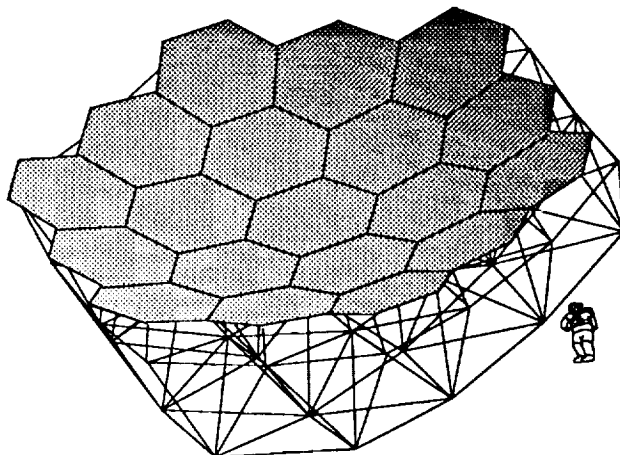NASA Technical Memorandum 102605

PRELIMINARY DESIGN APPROACH FOR LARGE HIGH PRECISION SEGMENTED REFLECTORS

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LARGE HIGH PRECISION SEGMENTED REFLECTORS

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Introduction

There is currently considerable interest in precision segmented reflectors for various future NASA missions. Potential applications for such reflectors include submillimeter astronomical observatories, microwave radiometers which 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 references 1 and 2, while typical structural requirements are presented in reference 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 (references 4, 5, 6, and 7 for example) have shown that 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 (reference 8), and it has been shown in reference 9 that well made trusses can provide a very precise framework for supporting reflective panel surfaces.

A common concept for constructing precision reflectors is to attach hexagonal reflector panels to a tetrahedral truss which has nearly equilateral triangular bays. In this paper, a rapid preliminary design procedure is described and results are presented which 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 preliminary design procedure-developed in this paper is based on the equivalent plate theory for trusses developed in reference 8 and verified in reference 10. Although the simple analysis used in this paper is not

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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 currently limited in size to 1 to 2 meters 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. As a first attempt at addressing this problem, this paper concludes by briefly presenting a concept for a family of larger modular panel segments. These modules are obtained by preassembling, on the ground, a series of small high precision hexagonal panels into a larger module supported by a stiff backup structure. The use of such panel modules for the construction of precision segmented reflectors may dramatically reduce on-orbit construction time, which is perhaps the major design driver for such reflectors.

Erectable Reflector
Description and Design Drivers

An example of a high precision reflector is shown in figure 1. This particular reflector is a 20 meter-diameter submillimeter astronomical observatory for deep space measurements of infrared radio frequencies as discussed in reference 1. This application requires that distortions in the reflector surface be no greater than 2 microns 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 (LaRC) is to achieve surface accuracies for large segmented reflectors on the order of 10 to 20 microns rms passively This represents about an order of magnitude improvement over the current state-of-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 ongoing research activities within NASA.

2

A typical reflector that is considered in this paper is shown in figure 2. This particular reflector has an effective circular diameter (the diameter of a circle with the same area as the reflector surface) of 16.6 meters and is composed of 19 hexagonal honeycomb panels which are 4.2 meters across (corner to corner). This is the maximum size panel which can fit into the Shuttle cargo bay. Note that although panel size may currently be limited to 1 or 2 meters as mentioned previously, this paper considers even larger panel sizes. As panel technology develops, larger panels (or alternatively, panel modules as described later in this paper) 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 figure 3 where a 36 meter-diameter reflector with 4.2 meter 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 meters), reflectors with a diameter greater than approximately 50 meters 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 figure 4 and is discussed in detail in reference 11. 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 which 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 larger panels in order to maintain accuracy. This increases packaging volume and 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 reference 9) 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) does 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 which 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 figure 5. 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, while the reverse is true for truss B. The shaded regions in figure 5 indicate the definition of rings as used in this paper. The example shown in the figure represents 3 rings of hexagonal panels. Note that ring number 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 Appendix A. The part count and reflector panel geometry as a function of number of rings is shown in figure 6. 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 greater than about 40 GHz, continuous (non-mesh) 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, packaging volume constraints and mechanical complexity have to date limited these deployable concepts to about 10 meters 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 which is required. Past studies have shown that assembly time and effort are directly related to the number of individual elements which must be assembled. As a result, part count is a major concern for large reflectors. As was shown in figure 6, the element part count increases significantly as the number of rings in a reflector increases. In reference 11 astronaut assembly times were estimated to be about 1/2 minute per strut, and about 10 minutes per panel. These estimates are based on two astronauts simultaneously performing a construction. Using these estimates, reflector assembly times are shown in figure 7.

Current space suit technology limits each individual astronaut EVA (extravehicular activity) to about 5 or 6 hours. 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 minutes 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 underway to reduce the time required to install individual panels from the current 10 minute estimate to 5 minutes. Such an improvement would reduce total construction time by about 35 percent. 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 next sections of the paper, weight, stiffness, and packaging volume will be examined as a function of the number of reflector rings in order to establish sensitivities to these parameters and to explore practical limitations on reflector sizes.

Parameters Used in Design Study

The basic structural parameters used in the preliminary design study are shown in figure 8. 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 minimum gage constraints. The face sheet thickness was chosen as 0.02 inches 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 and is discussed in reference 12. 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

5

ratio results in 1 meter panels that are 1 inch thick, 2 meter panels that are 2 inches 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 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 CTE graphite epoxy cores.

The modulus of the truss struts was chosen as 30 million 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 inches to ensure adequate toughness for handling and assembly. The strut material density of 0.06 lb/in^3 was chosen to be 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 a 1000 pound buckling load capacity (Pcr). The value of 1000 pounds 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 pounds results in a weight savings of less than 5 percent. It is believed that the constraint of 1000 pounds 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 lbs) was assumed to be the same as the joint clusters in the precision truss discussed in reference 7, however a 50 percent weight penalty was included to account for fixtures that would be required to attach the panels to the truss. In reference 7 the struts and the joints are one inch in diameter. Although the diameters of the truss struts considered in this paper vary from about one inch to two inches, 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 one inch in diameter are considered, they are assumed to be tapered at the ends to allow interfacing with a one inch 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, since 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 upon speed and ease of use rather than refined accuracy.

Frequency Analysis

The method of frequency analysis used herein is summarized in figure 9. The truss/panel reflector system is considered dynamically as an equivalent flat circular sandwich plate. It is shown in reference 5 that the effects of curvature are negligible on the lowest natural frequency of a free-free reflector. As in reference 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 figure 9. 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 reference 13. Results from this analysis were compared with more accurate results from a finite element analysis and the correlation is given in Appendix B. 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

7

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 Appendix A) to decrease the resulting plate frequency by approximately 20 percent. As discussed in Appendix B, 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 in Macintosh Microsoft Basic and a complete listing is presented in Appendix D. Self descriptive variables were used when possible in the program, and numerous comment statements are included. A simple flow chart of the program is given below.

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 figure 9

8

Reflector frequency from equation in figure 9
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

OUTPUT RESULTS
Output results to 19" monitor, allow viewing of several variables simultaneously
Import results to plotting program for graphical interpretation

Preliminary Design Study Results

The above program permits a wide range of reflector parameters to be studied and iterated upon 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 (4.2 meters)

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 meters in diameter. For these reflectors, the panel size was chosen to be fixed to the maximum size panel which can fit in the Shuttle cargo bay (4.2 meters), so as to minimize part count and on-orbit assembly time. The reflector weights are shown in figure 10 and indicate that large diameter reflectors are extremely heavy and are dominated by panel weight. On the other hand, reflectors 40 meters in diameter and smaller have weights which 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 11 shows the lowest natural frequency of the reflectors examined with the fixed panel size of 4.2 meters. 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 purposes. 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.

9

Reflector Weight as a Function of Number of
Rings for Fixed Diameters (15, 20, and 40 m)

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 meters. For each of these diameters the reflector weight was determined as a function of the number of reflector rings in order 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. Detailed listings of the computer generated results for each diameter are presented in Appendix E.

15 Meter-Diameter Reflector: In figure 12 weight results are shown for a 15 meter-diameter 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 which are smaller in diameter which can hence be thinner and lighter. The truss weight increases as the number of rings increases primarily because of the larger number of truss joints required. As shown by the equations in reference 12, for large diameter reflector trusses, the number of joints increases approximately as the square of the number of rings. For an unconstrained 15 meter-diameter reflector, the minimum total weight of about 2400 pounds occurs for a 3 ring reflector. However, very little weight penalty would result by using a 2 ring reflector with larger panels. For a 2 ring reflector the weight of the truss is 500 pounds, about 20 percent of the total reflector weight. Such a 2 ring truss has 45 joints which weigh 300 pounds and 156 struts which weigh 200 pounds.

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 meter-diameter reflector, considering a discrete number of rings, the best panel size from an assembly point of view would be 3.8 meters (N=2). This is the largest panel size for this case which can fit in the Shuttle cargo bay. For a 2 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 meters, 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 figure 7.

10

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 figure 13. To compare the fixed 15 meter-diameter 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 2 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 figure 12, the total weight does not increase dramatically for N=3 and N=4. However, for N=5 the reflector weight is approximately double the minimum weight of 2400 pounds. 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 C.

20 Meter-Diameter Reflector: Weight trends for a 20 meter-diameter reflector are shown in figure 14. The trends are similar to those for the 15 meter reflector shown in figure 12. The minimum weight for the 20 meter reflector is about 4200 pounds and occurs for the case of 4 rings. The maximum size panel that can fit in the Shuttle cargo bay for a 20 meter reflector is 3.6 meters, and occurs for a 3 ring truss. The number of panels required to assemble the 3 ring 20 meter reflector (37) is nearly double the number required for the 2 ring 15 meter 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 3 ring reflector (f = 16.3 Hz). With this constraint, for example, a 6 ring reflector weighs about 6300 pounds which represents a 50 percent weight increase over the 3 ring reflector.

40 Meter-Diameter Reflector As a final example, the weight trend for a very large reflector (40 meter-diameter) is shown in figure 15. The minimum weight for the 40 meter reflector is about 17,000 pounds for an 8 ring reflector. The largest reflector panel for this case that will fit in the Shuttle cargo bay is 3.9 meters and this occurs for a 6 ring reflector. From figure 15, 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 6 ring reflector to reduce part count and minimize on-orbit assembly time. For this case, a frequency constraint corresponding to the 6 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.

11

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 meter-diameter reflector was used to examine in further detail how each driver varies with panel size (or number of rings). For the 20 meter reflector, the four design drivers were examined for reflectors having 3, 4, 5, 6, 7 and 8 rings. Sample results of this investigation are presented in figure 16 for 3, 5, and 8 rings. The number of panels and struts is also presented in the table since these directly influence the assembly time and will have a major impact upon 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 non-weighted, normalized basis. This has been done and the results are presented in figure 17 for a range of number of reflector rings from 3 to 8. This range is considered because for less than 3 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 figure 17, 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 the all the drivers over a practical range of interest. For example, it can be seen that the weight varies by 30 percent and the volume varies by 70 percent over the range. The frequency increases by a factor of 2.5 over the minimum value while the assembly time is six times as large for 8 rings as it is for 3 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 which have a smaller packaging volume. In figure 16 it can be seen that the worst case volume (a 3 ring, 20 meter reflector) is approximately one sixth the volume of the Space Shuttle cargo bay 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 figure 17 that this driver has the most rapid variation over the range of rings considered. Thus, it would seem that attention should be focused upon developing reflector concepts with a minimum number of rings which keep within the packaging constraints of the Space Shuttle cargo bay. In the next section a new modular reflector concept is presented for constructing reflectors with fewer numbers of rings (reduced part count) that would require less assembly time.

12

Panel Module Concept

As indicated previously there are practical constraints such as manufacturing difficulty and cost, on how large individual panels can be fabricated. Thus, it may not be possible to individually fabricate to required accuracies some of the larger panels (> 2 meters) considered in this paper. To circumvent this problem, it is considered desirable to develop panel assemblies (herein called panel modules) made up of smaller single hexagonal panels. The geometries developed for these two new panel modules (3 hexagon modules and 7 hexagon modules) are shown in figure 18.

For the geometries shown, both the single panel and the panel modules attach to the same size truss at the same three points and each has the same area. In addition, each of the concepts shown in figure 18 can be used to form similar hexagonal shaped reflectors as shown in figure 19. The panel modules are basically crenelated hexagons since on a global scale they possess the same geometric characteristics as a single hexagon panel. The use of these built-up modules would have the effect of significantly reducing the number of components that would have to be assembled on orbit as compared with single smaller hexagon panel construction. Existing technology could be used to fabricate the smaller single panels which would make up each module. An example of how the use of modular panels can reduce part count and assembly time is shown in figure 19 for the case of an 8 ring reflector that is constructed from 217 single hexagon panels. Using 3 panel modules to construct the same reflector results in one fourth as many parts while using 7 panel modules results in one sixth as many parts. This reduction in part count results in a factor of 5 reduction in on-orbit assembly time. Further development and study of the panel module concept is currently underway.

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. A concept for a new family of hexagon panel modules which may permit a significant reduction in reflector part count was introduced. 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

13

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 meter 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) which 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. Truss stiffness could be a major factor if rapid reflector slewing maneuvers are required. In this case a deeper truss might be beneficial.

