Preliminary Design Considerations for 10–40 Meter-Diameter Precision Truss Reflectors

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A simplified preliminary design capability for erectable precision segmented reflectors is presented. This design capability permits a rapid assessment of a wide range of reflector parameters as well as new structural concepts and materials. The preliminary design approach was applied to a range of precision reflectors from 10 to 100 m in diameter while considering standard design drivers. The design drivers considered were weight, fundamental frequency, launch packaging volume, part count, and on-orbit assembly time. For the range of parameters considered, on-orbit assembly time was identified as the major design driver.

Introduction

CURRENTLY, there is considerable interest in precision segmented reflectors for various future NASA missions. Potential applications for such reflectors include submillimeter astronomical observatories, microwave radiometers that make atmospheric measurements necessary for studying the greenhouse effect, advanced communications antennas, and solar dynamic reflectors. The operational requirements for some of these missions are described in Refs. 1 and 2, and typical structural requirements are presented in Ref. 3.

A study of these missions indicates that common requirements for these reflectors are high surface accuracy and high stiffness for controllability. Numerous studies of precision reflectors[4–7] have shown that the best approach for meeting these requirements is to attach a number of thin, high precision reflector panel segments to a larger deep truss to form the complete reflector. It is well known that trusses possess high stiffness,[8,9] and it has been shown that well made trusses can provide a very precise framework for supporting reflective panel surfaces.[10]

A common concept for constructing precision reflectors is to attach hexagonal reflector panels to a tetrahedral truss that has nearly equilateral triangular bays. In this paper, a rapid preliminary design procedure is described and results are presented that indicate the major design drivers for such reflectors. Design drivers such as weight, frequency, packaging volume, part count, and assembly time, as related to various truss and panel input parameters, are considered. The paper culminates with a rational comparison of these drivers on a normalized basis, which yields considerable insight as to which design parameters are dominant. The preliminary design procedure developed in this paper is based on the equivalent plate theory for trusses developed in Refs. 8 and 9 and verified in Ref. 11. Although the simple analysis used in this paper is not highly accurate for predicting the natural frequencies of small-diameter trusses, the analysis is sufficiently accurate to permit a rapid assessment of various drivers for preliminary design purposes. The alternative to this simple approach is a finite element analysis, which can be time consuming for conducting conceptual design studies.

The truss/reflector systems considered in this paper are assumed to be erected on orbit by astronauts or by robots. Thus, the cost and time associated with on-orbit construction become important considerations. A major impediment to the construction of large segmented reflectors is the difficulty and cost of fabricating large precision panel segments. For example, high precision reflector panels are curently limited in size to 1–2 m because of these reasons. This limitation in panel size leads to a relatively high reflector part count, which in turn directly increases the required on-orbit construction time.

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Erectable Reflector Description and Design Drivers

An example of a high precision reflector is the 20-m-diam submillimeter astronomical observatory discussed in Ref. 1. Designed for deep space measurements of infrared radio frequencies, this application requires that distortions in the reflector surface be no greater than 2 μ root mean square (rms). It may not be possible to meet this accuracy requirement passively, and, thus, the required reflector surface accuracy may necessitate active control. A research goal at NASA Langley Research Center is to achieve surface accuracies for large segmented reflectors on the order of 10–20 μ rms passively. This represents about an order of magnitude improvement over the current state of the art for large ground application segmented reflectors. This level of accuracy would enable the passive measurement of radio frequencies in the 200-GHz range and would significantly reduce the amount of active control required to obtain high surface accuracy reflectors. The achievement of such accurate reflectors requires the development of very high precision and thermally stable truss support structures as well as high precision and stable panel segments to form the reflector surface. Both of these considerations are part of the ongoing research activities within NASA.

A typical reflector that is considered in this paper is shown in Fig. 1. This particular reflector has an effective circular diameter (the diameter of a circle with the same area as the reflector surface) of 16.6 m and is composed of 19 hexagonal

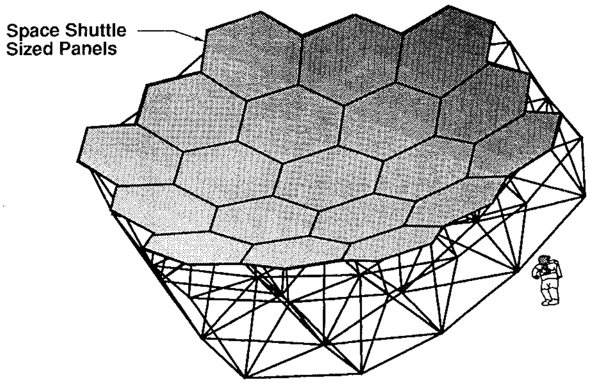

Fig. 1 A 16.6-m-diam two-ring reflector.

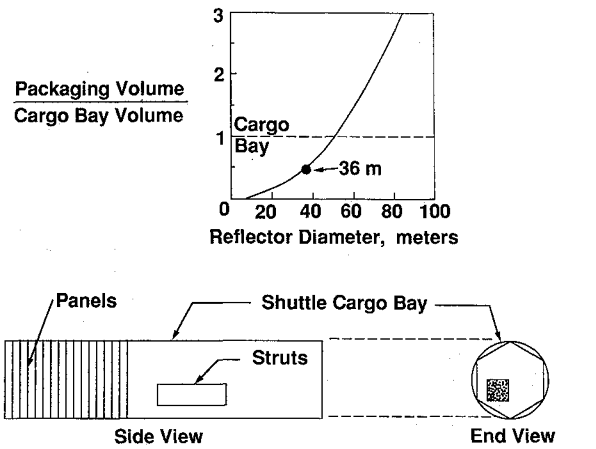

Fig. 2 Packaging of erectable reflector in Space Shuttle cargo bay.

honeycomb panels that are 4.2 m across (corner to corner). This is the maximum size panel that can fit into the Shuttle cargo bay. Note that although panel size may currently be limited to 1 or 2 m as mentioned previously, this paper considers even larger panel sizes. As panel technology develops, larger panels (or alternatively, panel modules that are preassembled on the ground) may become feasible. Thus, it is desirable to understand the design implications of a wide range of panel sizes.

