Vukobratovich, D. “Lightweight Mirror Design”
Optomechanical Engineering Handbook
Ed. Anees Ahmad
Boca Raton: CRC Press LLC, 1999

Lightweight Mirror Design

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5.1 Introduction

Design of lightweight mirrors is a complex problem, involving optimization of both mirror and mount. The expense and complexity of lightweight mirrors require careful consideration of design requirements. Scaling laws provide rapid estimates of mirror weight during preliminary design. A significant issue is the self-weight deflection of lightweight mirrors. The simplest type of lightweight mirror is the contoured back mirror, of which there are three types: double concave, single arch, and double arch. The sandwich mirror offers the best stiffness-to-weight of any lightweight mirror, but is complex to design and fabricate. Open back mirrors are low in stiffness, but are relatively easy to fabricate. Mounting must be considered as part of the mirror design problem.

Lightweight mirrors are used in optical systems for a variety of reasons. Some advantages of lightweight mirrors include shorter thermal equilibrium times, reduced weight, and lower system cost. Reduced self-weight deflection, and higher fundamental frequency are additional reasons for the use of lightweight mirrors. Lightweight mirrors are often defined as mirrors that are lighter in weight than comparable-size conventional mirrors. This is often a difficult definition to apply, since there is considerable variation in the weight of "conventional" mirrors. One traditional rule of thumb first suggested by Ritchey is that "conventional" mirrors are right circular cylinders, with a diameter-to-thickness ratio of 6:1. In addition, this rule of thumb assumes that the mirror material is solid optical glass. This rule of thumb is easy to calculate and therefore is quite popular.

A more controversial definition based on structural efficiency is suggested by Schwesinger. Schwesinger suggests that a mirror is a "lightweight" if it has greater stiffness than a solid right circular cylinder mirror of the same weight.[1] If there is no improvement in stiffness, then the "lightweight" mirror does not have any advantage over the same weight solid mirror. This definition of a lightweight mirror requires considerable insight into the elastic behavior of the mirror, and is not as popular as the first rule of thumb suggested above.

5.2 Estimating Mirror Weight

It is often desirable to estimate the weight of lightweight mirrors well in advance of detailed design. Scaling laws are used to estimate mirror weight based on mirror diameter. Caution is indicated in the use of scaling laws. Scaling laws are based on statistical analysis of existing mirrors. Attempts to extend the scaling laws beyond the range of statistical data are hazardous. There are often design constraints such as dynamic loads in the mirror that may cause the final design to depart significantly from the weight predicted by the scaling laws. Within these limitations, scaling laws are a useful tool, especially for performing parametric analysis.

Surveys of the open literature on mirrors that are in existence indicate that mirror weight is dependent on mirror diameter raised to a power. There is some controversy concerning the exponent in this scaling law. Ordinary engineering analysis suggests that mirror weight should scale as the cube of the mirror diameter. Weight per unit area is sometimes used as an index of lightweight mirror efficiency. Use of weight per unit area implies a scaling law based on the square of mirror diameter.

A survey of lightweight mirrors by Valente indicates that mirror weight varies approximately with the cube of the mirror diameter.[2] This survey included 61 mirrors from 0.24 to 7.5 m in diameter using a variety of materials. Valente's table of lightweight mirrors is shown in Table 5.1.

For conventional solid mirrors, Valente gives the following relationship:

$$W = 246\,D^{2.92}$$

where W = mirror weight (kg)
D = mirror diameter (m)

This relationship is shown in Figure 5.1.

For all lightweight mirrors, Valente gives the following relationship:

$$W = 82\,D^{2.95}$$

This relationship is shown in Figure 5.2.

For specific mirror types, other scaling relationships are used. Contoured mirrors are mirrors with a back contoured to improve stiffness and reduce weight. The weight of contoured mirrors is shown in Figure 5.3 and is given by:

$$W = 106\,D^{2.71}$$

Structured mirrors are mirrors with a sandwich or open back geometry. The weight of structured mirrors is given by:

$$W = 68\,D^{2.90}$$

This relationship is shown in Figure 5.4.

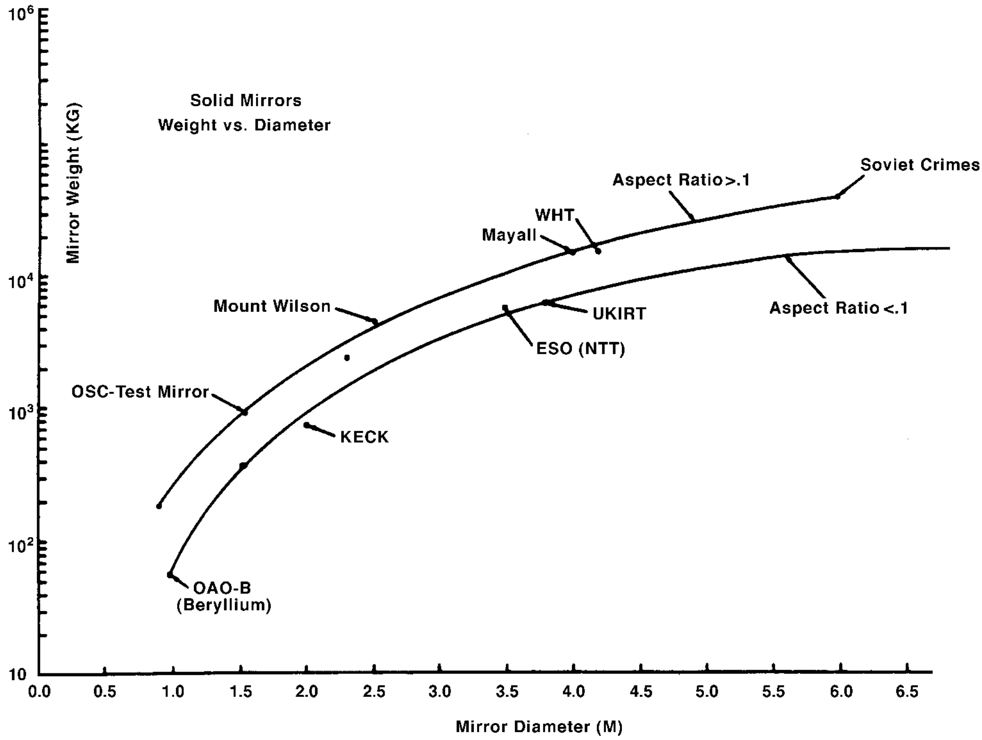
Beryllium mirrors are made in a variety of configurations. Beryllium mirrors are normally lighter than other types of mirrors regardless of the type of lightweight design. Weight of beryllium mirrors is given by:

$$W = 26\,D^{2.31}$$

TABLE 5.1 Lightweight Mirrors

Mirror	Year	Dia. (M)	Thick. (M)	Weight (KG)	Matl.	Config.	Misc.
IRAS	1983	0.60	0.09	12.6	Beryllium	Openback	annular ribs
Ball Relay	1989	0.60	0.06	9.07	Beryllium	Openback tri. cells	Hip process
P-E 40 inch	1989	1.02	0.05	18.14	Beryllium	Sandwich hex cells	2.265 in cells
P-E Scan	1975	.86 × .81	0.08	14.52	Beryllium	Openback sqr. cells	Flat λ/20
P-E second.	1975	1.65 × 1.02	0.08	53.5	Beryllium	Openback sqr. cells	f/0.67
Thematic Map.	1972	.406 × .508	0.04	1.86	Beryllium	Sandwich sqr. cells	Brazed
P-E 9.5 inch	1984	0.24	0.05	0.98	Beryllium	Sandwich hex cells	HIP process
P-E test	—	0.57	0.04	13.25	Beryllium	Double arch	—
P-E test	—	0.51	0.05	6.51	Beryllium	Double arch	1in circ cores
Hale	1950	5.0	0.60	13158	Pyrex	Openback	—
MMT	1979	1.8	0.30	567	F silica	Sandwich sqr. cells	6 mirrors
RCT	1965	1.3	0.15	200	Aluminum	Single arch	—
Spacelab UV	1979	0.92	0.15	100	Cervit	Double arch	—
Hubble	1990	2.48	0.30	773	ULE	Sandwich sqr. cells	—
Teal Ruby	1980	0.50	0.08	7.3	F silica	Sandwich hex cells	—
OAO-c	1972	0.82	0.13	48	F silica	Sandwich sqr. cells	—
U of Colorado	1979	0.41	0.05	9.98	Cervit	Double arch	f/2.5, 1/4λ
Steward Obs.	1985	1.8	0.36	703	Borosilicate	Sandwich hex cells	f/1.0
LDR test	1985	0.38	0.13	6.24	Borosilicate	Sandwich hex cells	sand-hexing
LDR test	1985	0.15	0.05	0.53	Vycor	Sandwich hex cells	air pressure
UTRC	1985	0.30	0.06	1.1	Glass TSC	Sandwich	Frit bonded
Ft. Apache	1986	3.5	0.46	1893	Borosilicate	Sandwich hex cells	—
NASA	1983	2.48	0.30	771	Glass	Sandwich sqr. cells	—
Los Alamos	1982	1.1 × 1.1	0.20	204	Tempax	Openback sqr. cells	—
SIRTF test	1983	0.51	0.089	16–25	quartz	Single arch	—
SIRTF test	1983	0.51	0.102	19–29	F silica	Double arch	f/4
Landsat-D	1979	0.42	0.07	9	ULE	Sandwich sqr. cells	—
GIRL	1985	0.50	0.074	25	Zerodar	Double taper	—
ISO	1985	0.64	0.075	20	F silica	Sandwich	machined
Hextek	1989	1.0	0.15	73	Borosilicate	Sandwich hex cells	f/0.5, meniscus
Hextek	1989	0.46	0.086	5.17	Borosilicate	Sandwich hex cells	—
Hextek	1989	0.38	0.076	7.71	Borosilicate	Sandwich hex cells	—
Shane 3 M.	1959	3.0	0.406	3856	Pyrex	Openback tri. cells	f/5
NASA 2.4 M.	1981	2.4	0.305	748	ULE	Sandwich sqr. cells	f/2.35
Milan 54 inch	1968	1.37	0.20	907.2	Aluminum	Single arch	—
Steward 68 cm.	—	0.68	0.10	25.4	Pyrex	Sandwich hex cells	—
Soviet test	1977	0.506	0.076	13.7	quartz	Openback hex cells	54 mm cells
Soviet test	1977	0.50	0.065	12.5	quartz	Sandwich hex cells	54 mm cells
Soviet test	1977	0.37	0.052	5.2	F silica	Sandwich hex cells	28 mm cells
Soviet test	1983	0.52	0.053	12.4	F silica	Sandwich	70 mm cells
Soviet test	1983	0.57	0.057	13.2	F silica	Sandwich	71 mm cells
Soviet test	1983	0.42	0.059	11.2	F silica	Sandwich	73 mm cells
Soviet test	1985	0.70	0.10	20	Al alloy	Openback	annular ribs
Schott test	—	1.143	0.159	204.12	F silica	Sandwich	—
OSC 16 in scope	1989	0.406	0.076	6.17	SXA	Single arch	—
OSC 12 in scope	1988	0.305	0.064	2.04	Aluminum	Double concave	Al foam core
OSC 12 in scope	1988	0.305	0.043	1.95	Aluminum	Double concave	Al foam core
AFCRL	1972	1.524	0.165	363	Cervit	Single arch	—

This relationship is shown in Figure 5.5.



	Aspect Ratio h/D <.1	Aspect Ratio h/D >.1
BEST FIT FUNCTION	HOERL FUNCTION $Y=(117.7).5457^{X}X^{4.738}$	PARABOLA FUNCTION $Y=2279-3721X+1721X^{2}$
GOODNESS OF FIT	.9951	.9974
POWER FUNCTION	$Y=98.78X^{2.867}$	$Y=2.46.1X^{2.917}$
GOODNESS OF FIT	.9452	.9957

FIGURE 5.1 Weight vs. diameter of solid mirrors. (From Valente, T.M. 1990. *Proc. SPIE 1340*, 47.)

5.3 Mirror Self-Weight Deflection

A parameter of considerable importance in the design of lightweight mirrors is the self-weight induced deflection. Self-weight deflection is important in terrestrial systems if the direction of the gravity vector changes. An example of an application of a system with a changing gravity vector is an astronomical telescope. As the telescope is pointed at objects at different zenith distances, the direction of the gravity vector acting on the optics changes.

Self-weight deflection in space optical systems is related to the change in optical figure upon gravity release in space and to the fundamental frequency of the mirror. Fundamental frequency is critical in determining the response of the mirror to random vibration during launch. Fundamental frequency and self-weight deflection are related by:[3]

$$f_n \approx \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$$

where f_n = fundamental frequency (Hz)

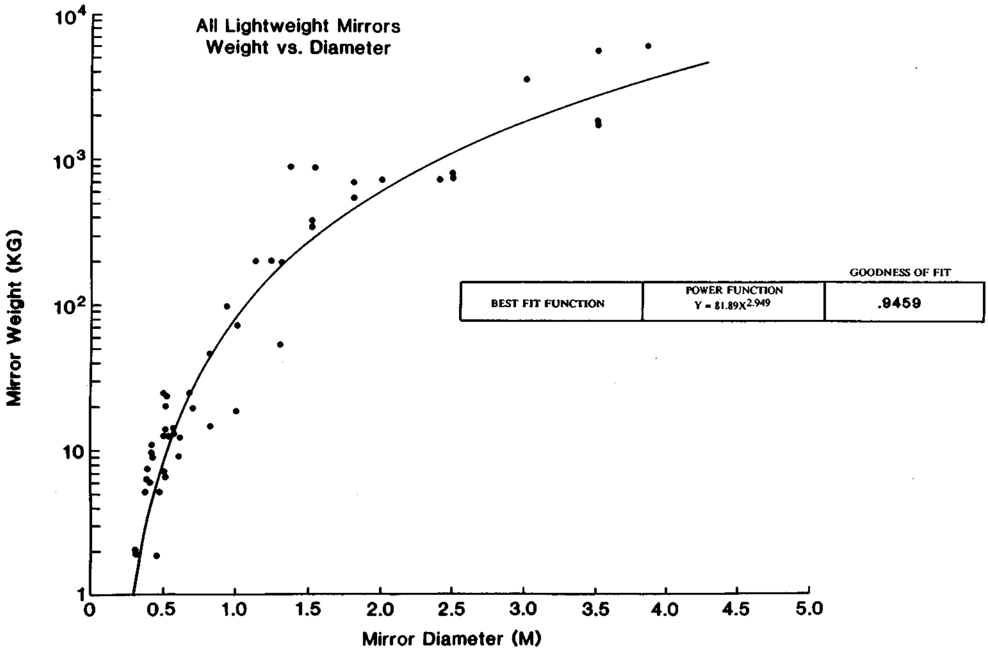


FIGURE 5.2 Weight vs. diameter of all lightweight mirrors. (From Valente, T.M. 1990. *Proc. SPIE 1340*, 47.)

g = acceleration due to Earth's gravity
δ = self-weight deflection of mirror

Self-weight deflection is normally calculated as normal to the mirror surface. For an axisymmetric mirror, the most common self-weight loading condition is the worst case of the gravity vector acting along the axis of symmetry. In this loading case the gravity vector is normal to the plane of the mirror and parallel to the optical axis. This loading condition is called the axial deflection case.

