

Selection of Glass Alternative Materials for Manufacturing Heliostat Reflective Facets

An Analytical Approach

Jean Schnaar-Campbell¹ , Johann Bredell¹ , and
Craig McGregor^{1,*} 

¹Stellenbosch University, Stellenbosch, SA

*Correspondence: Craig McGregor, craigm@sun.ac.za

Abstract. Traditional concentrating solar power plants utilize silvered glass reflective facets that account for a significant portion of the total plant cost. Several cost-effective non-glass reflectors have been developed. A majority of these novel reflectors are flexible, which necessitates exploring support structures to maintain their shape during use. This study presents a methodology for evaluating and selecting structural support materials for non-glass heliostat facets. The approach employs analytical models to efficiently screen a large pool of candidate materials based on strength, stiffness, and deflection due to gravity, using an existing glass reflector panel as a reference. During analysis, sandwich panels showed promise as viable support for the reflector. The most promising sandwich panel designs are further investigated using experimental testing. The analytical model predictions are found to align moderately well with physical test results, validating its utility for rapid material selection. The top-performing sandwich panel designs demonstrate the potential for cost savings and increased facet size compared to traditional glass facets. This research provides a framework for evaluating and selecting structural materials for non-glass heliostat facets, enabling improved cost-effectiveness and performance of concentrating solar power systems.

Keywords: Concentrating Solar Power, Sandwich Panels, Heliostat Facets, Material Selection, Novel Reflectors

1. Introduction

There is worldwide increasing demand for renewable electricity generation. However, the majority of renewable energy solutions are unreliable. Concentrating Solar Power (CSP) is capable of on-demand power generation [1]. In a traditional CSP plant, the reflective facets used are 7 % of the total cost of the plant [2]. The most common reflectors used in CSP plants are second surface silvered glass panels with a metal support structure. Alternate non-glass facets with a 25 % cost reduction potential have already been discovered [3], and developed for CSP applications [4]. However, a common feature among these reflector alternatives is that the reflectors are more flexible than glass.

Exploring alternative materials or composite panels to provide the structure of a facet panel could lead to increased heliostat performance and cost reduction.

This research aims to find a method to select materials for the structural support of glass alternative heliostat facets. The method is based on an analytical model that requires the non-glass facets must have at least the same performance in terms of strength, stiffness, and own-weight deflection as a glass reference panel. Candidate materials are ranked according to cost per unit area. The method is then validated with physical testing. The results of the study will be used to select an optimal panel for the structural support of non-glass heliostat facets.

2. Viable reflectors

For CSP applications the main reflectors that have been considered are thin-film reflectors and polished metal reflectors [4]. Both thin film and metal reflectors have been considered in this study. Sundog solar's EverBright Mirror Film [5] was selected as a candidate. The film is 98 % specular reflective and features a scratch resistant, anti-soiling protective coating and an adhesive on the back. The selected polished metal reflector for this study was Alonod's MIRO-SUN [6]. The MIRO-SUN is a 0.5 mm thick polished aluminium reflector that features an anti-soiling coating. Below is Table 1 that shows the advantages and disadvantages of the alternate reflectors.

Table 1. Advantages and disadvantages of alternate reflectors.

Reflector	Thick	Reflective	Mass	Cost	Yield	Stiffness	Source
Unit	[mm]	[%]	[kg/m ²]	[\$/m ²]	[MPa]	[Nm ²]	
Glass	4	95	10	15.88	45	384	[7]
Metal	0.5	90	1.36	43.37	186	0.718	[6]
Thin film	0.05	98	0.07	10.00	Non-structural		[5]

3. Single material analytical model

The proposed analytical model involves defining a reference facet panel, typically an existing glass panel used in a functioning CSP plant, and comparing the bending strength, stiffness, and deflection due to gravity of alternative panels to this reference panel. It is assumed that if the alternate has equivalent or better mechanical properties than the reference, it can be used in the same applications as the reference panel. A diagram of the assumed load case for comparisons can be found in Figure 1.

