolute pressure). Inflation energy is not the governing parameter when selecting pressurization systems for lightly loaded and extremely lightly loaded structures. The governing parameter is some predetermined stress level established to prevent buckling or to strain the structure some given amount.

Valve and control system weights, as well as weight for positive explusion of cryogenic fluids, are not included in Figure A-20. These can be sizeable weight items for small inflation systems, especially for cryogenic systems.

The temperature of the inflation gas in the structure was assumed to be 70 F in all cases.

Two examples of sublimating powder systems are shown in Figure A-20. The points were calculated from data on Echo I and Echo II balloon inflation systems.

Structural Weight of Space Vehicles

The types of vehicles considered in this paper along with the important dimensions, design parameters, and total structural weight are shown in Figures A-21 through A-25. The total structural

weight of each vehicle was calculated, using the parametric data in the preceding sections, and then was tabulated on the bar charts (Figures A-21 through A-25) and in Table I. This total structural weight (W_{TOT}) is one of the important factors in determining typical values of the figure of merit,

$$F = \frac{pv}{W_{TOT}}$$

The types of unmanned vehicles considered were passive communication satellites (Figure A-21) made of a wire grid and photolyzable film, and solar concentrators (Figure A-22) made of aluminized Mylar. The shapes of the passive communication satellites considered were spherical and lenticular. The solar concentrator shapes were spherical and lenticular paraboloids.

Three shapes of manned vehicles were considered: a torus (Figure A-23), a dumbbell (Figure A-24), and a cylinder with hemispherical ends (Figure A-25). Each of these shapes was constructed of: (1) Mylar film, (2) square woven coated cloth, (3) coated filaments, and (4) a hypothetical film having the same strength-weight ratio as the square woven cloth, which is approximately three times the strength-weight ratio of Mylar. One additional special material, a neoprene-coated sapphire whisker, was used to filament wind a toroidal space station to determine the effect of extremely high-strength filaments on the total structural weight of a vehicle.

It should be pointed out once again that the total structural weight of each vehicle does not include the weight of meteoroid protection material or material for radiation protection.

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A-1. Jeppesen, N. L., et. al.: Final Report, Study on Methods of Structurally Evaluating Expandable Structures Having Potential Space Applications. GER-11187. Goodyear Aerospace Corporation, Akron, Ohio. 23 August 1963.

A-2. Kornhauser, M.: "Satellite Pressure Losses Caused by Meteoroid Impacts," Journal of the American Rocket Society, May 1960; 30:475-79.

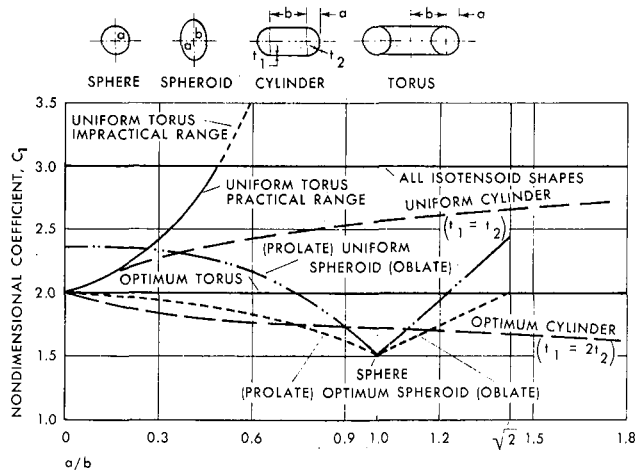


Figure A-1 – Effect of Shape and Method of Construction on Structural Weight

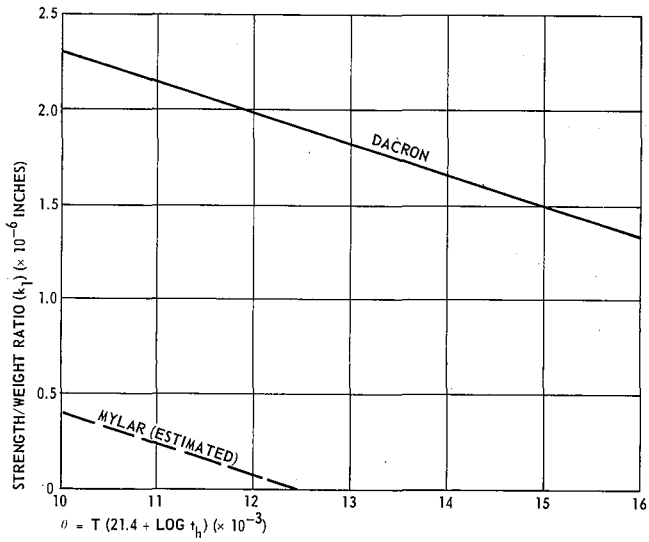


Figure A-2 – Strength-Weight Ratio versus Larson-Miller Parameter for Dacron and Mylar