4) On-orbit assembly time increases dramatically with increasing numbers of rings in a reflector. For example, a 3 ring reflector would require about 2 Shuttle based astronaut EVAs while a 6 ring reflector would require about 6 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 upon developing concepts for minimizing the number of rings in a reflector while staying within the packaging size limitations of the Space Shuttle or other applicable launch vehicle. 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.

6) By using panel modules which are built-up from a series of smaller single hexagon panels, the potential for significantly reducing reflector part count and assembly time exists. Further development of this concept should be pursued to enable the practical application of such modules.

14

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4. Ard, K. E.: Design and Technology Study for Extreme Precision Antenna Structures, Final Report for the Antenna Design Study for Extreme Precision Antenna Reflectors and Systems Using Advanced Materials and Techniques. NASA Contractor Report 174861, August 1985.

5. Hedgepeth, John M. and Adams, Louis R.: Design Concepts for Large Reflector Antenna Structures. NASA Contractor Report 3663, January 1983.

6. Mahoney, M. J. and Ibbott, A. C.: A Large Deployable Reflector Assembly Scenario, A Space Station Utilization Study. NASA JPL D-5942, November 1, 1988.

7. Collins, Timothy J. and Fichter, W. B.: Support Trusses for Large Precision Segmented Reflectors: Preliminary Design and Analysis. NASA TM 101560, March 1989.

8. Mikulas, Martin M., Jr.; Bush, Harold G.; and Card, Michael F.: Structural Stiffness, Strength and Dynamic Characteristics of Large Tetrahedral Space Truss Structures. NASA TM X-74001, March 1977.

9. Greene, W. H.: Effects of Random Member Length Errors on the Accuracy and Internal Loads of Truss Antennas. Journal of Spacecraft and Rockets, Vol. 22, No. 5, September-October 1985, Pg. 554.

10. Hollaway, L.; Farhan, A.; and Gunn, M. J.: A Continuum Vibrational Analysis for a Tetrahedral Space Platform. International Journal of Space Structures, Vol. 3, No. 2, 1988, Pg. 104.

11. Bush, Harold G.; Lake, Mark S.; Watson, Judith J.; and Heard, Walter L., Jr.: The Versatility of a Truss Mounted Mobile Transporter for In-Space Construction. NASA TM-101514, November 1988.

12. Manhart, P. K. and Rodgers, J. M.: Segmented Mirror Manufacturing and Alignment Tolerances (SMMAT). NASA JPL Publication 89-3, March 1, 1989.

13. Blevins, Robert D., Ph.D.: Formulas for Natural Frequency and Mode Shape. Robert E. Krieger Publishing Company, Reprint 1986.

14. Kenner, W. Scott; Rhodes, Marvin D.; and Fichter, W. B.: Component Evaluation and Preliminary Assembly Considerations for Large Space Truss Structures. NASA TM-102604, February, 1990.

Figure 1. 20 Meter-Diameter Submillimeter Astronomical Observatory.
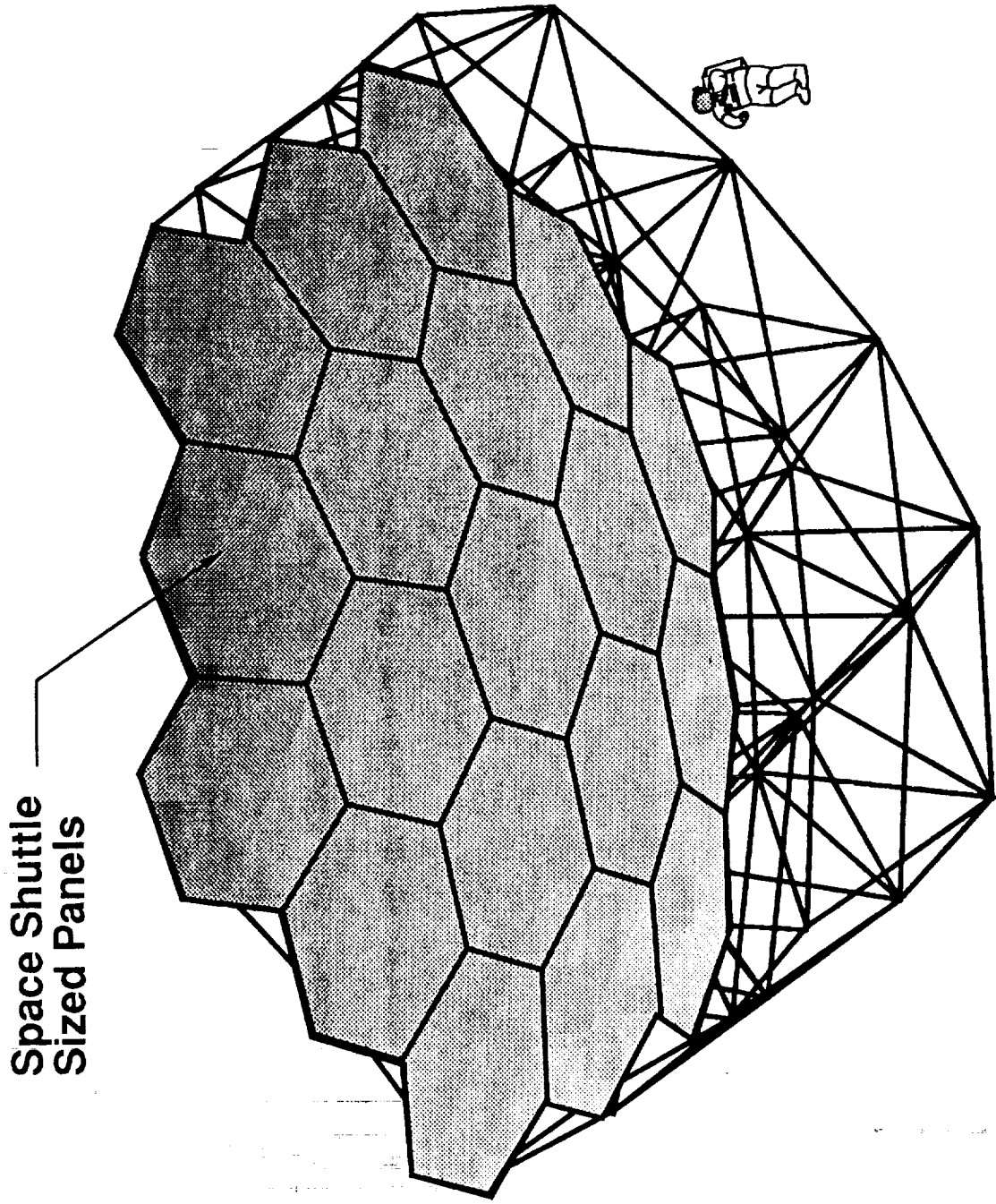
Space Shuttle
Sized Panels

Figure 2. 16.6 Meter-Diameter 2 Ring Precision Truss Reflector.
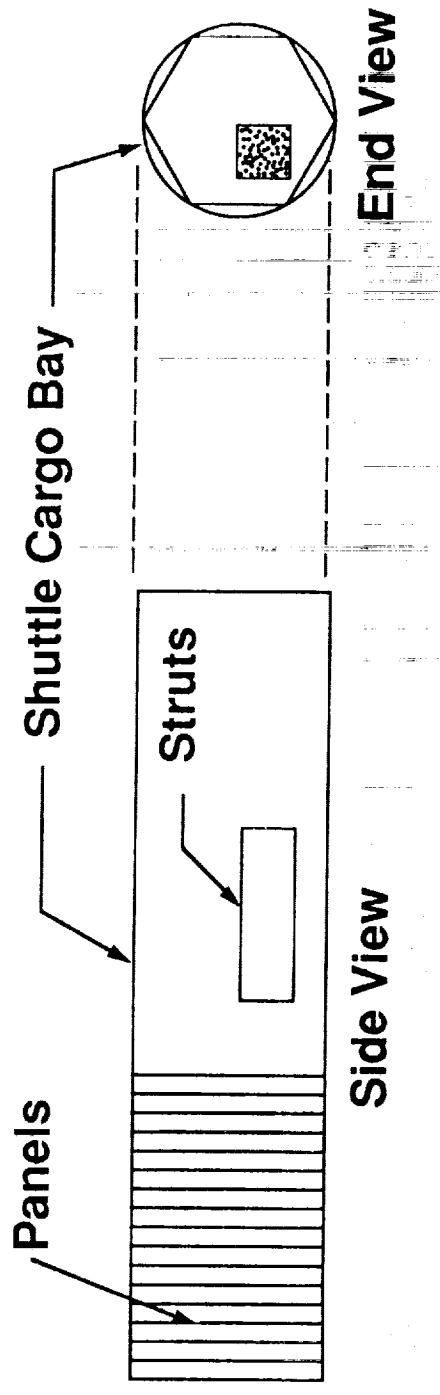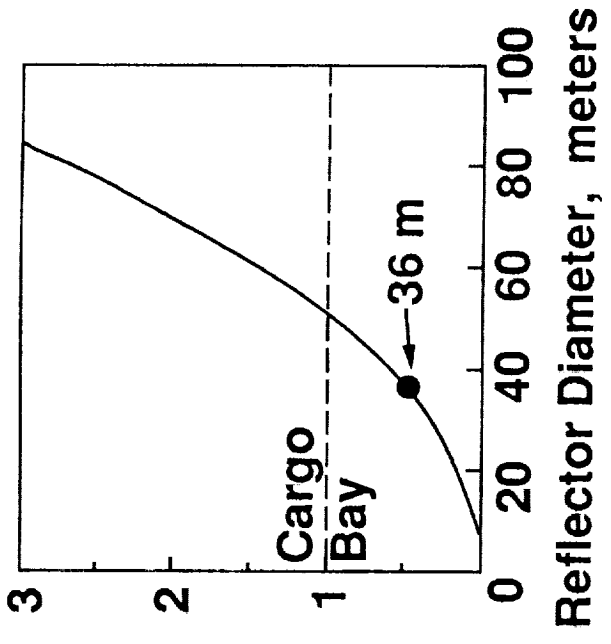
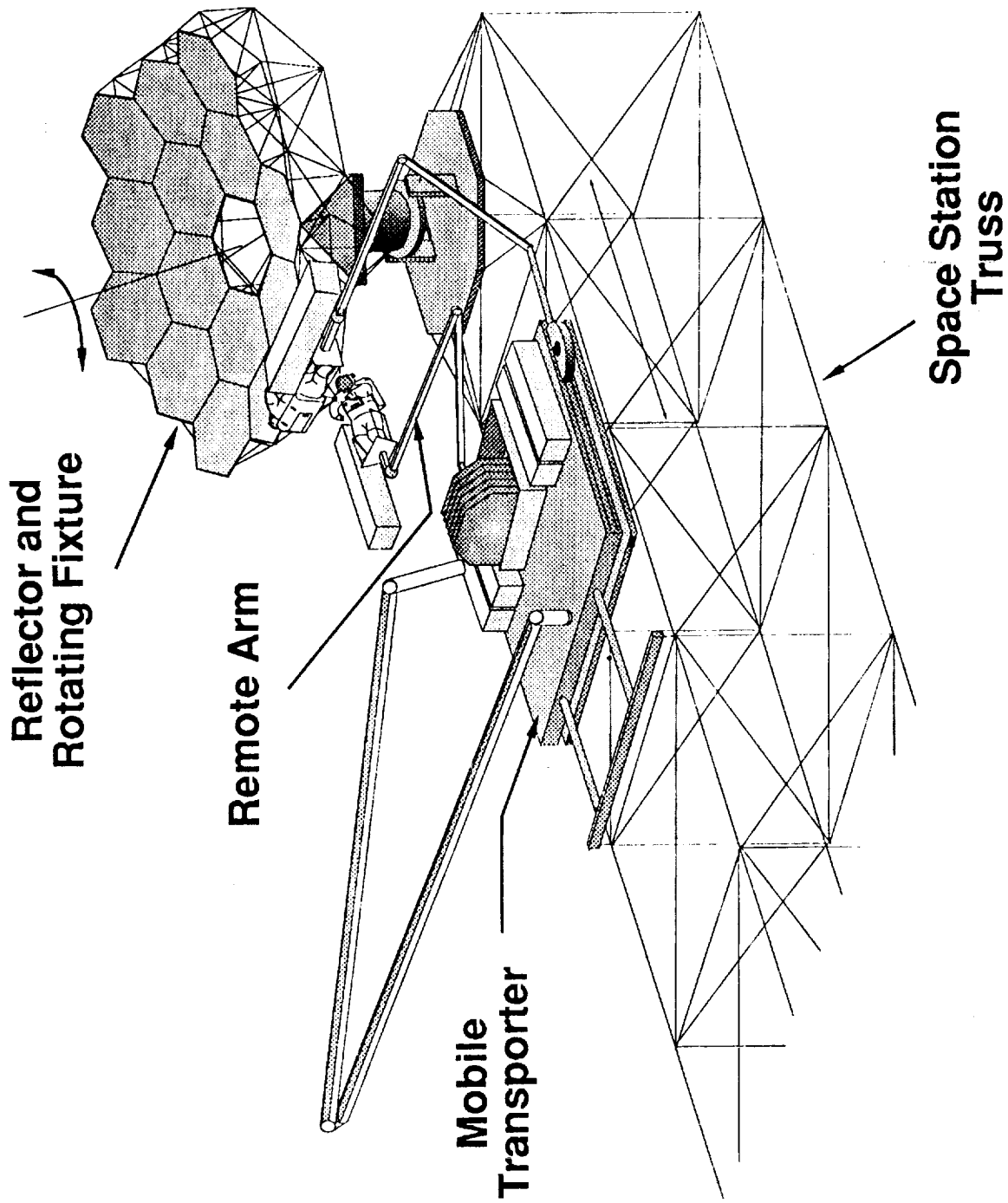Figure 3. Packaging of Erectable Reflector in Space Shuttle Cargo Bay.