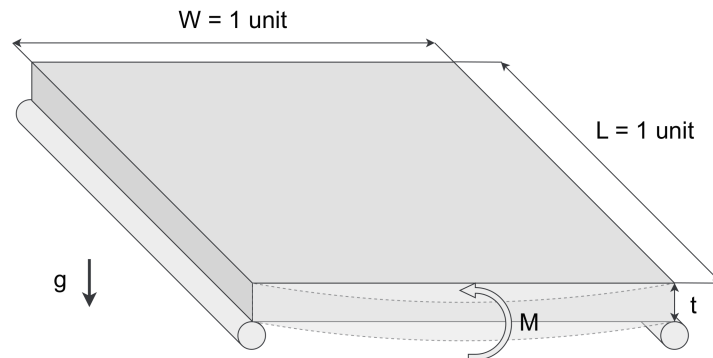


Figure 1. Diagram of the loads, supports and dimensions assumed for calculations.

To perform the comparison, consider the reference moment (M_{max}) that causes the reference panel to yield (Equation 1 [8]). The reference panel has a unit length (L), meaning the reference moment is a function of thickness (t) and yield stress (σ_y) only. The new panel can be compared to the alternate panel by setting both panels to the same reference moment, and calculating the required thickness of the alternate panel (Equation 2). This way the alternate panel can be assumed to resist the same loading as the reference panel.

$$M_{max} = \frac{1}{6}\sigma_y L t^2 = \frac{1}{6}\sigma_y t^2 \quad (1)$$

$$t_{alt} = t_{ref} \sqrt{\frac{\sigma_{y,ref}}{\sigma_{y,alt}}} \quad (2)$$

For the stiffness comparison, flexure stiffness (K) was used. Due to the selected reference panel, the flexure stiffness can be simplified to Equation 3. Where E is the modulus of elasticity and I is the second moment of area. The alternate panel can be compared to the reference panel by setting the same reference stiffness, and calculating the required thickness of the alternate panel. This way the alternate panel should resist the same loading as the reference panel. The required thickness for the alternate panel based on stiffness can be seen in Equation 4.

$$K = EI = \frac{1}{12}Et^3 \quad (3)$$

$$t_{alt} = t_{ref} \sqrt[3]{\frac{E_{ref}}{E_{alt}}} \quad (4)$$

The maximum deflection of the panel due to gravity (y_{max}) is calculated using Equation 5. Assuming both panels have the same deflection due to gravity, the required thickness of the alternate panel can be calculated using Equation 6. Where ρ is the density of the material and g is the gravitational acceleration.

$$y_{max} = \frac{5\rho g}{384EI} = \frac{5\rho g}{32Et^2} \quad (5)$$

$$t_{alt} = t_{ref} \sqrt{\frac{E_{ref}\rho_{alt}}{E_{alt}\rho_{ref}}} \quad (6)$$

Table 2. Short list best performing single materials.

Category	Material	Required thickness	Price	Mass
Unit		mm	\$/m ²	kg/m ²
Overall	Aerated concrete	29.10	1.43	18.92
Wood	Fir	4.70	1.77	1.67
Foam	Polypropylene foam	155.54	6.40	3.26
Metal	Cast Iron	4.42	16.36	32.23
Reference	Glass	4.00	15.88	10.00

The reference panel material is assumed to be glass with a thickness of 4 mm. The assumed mechanical properties is: density of 2500 kg/m³, modulus of elasticity of 72 GPa, and yield stress of 45 MPa. Granta Edupack [9] was used as the materials database for this study. For each material in this material database, the required thickness of a support panel is assumed to be the maximum of the thicknesses calculated

in Equations 2, 4 and 6. The cost of the material per unit area is then calculated based on the required thickness and the cost per unit volume of the material. The materials are then ranked based on cost per unit area. Table 2 shows some of the top materials.

4. Sandwich panel analytical model

A sandwich panel is a composite panel consisting of a core, and two skins. Sandwich panels have been proposed as potential support structures for heliostat facets [10]. The benefit of sandwich panels is that they can be stiff and strong while being light [11]. A diagram of a sandwich panel can be found in Figure 2.