The general approach for stowing an erectable segmented reflector in the Shuttle cargo bay is shown in Fig. 2, where a 36-m-diam reflector with 4.2-m panels is used as an example. Packaging volume requirements as a function of reflector diameter are shown at the top of the figure. It can be seen that the honeycomb sandwich panels dominate the volume requirements. Note that, for this panel size (4.2 m), reflectors with a diameter $\gtrsim 50$ m would require more than one Shuttle flight and, thus, might not be feasible. The ability to tightly package the panels is a major benefit of the erectable approach to constructing large segmented reflectors. Other packaging schemes may result in prohibitively high launch volume requirements for large reflectors. The general scenario for constructing an erectable structure on orbit is shown in Fig. 3 and is discussed in detail in Ref. 12. In this scenario, a mobile transporter together with two remote arms is used to position the astronauts.

The reflector is mounted on a rotating fixture, which facilitates assembly. Although it is desirable to keep the reflector panel segments as large as possible to minimize part count and assembly time, there are other factors that must be considered in the design process. For example, the fabrication cost of very high precision reflector panels increases rapidly with increased size. Also, the panel thickness must be greater for large panels to maintain accuracy. This increases packaging volume and

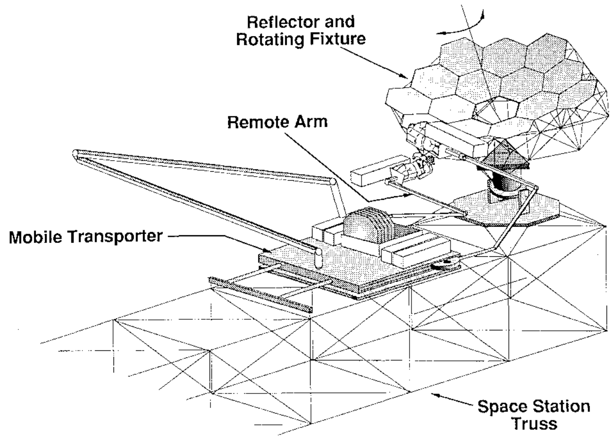

Fig. 3 Construction of precision segmented reflector using mobile transporter.

weight. The other major factor to be considered is the overall stiffness (which affects natural frequency) of the reflector, which is provided by the truss and is required for maintaining dynamic controllability.

The resulting design drivers for precision reflectors are weight, stiffness, launch vehicle packaging volume, and on-orbit assembly time. It is noted that an additional design driver for precision reflectors will be reflector surface accuracy. It is known (see Ref. 10) that fabrication errors in the lengths of the truss struts directly affect surface accuracy. However, experimental data relating nominal strut length to strut fabrication error (and, hence, reflector surface accuracy) do not exist. Furthermore, because panel technology is still developing, the relationship of panel size and panel weight to achievable surface accuracy is not well known. For these reasons, surface accuracy is not considered as a design driver in this paper. Total system cost, which includes fabrication, launch, on-orbit construction, and life cycle costs, may also be a major design driver. However, the costing of such complex factors is beyond the scope of this paper. Instead, the major purpose here is to present a simple preliminary design capability that will permit a rapid relative assessment of the various design drivers for different applications. This capability is intended to serve as an aid in conceptual design parametric studies.

Truss Geometry Definition and Part Count

Two geometric truss configurations that can be used for attaching hexagonal panels to a tetrahedral truss are shown in Fig. 4. Both of these geometries have been considered in the past for reflectors. In the current paper, attention is focused on the concept labeled truss B. This concept is considered superior for segmented reflectors because of the greater degree of symmetry exhibited by the segmented panel surface. In other words, the reflector panel geometry is closer to being circular. It is interesting that a regular symmetric truss (truss A) results in an irregular array of panels, whereas the reverse is true for truss B. The shaded regions in Fig. 4 indicate the definition of rings as used in this paper. The example shown in the figure represents three rings of hexagonal panels. Note that ring 1 of truss type B is defined to include the center triangle of the truss and the central panel. Equations defining the geometry and part count for reflectors with different numbers of rings are given in Ref. 13. The part count and reflector panel geometry as a function of number of rings is shown in Fig. 5. It can be seen from this figure that the number of struts and panels required for assembling a given diameter reflector changes dramatically with the size of the individual panels used. For large numbers of rings, the strut and panel part counts increase approximately as the square of the number of rings.

Reflector On-Orbit Assembly Time

Numerous concepts have been successfully developed for deploying relatively large mesh surface reflectors in space. The meshes used in these reflectors were developed specifically to be tightly packaged for launch and to be wrinkle free upon deployment. However, for applications involving radio frequencies $\gtrsim 40$ GHz, continuous (nonmesh) surface reflectors must be used to eliminate reflective losses. This situation leads to the requirement for solid panel reflectors. Numerous attempts have been made, and are continuing, to develop concepts for deploying solid panel segmented reflectors. However, to date, packaging volume constraints and mechanical complexity have limited these deployable concepts to about 10 m or less in diameter. Thus, to enable the construction of larger reflectors, erectable concepts are being considered wherein the reflector system is actually constructed on orbit from individual truss struts and reflector panel segments. The obvious major disadvantage of this approach is the astronaut assembly time or robotic capability that is required. Past studies have shown that assembly time and effort are directly related to the number of individual elements that must be assembled. As a result, part count is a major concern for large reflectors.

As was shown in Fig. 5, the element part count increases significantly as the number of rings in a reflector increases. In Ref. 12, astronaut assembly times were estimated to be about 0.5 min/strut, and about 10 min/panel. These estimates are based on two astronauts simultaneously performing a construction. Using these estimates, reflector assembly times are shown in Fig. 6.