Reflector and Rotating Fixture

Remote Arm

Mobile Transporter

Space Station Truss

Figure 4. Construction of Precision Segmented Reflector Using Mobile Transporter as an Assembly Aid.

19

Truss B

Truss Rings

N=1
N=2
N=3

Truss A

Corresponding
Panel
Rings
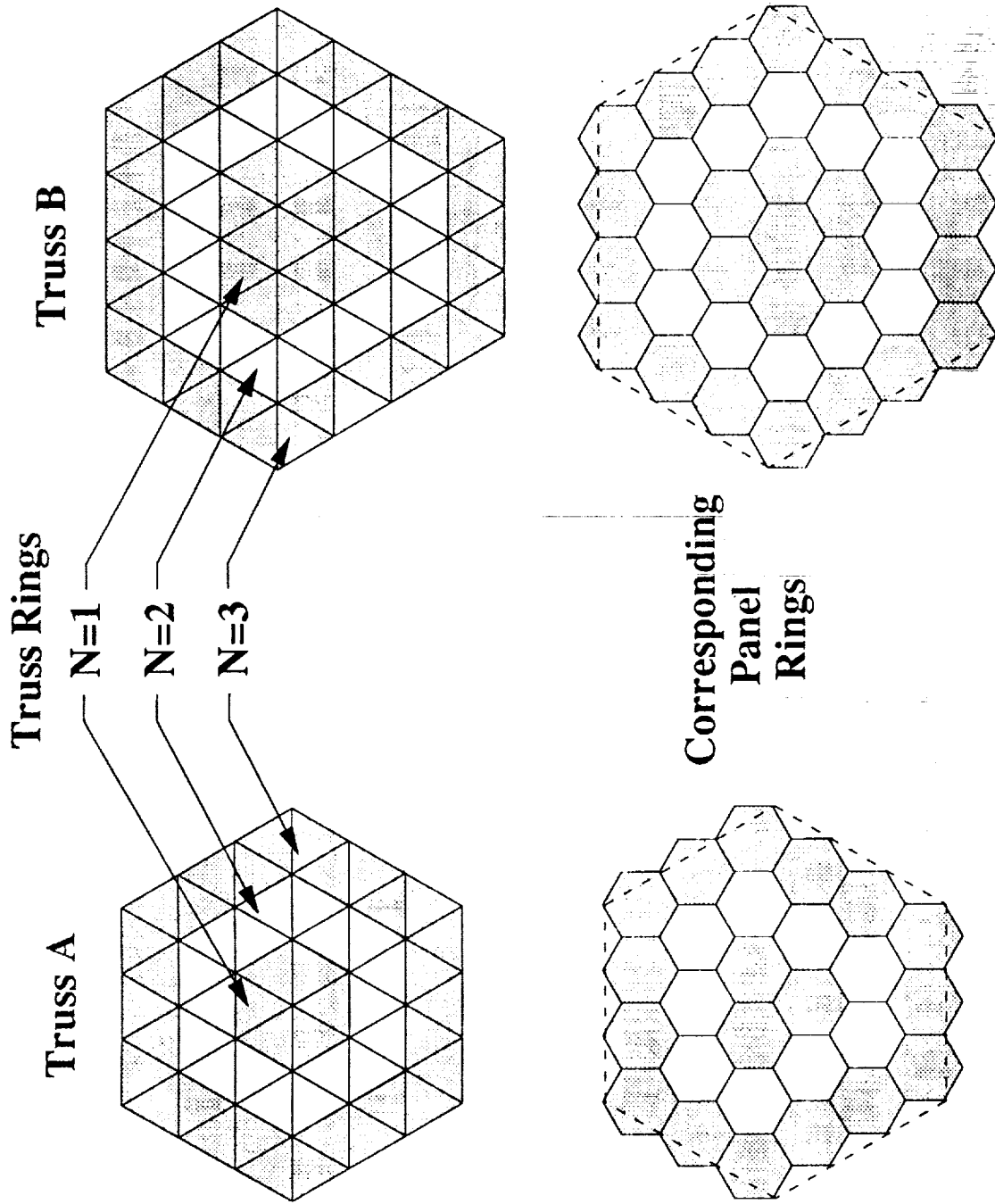
Figure 5. Truss And Panel Ring Definition for Two Types of Tetrahedral Trusses.

20

Figure 6. Part Count and Reflector Geometry as a Function of Number of Rings for Truss Type B.
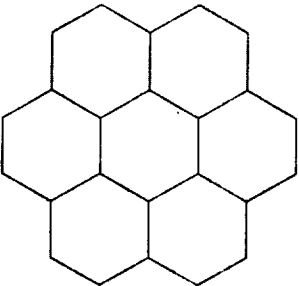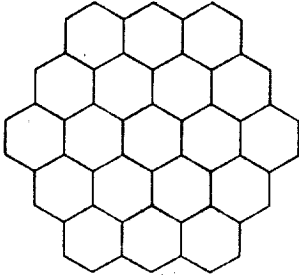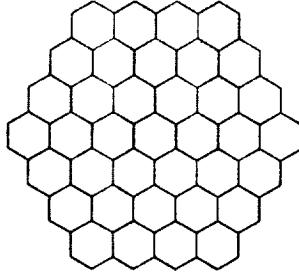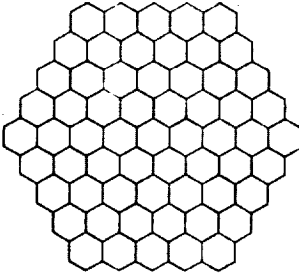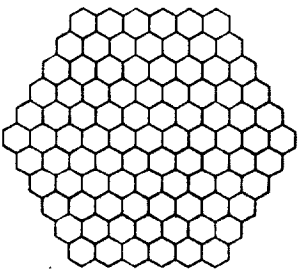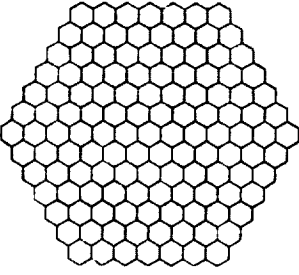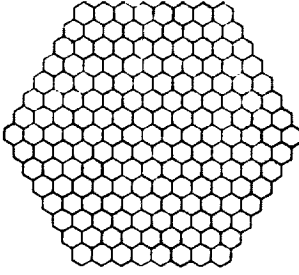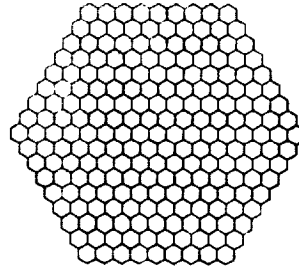
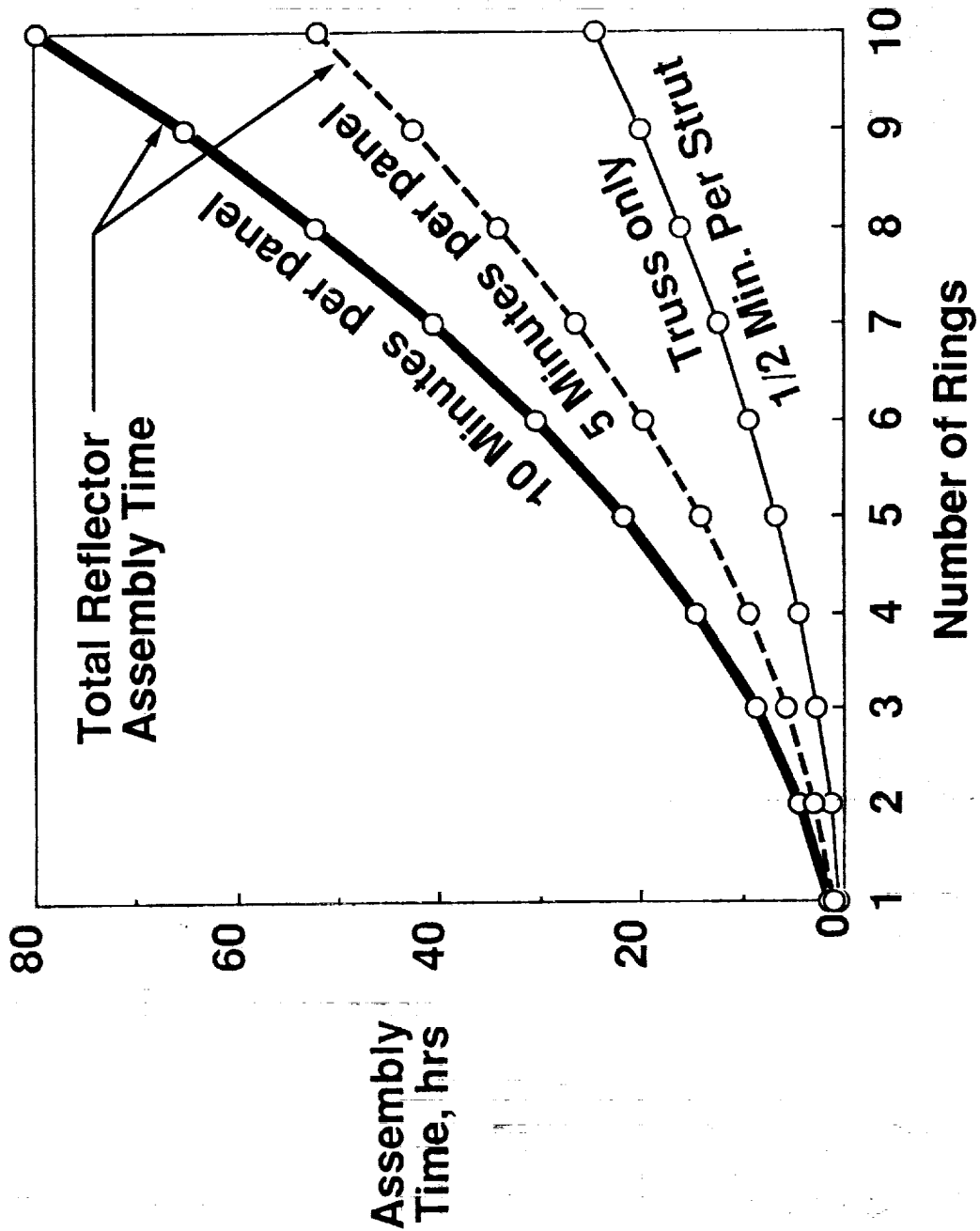| Ring	Panels	Struts
1	7	51
2	19	156
3	37	315
4	61	528
5	91	795
6	127	1116
7	169	1491
8	217	1920

21

Figure 7. Reflector On-orbit Assembly Time.

PANELS: (Graphite/Epoxy With Honeycomb Core)

Face Sheet Thickness - .02 in

Core Thickness   - .025 x Panel Width

Core Density   - 2 lb/ft^3

STRUTS: (Graphite/Epoxy)

Strut Modulus   - 30E6 lb/in^2

Minimum Thickness   - .04 in

Strut Density   - .06 lb/in^3

Strut Diameter   - (Determined From Pcr = 1000 lbs)

NODES: (Aluminum And Graphite/Epoxy)

Node Weight   - 6.6 lbs (50% Panel Attachment Penalty)

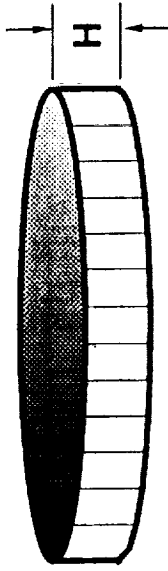Figure 8. Parameters Used in Reflector Preliminary Design Analysis.

SANDWICH PLATE

H
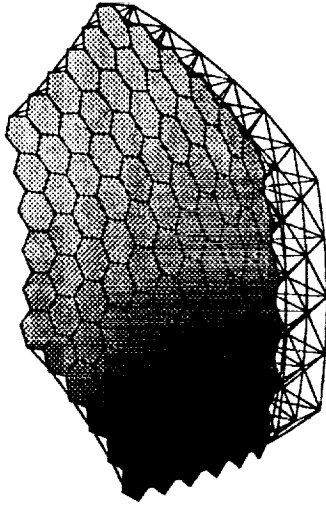
Plate Bending Stiffness

$$D = \frac{E_{face}\, t_{face}\, H^2}{2(1-\nu^2)} \quad \text{where} \quad H = \sqrt{\frac{2}{3}}\, L_{strut}$$

Truss Weight/Area

$$\left(\frac{W}{A}\right)_{truss} = \frac{(N_{struts})(W_{strut}) + (N_{nodes})(W_{node})}{A_{reflector\ surface}}$$

Plate Frequency

$$f = \frac{3.343}{D_{max}^2}\sqrt{\frac{gD}{\left(\frac{W}{A}\right)_{truss} + \left(\frac{W}{A}\right)_{panels}}}$$

REFLECTOR SYSTEM

Isotropic Face Equivalent Properties

$$E_{face} = \frac{E_{strut}}{3}$$

$$t_{face} = \frac{3\, A_{strut}}{\frac{\sqrt{3}}{2}\, L_{strut}}$$

$$\nu = 1/3$$

Figure 9. Reflector Preliminary Design Analysis.