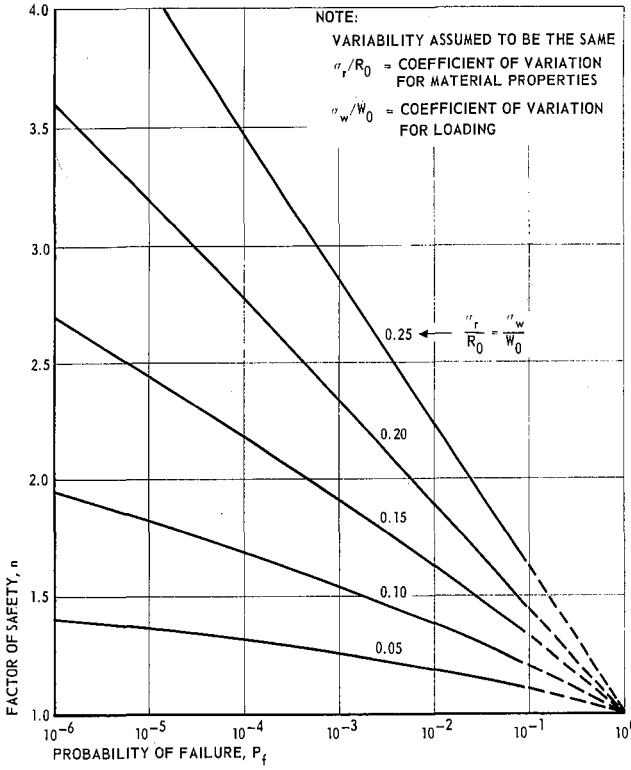


Figure A-3 – Minimum Required Factor of Safety versus Probability of Failure for Log-Normal Distribution of Load and Strength

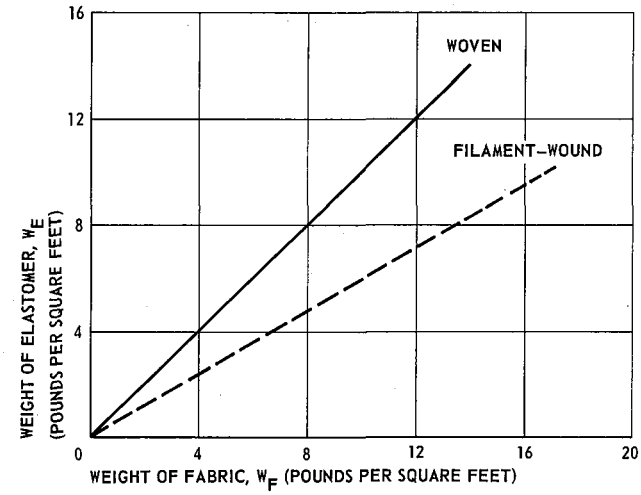


Figure A-4 – Weight of Elastomer versus Weight of Fabric for Dacron-Neoprene Material

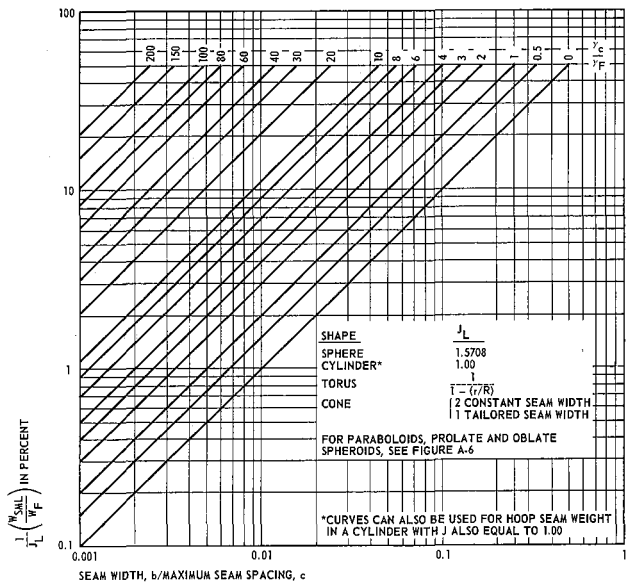


Figure A-5 – Longitudinal Seam Weight as Percentage of Surface Weight for Various Shapes of Revolution

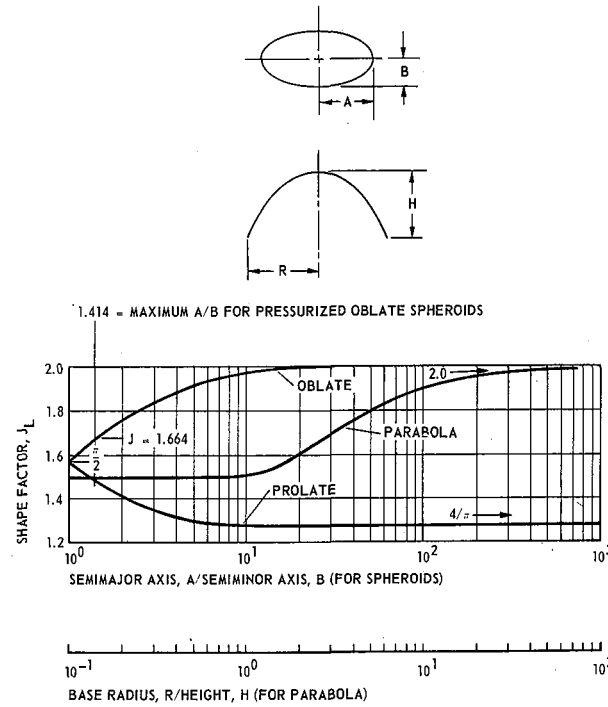


Figure A-6 – Seam Factor for Longitudinal Seams in Prolate and Oblate Spheroids and Paraboloids