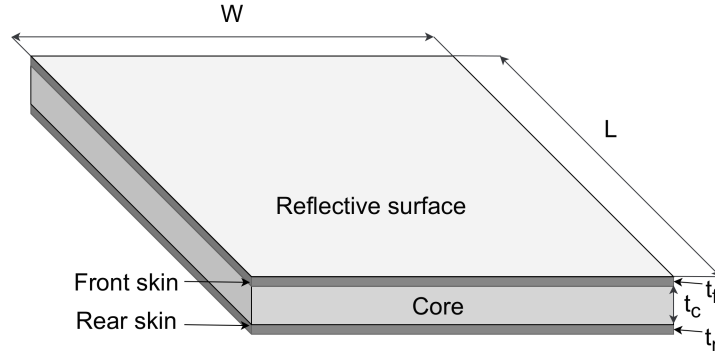


Figure 2. Diagram of a sandwich panel.

Instead of calculating the required thickness of a sandwich panel directly, a theoretical sandwich panel is created and the dimensions optimised. The strength, stiffness, and deflection due to gravity are calculated using Roark's Formulas for Stress and Strain, Section 11.4 [12]. The cost of the sandwich panel per unit area is calculated by adding the cost of the skin and core materials, using only the cost per volume of the material, and the material thickness.

4.1 Glass equivalent sandwich panel

The required resistance to bending, stiffness, and maximum deflection due to gravity is calculated from the same reference panel is the same glass panel used in Section 3. The most optimal thicknesses of the skin and core materials are found using the generalized reduced gradient (GRG) non-linear optimisation in Microsoft Excel [13], with the goal set as the cheapest cost per unit area, with equal or greater mechanical properties. An automated script was created to iterate this through all material combinations in the database. The best sandwich and core combination from the script are shown in Table 3.

4.2 Unsupported sandwich panel

From Table 3, it is possible to see that the sheets in the sandwich panels are extremely thin. This makes manufacturing more difficult and increases costs despite using minimal material. Due to this a new analysis was performed. A theoretical required strength, stiffness and deflection due to gravity was calculated for a 1.2 m x 6 m panel. This width was selected since many sheet materials are sold in 1.2 m rolls. The length was selected so that no extra support for the facet is required past the torque tube. The theoretical heliostat that will be created from this can be seen in Figure 3. Both heliostats has a reflective area of 36 m².

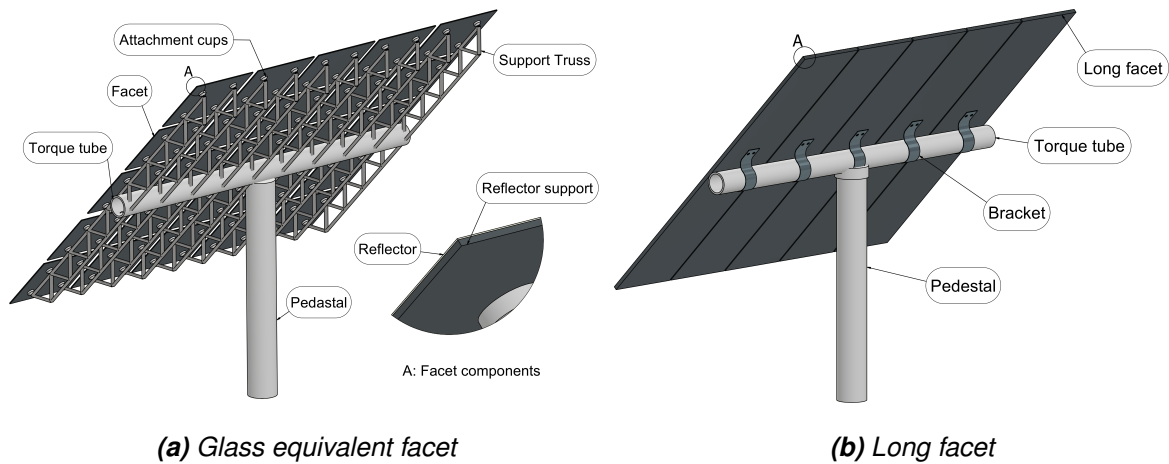


Figure 3. Diagrams of potential heliostats using different facets.