When the gravity vector acts normal to the axis of symmetry, the loading condition is called the radial deflection case. In this case, gravity is acting parallel to the plane of the mirror and normal to the optical axis. Although gravity acts parallel to the mirror surface, deflection normal to the mirror surface is induced by this loading condition.[4]

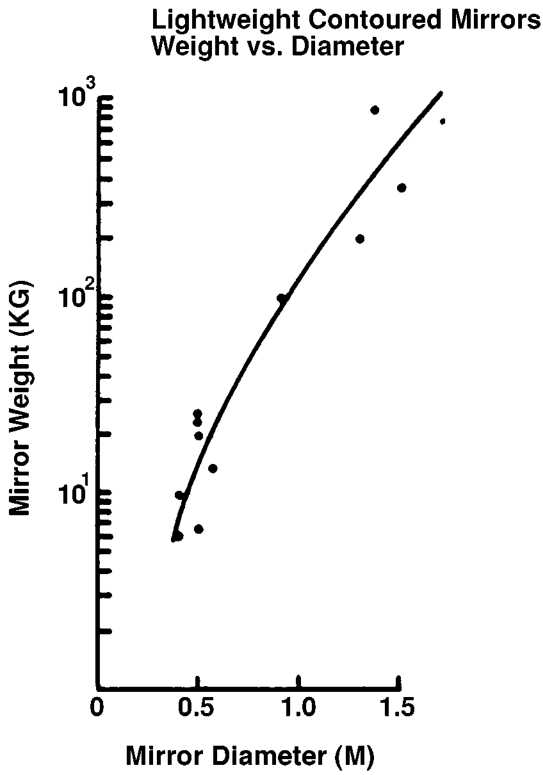
If the mirror is subjected to a loading condition in which the gravity vector is at an angle to the axis of symmetry, the resulting mirror surface deflections are given by:[5]

$$\delta_\theta \approx \sqrt{\left(\delta_A \cos\theta\right)^2 + \left(\delta_R \sin\theta\right)^2}$$

where δ_θ = mirror self weight deflection when gravity vector is at an angle to mirror axis
δ_A = mirror self-weight deflection in axial deflection case
δ_R = mirror self-weight deflection in radial deflection case
θ = angle between mirror axis and gravity vector

For most lightweight mirrors, the radial deflection is very small and is often ignored in preliminary estimates of performance. In some cases, calculation of the radial deflection is important. Such cases include very large mirrors, extremely lightweight systems, and systems used under high accelerations.

Self-weight deflection of mirrors in the axial loading condition is calculated using the classical plate theory. Caution is indicated in applying plate theory to lightweight mirrors. Classical plate



		GOODNESS OF FIT
BEST FIT FUNCTION	LINEAR HYPERBOLA $Y=-4409+2854X+1439/X$	.9967
POWER FUNCTION	$Y=106X^{2.712}$	.9728

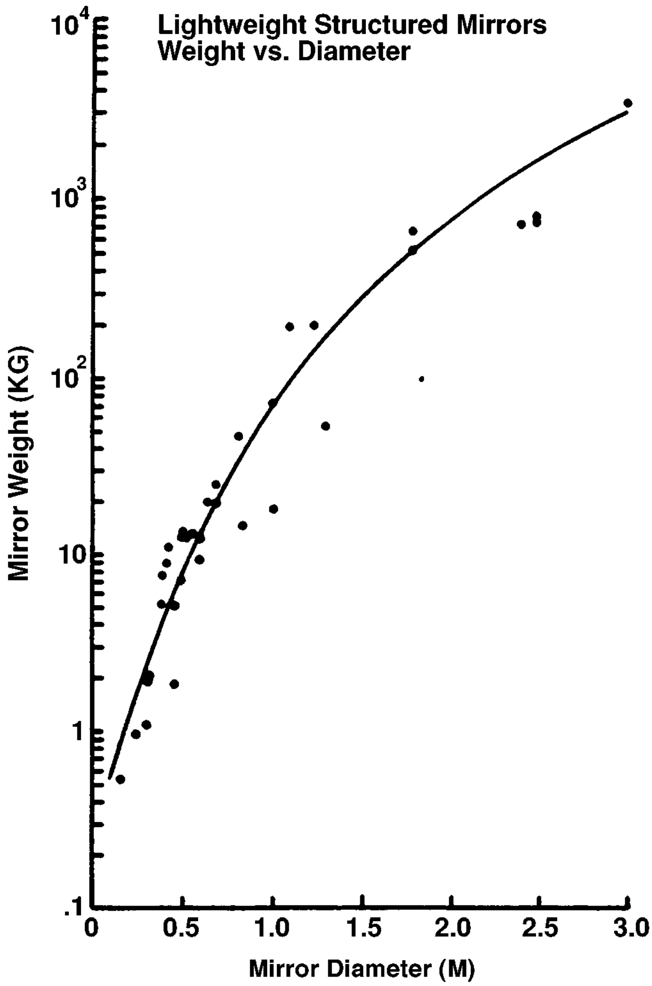
FIGURE 5.3 Weight vs. diameter of contoured mirrors. (From Valente, T.M. 1990. *Proc. SPIE 1340*, 47.)

theory assumes axisymmetric plane parallel plates, with a diameter-to-thickness ratio of 10:1 or more. Real lightweight mirrors may depart significantly from these assumptions. Shear deformations may play an important role in self-weight deflection and are ignored in classical plate theory. Shell action may become important if the mirror has significant surface curvature. Classical plate theory is an approximation and is used for preliminary design. More sophisticated design analysis using such techniques as finite element analysis is necessary for final design.

The general equation for axial deflection due to self-weight is[6]

$$\delta_A = C\frac{qr^4}{D}$$

where δ_A = axial deflection due to self-weight
C = support condition constant
q = weight per unit area of mirror
r = mirror radius
D = flexural rigidity of mirror

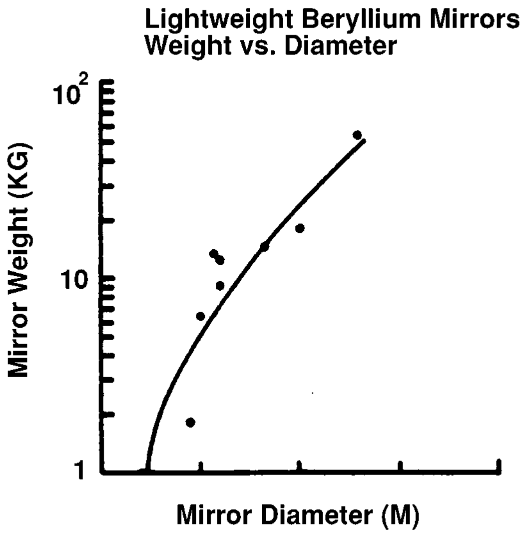


		GOODNESS OF FIT
BEST FIT FUNCTION	LOG NORMAL $Y=.177X10^{-3}\exp\left[(-8.739-\ln X)^{\frac{2}{5.996}}\right]$	.9512
POWER FUNCTION	$Y=106X^{2.712}$	.9499

FIGURE 5.4 Weight vs. diameter of structured mirrors. (From Valente, T.M. 1990. *Proc. SPIE 1340*, 47.)

Axial deflection due to self-weight is reduced by changing the support condition, reducing the weight per unit area of the mirror, or by increasing the flexural rigidity of the mirror. Weight per unit area of the mirror is determined by the mirror material density, the mirror structure, and mirror thickness. When material is removed from the cross section of the mirror, a lightweight structure is produced. Flexural rigidity is determined by the mirror material elastic modulus, mirror thickness, and mirror structure. Removing material from the cross section of the mirror to produce a lightweight structure influences flexural rigidity.

Another form of the general equation for axial deflection due to self-weight is



		BEST FIT FUNCTION
BEST FIT FUNCTION	PARABOLA FUNCTION Y=7.75-27.12X+45.41X^2	.8784
POWER FUNCTION	Y=26.19X$^{2.305}$	.8566

FIGURE 5.5 Weight vs. diameter of beryllium mirrors. (From Valente, T.M. 1990. *Proc. SPIE 1340*, 47.)

$$\delta_A = C\left(\frac{\rho}{E}\right)\frac{V_o}{I_o}r^4\left(1-\nu^2\right)$$

where δ_A = axial deflection due to self-weight
V_o = unit volume of mirror
ρ = mirror material density
E = mirror material elastic modulus
C = support condition constant
I_o = unit cross-sectional moment of inertia
r = mirror radius
ν = Poisson's ratio of mirror material

In the above axial deflection equation, the material parameter determining self-weight deflection is the ratio of mirror material density to elastic modulus. This material properties ratio, ρ/E, is the inverse specific stiffness of the material. This ratio does not change significantly for most common structural materials, and typically has a value of 386×10^{-9} m^{-1}. Significant exceptions to the rule that most materials have about the same specific stiffness are

Unusual Specific Stiffness Materials

Material	p/E × 10^{-9} m^{-1}
Beryllium	58.8
Silicon carbide	92.1
Metal matrix composite aluminum/silicon carbide	244
Graphite/epoxy composite	188

The ratio of unit volume to unit cross-sectional moment of inertia is the structural efficiency of the mirror cross section. This ratio, V_o/I_o, is a measure of the stiffness-to-weight independent of material properties. The structural efficiency is determined by the distribution of material in the cross section. High structural efficiency is achieved when the material in the cross section is distributed as far as possible from the neutral or bending axis. This condition is satisfied by the sandwich mirror. In the sandwich mirror, most of the mirror material is in the face plate and back plate of the mirror and is distant from the bending axis.

For a solid mirror the following conditions hold:

$$V_o = h$$

$$I_o = \frac{h^3}{12}$$

$$\frac{V_o}{I_o} = \frac{12}{h^2}$$

where V_o = unit volume of mirror
I_o = unit cross-sectional moment of inertia
h = mirror thickness

The support condition constant varies with the geometry of mirror support. There are no units associated with the support condition constant. The magnitude of the support condition constant depends on the location and number of the support forces. For a given number of support forces, there is an optimum location geometry to produce minimum axial deflection in the mirror due to self-weight. Increasing the number of support forces usually reduces axial deflection due to self-weight if the supports do not overconstrain the mirror.

A common mirror geometry is a right circular cylinder. One type of axial support for the right circular cylinder-shaped mirror consists of three points on a common diameter. The point supports are equal spaced around the diameter, and the diameter is some fraction of the mirror diameter. There is an optimum location for the three point supports for the right circular cylinder mirror, and that optimum location is a support diameter that is 0.68 of the mirror diameter. When the mirror is supported by three points on the 0.68 diameter, axial deflection due to self-weight is at a minimum for any three-point support. If the mirror is supported on this optimum three-point support, the maximum deflection is given by:

$$\delta_A = 0.02859\left(\frac{\rho}{E}\right)\frac{V_o}{I_o} r^4\left(1-\nu^2\right)$$

If a right circular cylinder mirror is supported by three points equally spaced at the edge, the maximum deflection is given by:

$$\delta_A = 0.11303\left(\frac{\rho}{E}\right)\frac{V_o}{I_o} r^4\left(1-\nu^2\right)$$

If a right circular cylinder mirror is supported by six points equally spaced at 0.68 of the mirror diameter, the axial deflection due to self-weight is at a minimum for a six-point support. The maximum deflection for this optimum six-point support is given by:

$$\delta_A = 0.0048973\left(\frac{\rho}{E}\right)\frac{V_o}{I_o} r^4\left(1-\nu^2\right)$$

Occasionally rectangular mirrors are used in optical systems. The optimum three-point support for a rectangular mirror consists of one point located at the middle of one long edge, and the other two points located at the corners of the opposite long edge. For this optimum support of a rectangular mirror, axial deflection due to self-weight is given by:[7]

$$\delta_A = \frac{1}{\psi^2}\left(\frac{\rho}{E}\right)\frac{V_o}{I_o} a^4\left(1-\nu^2\right)$$

$$\psi^2 = \frac{7\left(\frac{a}{b}\right)}{\left[1+0.461\left(\frac{a}{b}\right)^{13}\right]^{\frac{1}{13}}}$$

where: δ_A = axial deflection of mirror due to self-weight
ρ = mirror material density
E = mirror material elastic modulus
V_o = unit volume of mirror
I_o = unit cross-sectional moment of inertia
a = length of mirror
b = width of mirror
ν = Poisson's ratio for mirror material

These equations are used for rapid estimation of mirror axial self-weight deflection. More sophisticated calculations are typically required for final design. A significant disadvantage of these equations is neglect of shear deformations. A rough correction for shear effects is possible using a method developed by Nelson.[8] The shear correction is

$$\delta_{total} = \delta_{bending}\left(1+\frac{N\pi h^2}{A}\right)$$

where δ_{total} = total axial deflection including shear effects
$\delta_{bending}$ = axial deflection due to bending
N = number of point supports
h = mirror thickness
A = mirror surface area

5.4 Contoured Back Mirrors

Contoured back mirrors are mirrors with a back contour shaped to reduce weight, and in some cases self-weight deflection. Three types of contoured back mirrors are used: symmetric, single arch, and double arch. These three types of mirrors are shown in Figure 5.6. Contoured back mirrors offer reduction in weight up to 25% in comparison with a right circular cylinder, 6:1 aspect ratio solid mirror. Contoured back mirrors are low in fabrication cost and are relatively easy to mount. A significant disadvantage of contoured back mirrors is the variation in mirror thickness. This variation in mirror thickness causes different portions of the mirror to reach thermal equilibrium at different times following a change in temperature. The resulting variation in temperature with mirror thickness induces optical surface distortion. Contoured back mirrors are more sensitive to optical surface distortion due to temperature changes than other types of mirrors.

Axial deflection of contoured back mirrors is difficult to calculate using simple closed-form equations due to the substantial variation in mirror thickness and shell action of the curved contours of the mirror. Cho gives a scaling relation for contoured back mirrors of different sizes.[9] This scaling relation is used for mirrors of similar contours to scale deflections as the mirror size changes. This scaling relation is

$$\delta = \left(\frac{\rho}{\rho_{ref}}\right)\left(\frac{E_{ref}}{E}\right)\left(\frac{A}{A_{ref}}\right)\delta_{ref}$$

where δ_{ref} = axial deflection due to self-weight of reference mirror
δ = axial deflection due to self-weight of new mirror
ρ_{ref} = mirror material density of reference mirror
ρ = mirror material density of new mirror
E_{ref} = elastic modulus of reference mirror material
E = elastic modulus of new mirror material
A_{ref} = cross-sectional area of reference mirror
A = cross-sectional area of new mirror

Symmetric mirror shapes are used to minimize axial deformation due to self-weight when the gravity vector is perpendicular to the mirror optical axis. Bi-metallic bending of plated metal mirrors is reduced through the use of symmetric shapes. The front and back of the symmetric mirror are given equal radii, but of opposite sign. A symmetric mirror is either double concave or double convex. Normally a symmetric mirror is supported either by a small ring near the center or by multiple points at the edge.