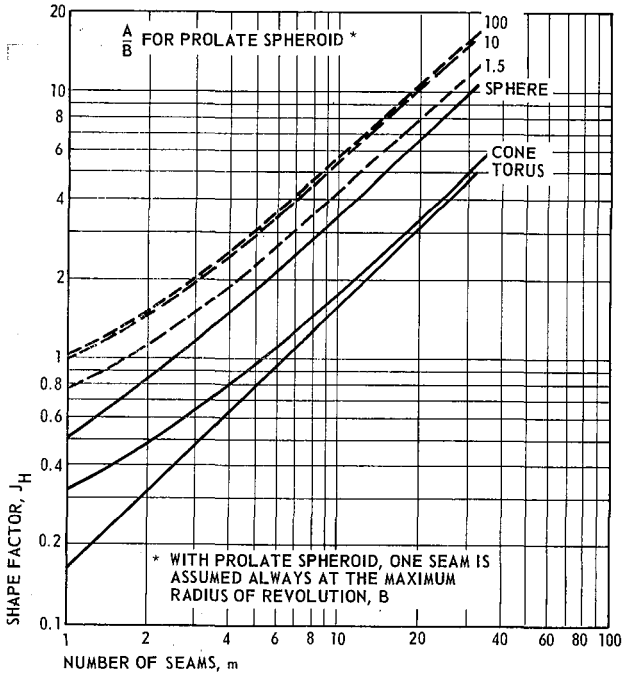


Figure A-7 – Hoop Seam Shape Factor

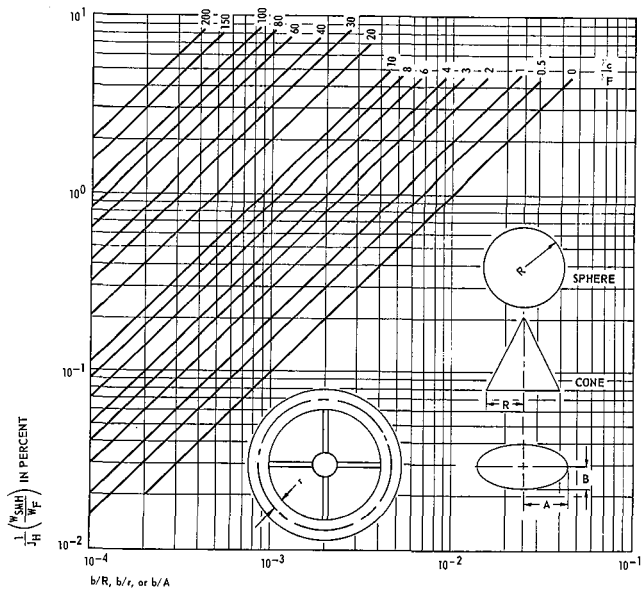


Figure A-8 – Hoop Seam Weight as Percentage of Surface Weight

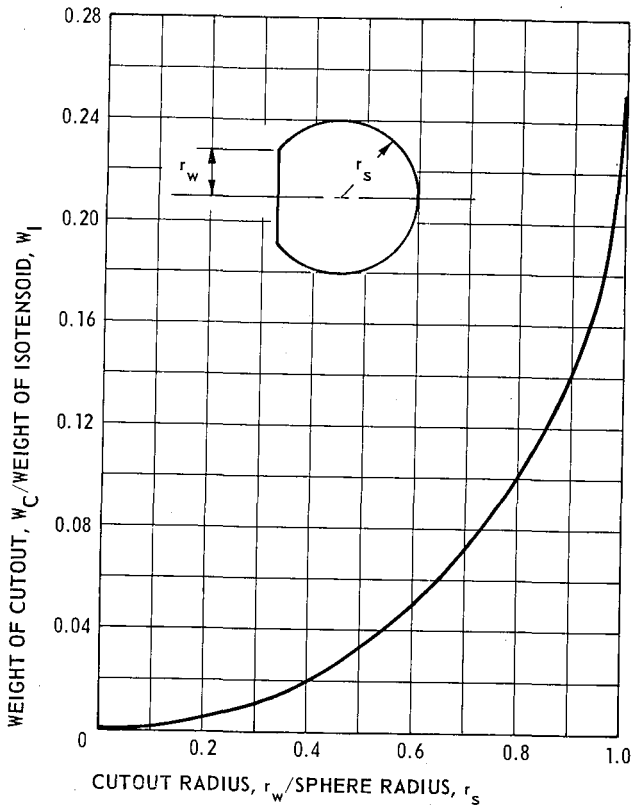


Figure A-9 – Cutout Reinforcement Weight for Isotensoid Pressurized Sphere

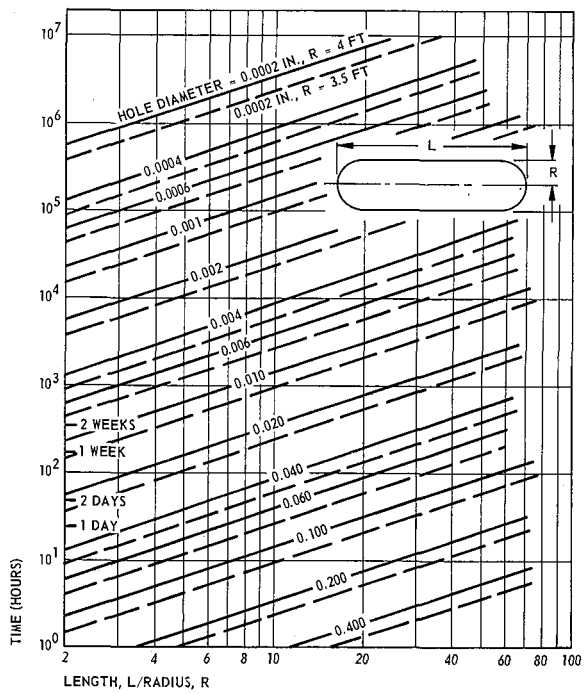


Figure A-10 – Time Required for Internal Pressure to Drop to 1/3 Initial Pressure due to Puncture in Cylindrical Module with Hemispherical End