The required strength was calculated from ASCE 7 [14], using the wind load calculation for flat panels, assuming a maximum operating wind speed of 20 m/s. The required stiffness and bending due to gravity was calculated assuming a maximum deflection of 30 mm over the length of the panel at peak wind load and due to gravity. This was selected since this deflection will lead to a slope error of less than 1 mrad rms. From these criteria a maximum bending moment of 642.6 Nm and flexural stiffness of 48.2 kNm² was calculated, with a maximum deflection due to gravity of 30 mm. The best performing sandwich panel is shown in Table 3.

Table 3. Best performing sandwich panels.

Type	Material		Thicknesses		Price	Mass
	Core	Skins	Core	Skins		
Unit			mm	mm	\$/m ²	kg/m ²
Glass eq.	PP foam 0.020	AISI 5140	14.00	0.03	1.17	0.765
Long	PP foam 0.020	AISI 5140	65.0	0.1	4.66	3.52

4.3 Interpreting analytical results

Extra interpretation is required due to the idealisations of the analytical screening. The best performing sandwich panels had plastic based foam cores. Polypropylene (PP) foam is uncommon in the local (South African) market. Polyvinyl chloride (PVC) foam (closed cell, 0.70) is used in large quantities for insulation, with low theoretical cost increase. The best performing skin materials were all steels, with little variation in mechanical performance between steel types. Galvanised mild steel was selected as this is the cheapest corrosion resistant material. Using the selected materials, and setting the material thicknesses to the nearest standard sheet size available, a set of possible viable facet panels were generated in Table 4. The cost of the glass equivalent panels are more expensive compared to the cost of silvered glass mirrors, however the long panels show promise for potential cost reduction. The remainder of the paper focuses on the two panels using Alanod MIRO-SUN [6] as the reflector.

Table 4. List of viable sandwich panels.

Material	Position	Thickness	Price	Mass
Unit		mm	\$/m ²	kg/m ²
Glass equivalent sandwich facet				
Alonod MIRO-SUN	Front skin	0.5	43.37	1.36
PVC foam (closed cell, 0.70)	Core	3	59.44	2.10
Galvanised mild steel	Rear skin	0.58	5.70	4.55
	Total	4.08	60.21	8.01
EverBright Mirror Film	Reflector	0.02	10.00	0.03
Galvanised mild steel	Both skins	0.58	5.70	4.55
PVC foam (closed cell, 0.70)	Core	3	11.14	2.10
	Total	4.18	32.54	11.23
Long sandwich facet				
Alonod MIRO-SUN	Front skin	0.5	43.37	1.36
PVC foam (closed cell, 0.70)	Core	16	59.44	11.20
Galvanised mild steel	Rear skin	0.58	5.70	4.55
	Total	4.08	108.50	17.11
EverBright Mirror Film	Reflector	0.02	10.00	0.03
Galvanised mild steel	Both skins	0.58	5.70	4.55
PVC foam (closed cell, 0.70)	Core	16	59.44	11.20
	Total	4.18	80.83	20.33

5. Physical testing

Physical testing was done on the panels to confirm the results of the analytical model. A three pin bending test was done on the sandwich panels to determine the flexure stiffness and yield moment of the panel. The setup can be seen in Figure 4a.