Self-weight axial deflection of the symmetric mirror is worse for equal weights than the single arch or double arch shape. Radial deflection of the symmetric mirror is much smaller than that of single arch or double arch mirror shapes. Therefore, the symmetric shape is often used when the operating position of the mirror is such that the gravity vector is perpendicular to the optical axis. The extremely small radial deflection of the symmetric mirror shape makes it attractive as a candidate space mirror. The very small radial deflection of the symmetric mirror minimizes residual optical surface error after gravity release.[10] This advantage is offset by the lower fundamental frequency of the symmetric mirror in comparison with other mirror shapes.

The symmetric shape is used to minimize bi-metallic bending effects in electroless nickel-plated metal mirrors.[11] The thermal coefficient of expansion of electroless nickel is either 12.5 or 15 × 10^{-6} m/m-K depending on whether the nickel is annealed after plating.[12] In comparison, the thermal coefficients of expansion of aluminum and beryllium are 23×10^{-6} and 11×10^{-6} m/m-K, respectively. Electroless nickel is normally plated onto the mirror surface to a thickness of 75 to 125 μm or more. The difference in thermal coefficient of expansion between substrate material and plating, and the relatively thick plating layer leads to bi-metallic bending effects when the temperature of the plated mirror is changed.

Plating both sides of a symmetric mirror shape with the same thickness of electroless nickel minimizes this bi-metallic bending, since equal and opposite bending forces are produced in the mirror. This technique of suppressing bi-metallic bending reduces bending deflection, but does

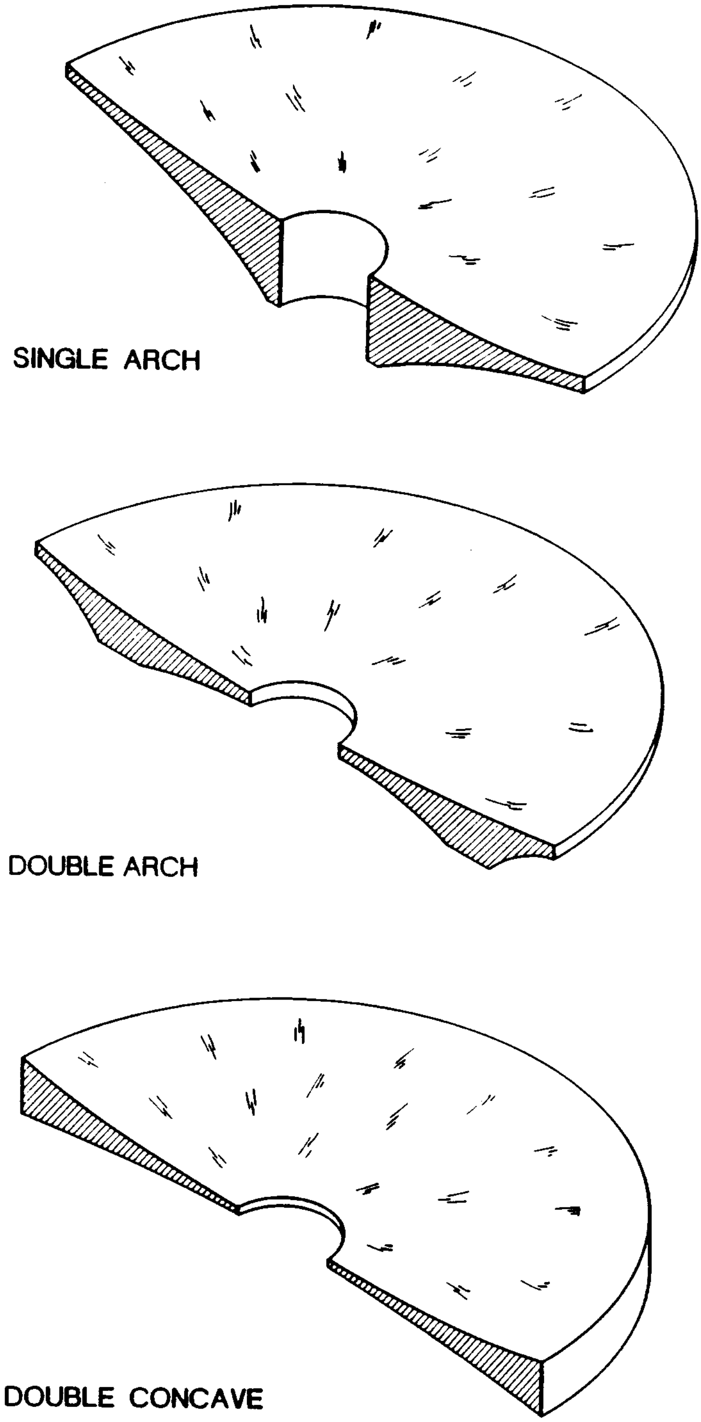


FIGURE 5.6 Types of contoured back mirrors.

not affect bending stress. Bending stress may still exceed the microyield strength of the material (microyield strength is defined as that amount of stress required to produce a permanent strain of 10^{-6} in the material; for maximum dimensional stability the rule of thumb is to keep all stress in the substrate below one half of the microyield strength of the material). If the microyield strength of the material is approached or exceeded due to bi-metallic bending, thermal hysteresis results, with poor optical figure stability.

Cho's studies include a 40-in.-diameter, 5-in.-thick (at the edge) double concave mirror.[13] The radius of curvature is 160 in., so the center thickness is 2.49 in. Mirror material is an aluminum/silicon carbide reinforced metal matrix composite, SXA™, with a density of 0.10 lb/in.^{3} and an elastic modulus of 16×10^6 lb/in.^{2}. The mirror support consists of either a continuous edge ring or multipoint supports spaced around the edge. Optical surface deflection in both zenith (axial) and horizontal (radial) for ring and multipoint supports is given by:

Self-Weight Deflection of 40 in. Double Concave Mirror
Surface Deformation (RMS Wave, 1 Wave = 633 nm)

Gravity Load	Ring	12–30°	6–60°	4–90°	3–120°
Zenith	0.282	0.284	0.292	0.402	0.883
Horizon	0.002	0.003	0.008	0.012	0.020

The single arch mirror is a contoured back mirror with a taper from a thick center to a thin edge. Three kinds of back contours or tapers are used. These are a straight taper,[14] producing a conical back, a convex back taper, and a parabolic taper.[15] The vertex of the parabolic taper is located either at the back of the mirror or the edge of the mirror.

Very good stiffness-to-weight is obtained with a single arch mirror shape using a parabolic taper, with the vertex of the parabola located at the mirror edge. Lower weight, but reduced stiffness, is obtained by locating the vertex of the parabola at the back of the mirror. Both the straight taper or conical back and convex back mirrors are inferior in stiffness-to-weight when compared against parabolic back mirrors.

Cho's studies include a series of single arch shapes, of 16 in. diameter, 3 in. thick, with different back tapers. In all cases the mirrors are SXA™, with a 48-in. optical radius of curvature and a concave shape. Typical edge thickness is 0.5 in. The following self-weight deflections of these mirrors are

Self-Weight Deflection of 16 in. Single Arch Mirrors (1 Wave = 633 nm)

Mirror Type	Horizon (RMS Waves)	Zenith (RMS Waves)	Mirror Weight (Lb)
Straight taper	0.003	0.007	27.3
Convex back	0.003	0.015	29.6
Parabola, vertex at edge	0.003	0.004	18.9
Parabola, vertex at back	0.003	0.012	16.2

For typical single arch mirror designs, the mirror center of gravity is either very close to the optical surface vertex or actually outside the mirror, beyond the vertex. The forward location makes center of gravity support in the radial direction very difficult. Since the mirror cannot easily be supported through the plane of its center of gravity, the optical surface develops astigmatism when the optical axis is in the horizontal position. This astigmatism in the axis horizontal position is a serious limiting factor for larger single arch mirrors. For typical single arch mirrors at a diameter of 1.2 m the self-weight-induced astigmatism is about 1 wave (1 wave = 633 nm) peak-to-valley in the axis horizontal position.[16]

The poor radial bending stiffness of the single arch mirror causes problems for both plated metal mirrors and in a dynamic environment. The very thin edge of the single arch mirror is subject to significant distortion due to bi-metallic bending effects when made of a nickel-plated metal. Vibration can excite the thin edge of the mirror, leading to blur in the final image.

Like all contoured back mirrors, the changing thickness of the single arch mirror causes distortion in the optical surface when the mirror is exposed to a rapid change in temperature. This distortion is exploited in some applications. In the single arch mirror used in the primary of the Mars Observer Camera, a radial temperature gradient is used to control focus. The temperature gradient is created by heating elements at center and edge of the mirror.[17]

The single arch mirror is relatively easy to produce. Typically the mirror is generated from a solid; alternately the mirror may be cast. The thin edge and cantilever form of the single arch complicate optical fabrication, driving up the production cost. Special blocking techniques are sometimes used to support the thin edge of the single arch mirror during polishing.

Ease of mounting is an important advantage of the single arch mirror. The single arch is normally mounted by a center hub support. The central hub may be bonded to an axial hole in the mirror. An athermal center hub mount uses a conical hole in the mirror, with the apex of the hole coincident with the back of the mirror. Figure 5.7 shows a bonded athermal single arch mirror mount. A conical mount is installed into the central conical hole and acts to pull the mirror into contact with a rear flange. The conical mount is provided with an axial spring preload.[18]

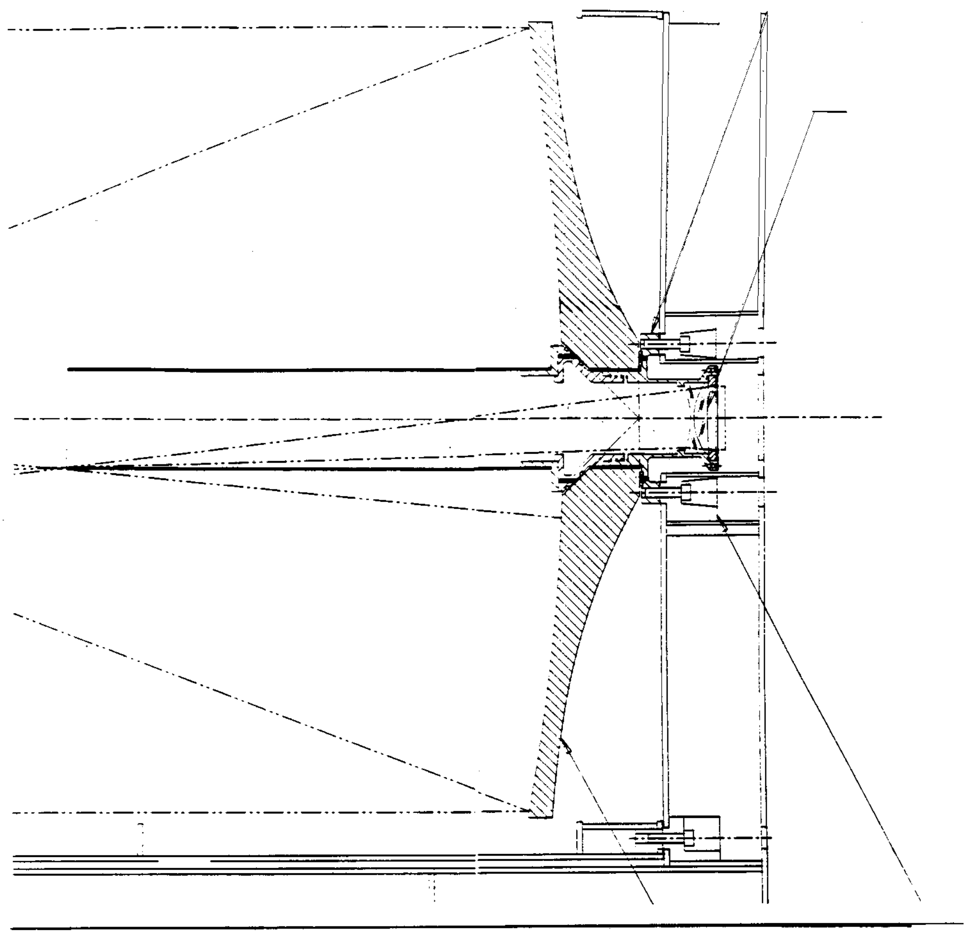


FIGURE 5.7 Single arch mirror with athermal center hub mount.

When compared to other types of lightweight mirror, such as the double arch or sandwich, the single arch mirror is relatively poor in stiffness-to-weight. The extremely low mass and simplicity of the central hub mount for the single arch mirror may make this type of mirror very competitive when the weight of mirror and mount are considered together. In particular, for diameters below 0.5 m the single arch is very competitive in performance and cost to other types of mirrors. An example of the single arch is the primary mirror for the Mars Observer Camera.

The double arch mirror is supported at the back on a ring intermediate in diameter between center and edge. The mirror thickness is reduced away from the support ring, so that both edge and mirror center are thin in comparison with the part of the mirror above the support ring. The cross section of the double arch mirror resembles a bridge with two piers with arches between the piers.[19] This is the source of the "double arch" name. A straight taper, convex taper, or parabolic taper is used on the inner and outer tapered sections of the mirror. The vertex of the parabolic taper may be located at either the edge or back of the mirror.

Stiffness-to-weight of the double arch mirror shape is the best of any contoured back mirror,[20] and is competitive with other types of lightweight mirrors. Optimization of the stiffness-to-weight of the double arch mirror requires selection of an optimum radius of support for the ring and an optimum taper for the mirror back. Cho's studies include four types of back contours: straight taper, convex taper, parabolic taper with vertex at edge, and parabolic taper with vertex at mirror back. All mirrors are made of SXA™, with a 40-in. diameter and 5 in. thick. Typical edge thickness is 0.5 in. Optical surface radius of curvature is 160 in. and the surface shape is concave. The ratio of support ring diameter to mirror diameter varied from 0.5 to 0.65, with both continuous ring and multiple point supports considered. Cho's results are as follows:

Double Arch Mirror Self-Weight Deflection (All Deflections in Units of RMS Waves, 1 Wave = 633 nm)

Mirror Shape	Mirror Weight (lb)	Axis	Ring	Support Location			
				12–30°	6–60°	4–90°	3–120°
Support Ring Ratio = 0.5							
Parabola back vertex	256	Zenith	0.021	0.021	0.023	0.078	0.253
		Horizon	0.027	0.027	0.028	0.046	0.119
Parabola edge vertex	254	Zenith	0.021	0.022	0.023	0.073	0.234
		Horizon	0.027	0.027	0.028	0.041	0.098
Straight taper	324	Zenith	0.069	0.070	0.070	0.092	0.229
		Horizon	0.021	0.021	0.022	0.044	0.136
Support Ring Ratio = 0.55							
Parabola back vertex	256	Zenith	0.004	0.004	0.016	0.090	0.291
		Horizon	0.020	0.020	0.021	0.041	0.097
Parabola edge vertex	254	Zenith	0.013	0.013	0.019	0.085	0.272
		Horizon	0.046	0.046	0.047	0.051	0.065
Straight taper	324	Zenith	0.036	0.036	0.038	0.083	0.260
		Horizon	0.025	0.025	0.026	0.043	0.109
Support Ring Ratio = 0.60							
Parabola back vertex	256	Zenith	0.045	0.046	0.050	0.114	0.352
		Horizon	0.019	0.018	0.021	0.045	0.103
Parabola edge vertex	254	Zenith	0.036	0.037	0.041	0.093	0.266
		Horizon	0.007	0.007	0.011	0.027	0.044
Straight taper	324	Zenith	0.006	0.007	0.017	0.085	0.279
		Horizon	0.008	0.008	0.012	0.038	0.094
Support Ring Ratio = 0.65							
Parabola back vertex	256	Zenith	0.065	0.066	0.072	0.133	0.368
		Horizon	0.006	0.006	0.012	0.033	0.052
Parabola edge vertex	254	Zenith	0.065	0.067	0.072	0.131	0.355
		Horizon	0.010	0.010	0.014	0.032	0.045
Straight taper	324	Zenith	0.043	0.044	0.049	0.114	0.343
		Horizon	0.007	0.007	0.012	0.037	0.187

Optimum support for the double arch depends on the number of supports and the contour of the back. If three supports equally spaced on a common diameter are used, as is normal practice with lightweight space mirrors, the optimum support diameter ratio is about 0.5. This is true regardless of back shape. The best stiffness-to-weight is obtained with a parabolic set of back contours. For the optimum shape, the outer parabola vertex is at the edge, and the inner parabola vertex is at the center.