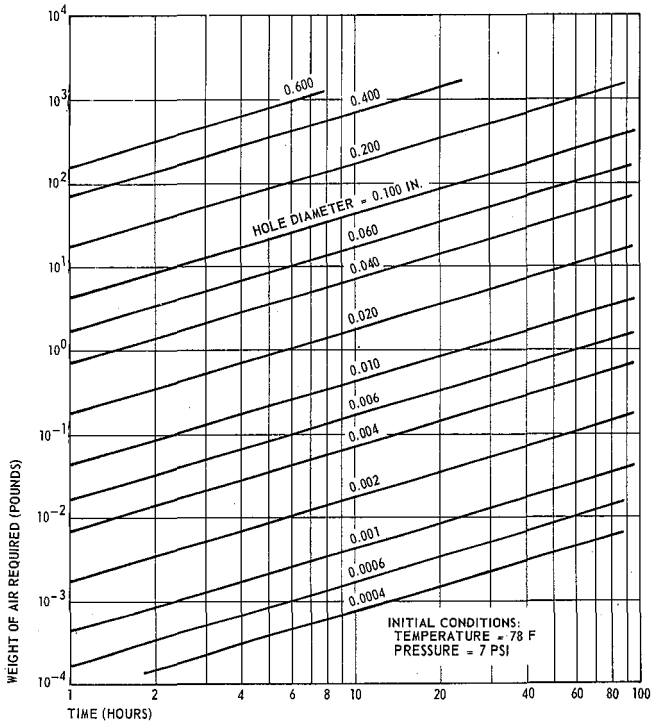


Figure A-11 – Weight of Emergency Replacement Air to Maintain Constant Station Pressure of 7 psi

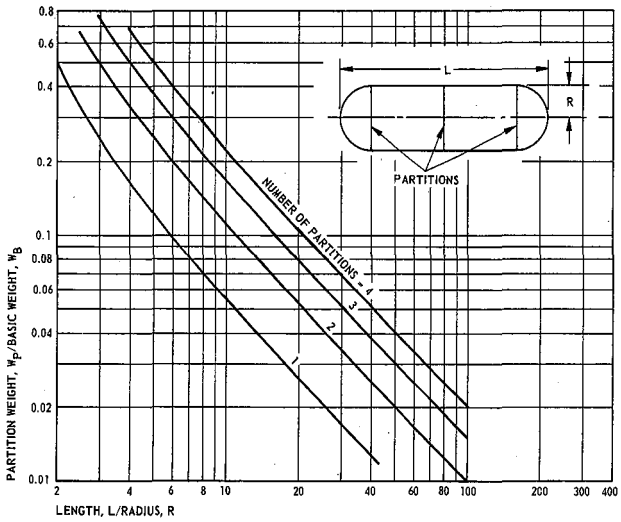


Figure A-12 – Partition Weight for Cylinders with Hemispherical End Using Uniform Sheet Construction

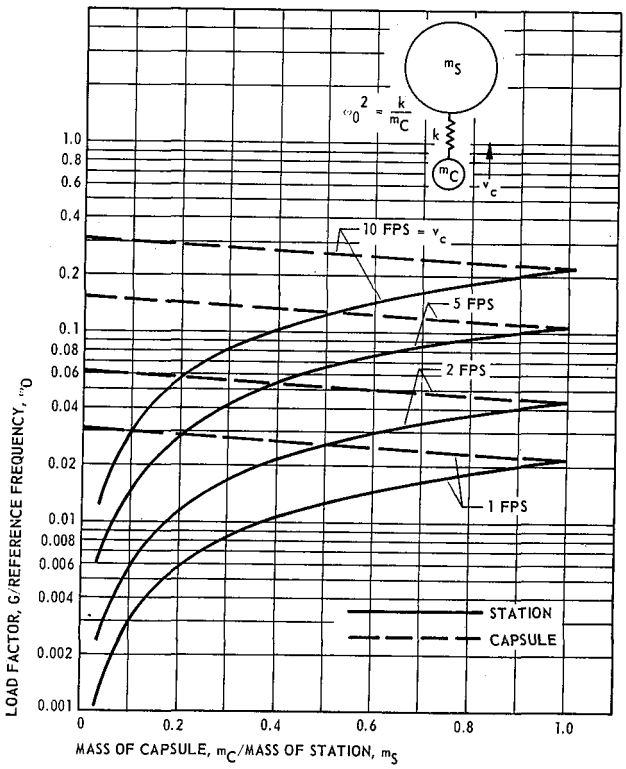


Figure A-13 – Dynamic Response of Space Station to Docking Loads Assuming Two-Mass, One-Spring System