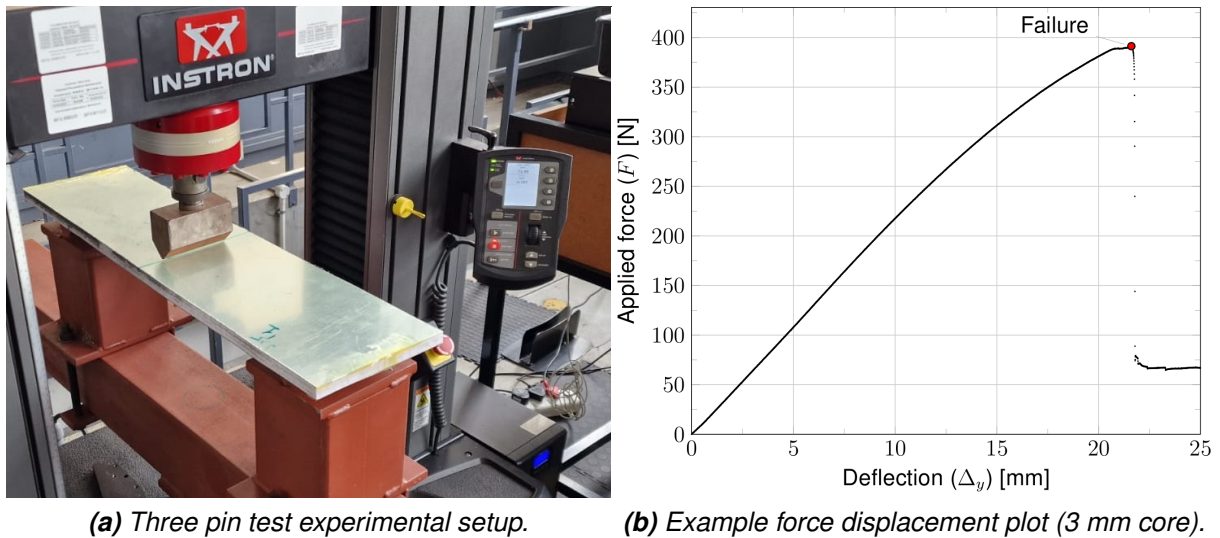


Figure 4. Experimental setup and sample results.

First two different panels were constructed using Alanod MIRO-SUN as the reflector namely; glass equivalent and long. See Table 4 for the material composition and thicknesses. The dimensions of the test samples are $L_{net} = 800$ mm and $W = 200$ mm (Figure 5). The plates that make up the samples were cleaned on the mating surfaces

and Ampreg resin and hardener [15] were used to glue all the layers together. Before any test were done, the samples were then left to cure for 48 hours.

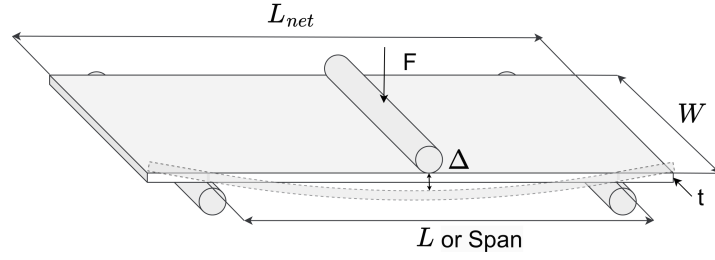


Figure 5. Three pin test dimensions.

To perform the tests, the three pin structure was attached on a 100 kN Instron universal testing machine. The sandwich panel was placed on the bottom two pins, separated by 600 mm (L) (Figure 5). The head was lowered till the top pin made contact with the test sample. The displacement was set to zero, and then the head was lowered at a rate of 0.25 mm/s. The force and displacement on the head was measured till the sample failed. This was repeated four times for each sample. An example force displacement graph from one of the tests can be found in Figure 4b. The maximum bending moment and stiffness was calculated using Equation 7 and Equation 8 respectively. The maximum moment M was calculated using the load F at failure, while stiffness K , used the force F and displacement Δ_y from the linear region of the force displacement curve. The results were averaged over all four samples for each test. The results of the physical testing are shown in Table 5.

$$M = \frac{1}{4}FL \quad (7)$$

$$K = EI = \frac{FL^3}{48\Delta_y} \quad (8)$$

6. Comparison of results

The results of the analytical model, and physical testing are compared in Table 5. The results show that the analytical model is a decent predictor of the physical maximum bending moment (M) of the sandwich panels. The stiffness (K) of the panel was correctly predicted for the 16 mm thick core panel, but the 3 mm core panel predicted a significant less stiff panel. This is likely due to a variation in material properties for a thin core, and requires further investigation. The analytical model is simpler to implement than a finite element analysis or repeated physical testing, making it a good tool for quickly comparing a large number of materials.

Table 5. Comparison of results.