A set of six individual supports equally spaced on a common diameter provides a support condition which closely approximates a ring support. This suggests that for critical applications the double arch should be supported by a six-point support. For this type of support the optimum support diameter ratio is 0.55. The optimum back contour associated with the six-point support is identical in form to that of the six-point support. The inner and outer portions of the double arch back contour are parabolas; the vertex of the outer parabola is at the edge, and the inner at the center.

Like the single arch, the variable thickness of the double arch causes the mirror to distort following a sudden change in temperature. The double arch mirror usually develops a more complex optical surface distortion than the single arch when a similar temperature gradient is introduced into the mirror. This suggests that the double arch mirror is not well suited for applications in which the temperature is changing rapidly. Owing to better radial stiffness, the double arch mirror is subject to less bi-metallic bending than the single arch mirror shape when used with plated metal.

Unlike the single arch, the center of gravity location of the double arch is usually below the optical surface vertex. In most applications, the plane of the center of gravity is accessible for mounting. Since the double arch is readily supported through the plane of its center of gravity, deflection of the optical surface producing astigmatism is limited when the optical axis is horizontal. The double arch is well suited for use in mirror sizes over 1 m. Double arch mirror designs up to 4 m diameter are discussed in the literature.[21]

Mounting of the double arch is significantly more complex than the single arch. Standard practice is to produce cylindrical pockets in the back of the mirror, at the support ring. These pockets extend axially into the mirror to sufficient depth to reach the plane of the center of gravity. Radial support forces act through the plane of the center of gravity, against the side wall of the pockets. Axial supports are provided either at the same pockets, or spaced in-between the pockets. A key design issue is the athermalization of the mounting hardware in the pockets. If the temperature is limited, a simple Invar ring is bonded into the pocket, coincident with the plane of the center of gravity. For larger temperature changes an athermal socket is used, typically with a single or double conical taper.[22] These athermal sockets are difficult to fabricate in the back of the mirror, and significantly increase the mirror cost. Performance of conical athermal sockets is very good. In a test performed at NASA Ames Research Center, a 0.5-m-diameter fused silica double arch mirror with three athermal conical sockets was taken from room temperature to about 10 K. Total change in figure over this range of temperature was about 0.1 wave RMS (1 wave = 633 nm).[23] Figure 5.8 is a schematic drawing of the athermal mount used in the NASA Ames tests.

The double arch mirror is easy to fabricate.[24] It is generated from a solid or casting. Support during polishing is often provided by a continuous compliant ring. Mirror stiffness is sufficient, and a special support is not required for the inner and outer portions of the mirror. Edge thickness is often reduced in an effort to minimize the weight. The edge thickness of a metal matrix composite double arch mirror may be only 3 mm. Such thin edges pose a significant risk during fabrication and handling of the mirror.

Mounting sockets are machined in the back of glass material double arch mirrors using diamond tools. Fixed abrasive tools with a shape corresponding to the required socket shape are used. Mushroom-shaped holes in the mirror back are produced by rotating the mirror as well as the tool during the socket generating process. A cylindrical hole is first cored out of the back of the mirror. A tool with a corresponding contour is then inserted into the socket, and the tool rotated about its axis. The tool is then de-centered relative to the socket. Once the tool is de-centered, the mirror is rotated about the axis of the socket. This causes the tool to sweep out a circle concentric with the socket axis, producing the mushroom-shaped hole.

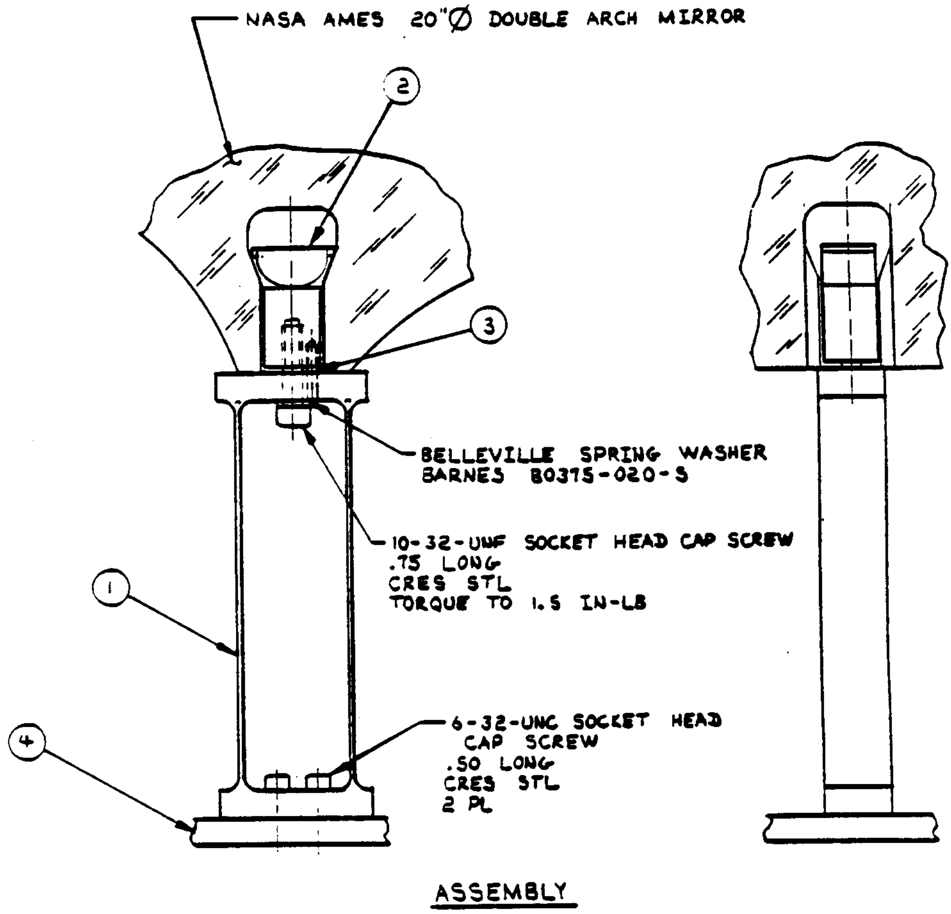


FIGURE 5.8 Athermal mounting socket for double arch mirror.

5.5 Sandwich Mirrors

Sandwich mirrors achieve the highest stiffness-to-weight ratios of any type of lightweight mirrors. Typically a sandwich mirror is from 40 to 20% of the weight of an equivalent solid right circular cylinder 6:1 diameter-to-thickness ratio mirror. Weight ratios below 20% are possible with the sandwich mirror, although cost and fabrication risk are high. The sandwich mirror is relatively expensive and difficult to fabricate. Mounting of the sandwich mirror is technically challenging, particularly when large loads must be accommodated in a dynamic environment. Thermal response of the sandwich mirror is controversial. For terrestrial applications there is the possibility of reduced thermal equilibrium time by ventilating the interior of the sandwich mirror.[25] Sandwich mirrors consist of a thin face sheet, a thin back sheet parallel to the face sheet, and a shear core connecting the two sheets. The shear core normally consists of thin ribs at right angles to the face and back sheets. These ribs intersect to form pockets between face and back sheets. The pocket geometry consists of triangular, square, or hexagonal cells.

Other types of shear cores are used. Cylindrical cell cores are used in machined sandwich mirrors. Tubular cells are employed in blow-molded borosilicate sandwich mirrors.[26] Foam cores are used to make ultralightweight sandwich mirrors.[27] The structural foam in the shear core is aluminum, metal matrix composite, or fused silica glass.

In the discussion of contoured back mirrors, the bridge analogy is used to explain the development of the double arch mirror. A beam analogy is likewise useful to explain the structural efficiency of the sandwich mirror. High stiffness is provided when the mass of a structure is distributed as far as possible from the neutral or bending axis. An I-beam distributes most of its mass in the flanges, which are far from the bending axis. Relatively little mass is placed in the web of the I-beam. In a similar fashion, the sandwich mirror places most of its mass in the face and back sheet, with as little mass as possible in the shear core. This provides a very efficient structure in bending.

Although the structure of a sandwich mirror is complex, the self-weight deflection of this type of mirror is readily calculated through the use of the concept of equivalent flexural rigidity.[28] The equivalent flexural rigidity of a sandwich mirror is the flexural rigidity of a solid plate of equal thickness. The flexural rigidity of a solid plate without a lightweight section is given by:

$$D_{solid} = \frac{Eh^3}{12\left(1-\nu^2\right)}$$

where D_{solid} = flexural rigidity of plate
E = elastic modulus of plate material
h = thickness of plate
ν = Poisson's ratio of plate material

In a similar manner, the flexural rigidity of a lightweight mirror is given by:

$$D_{lightweight} = \frac{Et_b^3}{12\left(1-\nu^2\right)}$$

where $D_{lightweight}$ = flexural rigidity of lightweight mirror
E = elastic modulus of mirror material
t_b = equivalent bending thickness of lightweight mirror
ν = Poisson's ratio of mirror material

The equivalent bending thickness is given by:

$$t_b^3 = \left(2t_f + h_c\right)^3 - \left(1 - \frac{\eta}{2}\right)h_c^3$$

where t_b = equivalent bending thickness of mirror
t_f = face sheet thickness
h_c = rib height
η = rib solidity ratio

The above equations assume that the face and back sheets are of equal thickness. A key parameter is the rib solidity ratio, which is a function of the rib thickness and pocket size. The size of the pockets in the shear core is expressed by the diameter of a circle that is tangent to all walls of the pocket. This circle is the inscribed circle. The rib solidity ratio is given by:

$$\eta = \frac{\left(2B + t_w\right)t_w}{\left(B + t_w\right)^2}$$

where η = rib solidity ratio
B = inscribed circle diameter
t_w = rib thickness

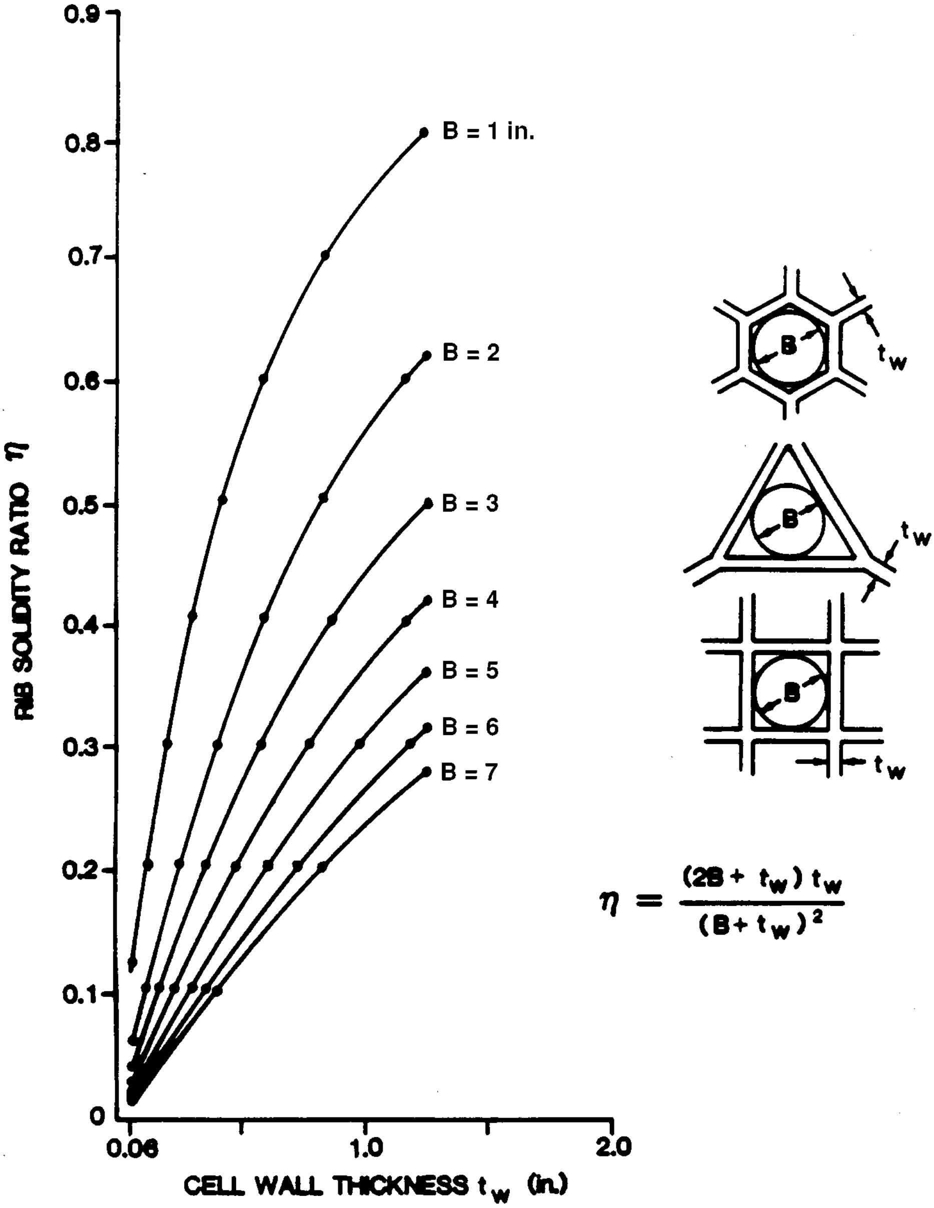


FIGURE 5.9 Rib solidity ratio. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

The rib solidity ratio is shown in Figure 5.9. The inscribed circle diameter for a triangular cell is

$$B_{triangular} = \frac{L}{\sqrt{3}}$$

where $B_{triangular}$ = inscribed circle diameter of triangular pocket
L = length of side of triangular pocket

The inscribed circle diameter for a square cell is

$$B_{square} = L$$

where B_{square} = inscribed circle diameter of square pocket
L = length of side of square pocket

The inscribed circle diameter for a hexagonal pocket is

$$B_{hexagonal} = \sqrt{3}\,L$$

where $B_{hexagonal}$ = inscribed circle diameter of hexagonal pocket
L = length of one of six sides of hexagonal pocket