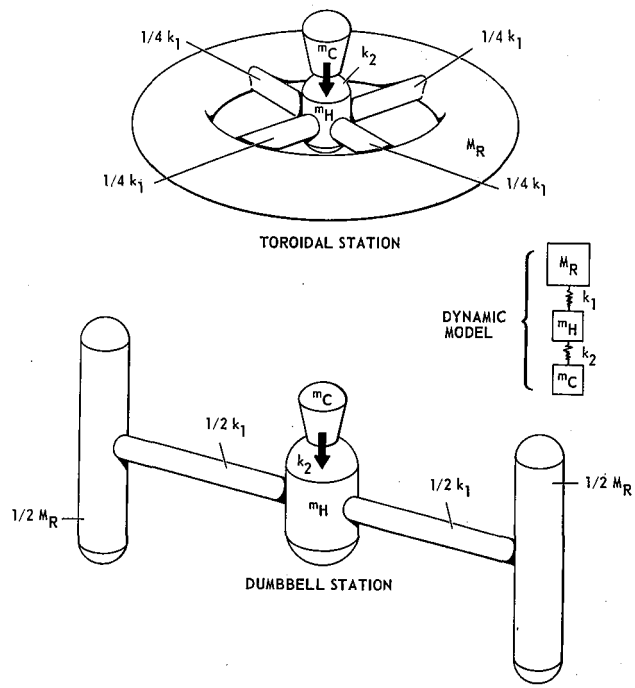


Figure A-14 – Two Typical Space Stations and Simulated Dynamic Model

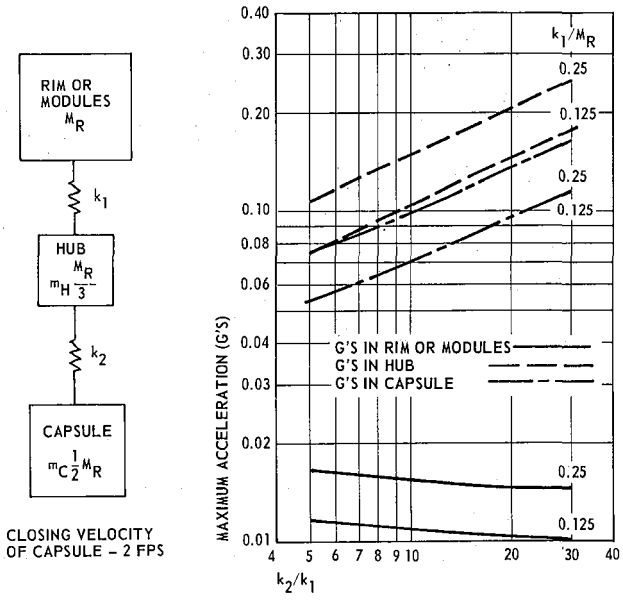
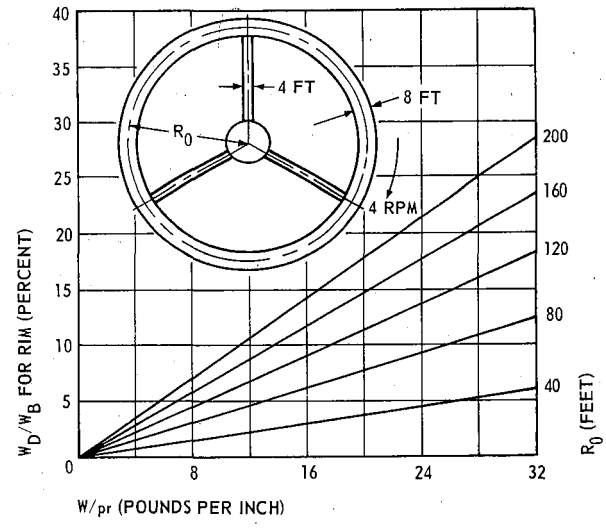


Figure A-15 – Dynamic Response of Space Station to Docking Loads Assuming Three-Mass, Two-Spring System



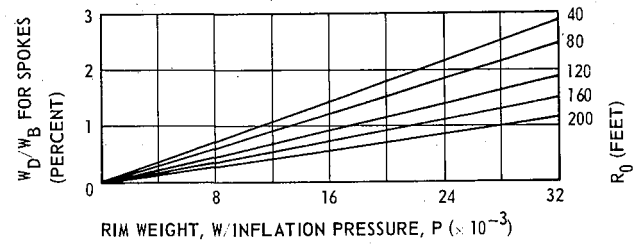


Figure A-16 – Effects of Artificial Gravity on Structure Weight for Particular Toroidal Configuration