Current space suit technology limits each individual astronaut extravehicular activity (EVA) to about 5 or 6 h. A further limitation is that a single Shuttle flight can only support 2 or perhaps 3 EVAs. However, it is envisioned that these large reflectors will be constructed from the Space Station Freedom, where perhaps more EVA time will be available for such major construction tasks. Assuming 10 min is required to assemble each panel, it can be seen from the figure that constructing a 6 or 7 ring reflector would indeed be a large construction effort requiring 6 or more EVAs. To improve this situation, research and development activities are under way to reduce the time required to install individual panels from the current 10-min estimate to 5 min. Such an improvement would reduce total construction time by about 35%. Nevertheless, the major consideration for construction time is still reducing the part count. Considering current limitations on EVA, it appears that feasible reflectors must be limited to about 3 or 4 rings. In the following sections of the paper, weight, stiffness, and packaging volume will be examined as a function of the number of reflector rings to establish sensitivities to these parameters and to explore practical limitations on reflector sizes.

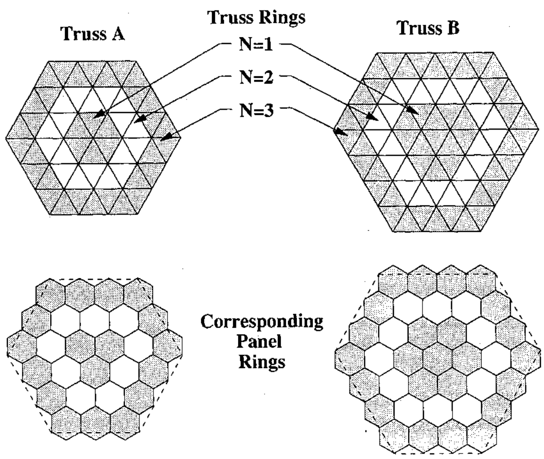

Fig. 4 Truss and panel ring definition for two types of tetrahedral trusses.

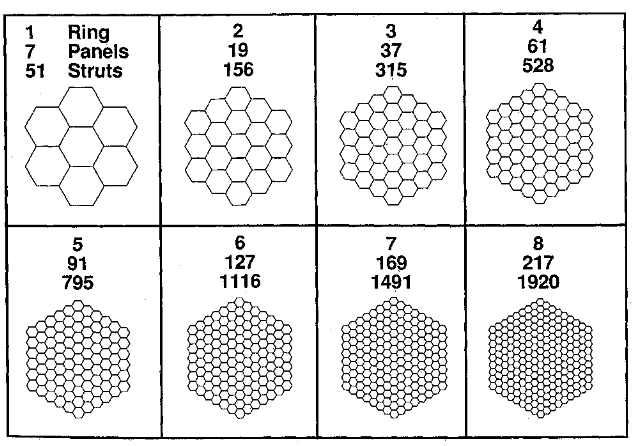

Fig. 5 Part count and geometry vs number of rings (truss type B).

Parameters Used in Design Study

The basic structural parameters used in the preliminary design study are shown in Table 1. The reflector surface panels were considered to be honeycomb sandwich with graphite epoxy face sheets. For lightly loaded sandwich panels, the maximum bending stiffness per unit weight will result when the face sheets are as thin as possible within maximum gauge constraints. The face sheet thickness was chosen as 0.02 in. to permit several plies of graphite epoxy to be used in the lay-up and to ensure that dimpling of the face sheet on the honeycomb core would not be a problem for high accuracy applications. Considerable research is being conducted on such high precision reflective panels.[14] The core thickness of the hexagonal panels was chosen such that the ratio of panel width (corner to corner) to core thickness stays constant as panel size changes. The constant ratio results in 1-m panels that are 1 in. thick, 2-m panels that are 2 in. thick, and so on. The increase in core thickness with panel size was believed to be necessary to maintain panel precision during manufacturing and during

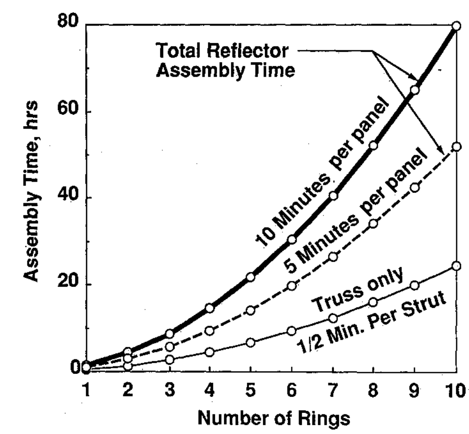

Fig. 6 Reflector on-orbit assembly time.

Table 1 Parameters used in reflector preliminary design analysis

Panels (graphite epoxy with honeycomb core)	
Face sheet thickness	0.02 in.
Core thickness	0.025 × panel width
Core density	2 lb/ft^3
Struts (graphite epoxy)	
Strut modulus	30E6 lb/in.^{2}
Minimum thickness	0.04 in.
Strut density	0.06 lb/in.^{3}
Strut diameter	(determined from Pcr = 1000 lb)
Nodes (aluminum and graphite epoxy)	
Node weight	6.6 lb (50% panel attachment penalty)

thermal loading, which occurs in space. The honeycomb core density of 2 lb/ft^3 was selected to be commensurate with lightweight aluminum cores but it was also estimated that this is about the same density that would result for low coefficient of thermal expansion (CTE) graphite epoxy cores.

The modulus of the truss struts was chosen as 30×10^6 psi assuming that high performance, low CTE graphite epoxy material would be used for high precision space structures applications. The minimum wall thickness of the struts was selected as 0.04 in. to ensure adequate toughness for handling and assembly. The strut material density of 0.06 lb/in.^{3} was chosen to be slightly greater than the density of graphite epoxy to allow for an impermeable coating, which will most likely be required for space applications. Although operating loads in space are quite low for most foreseeable applications, experience has shown that highly predictable and stable truss structures must be composed of strut elements with a reasonable Euler buckling load capacity. For example, initial imperfections in individual strut lengths can result in residual internal loads in redundant trusses. Similarly, variations in strut CTEs can result in internal load buildup, even under uniform changes in thermal loading. Having a sensible value of buckling load capacity in the struts minimizes the effect of these internal loads on structural performance.