24

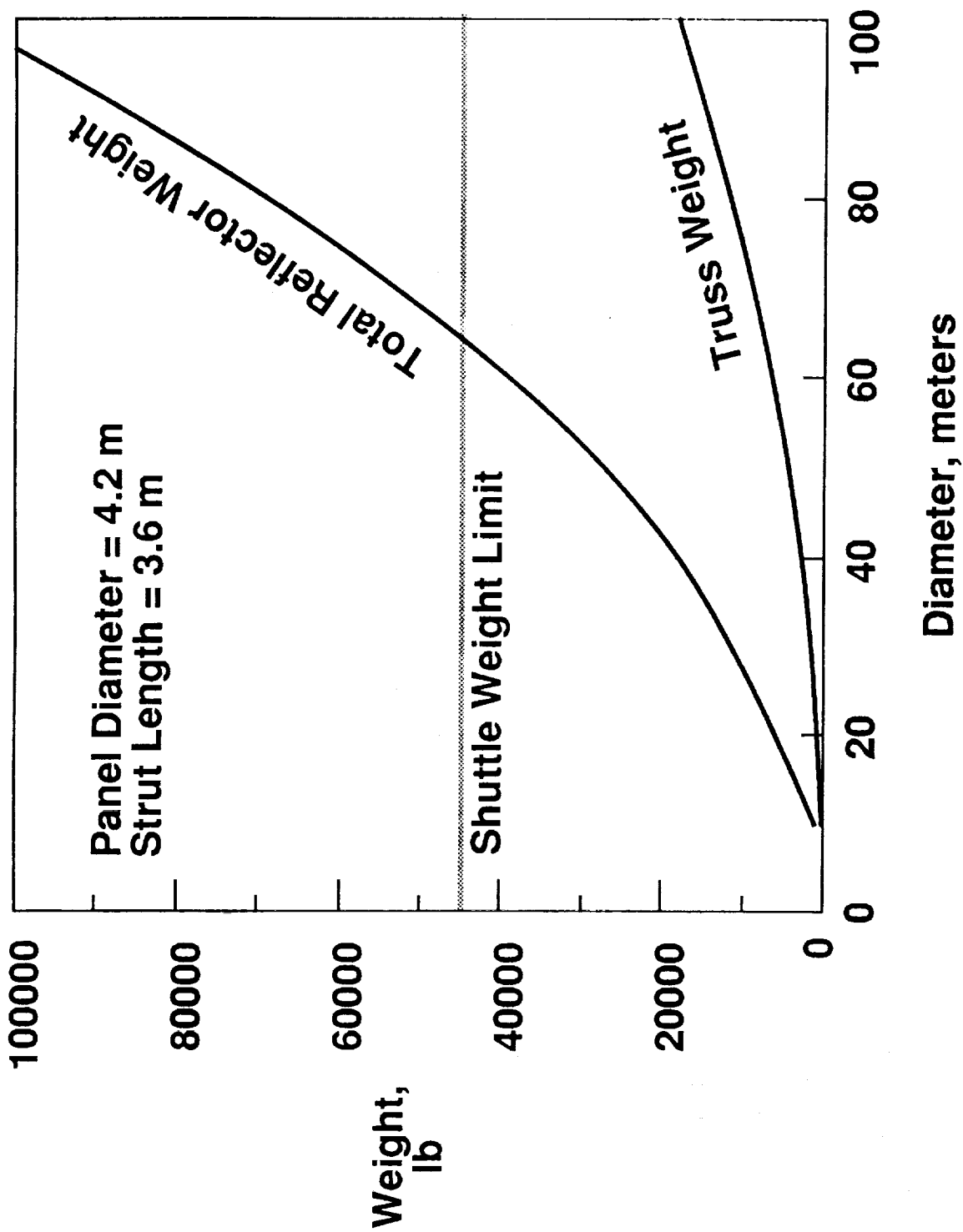Figure 10. Reflector Weight as a Function of Diameter for Fixed Truss Parameters.