Another parameter useful in analysis of lightweight sandwich mirrors is the cell pitch. This is the spacing of the cells in the shear core, or distance from center of inscribed circle to center of inscribed circle. The cell pitch is given by:

$$P = B + t_w$$

where P = cell pitch
B = inscribed circle diameter
t_w = rib thickness

The weight of a sandwich mirror is given by:

$$W = \rho A\left(2t_f + \eta h_c\right)$$

where W = sandwich mirror weight
ρ = mirror material density
A = area of the mirror
t_f = face sheet thickness
η = rib solidity ratio
h_c = rib height

Mehta has developed equations which optimize the distribution of mass in the face sheets and core of a sandwich mirror.[29] For a given overall height or weight, the optimum face sheet thickness is found for varying rib solidity which produces a mirror with the greatest possible flexural rigidity. Flexural rigidity is an important measure of stiffness, but is not necessarily a measure of stiffness-to-weight. For an optimum symmetric sandwich section:

$$t_f = \frac{W\left(\sqrt{1-\frac{\eta}{2}} - \sqrt{1-\eta}\right)}{\rho A\left\{2\left[\sqrt{1-\frac{\eta}{2}} - \sqrt{(1-\eta)^3}\right]\right\}}$$

where t_f = optimum face sheet thickness
W = mirror weight
η = rib solidity ratio
ρ = mirror material density
A = area of mirror

Figure 5.10 shows the relationship between face sheet thickness, rib thickness, mirror thickness, and inscribed circle diameter. In this figure, the minima of the curves represent mirrors with

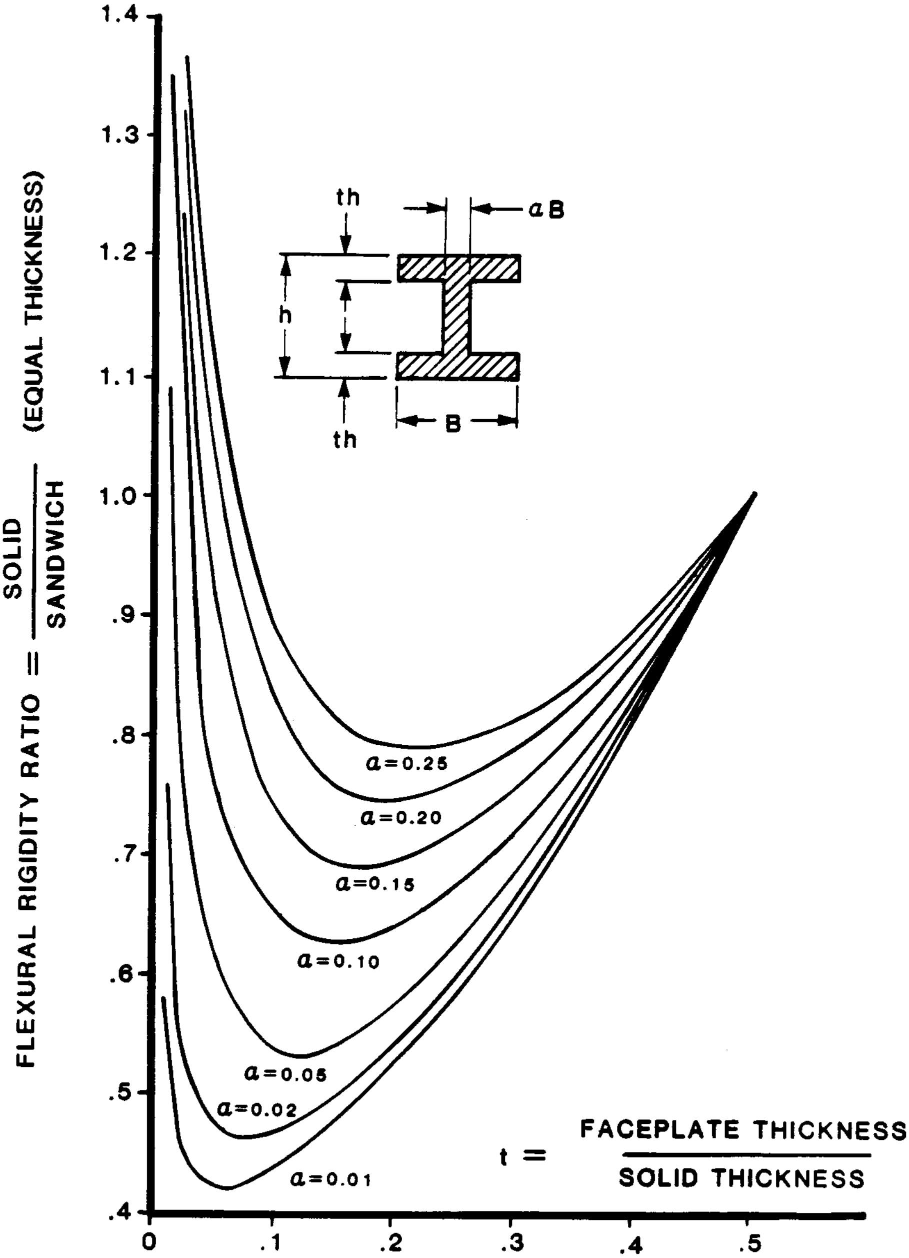


FIGURE 5.10 Symmetric sandwich mirror flexural rigidity. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

optimum stiffness-to-weight. The figure illustrates that sandwich mirrors are capable of better stiffness than solid mirrors of the same thickness.

The structural efficiency of the sandwich mirror is given by the ratio V_o/I_o. This is the ratio of unit volume to unit cross-sectional moment of inertia, and is given for the sandwich mirror by:

$$\frac{V_o}{I_o} = \frac{12\left(2t_f + \eta h_c\right)}{t_b^3}$$

where V_o = unit volume of mirror
I_o = unit cross-sectional moment of inertia
t_f = face sheet thickness of mirror
η = rib solidity ratio
h_c = rib height
t_b = equivalent bending thickness of mirror

Shear deflection is an important component of the deflection of sandwich mirrors. Corrections for shear in a sandwich mirror are difficult to implement in simple closed-form equations. One approximation including shear effects for self-weight deflection of circular lightweight mirrors on multiple point supports is[30]

$$\delta \approx 0.025\left(\frac{W}{A}\right)\left(\frac{r^4}{D}\right)\left(\frac{3}{n}\right)^2 + 0.65\left(\frac{W}{A}\right)\left(\frac{r^2}{S_c G A_o}\right)\left(\frac{3}{n}\right)$$

where δ = peak-to-peak surface deflection of mirror
W = mirror weight
A = area of mirror
r = radius of mirror
D = flexural rigidity of mirror
n = number of support points
S_C = shear coefficient
G = shear modulus
A_o = cross-section area/unit width

The shear relations are

$$S_c = \frac{A_w}{A_w + A_f} = \frac{1}{1 + \dfrac{4t_f}{\eta h_c}}$$

$$A_o = \frac{2Pt_f + h_c t_w}{P}$$

$$G = \frac{E}{2(1+\nu)}$$

where S_C = shear coefficient
A_W = area of rib
A_f = area of face sheet within pitch
t_f = face sheet thickness
η = rib solidity ratio
h_C = rib height
t_w = rib thickness
A_o = cross-section area/unit width
P = core pitch
G = shear modulus of mirror material
E = elastic modulus of mirror material
ν = Poisson's ratio of mirror material

For any given pocket geometry or cell pattern, the shear core has equal shear rigidity if the pitch is held equal. For sandwich mirrors, structural efficiency is independent of cell geometry.[31] The equivalence of different cell or pocket geometries is controversial.[32] Experience with actual mirrors indicates at best a very weak dependence on shear core geometry.

Another controversial area of sandwich mirror design is the use of an edge band. An edge band provides additional tangential stiffness, or stiffness in the direction of the circumference of the mirror. This additional stiffness helps prevent deformation of the mirror edge when the mirror surface changes radius. The edge band provides protection for the thin ribs of the sear core. In some applications the edge band is used to provide an anchor point for the mirror mount. The disadvantage of the edge band is the additional weight of the band at the edge of the mirror.

Contouring the back of a sandwich mirror provides additional weight reduction at a relatively small penalty in stiffness. Such contouring is expensive and may add to the cost of mirror fabrication. Contouring the back may also present mounting problems, and degrade the thermal response of the mirror for the same reasons given for contoured back mirrors. For these reasons, contoured back sandwich mirrors are relatively uncommon.

"Quilting" is an issue that is related to the optimization process in the design of a sandwich mirror. Quilting is a permanent pattern of deformation that is polished into the mirror during optical fabrication. This deformation is due to deflection of the face sheet of the mirror between the ribs under polishing pressure. When the surface of the mirror is viewed by an optical test the resulting deflection pattern resembles the squares of a quilt. This resemblance explains the use of the term "quilting".

Quilting creates surface errors that are periodic and of relatively small amplitude. These periodic surface errors act like a diffraction grating. In a diffraction-limited system, quilting scatters light from the central maximum of the diffraction disk. The reduction in energy due to quilting is given by:

$$\frac{I_I}{I_0} = \frac{4\pi^2\left(\frac{\delta_c}{2\lambda}\right)^2}{\left[1-2\pi^2\left(\frac{\delta_c}{2\lambda}\right)^2\right]\left[1-4\pi^2\left(\frac{\delta_c}{2\lambda}\right)^2\right]}$$

where I_I = energy in central maximum with quilting
I_0 = energy in central maximum without quilting
δ_C = face sheet deflection due to quilting
λ = wavelength

The relationship between quilting deflection and reduction of energy in the central maximum of the diffraction disk is shown in Figure 5.11.

The quilting deflection due to polishing pressure is given by:

$$\delta_c = \frac{PB^4}{\psi\left(\frac{Et_f^3}{12\left(1-\nu^2\right)}\right)}$$

where δ_C = face sheet deflection due to quilting
P = polishing pressure
B = inscribed circle diameter
E = mirror material elastic modulus

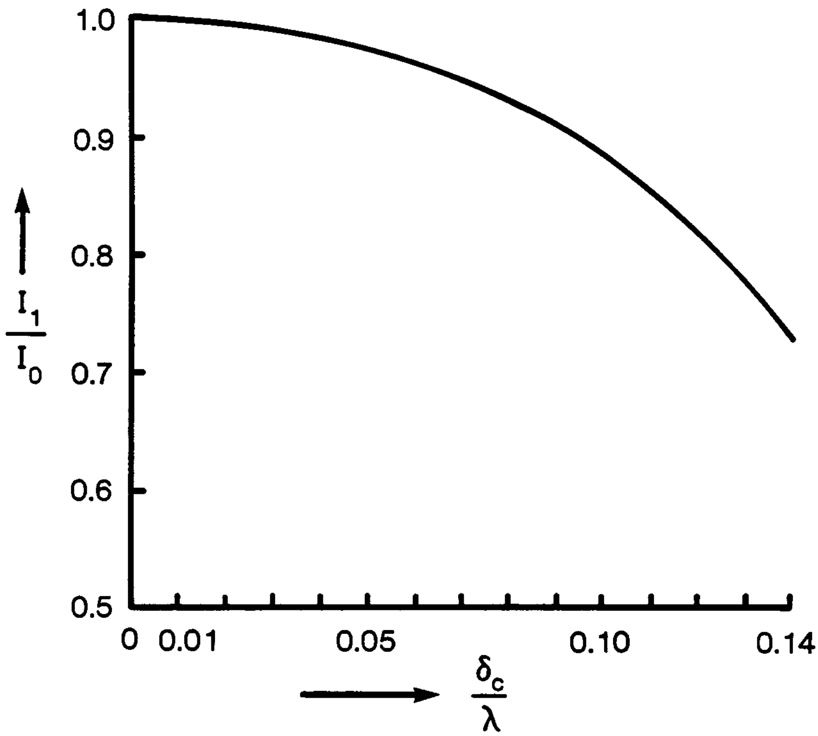


FIGURE 5.11 Reduction in energy of the central maximum of the diffraction disk due to quilting.

t_f = face sheet thickness
ν = Poisson's ratio for mirror material
ψ = geometric quilting constant

The geometric quilting constant depends upon the shape of the cell or pocket in the mirror and is given by:

Geometric Quilting Parameters

Cell Geometry	Ψ
Triangular	0.00151
Square	0.00126
Hexagonal	0.00111

Figure 5.12 shows examples of quilting for a sandwich mirror with triangular cells.

Even at modest levels of weight reduction, quilting becomes a significant issue in lightweight mirror design. Quilting is relatively independent of cell shape. As indicated by the above table, quilting varies by a factor of less than 1.5 between hexagonal and triangular cells. This is shown in Figure 5.13. More important is the inscribed circle diameter and face sheet thickness.

Several solutions are suggested for quilting. Reducing polishing pressure is the simplest solution. Quilting is linearly dependent on polishing pressure. Reducing polishing pressure directly reduces quilting. The drawback to this idea is that polishing time is also linearly proportional to polishing pressure. Reducing polishing pressure increases polishing time, and therefore increases polishing cost. Increased polishing time increases the possibility of the mirror developing high spatial frequency errors called "ripple". High spatial frequency errors affect the mirror the same way as quilting.

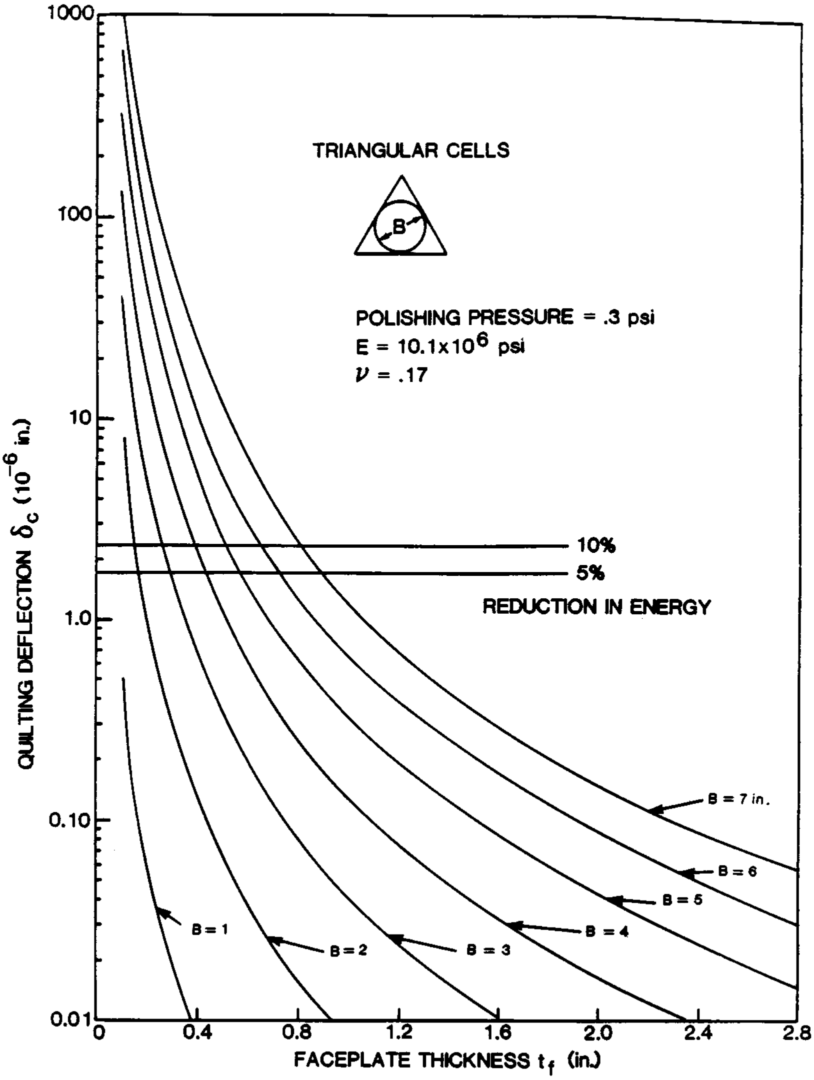


FIGURE 5.12 Example quilting deformation of triangular cell. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

Another possible solution is pressurization of the cells with a gas. This produces a pressure equal and opposite to the polishing pressure. This approach is expensive, and may be difficult to implement in a sandwich mirror with multiple holes in the core and back sheet. Polishing pressure is normally not uniform across the polishing tool, which limits the utility of the deflection compensation due to internal pressure. Part of the quilting deflection is due to poorly understood thermal effects, and pressurization does not reduce these effects.