For the current study, the diameters of the truss struts were determined by constraining the struts to have at least 1000-lb buckling load capacity (*Pcr*). The value of 1000 lb was arrived at by conducting a sensitivity study of reflector weight as a function of strut buckling load. For example, reducing the buckling load constraint to 250 lb results in a weight savings of $<5\%$. It is believed that the constraint of 1000 lb results in a robust strut that would be useful for both handling and operational purposes. For applications with extreme weight restrictions, it might be necessary to study the value of the buckling load constraint in more depth. Applying a buckling load constraint has the additional benefit of ensuring that individual strut vibration frequencies are not extremely low. For the 1000-lb load constraint, a check of the frequencies indicated that, for the range of truss parameters considered in this paper, the strut frequencies are always higher than the lowest natural frequency of the complete reflector. This is desirable because it minimizes coupling between the local strut and global reflector modes.

The weight of the nodal joint clusters (6.6 lb) was assumed to be the same as the joint clusters in the precision truss discussed in Ref. 7; however, a 50% weight penalty was included to account for fixtures that would be required to attach the panels to the truss. In Ref. 7, the struts and the joint are 1 in. in diameter. Although the diameters of the truss struts considered in this paper vary from about 1 to 2 in., the joints are considered to be the same for all trusses. The justification for this is that the trusses are extremely lightly loaded and, thus, high stresses in the joints will not occur. For this reason, when struts larger than 1 in. in diameter are considered, they are assumed to be tapered at the ends to allow interfacing with a 1-in. joint.

Preliminary Design Approach

A primary purpose of this paper is to present a rapid procedure for evaluating reflector design drivers for different values of truss and panel input parameters. No attempt is made to optimize the reflectors because it is extremely difficult to establish an absolute objective function. Instead, the perceived reflector design drivers are calculated, and an attempt is made to present these in a fashion that gives insight into which are the major drivers and to determine what can be done from a conceptual point of view to improve the overall design. As mentioned previously, the design drivers considered in this paper are weight, Shuttle packaging volume, reflector lowest natural frequency (stiffness), and on-orbit assembly time. In the design of space structures, a natural frequency requirement is difficult to establish. Generally, there is a requirement that the lowest natural frequency of the spacecraft be kept above the control bandwidth frequency, which, in most cases, is readily achievable. Aside from this requirement, it is generally accepted that stiffer is better from an overall performance point of view. Because there are generally no precise frequency requirements, established for reflectors, simple and approximate methods were used for the frequency analyses of this paper. The emphasis of the preliminary design approach presented here is on speed and ease of use rather than refined accuracy.

Frequency Analysis

The method of frequency analysis used herein is summarized in Fig. 7. The truss/panel reflector system is considered dynamically as an equivalent flat circular sandwich plate. It is shown in Ref. 5 that the effects of curvature are negligible on the lowest natural frequency of a free-free reflector. As in Ref. 8, the stiffness of the upper and lower surfaces of the truss are treated as isotropic faces, and the equivalent properties for the face modulus, thickness, and Poisson's ratio are given on the left of Fig. 7. The resulting expression for the plate bending stiffness D of an equivalent sandwich plate is given in the upper right of the figure. The weight per unit area for the truss is simply the total weight of the truss (struts + nodes) divided by the reflector area. The lowest natural frequency of the reflector f is given by the equation in the lower right of the figure and is the lowest free-free frequency of a circular plate, as given in Ref. 15.

Results from this analysis were compared with more accurate results from a finite element analysis, and the correlation is given in Appendix A. It is noted that the maximum diameter Dmax of the reflector surface was selected for use as the circular plate diameter in the preliminary design analysis. The maximum diameter is equivalent to the maximum dimension of the reflector. This selection was made only because it provides better correlation with the more refined finite element analyses. An alternative to using the maximum diameter would be to use the diameter of a circle having the same area as the reflector surface. The circular plate frequency is a function of the square of the diameter, and using the maximum diameter can be shown (see equations in Ref. 13) to decrease the resulting plate frequency by approximately 20%. As discussed in Appendix A, transverse shearing and rotary inertia effects are significant for trusses with a low number of rings, and the maximum diameter selection was made in a heuristic sense to somewhat account for these. The simplified analysis used in this paper is considered quite adequate for preliminary conceptual design studies because all trends are accurately predicted for practical numbers of truss rings. It is noted that the exact surface area is used for weight calculations. Also, reflector diameters are reported as effective circular area diameters to best indicate the wave collection capability of a particular hexagonal-segmented reflector surface.

Computer Program

To obtain numerical results for a wide variety of reflector parameters, a preliminary design computer program was written. A complete listing of this program appears in Ref. 16, and a simple flow chart of the program is given in the following.

Input Reflector Parameters

- Panel face sheet—thickness, density
- Panel core—density
- Strut—density, modulus
- Node weight
- Strut buckling load constraint value
- Frequency design constraint (if desired)
- Effective diameter of reflector
- Number of reflector rings
- Weight of node
- Normalized truss depth parameter (see Appendix B)
- Number of truss rings
- Strut initial thickness

Calculate
Panel width
Length of strut
Core thickness
Number of struts, nodes, and panels
Maximum reflector diameter
Diameter of strut from Euler buckling equation
Weight of panels
Weight of struts
Total weight of reflector
Truss bending stiffness from equation in Fig. 7
Reflector frequency from equation in Fig. 7
Volume of struts and panels
Check Frequency Constraint
Increase strut wall thickness (0.001-in. increments)
Recalculate weight and reflector frequency
Repeat until design frequency is met

Preliminary Design Study Results

This program permits a wide range of reflector parameters to be studied and iterated on in a relatively short period of time. A variety of reflector parameters were studied and selected results are presented in the following sections to demonstrate the usefulness of the computer program and to compare different design drivers.