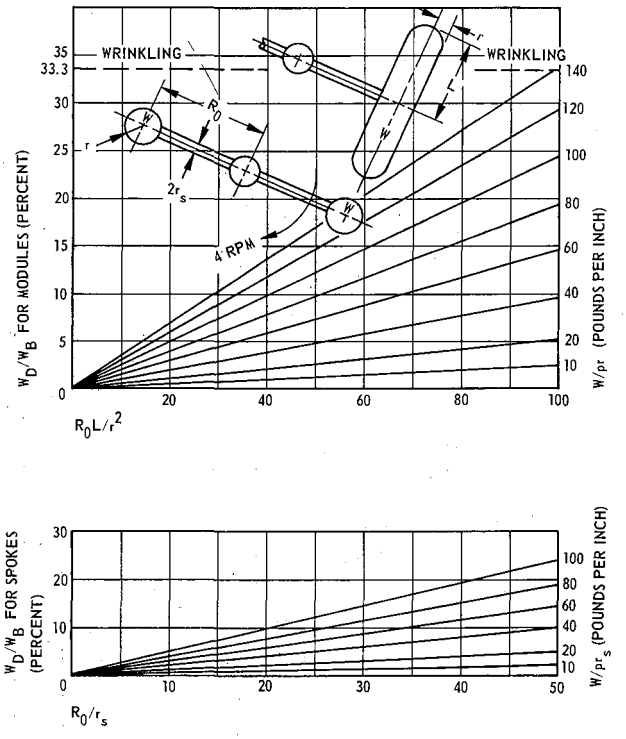


Figure A-17 – Effect of Artificial Gravity on Structure Weight for Dumbbell Space Station

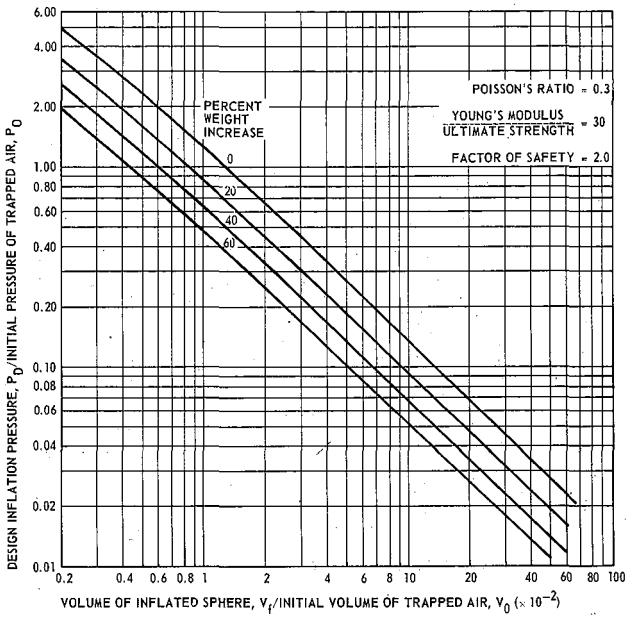


Figure A-18 – Effect of Trapped Air Deployment on Structural Weight of Inflated Sphere