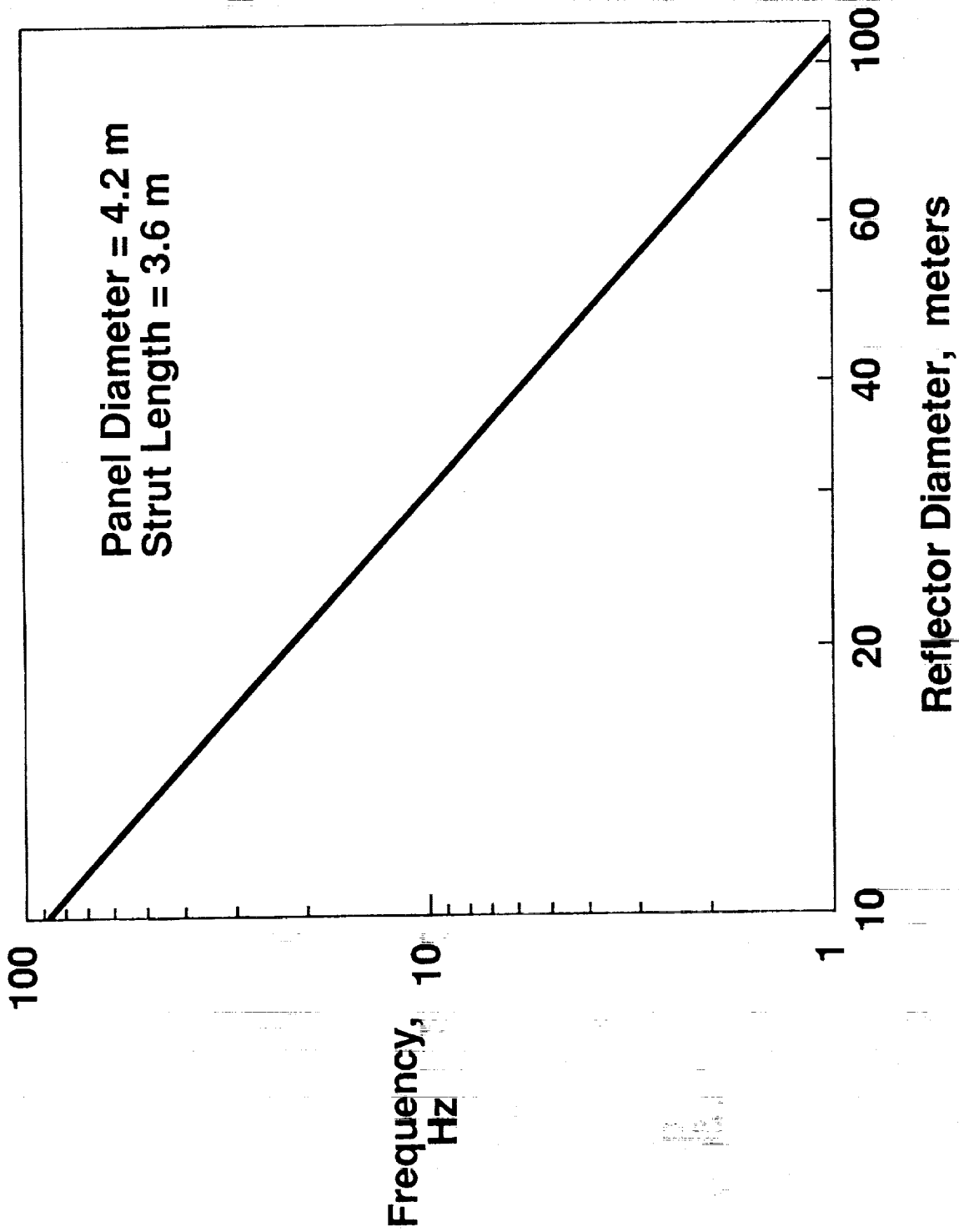Figure 11. Reflector Frequency as a Function of Diameter for Fixed Truss Parameters.
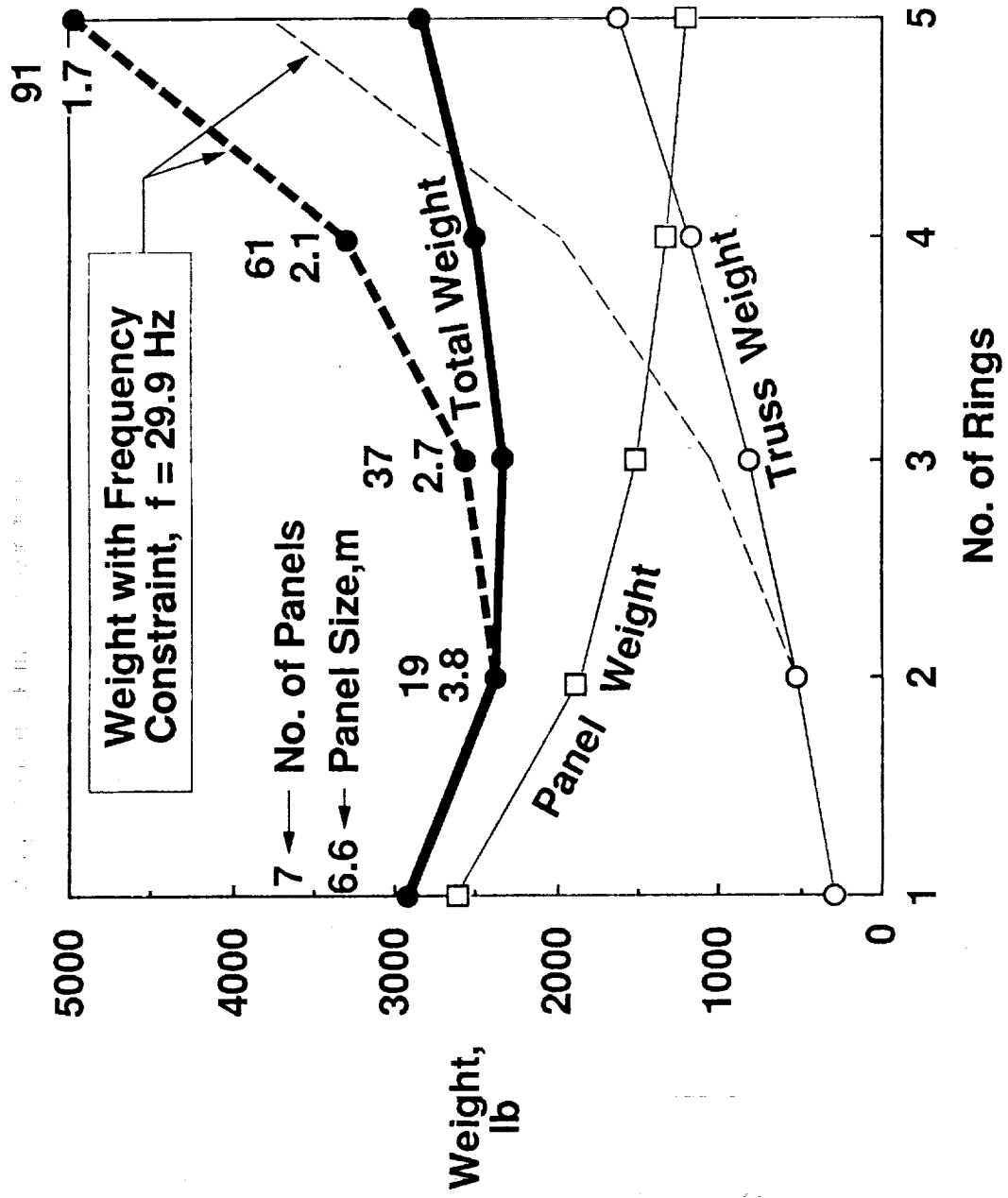
Figure 12. Weight of 15 Meter-Diameter Reflector With and Without Frequency Constraint.
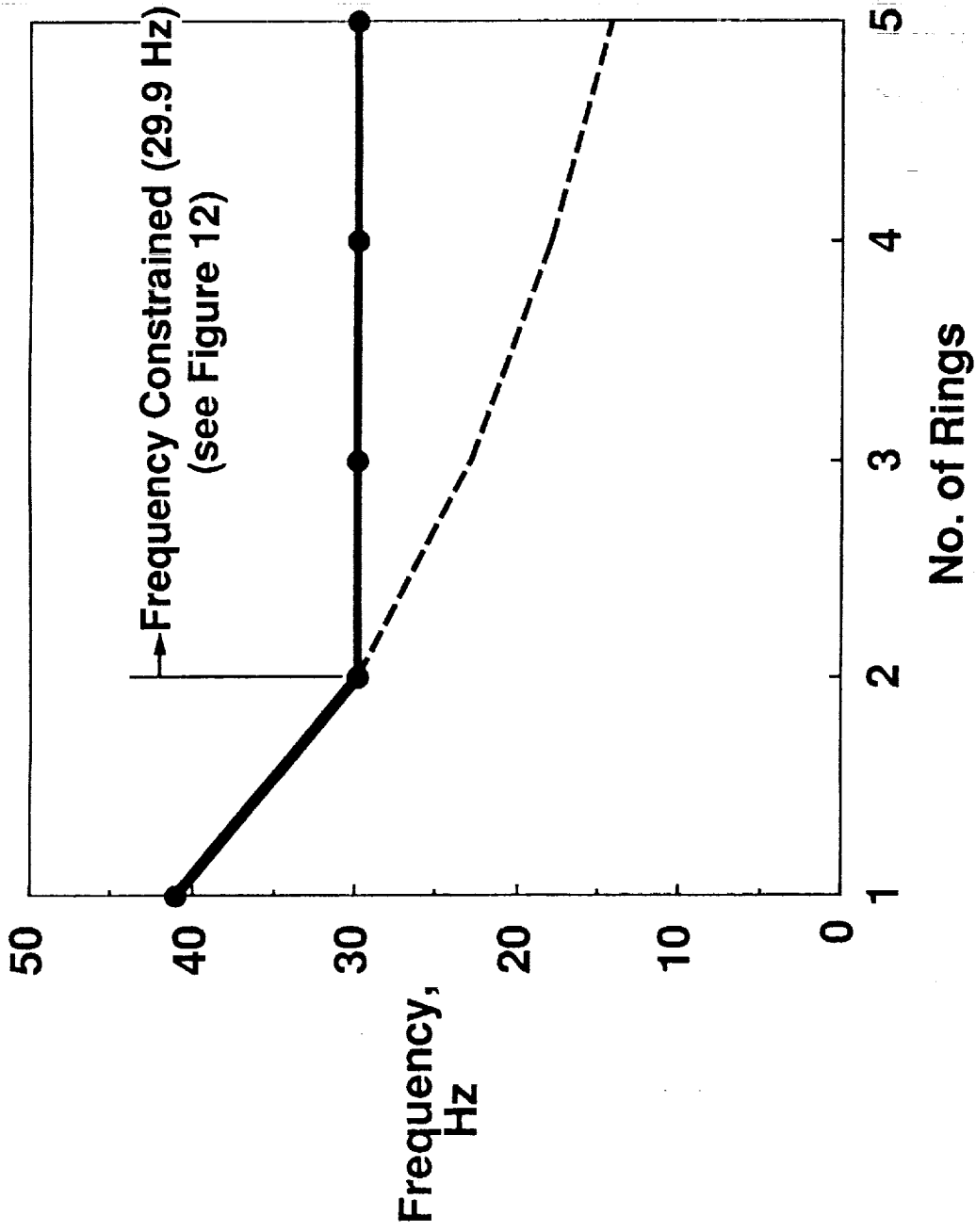
Figure 13. Lowest Natural Frequency of 15 Meter-Diameter Reflector.
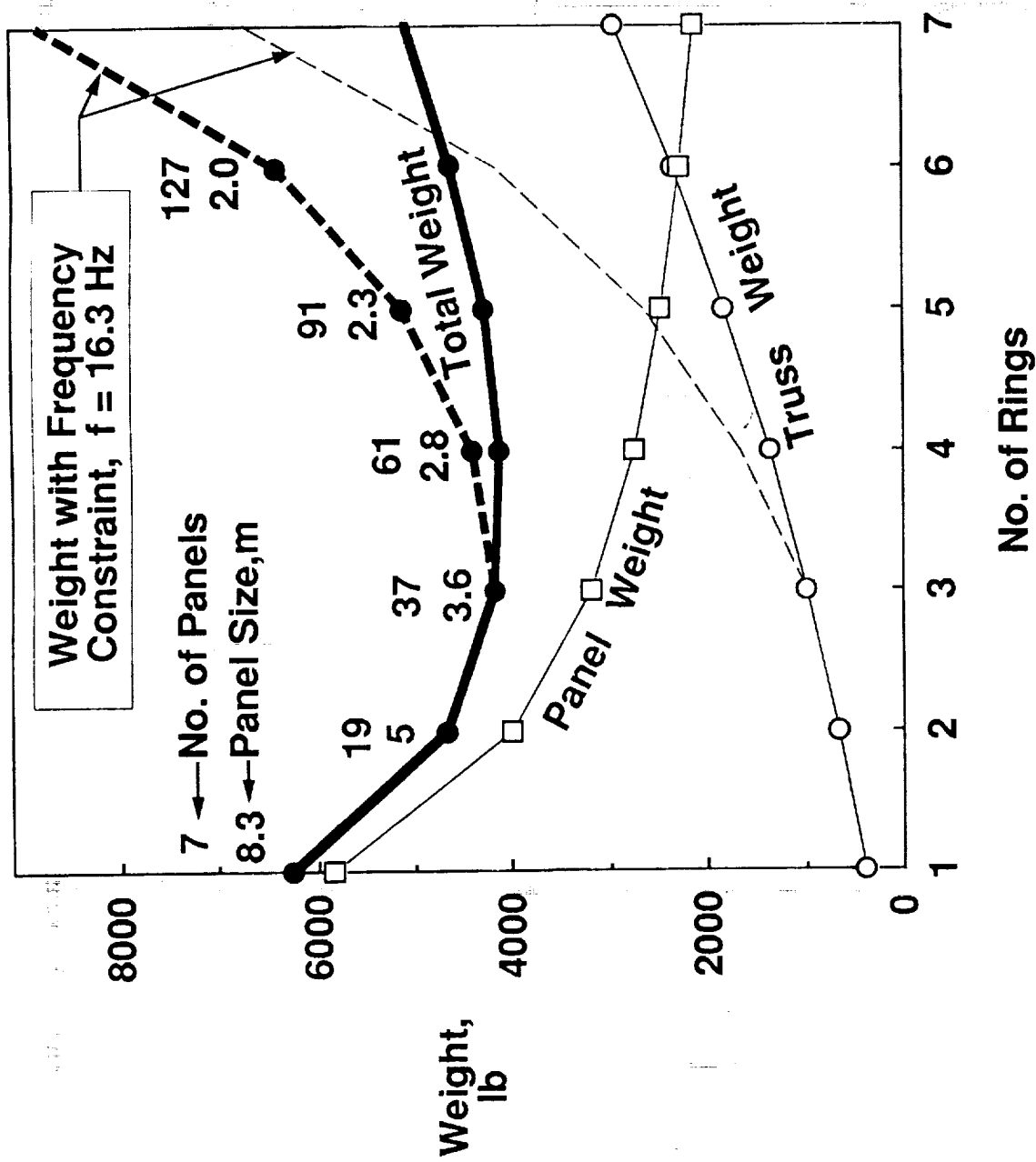
Figure14. Weight of a 20 Meter-Diameter Reflector With and Without a Frequency Constraint.
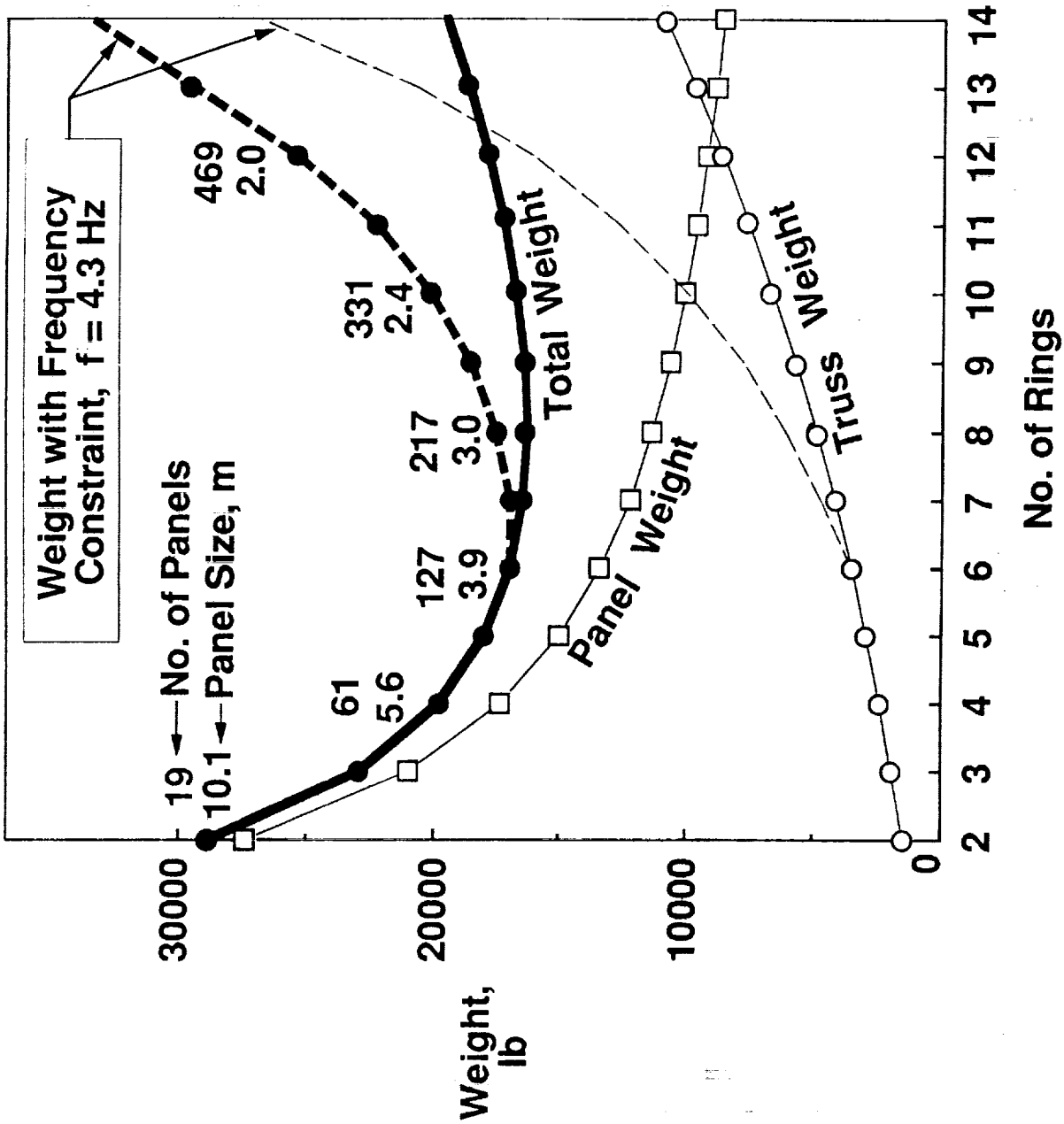
Figure 15. Weight of 40 Meter-Diameter Reflector With and Without a Frequency Constraint.

	3.6 m panels (3 Ring)	2.3 m panels (5 Ring)	1.5 m panels (8 Ring)
Weight	4200 lb	4300 lb	5600 lb
Frequency (Unconstrained)	16 Hz	11 Hz	7 Hz
$\dfrac{\text{Volume}}{\text{Volume Cargo Bay}}$	.15	.11	.09
Number of Panels	37	91	217
Number of Struts	315	795	1920
Construction Time	9 hrs	22 hrs	52 hrs

Figure 16. Table of Preliminary Design Results for a 20 Meter-Diameter Reflector.
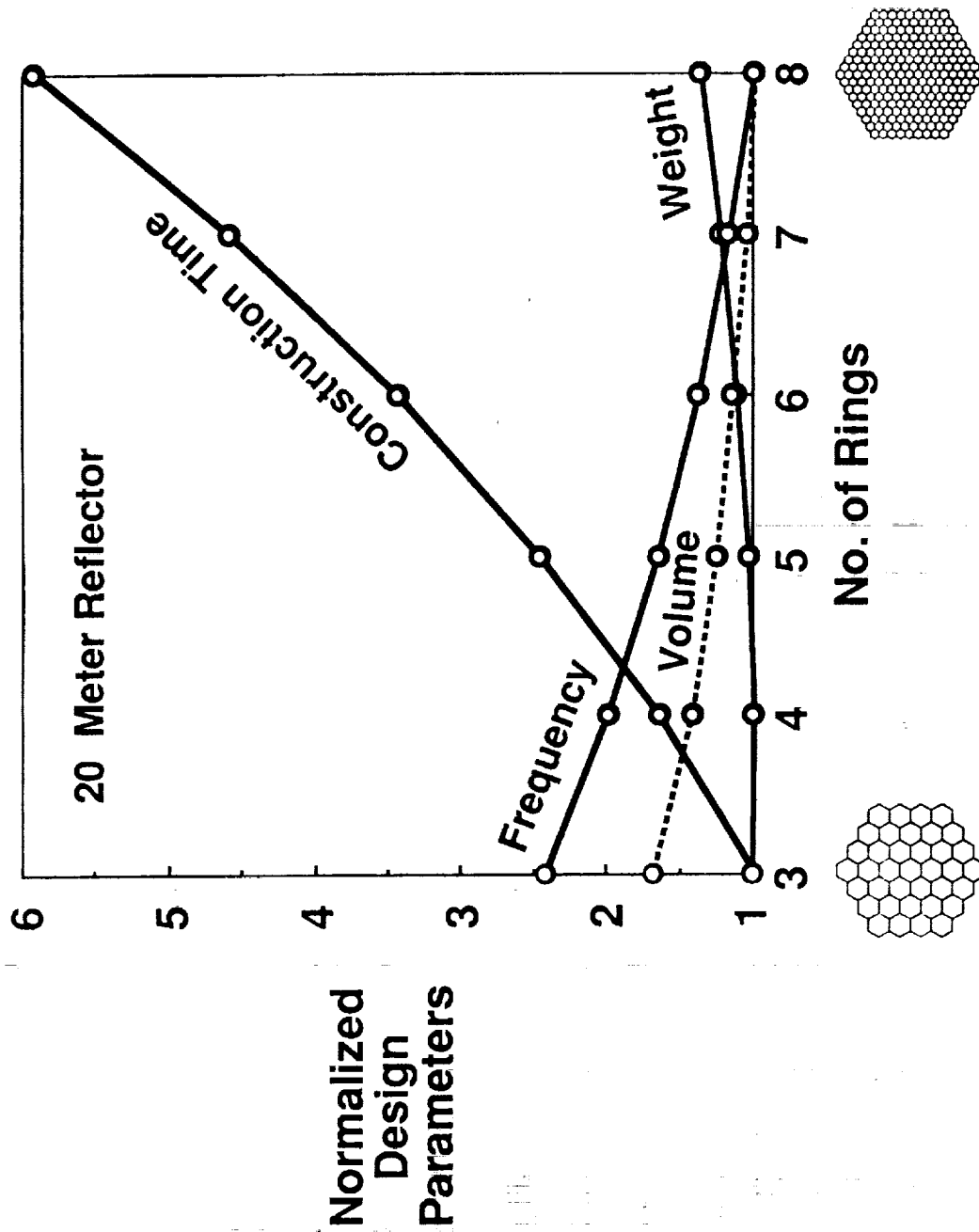
Figure 17. - Normalized Design Parameters For A 20 Meter Diameter Reflector