Reflector Weight as a Function of Reflector Diameter for Fixed Panel Size

To examine the weight of precision reflector structures over a wide range of sizes, the preliminary design procedure was applied to reflectors up to 100 m in diameter. For these reflectors, the panel size was chosen to be fixed to the maximum size panel that can fit in the Shuttle cargo bay (4.2 m), so as to minimize part count and on-orbit assembly time.

The reflector weights are shown in Fig. 8 and indicate that large-diameter reflectors are extremely heavy and are dominated by panel weight. On the other hand, reflectors 40 m in diameter and smaller have weights that are less than one half of the Shuttle weight limit and, thus, from a weight point of view, appear to be practical. The need for more than one Shuttle flight, due to weight or volume considerations, may or may not be crucial to reflector design.

Figure 9 shows the lowest natural frequency of the reflectors examined with the fixed panel size of 4.2 m. The lowest natural frequency is relatively low for larger reflectors. For some applications, it may be necessary to constrain the lowest natural frequencies to higher values for control purpose. To achieve higher natural frequencies, it will be necessary to provide a stiffer support truss. In the next section, constraining the lowest natural frequency to a higher value is shown to significantly increase the weight of the support truss.

Reflector Weight as a Function of Number of Rings for Fixed Diameters

To investigate the effect of panel size and of a frequency constraint on reflector design, three reflector diameters were chosen for study: 15, 20, and 40 m. For each of these diameters, the reflector weight was determined as a function of the number of reflector rings to determine the effect of panel size on reflector design. Recall that panel size decreases as the number of rings for a fixed diameter reflector increases. For each fixed diameter, reflector weight was examined both with and without a constraint on the lowest natural frequency of the reflector.

15-Meter-Diameter Reflector

In Fig. 10, weight results are shown for a 15-m-diam reflector as a function of the number of rings. As indicated in the figure by the heavy solid curve, the reflector weight is not a strong function of the number of rings.

The reason for this is the offsetting weight trends exhibited by the panels and the truss as a function of number of rings. The total panel weight decreases as the number of rings increases while the weight of the support truss increases. The panel weight decreases because the panel thickness is constrained to be a linear function of panel size to maintain stiffness for surface accuracy. Reflectors with larger numbers of rings have panels that are smaller in diameter, which can be thinner and lighter. The truss weight increases as the number

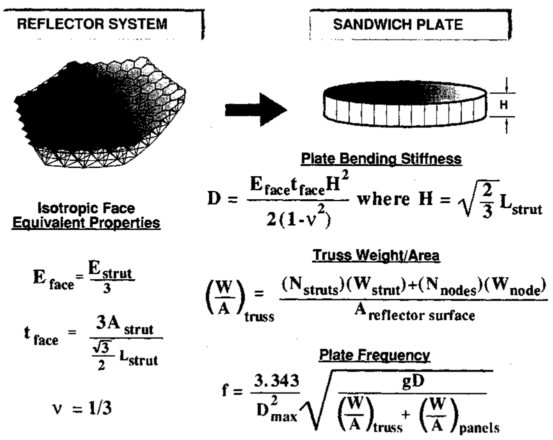

Fig. 7 Reflector preliminary design analysis.

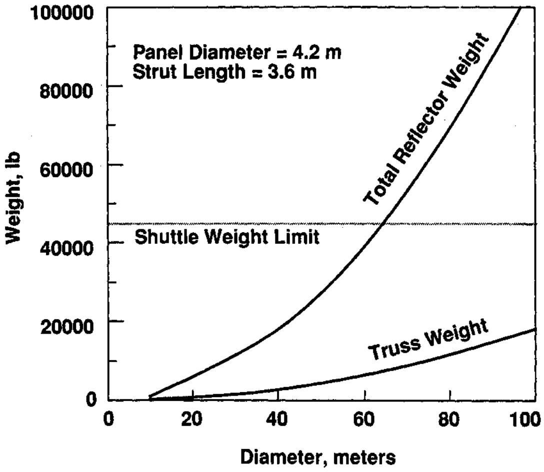

Fig. 8 Reflector weight as a function of diameter for fixed reflector parameters.

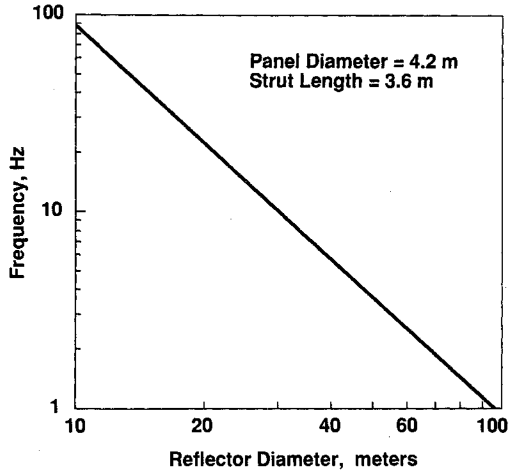

Fig. 9 Reflector frequency as a function of diameter for fixed reflector parameters.

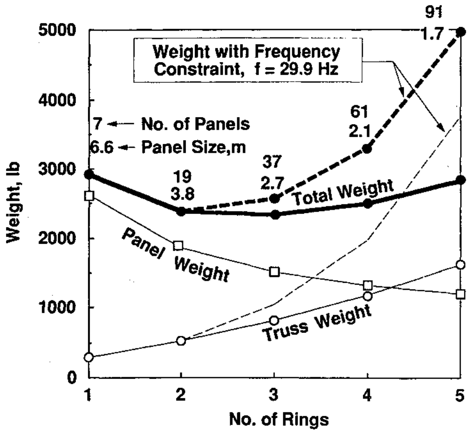

Fig. 10 Weight of 15-m-diam reflector with and without frequency constraints.