If there are holes in the back sheet of the mirror, supports can be run through these holes to provide additional stiffness for the face sheet. These supports are called "quilting posts". Use of these supports is dependent on access through the back of the mirror. Adjusting the posts to provide just the right amount of support is difficult. If not properly adjusted the posts may act as hard points to cause another quilting pattern corresponding to the post location.

Filling the shear core with an incompressible fluid to limit quilting is sometimes attempted. This approach has not proven successful. Handling a fragile lightweight sandwich mirror when

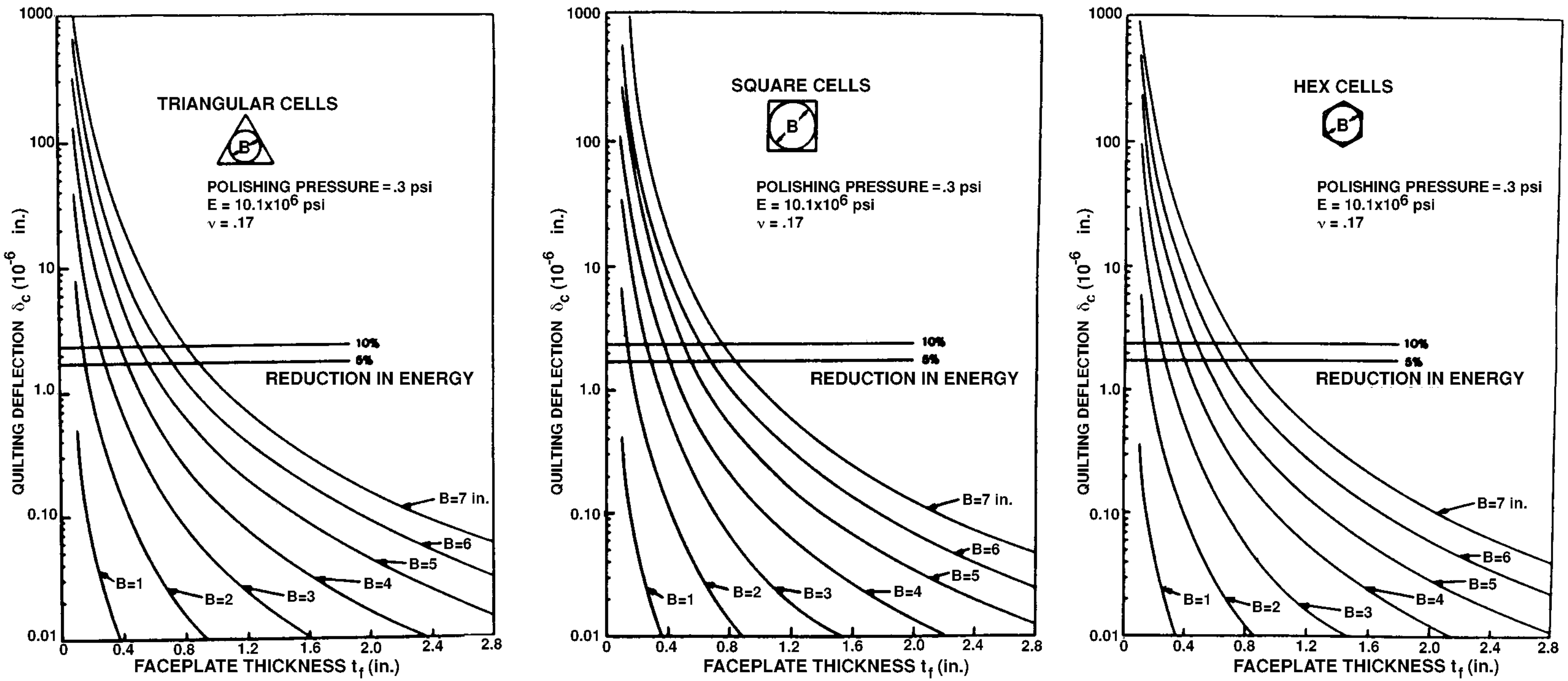


FIGURE 5.13 Quilting compared in triangular, square, and hexagonal cells. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

filled with fluid is very hazardous. The cells must be completely full of fluid and sealed relative to each other for this approach to succeed.

One solution to the quilting problem is local polishing of the face sheet above each cell. Polishing is normally done by hand. This approach is the most common. Although tedious, hand polishing of lightweight sandwich has been successful even on very large and lightweight mirrors.

Quilting posts require holes in the back sheet of the mirror. These holes represent discontinuities in the back sheet. Holes reduce the bending stiffness of the back sheet. As a rule of thumb, if the total area of the holes is less than 10% of the back sheet area, the mirror flexural rigidity is not significantly affected.

Holes in the shear core are also occasionally necessary. Holes located in the ribs between cells should be placed at the midplane of the mirror. In this position the holes have the smallest effect on flexural rigidity. Holes in the shear core should be circular to minimize stress concentration. Holes in the ribs should be less than 1/3 the height of the ribs. As a rule of thumb, the total area of the holes in the ribs should be less than 10% of the total rib surface area.

Lightweight sandwich mirrors are made by assembly from pieces, casting, or machining from a solid. The earliest technique for making a sandwich mirror was assembly from pieces, using a Bakelite adhesive. Ritchey pioneered this method, but was not successful in its use.[33]

Corning and others developed techniques for fusing together fused silica sections to produce a lightweight sandwich mirror. Typically the face and back sheets are made as individual plates. The shear core is made of slotted ribs. The assembly is placed into a furnace and fused together. The ribs of shear core produced by this assembly technique resemble the wall of an egg carton. This resemblance leads to the name “egg crate mirror”.[34] Richard and Malvick demonstrated that the flexural behavior of egg crate mirrors is highly isotropic despite the apparent poor adhesion in the slotted sections of the core.

A similar fusion technique is used by Hextek to produce “blow molded” sandwich mirrors. In this technique the shear core is produced from tubes of glass close packed between face and back sheet. During the fusing process an inert gas is blown into the tubes. This gas causes the tubes to expand and adhere to each other. The cells produced in the shear core by this process are roughly hexagonal.[35]

Shear cores are sometimes produced by machining cells or pockets in a monolithic blank. The blank is placed between face and back sheet, and the assembly is fused in an oven. This technique is often used to produce hexagonal pockets. The shear core is normally machined using classic glass fabrication methods.[36] Recently Eastman Kodak developed a technique for machining the shear core by water jet cutting.[37]

Frit bonding is another technique for assembling lightweight sandwich mirrors. Frits are special glass materials that act as cements when heated.[38] A torch is used to heat the frits during assembly. Bulk heating often creates an undesirable sag of the face sheet between the ribs during fusing. Fritting avoids the bulk heating of the mirror. Extremely lightweight sandwich mirrors are often assembled using frits to avoid this bulk heating.

Lightweight sandwich mirrors are sometimes machined from a single solid blank. This approach is extremely expensive and involves substantial risk. The possibility of damage to the mirror during the machining operations is very high. The main advantage of this method is the high degree of uniformity of the resulting mirror, since it is produced from a single piece of material. This technique was pioneered in the 1960s. Mirrors up to 1.8 m diameter were produced in materials such as Cer-Vit using this method. Today REOSC in France is the main proponent of this technique. An example of such a mirror is the primary mirror of the ISO (Infrared Satellite Observatory).

Glasses which melt at relatively low temperatures are used to cast lightweight mirrors.[39] Casting was used to produce the 5-m primary mirror for the Hale telescope at Mt. Palomar. Casting is often combined with the use of spinning furnace to produce a near-net optical surface shape.[40]

There are a number of significant problems with the casting process. Hydrostatic pressures on the molds for the shear cores are substantial. The core molds may break loose, as occurred during the Mt. Palomar casting operation. Pressure may cause the walls of the shear core to deform. This may produce ribs of uneven thickness, which reduces the flexural rigidity of the mirror. At high casting temperatures borosilicate glass is very chemically active and may react with the mold material. This chemical reaction leads to undesirable properties in the glass. These properties include compositional inhomogeneity, staining of the surface, and de-vitrification of the glass.

Bubbles rise to the surface of the mirror during the casting process. The very high viscosity of the molten glass causes the bubbles to remain, creating voids in the surface of the mirror. Removal of the molds following casting is difficult. One technique is the use of water-soluble mold materials. Molds made of such materials are flushed out of the core using high pressure water jets.

Beryllium sandwich mirrors are produced using a particle metallurgy process. This process is high isostatic pressing or the HIP process. The HIP process is somewhat similar to casting in that molds are used. Copper molds are the most common. The copper molds are placed in a steel canister containing beryllium powder. Heat and pressure are used to consolidate the powder. Acid is then used to remove the copper molds from the mirror.[41] Thermal coefficient of expansion differences between the steel canister, copper mold, and beryllium power may cause cracking of the mirror during this process.

Other metals such as aluminum are used to make lightweight sandwich mirrors by casting, welding, or brazing. Brazing is the most common assembly method for producing metal sandwich mirrors, and is used with aluminum, metal matrix composite (SXA™), and beryllium. Adhesive bonding is not generally used due to the extreme difference in thermal coefficient of expansion between adhesive and metal mirror material.

Mounting lightweight sandwich mirrors requires incorporation of special mounting features into the mirror. Such features consist of solid or near-solid cells. Alternately local regions of high density are placed at the perimeter of the mirror or on the back sheet. On blow-molded and cast mirrors it is common to incorporate mounting surfaces in open bottom cells.[42] These mounting surfaces are coincident with the neutral or bending axis of the mirror. Such mounting surfaces are used to carry loads in the plane of the center of gravity of the mirror.

Frit bonding is used to attach mounting features to the surface of the mirror. Pads are attached to the edge band or the back sheet of the mirror. An alternate method uses conventional adhesives to attach the mounting pads. The use of adhesives requires great caution since the thermal coefficient of expansion of most adhesives is much higher than that of common mirror materials. In an extreme case, a large change in temperature may induce failure in the bond between mirror and pad.[43]

Most mounting geometries are kinematic. Principal concerns are differences in the thermal expansion coefficient between mirror and mount, and lack of co-planarity in the mounting pads. Flexures are often used to isolate the mirror from expansion or contraction of the mount. If the mounting surfaces are not in the same plane, moments are introduced into the mirror. These moments are reduced by adding additional degrees of rotational compliance in the mirror mounts.

5.6 Open-Back Mirrors

Open-back mirrors consist of a thin face sheet with an array of ribs on the back side of the face sheet. These ribs intersect to form pockets in the back of the mirror. Unlike the sandwich mirror these pockets are completely open in the back. Open-back mirrors are a traditional means of producing lightweight mirrors. Normally open-back mirrors are comparable in weight to sandwich mirrors, with a weight reduction of 30 to 40% of the same diameter 6:1 diameter-to-thickness ratio right circular cylinder mirror. Extremely lightweight mirrors are produced using the open back geometry, with weight reductions in some case below 20%. Stiffness-to-weight ratio of the open-back mirror is poor, and is inferior to both sandwich and contoured back mirrors. Thermal behavior of the open-back mirror is very good, due to the favorable ratio of volume-to-surface area. In addition, all portions of the mirror are relatively thin, producing short thermal time constants. The open-back mirror is normally lower in cost than the sandwich mirror, but higher in cost than the contoured back mirror. Mounting of open-back mirrors is relatively easy.

The cells or pockets of the shear core of a open-back mirror are open in the back. This open geometry is responsible for the term "open-back" mirror. Normally, cell or pocket geometry is similar to that found in sandwich mirrors. Triangular, square, and hexagonal cells or pockets are used in the shear core of open-back mirrors. Circular pockets are produced in mirrors machined from a solid blank. Other cell geometries are produced by combinations of radial and concentric circular ribs. This type of geometry is not common.

A beam analogy is useful in understanding the bending behavior of the open-back mirror. The open-back mirror is comparable to a T-shaped beam. A T-shaped beam lacks symmetry about its neutral or bending axis. Such a beam is poor in structural efficiency in comparison with an I-shaped beam. This analogy is extended to explain the poor bending stiffness of the open back in comparison with a sandwich mirror.

The stiffness of an open-back mirror is determined using an approach very similar to that used in finding the stiffness of a sandwich mirror. Many of the equations used in calculating the bending of an open-back mirror are identical to those used for the sandwich mirror. Only those equations which are unique to the open-back mirror are presented here. Like the sandwich mirror, the stiffness of an open-back mirror is determined by calculating the flexural rigidity of a solid mirror of equivalent stiffness. The flexural rigidity of a lightweight open-back mirror is given by:

$$D_{lightweight} = \frac{Et_b^3}{12\left(1-\nu^2\right)}$$

where $D_{lightweight}$ = flexural rigidity of lightweight mirror
E = elastic modulus of mirror material
t_b = equivalent bending thickness of lightweight mirror
ν = Poisson's ratio of mirror material

The equivalent bending thickness of a lightweight open-back mirror is given by:

$$t_b^3 = \frac{\left[\left(1-\frac{\eta}{2}\right)\left(t_f^4 - \frac{\eta h_c^4}{2}\right) + \left(t_f + h_c\right)^4 \frac{\eta}{2}\right]}{\left(t_f + \frac{\eta h_c}{2}\right)}$$

where t_b = equivalent bending thickness of lightweight mirror
η = rib solidity ratio
t_f = face sheet thickness
h_c = rib height, measured from mirror back to back of face sheet

The same rib solidity ratio relationships are used for the open-back mirror as are used for the sandwich mirror. The rib solidity ratio is combined with the face sheet thickness and rib height to find the mirror weight. The weight of an open-back mirror is given by:

$$W = \rho A\left(t_f + \eta h_c\right)$$

where W = sandwich mirror weight
ρ = mirror material density
A = area of the mirror

t_f = face sheet thickness
η = rib solidity ratio
h_c = rib height

Mehta developed a relationship which is used to optimize the flexural rigidity of an open-back mirror. The lack of symmetry in an open-back mirror results in a significantly more complex relationship for optimization than is used for the sandwich mirror. This complex relationship is normally solved numerically. The relationship for an optimum open-back mirror is

$$4\left(t_f + \frac{\eta h_c}{2}\right)\left[\left(1-\frac{\eta}{2}\right)\left(t_f^3 - \frac{h_c^3}{2}\right) + \frac{(\eta-1)(t_f+h_c)^3}{2}\right]$$
$$-\left(\frac{1}{2}\right)\left[\left(1-\frac{\eta}{2}\right)\left(t_f^4 - \frac{\eta h_c^4}{2}\right) + \frac{\eta(t_f+h_c)^4}{2}\right] = 0$$

where t_f = face sheet thickness
η = rib solidity ratio
h_c = rib height

Figure 5.14 shows the relationship between face sheet thickness, rib thickness, inscribed circle diameter, and mirror thickness. In this figure the optimum mirror designs for the best stiffness-to-weight are found to the left, at the bottom of the curves. Unlike the sandwich mirror, the open-back mirror never exceeds the flexural rigidity of a solid mirror of equal thickness.