Figure 18. Panel Modules for Constructing a Segmented Precision Reflector Surface.

Panel Support Points
(same for both single panel and modules)

Truss Struts
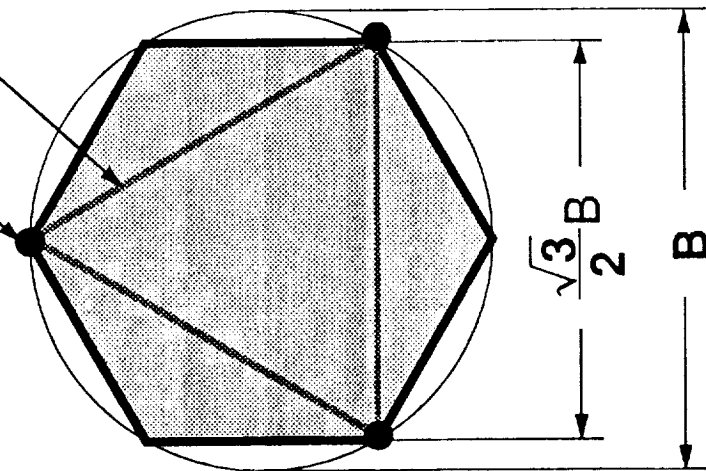
Single Hexagon Panel

$\frac{\sqrt{3}}{2}B$

B
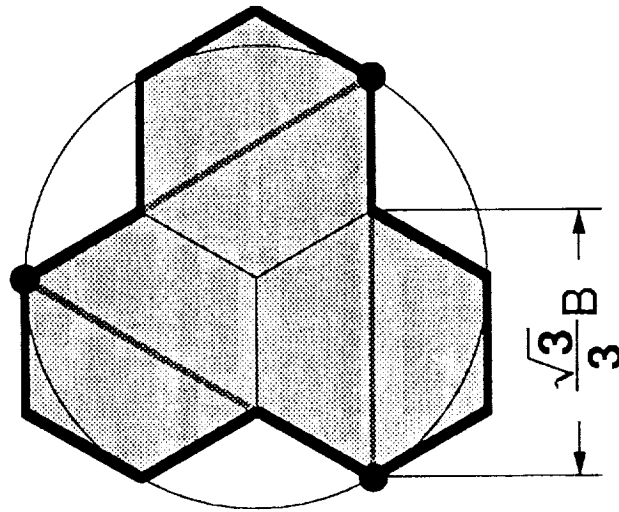
3 Hexagon Panel Module

$\frac{\sqrt{3}}{3}B$

$$\text{Area} = \frac{3\sqrt{3}}{3}B^2$$
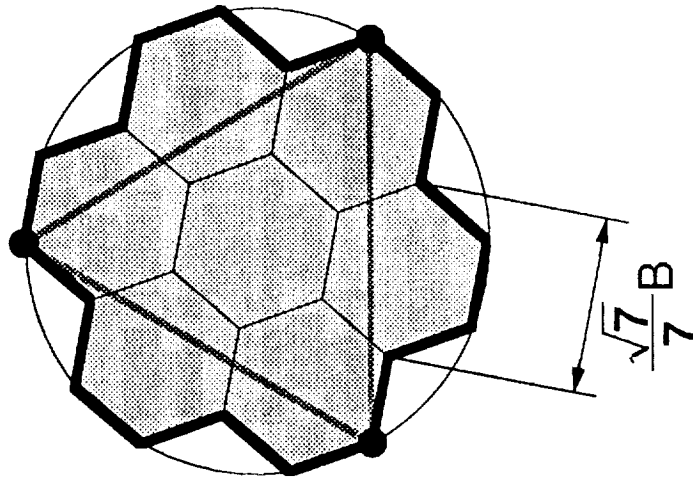
7 Hexagon Panel Module

$\frac{\sqrt{7}}{7}B$

33

Single Panels

3 Panel Modules

7 Panel Modules
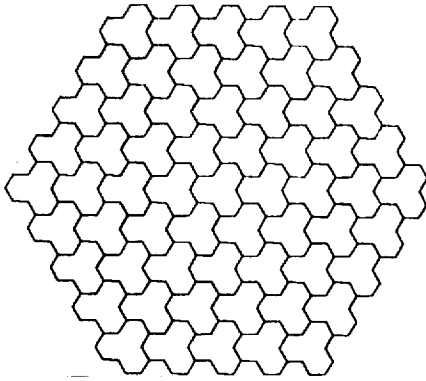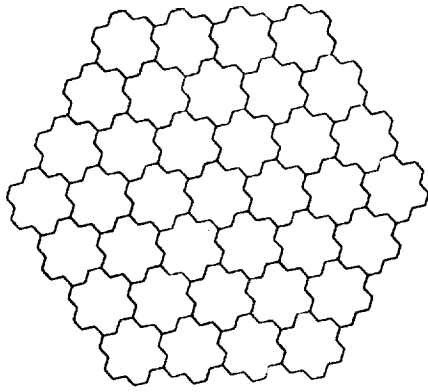
Single Panels	3 Panel Modules	7 Panel Modules
8 Rings	4 Rings	3 Rings
217 Panels	61 Modules	37 Modules
1920 Struts	528 Struts	315 Struts
2137 Total Parts	589 Total Parts	352 Total Parts

Assembly Time		
52 hours	15 hours	9 hours

Figure 19. Comparison of Same Diameter Reflector Constructed from Single Panels And 3 Or 7 Panel Modules.

Appendix A: Reflector Part Count and Geometry

In this Appendix, tables are presented which give the tetrahedral truss and corresponding segmented reflector component part count and geometry. The equations presented here were developed in reference 14. They have been reproduced for the convenience of the reader. The nomenclature "Truss A" and "Truss B" refer to the reflectors shown in figure 5.

Figure A1 shows component counts for the panels, struts, and nodes which compose the two types of tetrahedral trusses shown in figure 5. Component counts are given for the truss surface (the portion of the truss which interfaces with the reflector panels), the truss core, and the truss bottom. Figure A2 shows panel and reflector geometrical relations for the two types of trusses.

	Truss A	Truss B
Panels	$N_P = 3r^2$	$N_P = 3r^2 + 3r + 1$
Surface struts	$N_{SS} = 9r^2 + 3r$	$N_{SS} = 9r^2 + 12r + 3$
Core struts	$N_{CS} = 9r^2$	$N_{CS} = 9r^2 + 9r$
Bottom struts	$N_{BS} = 9r^2 - 6r$	$N_{BS} = 9r^2 + 3r - 3$
Struts (total)	$N_S = 27r^2 - 3r$	$N_S = 27r^2 + 24r$
Redundant struts	$N_{RS} = 9r^2 - 12r + 3$	$N_{RS} = 9r^2 - 3r - 3$
Surface nodes	$N_{SN} = 3r^2 + 3r + 1$	$N_{SN} = 3r^2 + 6r + 3$
Bottom nodes	$N_{BN} = 3r^2$	$N_{BN} = 3n^2 + 3n$
Nodes (total)	$N_N = 6r^2 + 3r + 1$	$N_N = 6r^2 + 9r + 3$
Truss struts + nodes	$N_T = 33r^2 + 1$	$N_T = 33r^2 + 33r + 3$

r is the number of rings

Figure A1. Component Count Expressions for the Two Types of Trusses Shown in Figure 5.

Single panel

$$D_{eff} = b\sqrt{\frac{3\sqrt{3}}{2\pi}}$$

$$A_p = \frac{3\sqrt{3}}{8}b^2$$
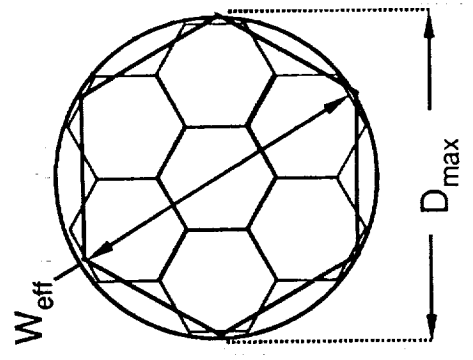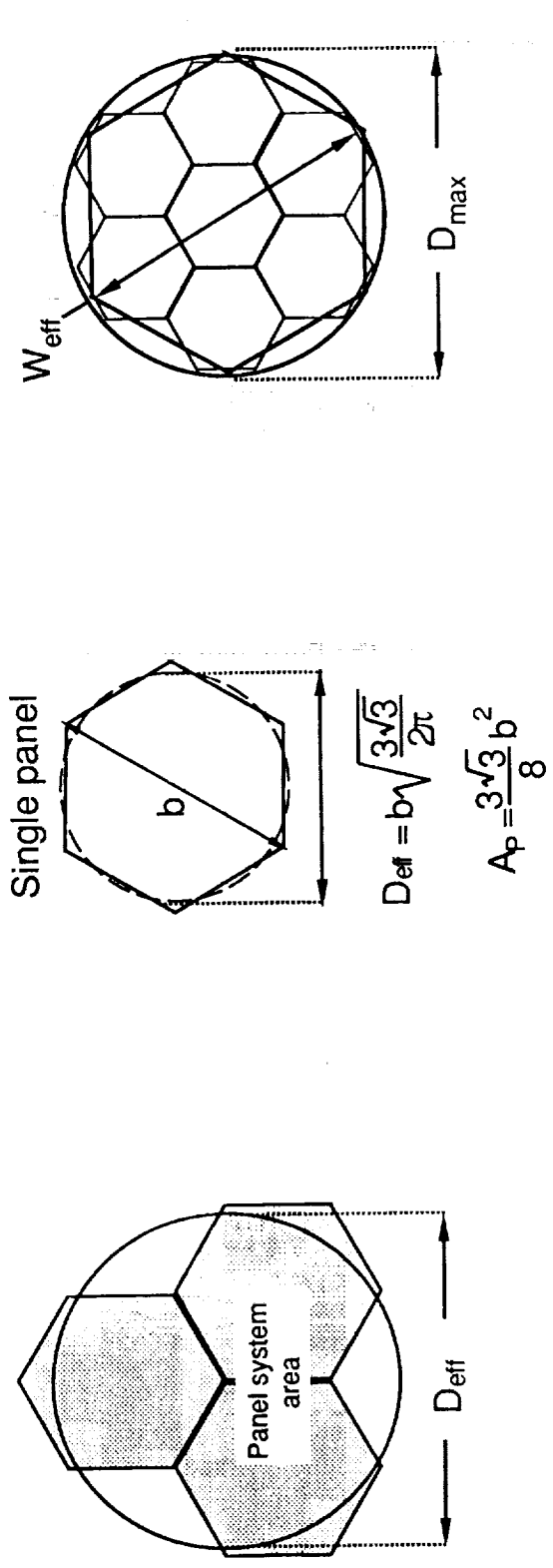
	Truss A	Truss B
Reflector Diameter	$D_{max} = b\sqrt{3r^2 + 1}$	$D_{max} = b\sqrt{3r^2 + 3r + 1}$
Width of equivalent hexagon	$W_{eff} = br\sqrt{3}$	$W_{eff} = b\sqrt{3r^2 + 3r + 1}$
Diameter of equivalent circle	$D_{eff} = 3br\sqrt{\frac{\sqrt{3}}{2\pi}}$	$D_{eff} = b\sqrt{(\frac{3\sqrt{3}}{2\pi}b^2)(3r^2 + 3r + 1)}$
Reflector Surface Area	$A_{PS} = b^2 r^2 \frac{9\sqrt{3}}{8}$	$A_{PS} = (\frac{3\sqrt{3}}{8}b^2)(3r^2 + 3r + 1)$

r is the number of rings

Figure A2. Panel and Reflector Geometrical Relations for the Two Types of Trusses Shown in Figure 5.