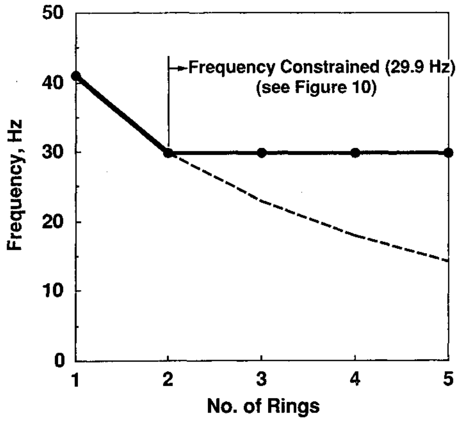

Fig. 11 Lowest natural frequency of 15-m-diam reflector.

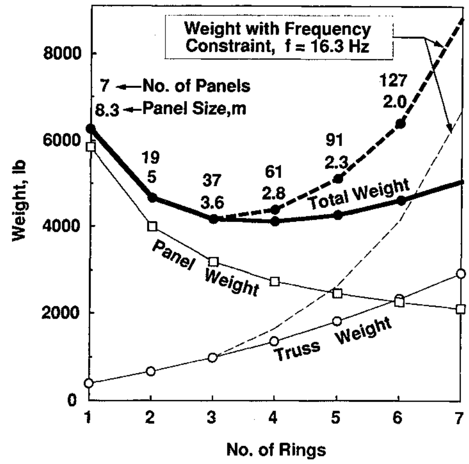

Fig. 12 Weight of a 20-m-diam reflector with and without frequency constraint.

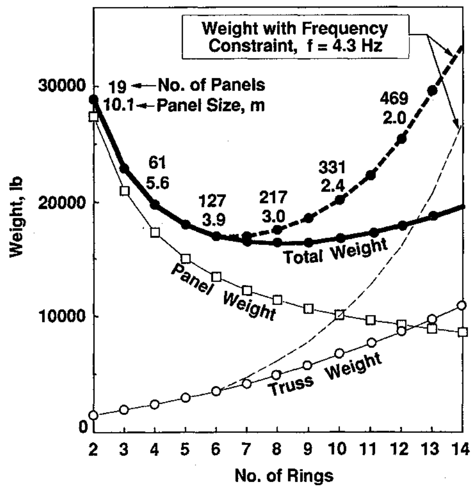

Fig. 13 Weight of a 40-m-diam reflector with and without frequency constraint.

of rings increases, primarily because of the larger number of truss joints required. As shown by the equations in Ref. 13, for large-diameter reflector trusses, the number of joints increases approximately as the square of the number of rings. For an unconstrained 15-m-diam reflector, the minimum total weight of about 2400 lb occurs for a three-ring reflector. However, very little weight penalty would result by using a two-ring reflector with larger panels. For a two-ring reflector, the weight of the truss is 500 lb, about 20% of the total reflector weight. Such a two-ring truss has 45 joints that weigh 300 lb and 156 struts that weigh 200 lb.

For a small number of rings, the individual panel size is large and the part count is relatively low. As the number of rings increases, the panel size decreases and the part count increases significantly. As mentioned previously, for on-orbit construction considerations, it is desirable to keep the part count low, but practical manufacturing considerations may limit the size of individual panels. For a 15-m-diam reflector, considering a discrete number of rings, the best panel size from an assembly point of view would be 3.8 m ($N = 2$). This is the largest panel size for this case that can fit in the Shuttle cargo bay. For a two-ring reflector, 19 panels would have to be assembled on orbit, as indicated in the figure. If the maximum panel size were limited to 2.1 m, the number of panels to be assembled would more than triple to 61. This would have a significant impact on assembly time, as was shown in Fig. 6. As the number of rings increases, the total reflector weight increases and the truss bending stiffness decreases. As a result, the lowest natural frequency of the reflector decreases. This is shown by the dashed line in Fig. 11.

To compare the fixed 15-m-diam reflectors on an equal stiffness basis, the lowest natural frequency of the reflectors with a higher number of rings ($N > 2$) was constrained to that of the two-ring reflector ($f = 29.9$ Hz). To meet this frequency constraint, the truss strut wall thickness was increased to obtain the desired stiffness. As can be seen by the heavy dashed line in Fig. 10, the total weight does not increase dramatically for $N = 3$ and 4. However, for $N = 5$, the reflector weight is approximately double the minimum weight of 2400 lb. For this case, the weight of the truss necessary to meet the frequency constraint begins to dominate the total reflector weight. Thus, in reflector designs with stringent frequency requirements, there will be a trade to be made between panel size (number of rings) and total weight and reflector construction time. An alternate, lower weight penalty approach for increasing the frequency is to increase the truss depth. This approach is discussed in Appendix B.

20-Meter-Diameter Reflector

Weight trends for a 20-m-diam reflector are shown in Fig. 12. The trends are similar to those for the 15-m reflector shown in Fig. 10. The minimum weight for the 20-m reflector is about 4200 lb and occurs for the case of four rings. The maximum size panel that can fit in the Shuttle cargo bay for a 20-m reflector is 3.6 m and occurs for a three-ring truss. The number of panels required to assemble the three-ring 20-m reflector (37) is nearly double the number required for the two-ring 15-m reflector (19). To compare the effect of smaller panel size on reflector weight on an equal stiffness basis, the reflector frequency was constrained to be that of the frequency for the three-ring reflector ($f = 16.3$ Hz). With this constraint, for example, a six-ring reflector weighs about 6300 lb, which represents a 50% weight increase over the three-ring reflector.

Table 2 **Preliminary design results for a 20-m-diam reflector**

Design drivers	3.6-m panels, three ring	2.3-m panels, five ring	1.5-m panels, eight ring
Weight, lb	4200	4300	5600
Frequency, Hz			
(unconstrained)	16	11	7
Volume			
Volume cargo bay	0.15	0.11	0.09
Number of panels	37	91	217
Number of struts	315	795	1920
Construction time, h	9	22	52

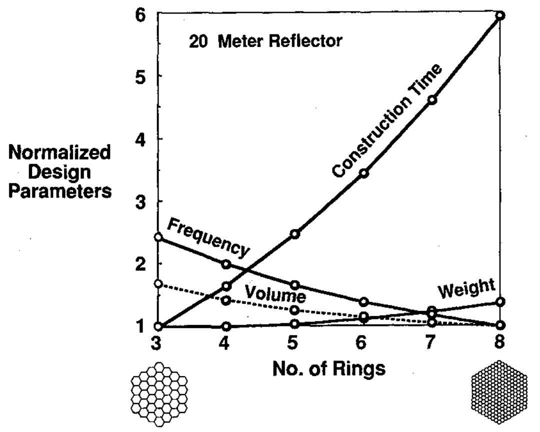

Fig. 14 Normalized design parameters for a 20-m-diam reflector.