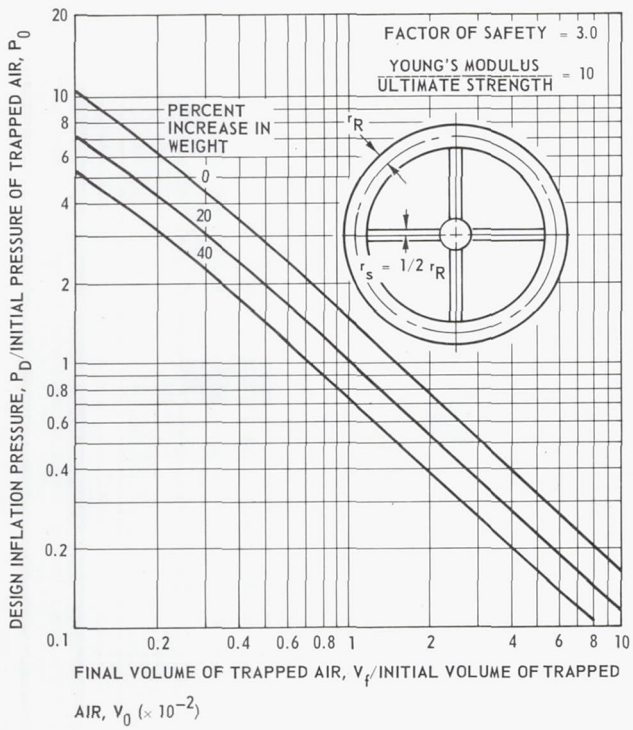


Figure A-19 – Effect of Trapped Air Deployment of Toroidal Space Station

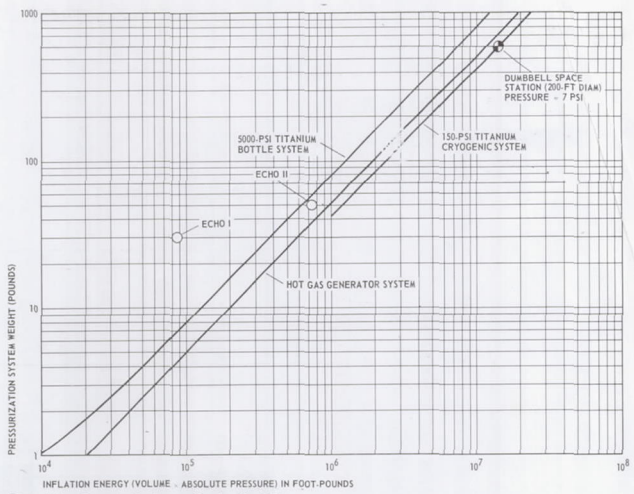
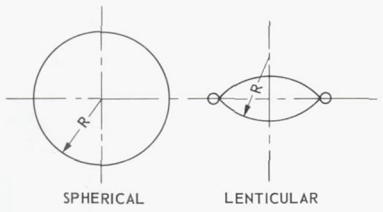
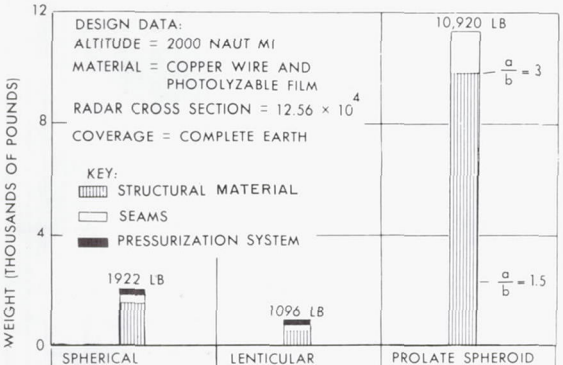


Figure A-20 – Pressurization Systems Based on One Filling





	SPHERICAL	LENTICULAR	PROLATE SPHEROID
STABILIZATION	NONE	GRAVITY GRADIENT	SPIN
DEPLOYMENT	INTERNAL PRESSURE	INTERNAL PRESSURE	SPIN
RADIUS	200 FT	200 FT	272 AND 147 FT

Figure A-21 – Structural Weight of Inflatable Passive Communication Satellites

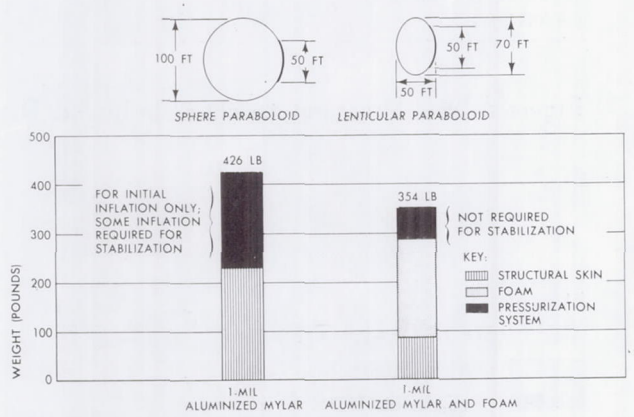
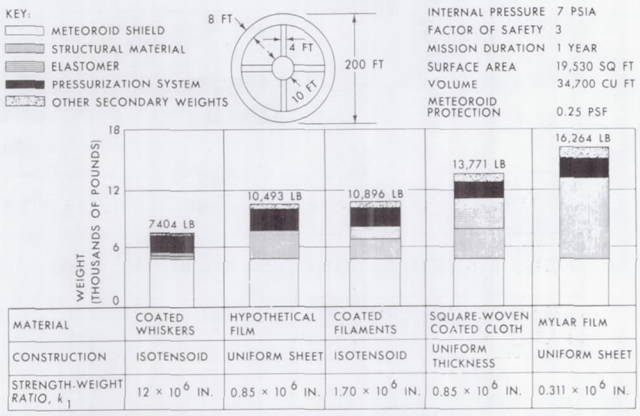
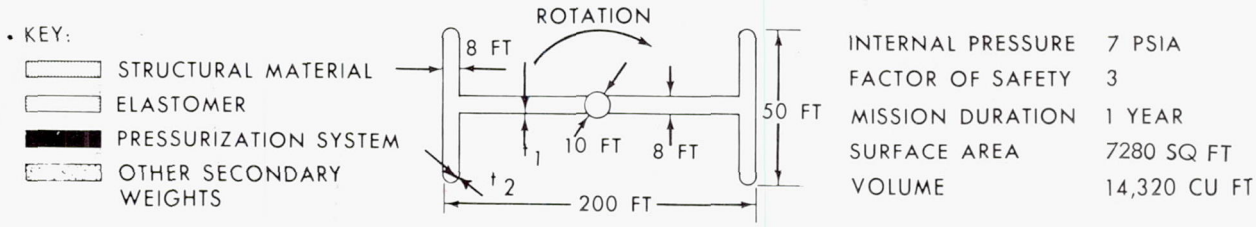