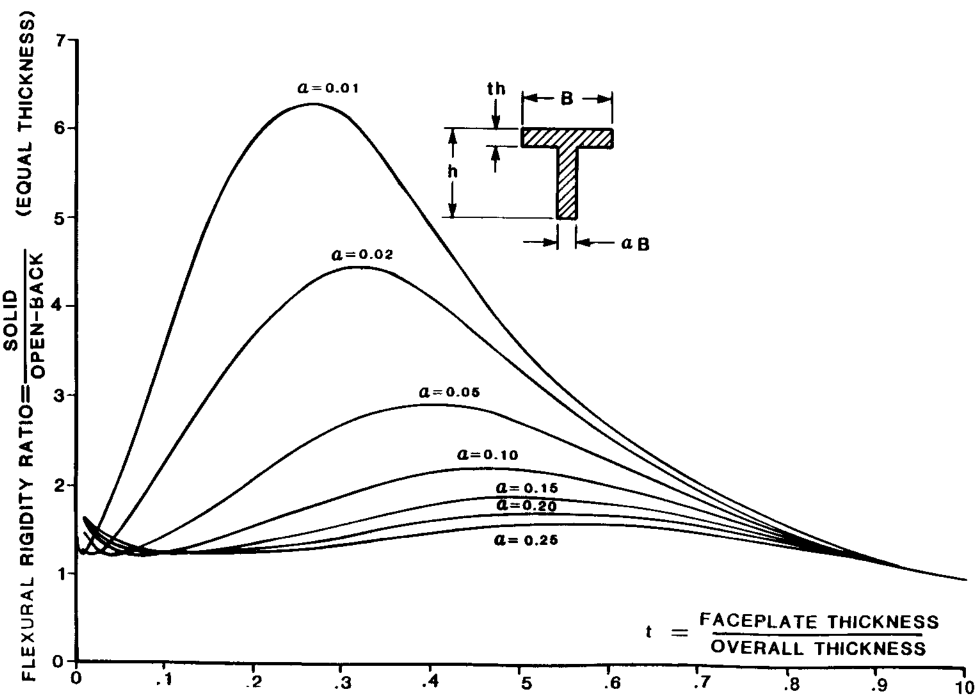


FIGURE 5.14 Open-back mirror flexural rigidity. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

The structural efficiency of an open-back mirror is given by:

$$\frac{V_o}{I_o} = \frac{12\left(t_f + \eta h_c\right)\left(t_f + \frac{\eta h_c}{2}\right)}{\left[\left(1 - \frac{\eta}{2}\right)\left(t_f^4 - \frac{\eta h_c^4}{2}\right) + \left(t_f + h_c\right)\frac{\eta}{2}\right]}$$

where V_o = unit volume of mirror
I_o = unit cross-sectional moment of inertia
t_f = face sheet thickness of mirror
η = rib solidity ratio
h_c = rib height

Shear deformation is an important component of the bending behavior of an open-back mirror. Self-weight deflection of an open-back mirror is computed using the same equation employed for sandwich mirrors. The lack of symmetry in the open-back mirror leads to a more complex set of equations for the shear coefficient in the deflection equation. The shear coefficient is given by:

$$S_c = \frac{u}{D_1 D_2 D_3 D_4}$$

$$u = 10\left(1 + \nu\right)\left(1 + 4m\right)^2$$

$$D_1 = 12 + 96m + 276m^2 + 192m^3$$

$$D_2 = \nu\left(11 + 88m + 248m^2 + 216m^3\right)$$

$$D_3 = 30n^2\left(m + m^2\right)$$

$$D_4 = 10\nu n^2\left(4m + 5m^2 + m^3\right)$$

$$m = \frac{Bt_f}{ht_w}$$

$$n = \frac{B}{h}$$

where S_C = shear coefficient
ν = Poisson's ratio for mirror material
B = inscribed circle diameter
t_f = face sheet thickness
h = total mirror thickness
t_w = rib thickness

There is a consensus in the U.S. optical engineering community that the optimum cell or pocket geometry for the open-back mirror is triangular. Other geometries provide less torsional resistance than the triangular geometry. Square cells are considered acceptable, and hexagonal cells are thought to provide inferior shear stiffness.

Open-back mirrors in the past were produced with a radial rib pattern. Concentric ribs (sometimes called intercostal ribs) joined the radial ribs to produce semicircular pockets or cells. This type of construction is used in the 0.6-m-diameter beryllium primary mirror for the IRAS.[44] This type of cell geometry is relatively easy to produce using older machine tools that are not numerically controlled. There is agreement in the U.S. optical community that this type of shear core is inferior in stiffness at comparable weight to shear cores using straight ribs and conventional cell shapes.

The open-back mirror is inferior in stiffness at comparable weights to the sandwich mirrors. Simple analysis of flexural rigidity without a shear correction sometimes indicates that superior stiffness is obtained in open-back mirrors by increasing the depth of the shear core or ribs. When a correction for shear is included for such very deep structures, shear effects are found to increase deflection. Very deep open back structures are not as efficient as comparable weight sandwich structures.

The stiffness of open-back mirrors is comparable with conventional solid mirrors of equal weight or equal thickness. For most applications the open back does not offer any significant advantage in stiffness when compared with the much lower-in-cost solid mirror. Open-back mirrors are used to provide shorter thermal equilibrium times than solid mirrors. If stiffness is not important, as is often the case for space applications, the open back may provide a reduction in weight in comparison with other mirror types.

Open-back mirrors are sometimes used to provide extremely lightweight mirrors. Two modifications used in open-back structures to further reduce weight are tapered backs and cylindrical holes located in the junction of the ribs. Both modifications are undesirable.

Open-back mirrors are low in bending stiffness. Tapering the ribs in the vertical direction tends to further reduce bending stiffness. This reduction is at the mirror edge, which is subject to the great deflection. If a reduction in weight is considered important, a better solution is redesign of the ribs or face sheet. One option is the use of thinner ribs.

Cylindrical holes located at the junction of the ribs interrupt the continuity of the ribs. This interruption significantly reduces the stiffness of the ribs. Any reduction in weight is offset by a decrease in the overall stiffness of the mirror.[45] Such cylindrical holes are sometimes used to improve the thermal equilibrium time of the mirror. Normally the rib junctions are the thickest portion of the shear core. Cylindrical holes reduce the effective thickness of the junction. This practice is questionable, since comparable thickness areas exist in the junction of rib and face sheet. The face sheet and rib junction is likely to be more critical to thermal response time of the mirror than the rib junctions.

Quilting effects in open-back mirrors are identical to those experienced in sandwich mirrors. Open-back mirror designs sometimes feature a pattern of “subribs” on the back of the face sheet between the ribs of the shear core. These subribs are intended to provide additional stiffness to the face sheet to help minimize quilting under polishing loads. Such relatively shallow ribs provide little additional stiffness. The weight of such subribs is better applied to increasing the thickness of the face sheet.

Open-back mirrors are often polished face down on the polishing lap. The pockets or cells of the shear core are provided with weight to offset the deflection of the face sheet under polishing pressure. Lead shot, for example, is used to load the cells. This method of reducing quilting effects appears to work for small, relatively stiff mirrors about 0.5 m in diameter.[46] The efficiency of this method for larger mirrors is controversial.

Open-back mirrors are produced by casting or machining from a solid blank. Casting is the oldest approach and was successfully employed for the primary mirror of the 5-m Hale telescope at Mt. Palomar. Machining is used to produce both metal and glass mirrors. Welding of metal mirrors to produce an open-back section is still largely experimental,[47,48] although large, low precision solar simulator mirrors have been produced this way.

Casting of open-back mirrors requires the use of a relatively low melting temperature glass, such as a borosilicate, or the use of a metal. Cast open-back mirrors are vulnerable to the same problems as discussed for sandwich mirrors. One advantage of the casting process for open-back mirrors is suppression of surface bubbles. Bubbles are suppressed by casting the mirror upside down, that is, with the face sheet down. The core mold is suspended above the mirror and then plunged into the molten material. This casting method is expensive and requires handling of the mold inside the furnace.

Machining of open-back mirrors from a solid is now common in the U.S. optical industry.[49] This approach allows the use of materials that cannot be cast. Machining from a solid is sometimes used as a way of minimizing quilting effects. The mirror optical surface is produced before the mirror is machined into a lightweight configuration. The mirror is then machined into an open-back geometry. There are two very serious difficulties with this approach: residual stress in the mirror and breakage during machining. Residual stress in the mirror is released during the machining of the mirror into the open-back configuration. This residual stress may produce mirror optical surface errors larger than the expected quilting errors from ordinary polishing. Any polishing to remove errors due to residual stress introduces the possibility of quilting. Breakage of the mirror during machining is a significant possibility. Such breakage occurs at the worst possible time, which is after the optical surface figure is produced. Some machining techniques break as many as one out of three mirrors.

Beryllium lightweight open-back mirrors are produced by machining from solid billets.[50] Older vacuum hot-pressed or VHP beryllium billets often are flawed, with internal voids. Such voids interrupt the continuity of the ribs and greatly reduce the stiffness of the mirror. This is an expensive procedure, since as much as 80% or more of the billet is removed during machining. Massive machining puts considerable stress into the beryllium mirror. Very rigorous heat treatment is necessary to remove this residual stress.

Open-back mirrors are straightforward to mount. Open-back mirrors are mounted by attachments at the edge of the mirror, or through the use of the interior of the cells or pockets. Mounting features are attached to either the rib sides or bottom of the pockets. The thickness of the face sheet is sometimes increased in the area of the cells used for mounting. The center of gravity of the open-back mirror is normally close to the bottom of the cell. A relatively small increase in face sheet thickness in the cells used for mounting brings the bottom of the cell into coincidence with the plane of the center of gravity of the mirror. This provides a very favorable location for mounting.

Although the open-back mirror mounting geometry is very favorable, the low stiffness requires attention in the design of the mount. Open-back mirrors are more sensitive to applied forces and moments than sandwich or contoured back mirrors. Particular care is necessary to minimize moments induced in the mirror due to alignment errors between mounts. One approach is to provide a universal joint between the point of attachment to the mirror and the mount at each mounting point. This universal joint consists of an ordinary ball and socket or a multiple degree of freedom flexure assembly.

5.7 Comparison of Mirror Performance

Lightweight mirrors are selected on the basis of the following criteria identified by Valente and Vukobratovich:[51]

1. Self-weight induced deflection
2. Efficiency of mirrors of equivalent weight, where efficiency is defined as a function of self-weight induced deflection and mirror thickness
3. Ease of fabrication

In the study performed by Valente and Vukobratovich, 1-m fused silica lightweight mirrors of single arch, double arch, sandwich, and open-back geometries were compared. Mirror thickness

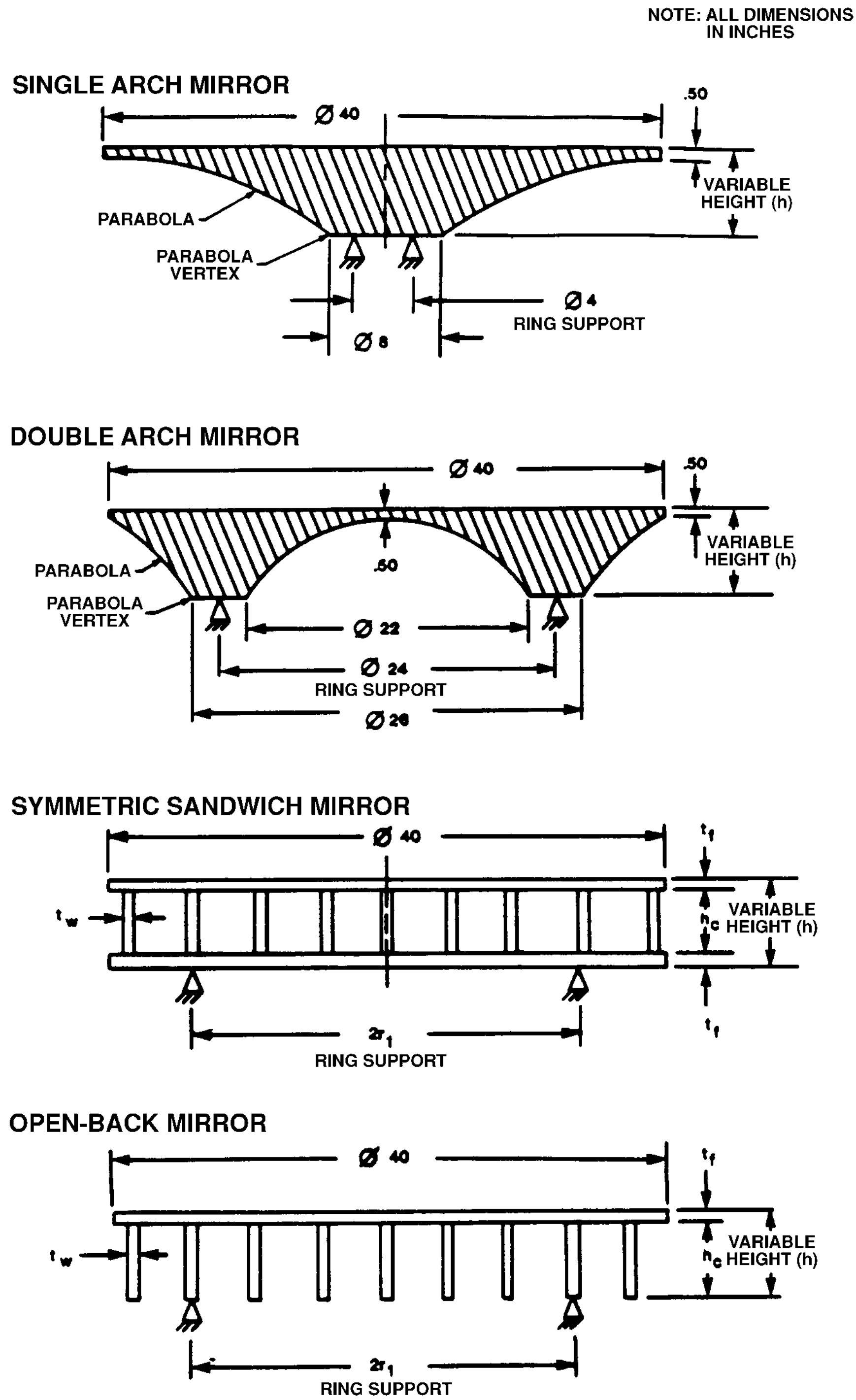


FIGURE 5.15 Mirror geometries used in mirror performance comparison. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.