40-Meter-Diameter Reflector

As a final example, the weight trend for a very large reflector (40-m diameter) is shown in Fig. 13. The minimum weight for the 40-m reflector is about 17,000 lb for an eight-ring reflector. The largest reflector panel for this case that will fit in the Shuttle cargo bay is 3.9 m and this occurs for a six-ring reflector. From Fig. 13, it can again be seen that the total weight is not strongly sensitive to the number of rings. Thus, very little weight penalty would result from selecting the six-ring reflector to reduce part count and minimize on-orbit assembly time. For this case, a frequency constraint corresponding to the six-ring reflector ($f = 4.3$ Hz) was applied. Again, the application of such a constraint significantly increases the total reflector weight as the number of rings increases.

Design Drivers for a 20-Meter-Diameter Reflector

To determine which of the four design drivers (weight, frequency, packaging volume, and on-orbit assembly time) will dominate the final reflector design, the fixed 20-m-diam reflector was used to examine in further detail how each driver varies with panel size (or number of rings). For the 20-m reflector, the four design drivers were examined for reflectors having three to eight rings. Sample results of this investigation are presented in Table 2 for three, five, and eight rings.

The number of panels and struts is also presented in the table because these directly influence the assembly time and will have a major impact on reflector fabrication costs. Since these design drivers are so different in nature, a complete mission system and costing study would have to be conducted to compare them for the purpose of an absolute design decision. However, considerable insight into which of the drivers are dominant can be obtained by comparing them on a nonweighted, normalized basis. This has been done and the results are presented in Fig. 14 for a range of number of reflector rings from three to eight. This range is considered because for less than three rings the individual reflector panels are too large for packaging in the Shuttle cargo bay, and for a large number of rings, the assembly times become prohibitive.

For each parameter in Fig. 14, the results have been normalized with respect to the minimum value of that parameter in the range of rings considered. These curves present a relative comparison of the maximum variation of all of the drivers over a practical range of interest. For example, it can be seen that the weight varies by 30% and the volume varies by 70% over the range. The frequency increases by a factor of 2.5 over the minimum value, and the assembly time is six times as large for eight rings as it is for three rings. All of these trends favor the smaller number of rings except the packaging volume. Recall that a larger number of rings results in thinner panels that have a smaller packaging volume. In Table 2, it can be seen that the worst case volume (a three-ring, 20-m reflector) is approximately one-sixth the volume of the Space Shuttle cargo bay, and so it is unlikely that volume will be a major design driver. The most likely major driver of the four considered in this paper is on-orbit assembly time. It can be seen from Fig. 14 that this driver has the most rapid variation over the range of rings considered. Thus, it would seem that attention should be focused on developing reflector concepts with a minimum number of rings that keep within the packaging constraints of the Space Shuttle cargo bay. A modular panel concept for constructing reflectors with fewer numbers of rings (reduced part count) that would require less assembly time is presented in Ref. 16.

Concluding Remarks

A simplified preliminary design method for precision reflectors has been presented and demonstrated. This design method is approximate but provides the capability for a rapid assessment of a wide range of reflector parameters as well as new structural concepts and materials. Four major design drivers for precision segmented reflectors (weight, packaging volume, stiffness, and on-orbit assembly time) were studied. From the results presented, the following conclusions can be drawn:

1) The weight of segmented reflectors is not a strong function of the number of rings over practical ranges (with perhaps the exception of reflectors with a high natural frequency constraint). This is a result of the fact that the support truss and the reflector surface panels have offsetting weight trends as the number of rings increases. However, total reflector weight will be a major factor in the performance of a complete spacecraft, and, accordingly, efforts should continue to reduce reflector weight.

2) Launch vehicle reflector packaging volume is dominated by the surface panels rather than the truss struts. The packaging volume of a 20-m reflector is only one-sixth of the Space Shuttle cargo bay volume. For a reflector of fixed diameter, packaging volume is the only parameter (of the four considered) that decreases as the number of rings in a reflector increases. This is because a larger number of rings results in smaller and, hence, thinner panels.

3) Constraining the lowest natural frequency of a reflector to some minimum value requires a stiffer reflector support truss. Such a constraint can significantly increase the total reflector weight if the stiffness is obtained by adding material to the truss struts. The lowest natural frequency of reflectors can be increased with a smaller weight penalty by increasing the depth of the support truss (see Appendix B), however, the lack of specific design requirements on frequency make it difficult to study frequency as a design driver.

4) On-orbit assembly time increases dramatically with increasing numbers of rings in a reflector. For example, a three-ring reflector would require about two Shuttle-based astronaut EVAs, whereas a six-ring reflector would require about six EVAs (possibly a prohibitive number).