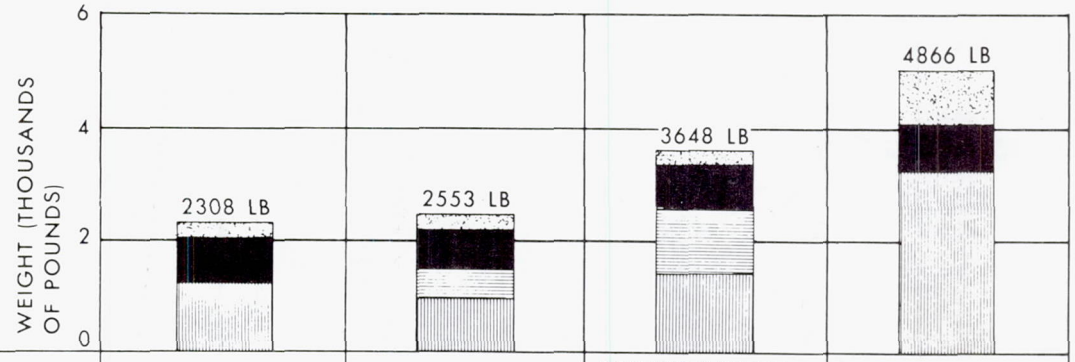
Figure A-22 – Structural Weight of Inflatable Solar Concentrators



MATERIAL	COATED WHISKERS	HYPOTHETICAL FILM	COATED FILAMENTS	SQUARE-WOVEN COATED CLOTH	MYLAR FILM
CONSTRUCTION	ISOTENSOID	UNIFORM SHEET	ISOTENSOID	UNIFORM THICKNESS	UNIFORM SHEET
STRENGTH-WEIGHT RATIO, k_1	12×10^6 IN.	0.85×10^6 IN.	1.70×10^6 IN.	0.85×10^6 IN.	0.311×10^6 IN.

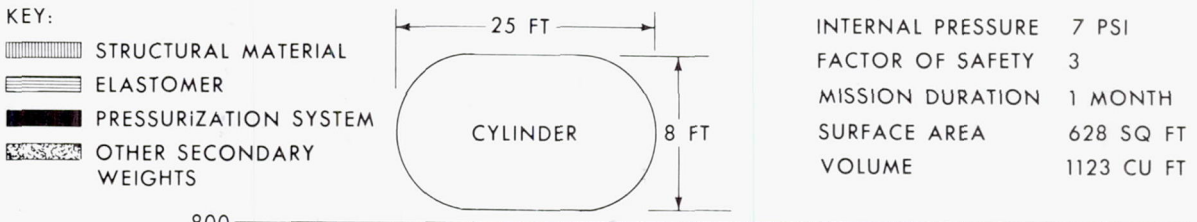
Figure A-23 – Structural Weight of Inflatable Toroidal Space Stations

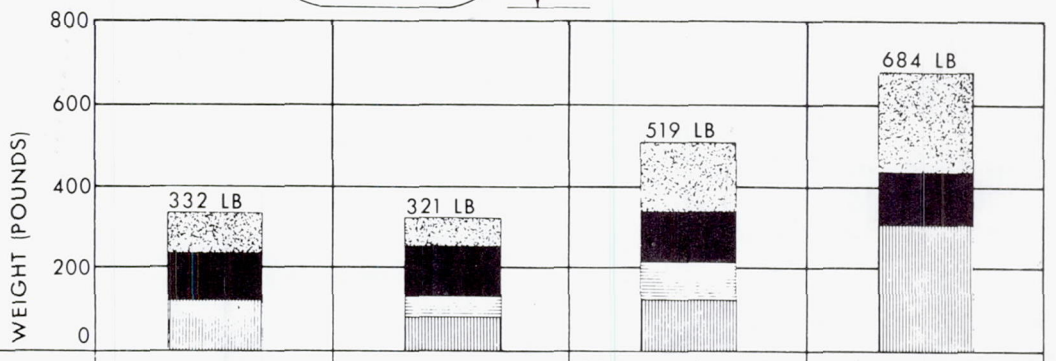




MATERIAL	HYPOTHETICAL FILM	COATED FILAMENTS	SQUARE-WOVEN COATED CLOTH	MYLAR FILM
CONSTRUCTION	UNIFORM SHEET, $t_1 = 2t_2$	ISOTENSOID	UNIFORM THICKNESS, $t_1 = 2t_2$	UNIFORM SHEET, $t_1 = 2t_2$
STRENGTH-WEIGHT RATIO, k_1	0.85×10^6 IN.	1.70×10^6 IN.	0.85×10^6 IN.	0.311×10^6 IN.

Figure A-24 – Structural Weight of Inflatable Dumbbell Space Stations





MATERIAL	HYPOTHETICAL FILM	COATED FILAMENTS	COATED SQUARE-WOVEN CLOTH	MYLAR FILM
CONSTRUCTION	UNIFORM SHEET	ISOTENSOID	UNIFORM THICKNESS	UNIFORM SHEET
STRENGTH-WEIGHT RATIO, k_1	0.835×10^6 IN.	1.67×10^6 IN.	0.835×10^6 IN.	0.304×10^6 IN.

Figure A-25 – Structural Weight of Inflatable Cylindrical Space Stations