Appendix B. 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 2 to 8. The reflector input parameters were as follows:

- Strut length	78.74 in	(2 m)
- Strut diameter	1.0 in	(2.54 cm)
- Strut wall thickness	0.06 in	(.0236 cm)
- Reflector mass per unit area	0.82 lb/ft^2	(4 kg/m^2)
- Node weight	4.4 lbs	(2.0 kg)
(no attachment penalty)		

The strut modulus and density was the same as presented in figure 8. 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 figure B1.

Again, the maximum diameter of the reflector panel surface was used in the preliminary design analysis to heuristically improve the overall comparison. The maximum diameter of the reflector is 10 percent greater than the effective circular area diameter for any number of rings as can be seen from the equations presented in reference 14. 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 percent. 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 which begin to influence the response.

The curve labeled stiff core in figure B1 represents a finite element analysis where the core members were intentionally made extremely stiff in order 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 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

38

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.
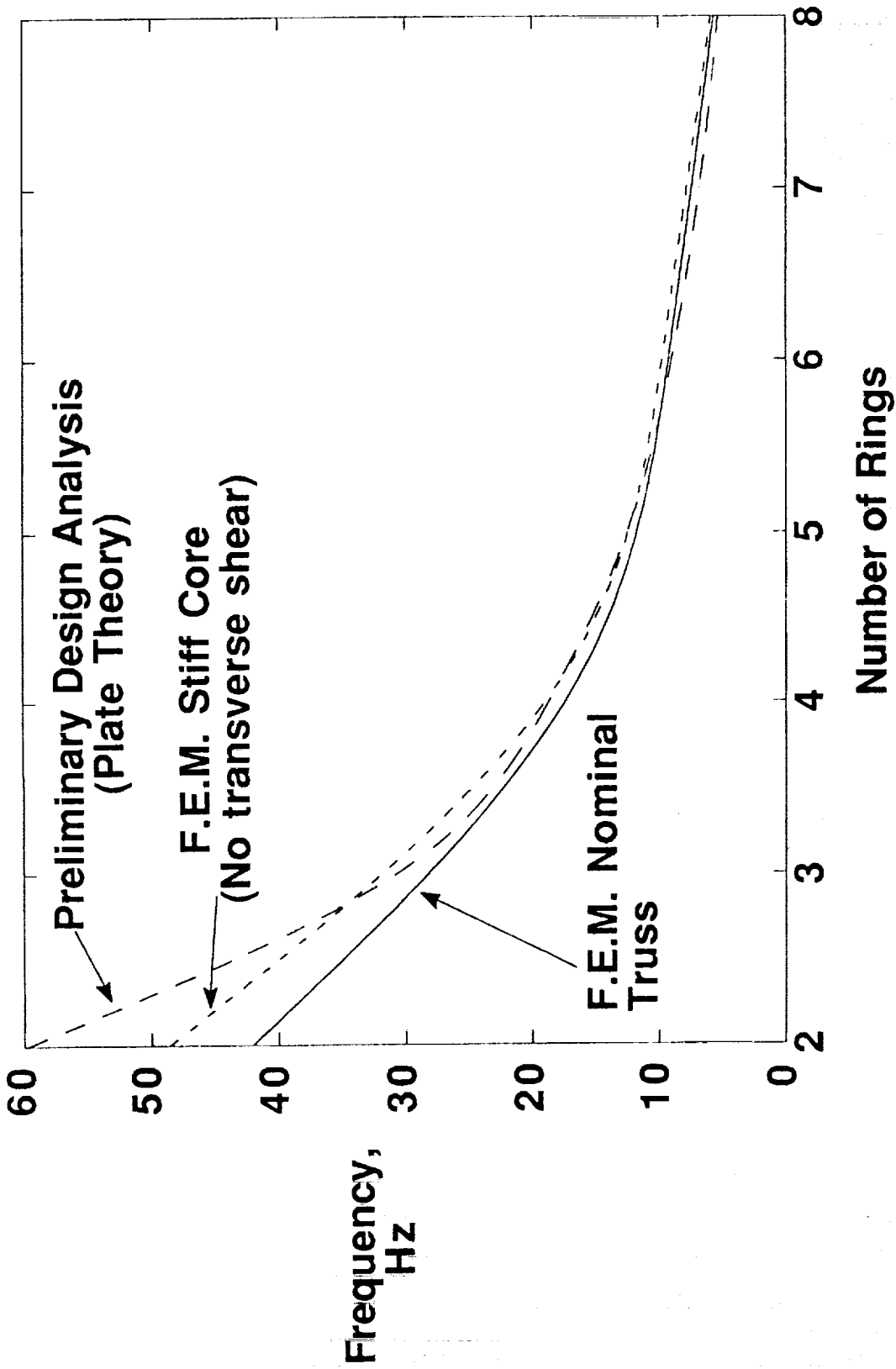
Figure B1. Comparison Between Preliminary Design Analysis and Finite Element Analysis.

40

Appendix C: 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 figure C1, and are included in the preliminary design program in Appendix C. 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 figure C2 for the 20 meter reflector of figure 14. For a seven ring truss doubling the truss depth reduces the total reflector weight by approximately 50 percent. 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 struts.

41
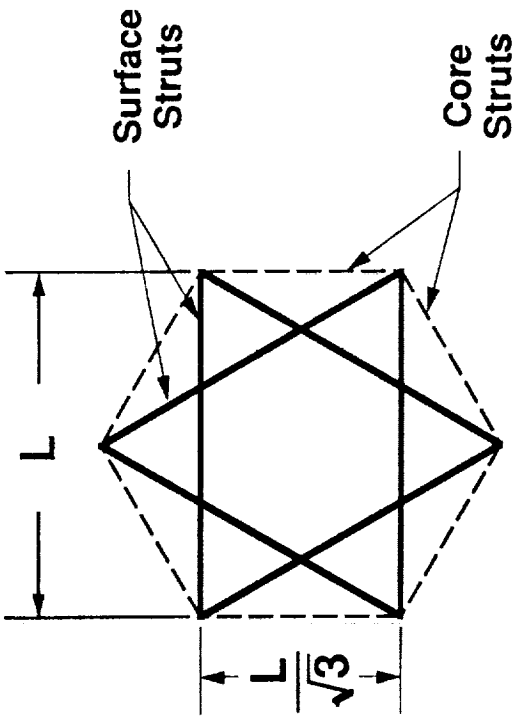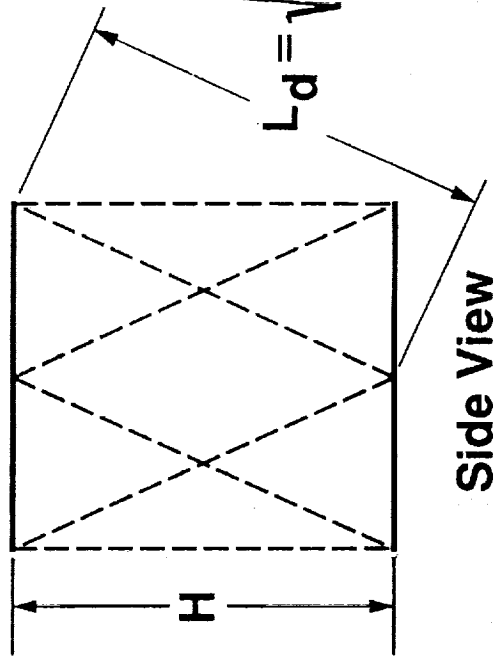
Assume $H = \beta \sqrt{\dfrac{2}{3}} L$ such that when

$\beta = 1$, H is the depth of a tetrahedral truss with equal length struts.

From reference 8 the plate bending stiffness is then found to be:

$$D = \frac{\sqrt{3}}{4} EAL \beta^2$$

And from Ref. 12 the total strut length is found to be:

$$Lstrut = (18n^2 + 15n)L + (9n^2 + 9n)L\sqrt{\frac{1+2\beta^2}{3}}$$

Perspective View

Top View

Surface Struts

Core Struts

L

$\dfrac{L}{\sqrt{3}}$

Side View

$$L_d = \sqrt{\frac{L^2}{3} + H^2} = L\sqrt{\frac{1+2\beta^2}{3}}$$

H
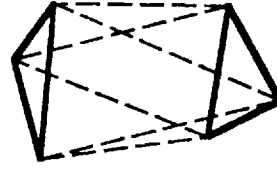
Figure C1. Characteristics of a Tetrahedral Truss with Variable Length Core Struts.
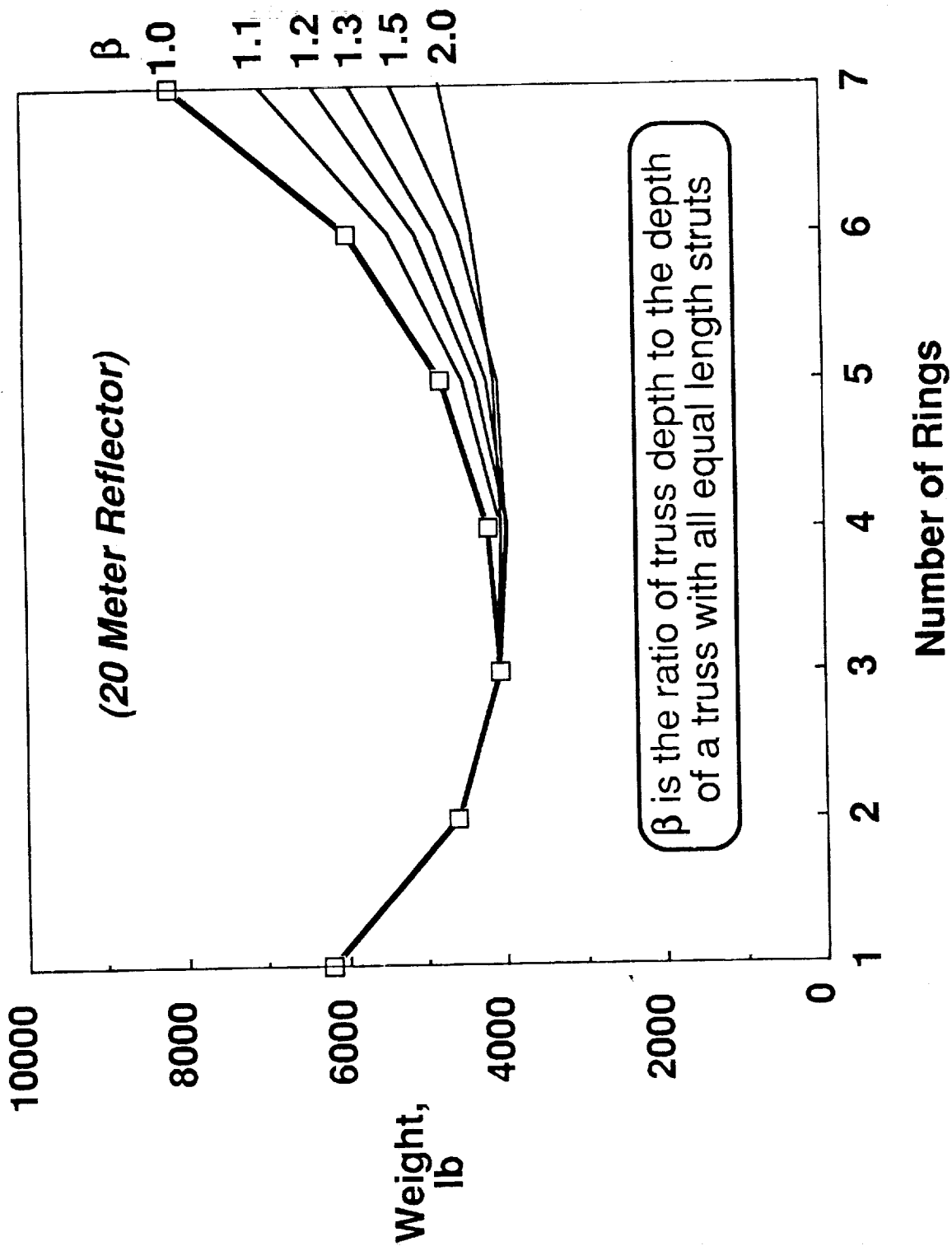
Figure C2. Effect of Truss Depth on Reflector Weight.