5) A normalized comparison of perceived design drivers as a function of the number of reflector rings indicates that on-orbit assembly time will be a major design driver. Attention should be focused on developing concepts for minimizing the number of rings in a reflector while staying within the packag-

Table A1 Reflector input parameters for finite element analysis

Strut length	78.74 in.	(2 m)
Strut diameter	1.0 in.	(2.54 cm)
Strut wall thickness	0.06 in.	(0.0236 cm)
Reflector mass per unit area	0.82 lb/ft^2	(4 kg/m^2)
Node weight (no attachment penalty)	4.4 lb	(2.0 kg)

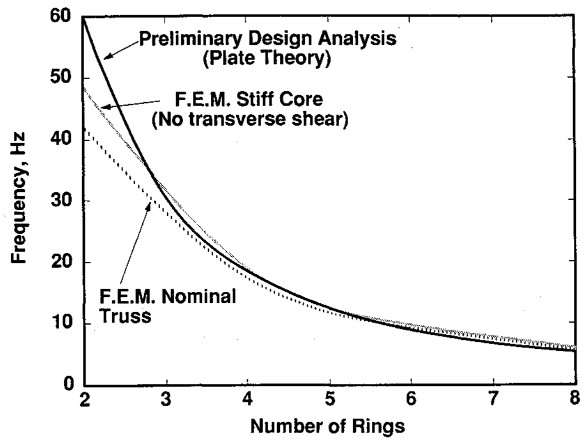

Fig. A1 Comparison between preliminary design analysis and finite element analysis.

ing size limitations of the Space Shuttle or other applicable launch vehicles. Although the emergence of automated in-space assembly methods may reduce the importance of assembly time lines, it is likely that reduced part count will still provide an improved assembly scenario.

Appendix A: Analysis Comparison

As discussed previously, the frequency analysis used in the preliminary design method was selected on the basis of simplicity and ease of use rather than high accuracy. However, to provide insight into the range of applicability of the method, a comparison was made with essentially exact results from a finite element analysis. The example chosen for comparison was a reflector with the strut length fixed while the number of rings was varied from two to eight. The reflector input parameters are listed in Table A1.

The strut modulus and density was the same as presented in Table 1. The finite element analysis was conducted using MSC/NASTRAN. Since only the lowest free-free natural frequency was investigated, rod elements were used for the truss. The results of this comparison are presented in Fig. A1.

Again, the maximum diameter of the reflector panel surface was used in the primary design analysis to heuristically improve the overall comparison. The maximum diameter of the reflector is 10% greater than the effective circular area diameter for any number of rings, as can be seen from the equations presented in Ref. 13. Since the frequency varies as the square of the diameter, the use of the larger maximum diameter results in decreasing the frequencies by approximately 20%. This decrease was found to yield a better correlation between the finite element and preliminary design results. The proper diameter to use for a large number of rings would probably be the effective diameter that provides the correct mass moment of inertia of the reflector about a major diameter. However, for a low number of rings, there is no simple approach to rationally account for transverse shearing and rotary inertia effects that begin to influence the response.

The curve labeled stiff core in Fig. A1 represents a finite element analysis where the core members were intentionally made extremely stiff to eliminate transverse shearing effects. This accounted for about one half of the difference between the finite element results and the simplified analysis for a small number of rings. The remainder of the difference is believed to

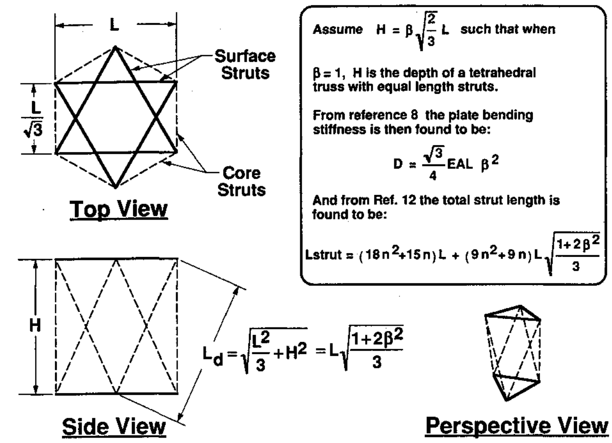

Fig. B1 Characteristics of a tetrahedral truss with different length core members.

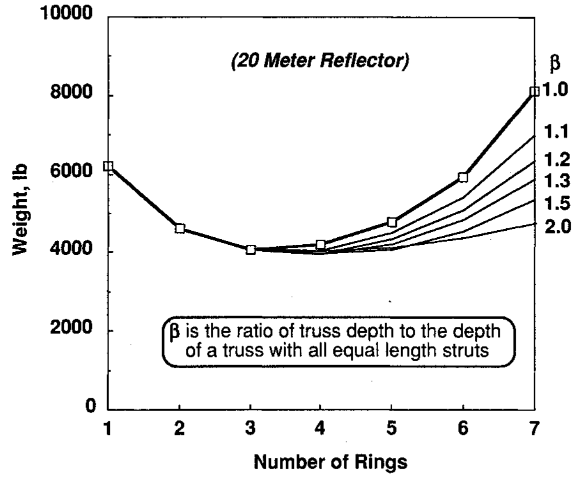

Fig. B2 Effect of truss depth on total weight of frequency constrained reflectors.

be caused by the neglect of rotary inertia effects in the preliminary design analysis. A one-dimensional truss beam was studied to explore this effect. For the one-dimensional beam, it was relatively simple to eliminate rotary inertia effects from the finite element analysis. Results of those studies indicated that, indeed, rotary inertia effects were the primary difference in the analyses.

Appendix B: Effect of Core Depth on Reflector Weight

As was mentioned previously, the application of a frequency constraint to a segmented reflector with a fixed panel size can result in a significant weight increase. The reason for this is that the increase in truss stiffness required to raise the natural frequency is usually obtained by adding more material to the struts. In general, this is an inefficient means for achieving stiffness. An alternate approach is to hold the length of the truss surface struts fixed while increasing the length of the truss core struts (core struts connect the upper and lower truss surfaces). This increases the truss depth (and stiffness) with very little increase in truss weight. To explore this approach, equations were developed for the equivalent plate stiffness D and total strut length, Lstrut, of trusses with core members of different length than the surface members. These equations are presented in Fig. B1 and are included in the preliminary design program in Ref. 16. Beta = 1 corresponds to all struts in the truss being equal length (core strut length = surface strut length). Truss depth increases linearly with beta.

An example of how increasing the core depth can reduce reflector weight for frequency constrained reflector designs is shown in Fig. B2 for the 20-m reflector of Fig. 12. For a seven-ring truss, doubling the truss depth reduces the total reflector weight by ~ 50%. For smaller numbers of rings, the weight savings are not as great. This approach should be seriously considered as an alternative to adding more material to the truss strut.

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