Core thickness mm	Analytical				Physical test	
	M	M error	K	K error	M	K
	Nm	%	Nm ²	%	Nm	Nm ²
3	62.28	7.15	69.27	26.00	58.12	93.61
16	334.81	4.46	1588.70	7.49	320.52	1477.96

An analytical approach for evaluating and selecting structural support panels for non-glass heliostat facets was created. The model was used to efficiently screen a large

database of candidate materials using an existing glass reflector panel as a reference. Sandwich panels showed improved performance compared to single material panels. The analytical model predictions were validated through physical testing of the most promising sandwich panel designs. The most promising sandwich panel designs will be tested further in future research.

Data availability statement

Data used is based on publicly available data from Ansys Granta Edupack [9].

Author contributions

Jean Schnaar-Campbell: Conceptualization, Methodology, Software, Validation, Analysis, Investigation, Writing Original Draft, Visualization. Johann Bredell: Supervision, Writing Review & Editing. Craig McGregor: Supervision, Writing Review & Editing.

Competing interests

The authors declare that they have no competing interests.

References

- [1] W. Bank, *Concentrating solar power clean power on demand 24/7*, 2020. [Online]. Available: <https://pubdocs.worldbank.org/en/849341611761898393/WorldBank-CSP-Report-Concentrating-Solar-Power-Clean-Power-on-Demand-24-7-FINAL.pdf>.
- [2] G. J. Kolb, C. K. Ho, T. R. Mancini, and J. A. Gary, *Sandia report power tower technology roadmap and cost reduction plan*, 2011. [Online]. Available: <https://www.osti.gov/servlets/purl/1011644>.
- [3] J. Yellowhair and C. E. Andraka, "Evaluation of advanced heliostat reflective facets on cost and performance", *Energy Procedia*, 2014. DOI: [10.1016/j.egypro.2014.03.029](https://doi.org/10.1016/j.egypro.2014.03.029).
- [4] J. Coventry, J. Campbell, and C. J. Hall, *Heliostat cost down scoping study-final report ocean wave power view project vortex tube view project*, 2013. [Online]. Available: <https://www.researchgate.net/publication/312214094>.
- [5] SunDog Solar, *Everbright*. [Online]. Available: <https://www.sundogsolartech.com>.
- [6] Alanod, *Alanod miro-sun*. [Online]. Available: <https://alanod.com/en/industries/solar/reflective-surfaces>.
- [7] P. Kurup, S. Akar, C. Augustine, and D. Feldman, *Initial heliostat supply chain analysis*, 2022. [Online]. Available: www.nrel.gov/publications.
- [8] R. Budynas and J. Nisbett, *Shigley's Mechanical Engineering Design*. McGraw-Hill Education, 2015, ISBN: 9789814595285.
- [9] ANSYS Inc., *Granta edupack*, 2023. [Online]. Available: www.ansys.com/materials.
- [10] STERG, *Helio100 promises cost-effective solar power breakthrough*, Aug. 2015. [Online]. Available: https://www.sun.ac.za/english/archive/Lists/English_News_Archive_110518/DispForm.aspx?ID=2863&ContentTypeId=0x010019F8BC5373DFA740B008FC720EA25DE601008842D5DFBB60F541BF61E7750F3D6BA5.
- [11] P. Sivalingam, K. Vijayan, S. Mouleeswaran, and V. Vellingiri, "On the tensile and compressive behavior of a sandwich panel made of flax fiber and agglomerated cork", *Institution of Mechanical Engineers*, 2022. DOI: [10.1177/14644207211043227](https://doi.org/10.1177/14644207211043227).
- [12] R. J. Roark, W. C. Young, and R. G. Budynas, *Roark's formulas for stress and strain*. McGraw-Hill, 2002, pp. 435–439, ISBN: 007072542X.
- [13] Microsoft Corporation, *Microsoft excel*, version 2019 (16.0), Sep. 24, 2018. [Online]. Available: <https://office.microsoft.com/excel>.

- [14] American Society of Civil Engineers, *ASCE 7* - 22. Jul. 2022, ISBN: 0784415781.
- [15] AMT Composites, *Ampreg 30 slow*. [Online]. Available: <https://www.amtcomposites.co.za/product/ampreg-30-slow-6-3kg/>.