APPENDIX D.   Reflector Preliminary Design Program

SEGMENTED REFLECTOR WEIGHT, DISCRETE RINGS -
TRUSS STRUCTURE WITH HONEYCOMB PANELS

Programmed in Microsoft Quick Basic---
---Macintosh Version---

WEIGHT = Wpanels + Wstruts + Wnodes
Wpanels = Wcore + Wface sheets
Subscripts c - Core,   f - Face Sheets,   s - Struts

PRINT OUTPUT TITLES TO SCREEN
'--NOTE-- Print title statements must be outside Data loop so they are only printed once
WRITE " N","   DEF","Wcore"," Wfaces"," Wstruts","Wnodes","WT","   WovA","NSTRUTS","NNODES","
NPNLS"," f","   B","   LStr ","   DStr"," AStr","   TC","   TS"
PRINT

SET UP OUTPUT FILE
OPEN "SEG.REF.WT.DATA.16" FOR OUTPUT AS #1
T$=CHR$(9): ' Define ASCII Tab Character for Column Spacing in Files
PRINT OUTPUT TITLES TO FILE*
PRINT #1, "Rings";"   DEFF,m";T$;"   Wpanels";T$;"   WTRUSS";T$;" WTRUSSuncon";T$;"
WEIGHT,lb";T$;"WEIGHTuncon";T$;"   WovA,kg/m^2";T$;"   Nstrut";T$;"   Nnodes";T$;
PRINT #1,"   Npanels";T$;" frequency";T$;" frequncon";T$;"   B,m";T$;"   LStrut,m";T$;"
DStrut,in";T$;"   Astrut,in^2";T$;"   TCore,in";T$;"   TStrut,in";T$;" VOLOVCB"

INPUT QUANTITIES*   (To be modified for each program run)
'   * UNITS OR COMMENTS*
TF=.02   :   ' Face Sheet Thickness, Inches
RHOF=.06   :   ' Face Sheet Density ,lbs/in^3
RHOC=2*(1/1728) : '   Core Density, lbs/ft^3 converted to lbs/in^3
'TS---NOTE--- TS Must be inside N Data loop because of Frequency resizing constraint
ES=3E+07 :   ' Strut Modulus, psi
RHOS=.06   :   '   Strut Density, lbs/in^3
PCR=1000   :   '   Buckling Load Constant In Struts, lbs.
FD=29.9:   '   Design Frequency, Hz
DEFF=15*3.28084*12   : 'Effective Diameter of Hexagonal Reflector, meters converted to inches
WNODE=4.4*1.5   : '   Node Weight, lbs.---50% Weight penalty for panel attachment hardware---
Beta=1:   '   Normalized Truss Depth (Beta=1 For All Equal Length Struts)
PI=3.14159
SQG=386^.5 :   '   Square Root of gravitational constant g
DATA 1,2,3,4,5,6,7,8,9,10,11,12,13,14:   ' Input For Number of Rings, N
FOR I=1 TO 14
READ N
TS=.04   :   '   Strut Initial Wall Thickness, Inches

••••BEGIN CALCULATIONS•••

B=DEFF/(3*SQR(3)/(2*PI)*(3*N*N+3*N+1))^.5   :   ' Panel Maximum Diameter,inches
'B=165.35 ' Alternate input to fix panel size--must comment out DEFF input above
TC=B/(3.28084*12) : 'TC is in inches; ----- B/TC=constant--(TC=1 inch For 1 meter Panel)
LS=SQR(3)/2*B: '   Strut Length, inches
NSTRUTS=27*N*N+24*N
NNODES=6*N*N+9*N+3
NPANELS=3*N*N+3*N+1
'*SL is the Total Length of the Struts in truss, inches*
SL=(18*N*N+15*N)*LS+(9*N*N+9*N)*LS*((1+2*Beta*Beta)/3)^.5
AREA= 3*SQR(3)/8*B*B*(3*N*N+3*N+1): ' Total Area of Hexagonal Panels
DEFF=B*(3*SQR(3)/(2*PI)*(3*N*N+3*N+1))^.5
DMAX=B*SQR(3*N^2+3*N+1)
WCORE=AREA*RHOC*TC: ' Total Weight of Panel Core, pounds
WFACES=2*AREA*RHOF*TF: ' TotalWeight of Panel Faces, pounds
WNODES=NNODES*WNODE: ' Total Weight of Nodes, pounds
DS=2*(PCR*LS^2/(PI^3*TS*ES))^.3333333: ' Diameter of Strut, inches
WSTRUTS=PI*SL*RHOS*DS*TS: ', pounds
WEIGHT=WCORE+WFACES+WSTRUTS+WNODES: ' Total Weight of Reflector
Dt=1.732051*PI/4*ES*DS*TS*LS*Beta*Beta: 'Bending Stiffness of Truss
F=(4*.8357*SQG/(DMAX*DMAX))*SQR(Dt/(WEIGHT/AREA)):'Frequency of Reflector, Hz
AREAST=PI*DS*TS:'   Cross-SectionalArea of strut (approximate)

••••RESIZING LOOP IF DESIGN FREQUENCY CONSTRAINT ACTIVE•••   )

DIS=DS-2*TS: ' Inside Diameter of Strut
' Save WSTRUTS , WEIGHT, & F as unconstrained values for printing before resizing
WSTRUTSuncon=WSTRUTS
WEIGHTuncon=WEIGHT
Funcon=F
WHILE F-FD<0: '-------- Frequency constraint
DS=DS+.001:   '-------- Hold inside diameter constant and increase outside diameter
AREAST=PI*(DS*DS-DIS*DIS)/4: '-----Use exact area calculation for strut
WSTRUTS=SL*RHOS*AREAST
WEIGHT=WCORE+WFACES+WSTRUTS+WNODES
Dt=1.732051/4*ES*AREAST*LS*Beta*Beta
F=(4*.8357*SQG/(DMAX*DMAX))*(Dt/(WEIGHT/AREA))^.5
TS=(DS-DIS)/2
WEND
'*Calculate WovA the Mass per Unit Area of Reflector in Kg/m^2*
WovA=(WEIGHT/2.2)/(PI*(DEFF/(12*3.28084))^2/4): 'Kg/m^2
'*Calculate volume of panels and truss divided by Shuttle cargo bay volume*
VOLOVCB=(AREA*(TC+1.5)+NSTRUTS*DS^2*LS)/1.7E+07:'1.5" added to TC for packaging penalty

••••PRINT TO SCREEN•••

' *Note, DEFF,B, and LS Converted Back to Meters For Printing*
PRINT USING "##"; N,
PRINT USING "######"; DEFF/(12*3.28084),
PRINT USING "#######"; WCORE,WFACES,WSTRUTS,WNODES,WEIGHT,
PRINT USING "######.#";WovA,
PRINT USING "########";NSTRUTS,NNODES,NPANELS,
PRINT USING "#####.#"; F,B/(3.28084*12),LS/(3.28084*12),
PRINT USING "#####.##"; DS,AREAST,TC,
PRINT USING "#####.###"; TS

```
PRINT
'PRINT VOLOVCB:  '"Optional screen print statement; Eliminate leading colon to print"
```

```
'***PRINT  TO  FILE***
PRINT #1, USING "###### &"; DEFF/(12*3.28084),T$,
PRINT #1, USING "####### &";
WCORE+WFACES,T$,WSTRUTS+WNODES,T$,WSTRUTSuncon+WNODES,T$,WEIGHT,T$,WEIGHTuncon,T$,
PRINT #1, USING "######.# &";WovA,T$,
PRINT #1, USING "########&";NSTRUTS,T$,NNODES,T$,NPANELS,T$,
PRINT #1, USING "#####.# &"; F,T$,Funcon,T$,B/(3.28084*12),T$,LS/(3.28084*12),T$,
PRINT #1, USING "#####.## &"; DS,T$,AREAST,T$,TC,T$,
PRINT #1, USING "#####.###&"; TS,T$,VOLOVCB
NEXT I
CLOSE #1
END
```

Appendix E. Computer Program Printout

Rings	DEFF,m	Wpanels	WTRUSS	WTRUSSuncon	WEIGHTlbs	WEIGHTuncon	WovA,kg/m^2	Nstrut	Nnodes	Npanels
1	15	2634	293	293	2927	2927	7.50	51	18	7
2	15	1857	529	529	2386	2386	6.10	156	45	19
3	15	1517	1060	823	2577	2340	6.60	315	84	37
4	15	1327	1978	1188	3305	2515	8.50	528	135	61
5	15	1206	3763	1627	4969	2833	12.80	795	198	91
6	15	1121	8424	2143	9546	3264	24.60	1116	273	127

	frequency	frequncon	B,m	LStrut,m	DStrut,in	Astrut,in^2	TCore,in	TStrut,in	VOLOVCB
1	41.00	41.00	6.20	5.40	2.130	0.270	6.230	0.040	0.128
2	29.90	29.90	3.80	3.30	1.530	0.190	3.780	0.040	0.088
3	29.90	22.90	2.70	2.30	1.300	0.290	2.710	0.076	0.071
4	29.90	17.90	2.10	1.80	1.230	0.480	2.110	0.139	0.062
5	29.90	14.30	1.70	1.50	1.340	0.870	1.730	0.257	0.057
6	29.90	11.60	1.50	1.30	1.750	1.980	1.460		

Table E1. Computer Program Output for 15 Meter-Diameter Reflector of Figure 12.

Rings	DEFF,m	Wpanels	WTRUSS	WTRUSSuncon	WEIGHTlbs	WEIGHTuncon	WovA,kg/m^2	Nstrut	Nnodes	Npanels
1	20	5854	401	401	6254	6254	9.00	51	18	7
2	20	4012	672	672	4684	4684	6.80	156	45	19
3	20	3206	989	989	4195	4195	6.10	315	84	37
4	20	2756	1662	1371	4418	4127	6.40	528	135	61
5	20	2468	2647	1825	5115	4293	7.40	795	198	91
6	20	2269	4158	2353	6427	4621	9.30	1116	273	127
7	20	2122	6700	2956	8822	5078	12.80	1491	360	169
8	20	2010	11637	3636	13647	5646	19.70	1920	459	217

frequency	frequncon	B,m	LStrut,m	DStrut,in	Astrut,in^2	TCore,in	TStrut,in	VOLOVCB
26.70	26.70	8.30	7.20	2.580	0.320	8.310	0.040	0.287
20.40	20.40	5.00	4.40	1.850	0.230	5.050	0.040	0.193
16.30	16.30	3.60	3.10	1.480	0.190	3.620	0.040	0.152
16.40	13.40	2.80	2.40	1.310	0.250	2.820	0.065	0.129
16.30	11.10	2.30	2.00	1.220	0.360	2.310	0.102	0.114
16.30	9.30	2.00	1.70	1.220	0.530	1.950	0.159	0.105
16.30	7.90	1.70	1.50	1.320	0.840	1.690	0.251	0.100
16.30	6.70	1.50	1.30	1.560	1.470	1.490	0.407	0.100

Table E2. Computer Program Output for 20 Meter-Diameter Reflector of Figure 14.

Rings	DEFF,m	Wpanels	WTRUSS	WTRUSSuncon	WEIGHTlbs	WEIGHTuncon	WovA,kg/m^2	Nstrut	Nnodes	Npanels
1	40	42154	1013	1013	43167	43167	15.60	51	18	7
2	40	27424	1487	1487	28911	28911	10.50	156	45	19
3	40	20977	1934	1934	22910	22910	8.30	315	84	37
4	40	17371	2415	2415	19786	19786	7.20	528	135	61
5	40	15070	2951	2951	18021	18021	6.50	795	198	91
6	40	13474	3550	3550	17024	17024	6.20	1116	273	127
7	40	12302	4699	4217	17001	16520	6.10	1491	360	169
8	40	11406	6122	4954	17528	16361	6.30	1920	459	217
9	40	10698	7846	5764	18545	16462	6.70	2403	570	271
10	40	10125	9992	6647	20117	16772	7.30	2940	693	331
11	40	9651	12683	7605	22335	17256	8.10	3531	828	397
12	40	9253	16149	8638	25403	17891	9.20	4176	975	469
13	40	8914	20702	9746	29617	18661	10.70	4875	1134	547
14	40	8622	26905	10931	35528	19554	12.90	5628	1305	631

	frequency	frequncon	B,m	LStrut,m	DStrut,in	Astrut,in^2	TCore,in	TStrut,in	VOLOVCB
1	9.10	9.10	16.60	14.40	4.100	0.520	16.620	0.040	2.105
2	7.30	7.30	10.10	8.70	2.940	0.370	10.090	0.040	1.355
3	6.20	6.20	7.20	6.30	2.360	0.300	7.230	0.040	1.026
4	5.40	5.40	5.60	4.90	1.990	0.250	5.630	0.040	0.841
5	4.80	4.80	4.60	4.00	1.750	0.220	4.610	0.040	0.723
6	4.30	4.30	3.90	3.40	1.560	0.200	3.900	0.040	0.640
7	4.30	3.90	3.40	2.90	1.440	0.230	3.380	0.052	0.581
8	4.30	3.50	3.00	2.60	1.360	0.260	2.990	0.065	0.535
9	4.30	3.20	2.70	2.30	1.300	0.310	2.670	0.082	0.500
10	4.30	2.90	2.40	2.10	1.260	0.370	2.420	0.103	0.471
11	4.30	2.70	2.20	1.90	1.250	0.450	2.210	0.129	0.449
12	4.30	2.40	2.00	1.80	1.260	0.560	2.030	0.163	0.431
13	4.30	2.20	1.90	1.60	1.290	0.700	1.880	0.207	0.418
14	4.30	2.10	1.80	1.50	1.360	0.910	1.750	0.263	0.409

Table E3. Computer Program Output for 40 Meter-Diameter Reflector of Figure 15.

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16. Abstract

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 meters to 100 meters 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. A family of modular panels is introduced which can significantly reduce the number of reflector parts and the on-orbit assembly time.

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