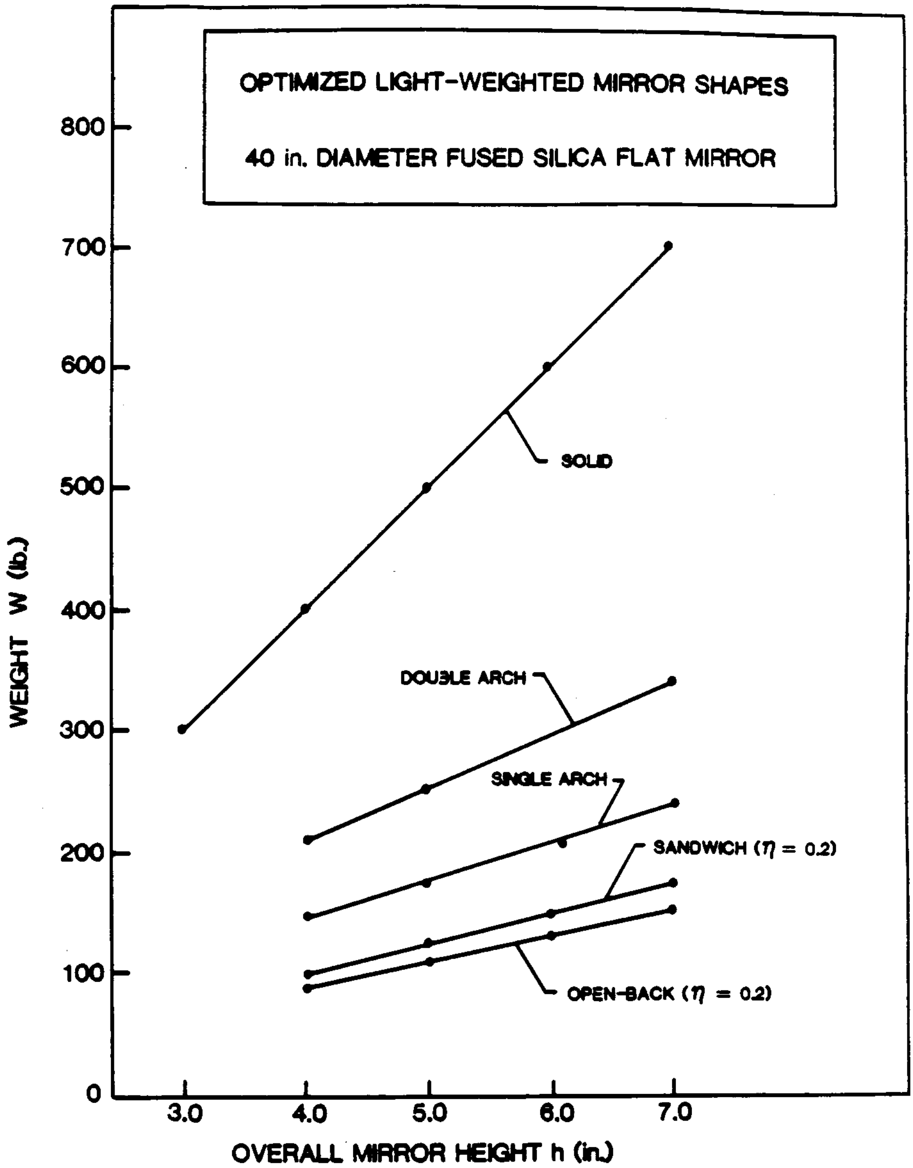


FIGURE 5.16 Mirror weight vs. mirror thickness. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

and weight were varied, with self-weight deflection computed for each variation in parameter. Figure 5.15 shows the mirror geometries used in this study. Figure 5.16 gives the mirror weight vs. mirror thickness or height. Certain reasonable assumptions were made in this study about detailed mirror parameters such as the rib solidity ratio, face sheet thickness, and so forth.

Figure 5.17 is a plot of the mirror height vs. self-weight deflection for the mirrors in the study. The worst deflection, and therefore the worst performance for a given height, is provided by the single arch mirror. The next best performance is obtained by the double arch. Significantly better at constant height than the double arch mirror is the solid mirror. Comparable deflection to the solid mirror is provided by the open-back mirror. Finally, the minimum deflection for a given height is provided by the sandwich mirror.

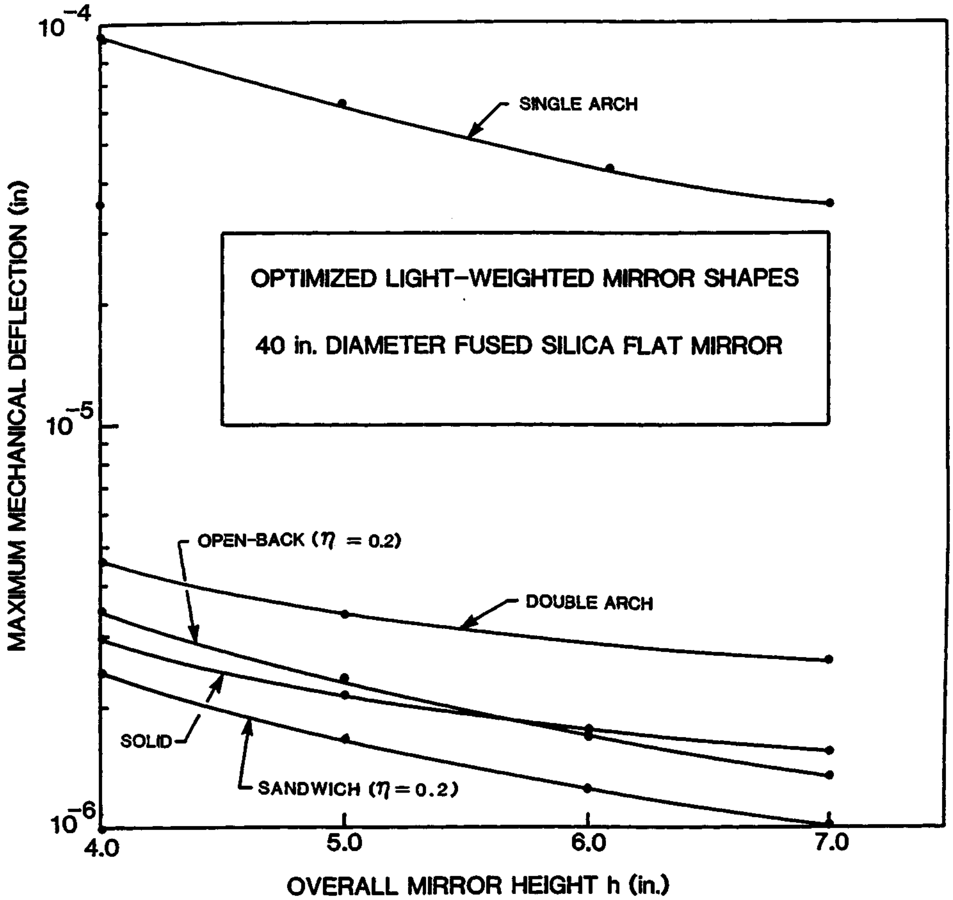


FIGURE 5.17 Mirror self-weight deflection vs. mirror height. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

Mirrors are sometimes produced by machining from solid blanks. For minimum self-weight deflection at a constant height, the best mirrors (excluding the sandwich mirror, which is not easily machined from a solid) are the open-back and solid mirrors. For comparable height, the open-back will be lighter than the solid mirror.

Different rankings are produced when mirror weight is plotted against self-weight deflection in Figure 5.18. Maximum deflection, and therefore the worst performance, is obtained with the single arch design. At comparable weights the solid mirror and open-back are the next best in performance. These two types of mirrors are virtually identical in deflection at comparable weights. Next best is the double arch mirror. Minimum deflection is provided by the sandwich mirror.

Both the deflection vs. height and deflection vs. weight charts indicate that best performance, in the sense of minimum self-weight deflection, is provided by the sandwich mirror. Use of a sandwich mirror may not always be possible, due to fabrication or cost concerns. Next best in minimizing deflection for a given weight is the double arch. At comparable weights, the solid and open-back mirrors provide the same self-weight deflection. If weight is an issue, use of an open-back mirror provides no stiffness advantage over a solid mirror of identical weight. Selection of an open-back mirror is often based on other criteria than stiffness and weight. Worst in performance is the single arch mirror. Mirror efficiency, defined as the total mirror height divided by the mirror self-weight deflection, is given in Figure 5.19.

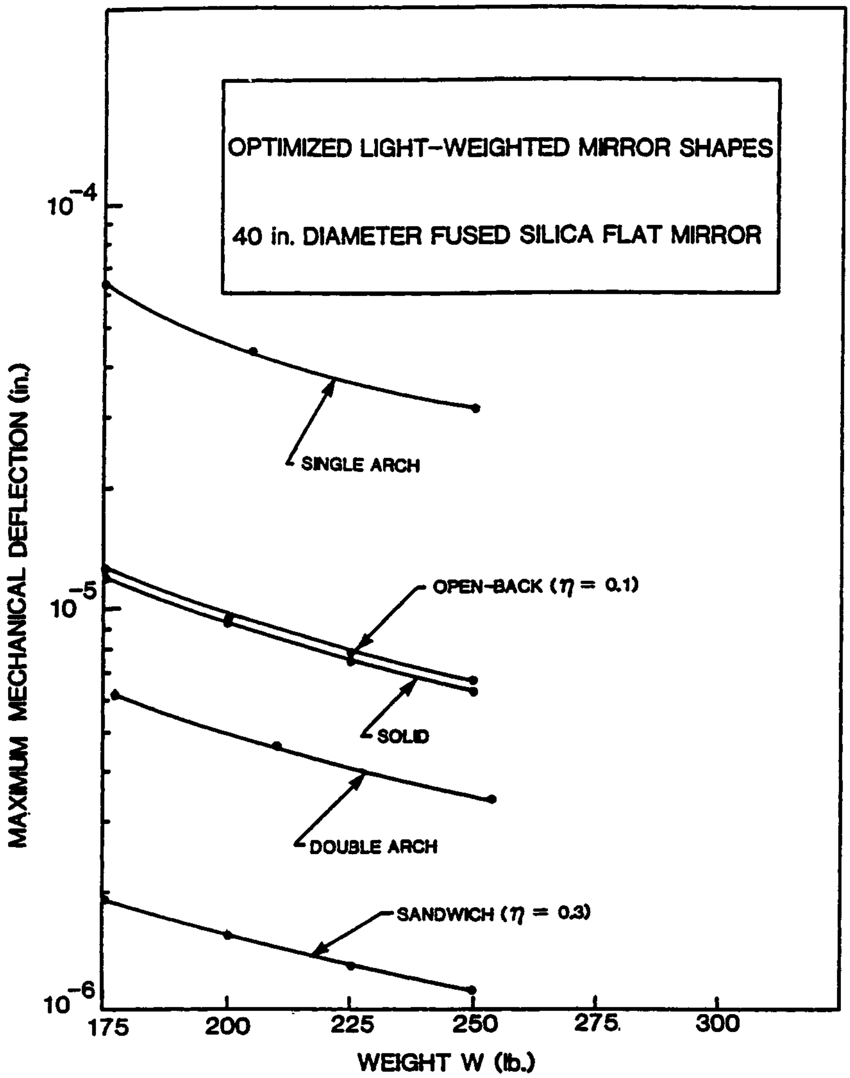


FIGURE 5.18 Mirror self-weight deflection vs. mirror weight. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

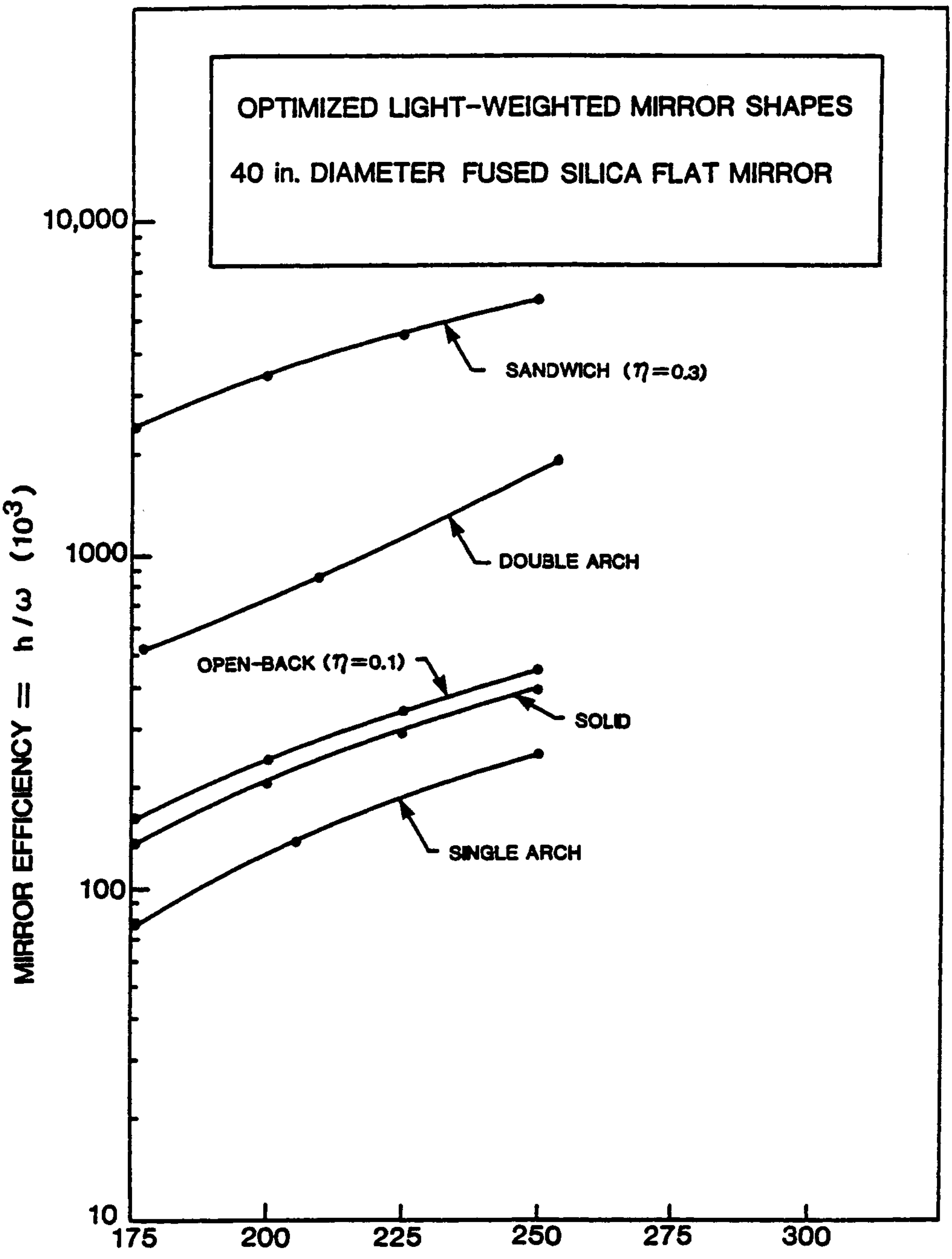


FIGURE 5.19 Mirror efficiency (total mirror height divided by mirror self-weight deflection) vs. mirror weight. (From Valente, T.M. and Vukobratovich, D. 1989. *Proc. SPIE 1167*, 20.)

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