MODELLING OF RADIATIVE HEAT TRANSFER OF A SQUARE TRIHEDRAL DESIGN RADIATOR PANEL ON THE LUNAR SURFACE

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ABSTRACT

In this study, a radiative thermal model is developed which can be used to predict the heat transfer performance of radiator panels on the lunar surface. It includes the effects of all environmental sources of radiation experienced on the lunar surface, namely direct and indirect solar radiation, lunar albedo, and infra-red thermal heating from the lunar surface. Ray tracing is used to capture the direct solar irradiation and also to determine surface-to-surface view factors. The radiation network methodology is employed to determine the contribution from diffuse reflections.

This model is applied to a retro-reflector-like trihedral design radiator panel. A trihedral is defined as a shape with three sides which meet at a common point. This form aims to minimise the negative effects of environmental radiation while maximising the heat transfer to deep space. For the geometry and conditions studied here, the radiator panel shows good potential as an effective lunar radiator when compared to a flat plate. A change in radiative dissipative performance is observed across the lunar day when the apex angle is varied, with smaller apex angles showing greater rates of heat transfer in the morning compared to larger angles which work best at noon. The orientation of the radiator on the lunar surface was also investigated. It was found that heat transfer rate was largely unaffected by the azimuth angle, however varying the inclination angle from horizontal to vertical causes a significant reduction due to increased view factors to the lunar surface. Further work is required to fully characterise this design, including increasing the number of trihedral elements and the inclusion of thermal conduction in the model.

NOMENCLATURE

\( A \)  surface area \((\text{m}^2)\)
\( a \)  surface area of mesh face \((\text{m}^2)\)
\( D \)  lunar disc diameter \((\text{m})\)
\( F \)  view factor \((-)\)
$G_{SC}$  solar constant (1367 W/m$^2$)

$h$  hexagon side length (m)

$J$  radiosity (W/m$^2$)

$N$  total number of rays (-)

$n$  total number of rays which hit another surface (-)

$q$  heat transfer rate (W)

$q''$  heat flux (W/m$^2$)

$R$  random number (-)

$T$  temperature (K)

$\alpha$  solar absorptivity (-)

$\beta$  surface inclination angle (rad, °)

$\gamma_n$  polar angle relative to surface normal vector (rad, °)

$\varepsilon$  infra-red emissivity (-)

$\theta$  angle of incidence (rad)

$\sigma$  Stefan-Boltzmann constant ($5.67 \times 10^{-8}$ W/(m$^2$K$^4$))

$\phi$  azimuth angle relative to true North (rad, °)

$\phi_n$  azimuth angle relative to surface normal vector (rad)

$\varphi$  apex angle (rad, °)

**INTRODUCTION**

Radiator panels make up one of the most critical components of the thermal management system of almost all spacecraft$^1$. They are the final subsystem that must reject excess heat by infra-red (IR) radiation to deep space in order to keep the spacecraft operating within its thermal control limits. Radiators conventionally consist of flat, plate-like panels mounted to the side of a spacecraft with special surface coatings in order to maximise emission and minimise absorption from environmental sources, such as the Sun$^2$.

With the requirements to dissipate increasingly greater heat fluxes generated by on-board high power and miniaturised electronics$^3,4$, numerous concepts and designs have been developed in recent years to enhance radiator performance. These include improved thermo-optical coatings$^5$, variable emissivity devices using shape memory alloys$^6$, origami-based designs$^7$, or the inclusion of fractal-like geometry in order to vary the effective surface properties$^8$. This method refers to the apparent increase in absorption and emission properties of a cavity relative to a flat surface caused by multiple internal reflections$^9,10$.

In this study, an array of trihedral elements is investigated in terms of its potential as an effective passive radiator panel for spacecraft applications. Trihedral defines a type of geometric shape consisting of three faces that meet at a common point. When the three faces are orthogonal, they make up a retro-reflective corner reflector, a form widely used in optics and radar ranging applications, whereby any incident rays are returned to the source of emission$^{11,12}$. Concerning radiative heat transfer, the array offers some advantages in terms of increased surface area per
footprint area when compared to a flat plate, reduced environmental heat loads on some surfaces due to self-shading, and its exploitation of the aforementioned cavity effect.

The objective of this paper is to present the development of a model that can accurately predict the thermal performance of a trihedral design radiator, as well as some initial results. There is a particular focus on its deployment on the lunar surface as part of the thermal management system of a lunar lander or separate science package.

TRIHEDRAL GEOMETRY

The geometry of one trihedral cell is shown in Figure 1. It is defined by three sloping quadrilateral faces that meet at a common vertex, the apex point. Each of these faces has one vertex on the hexagonal base plane, and two vertices on an intermediate plane. Six triangular faces, which are perpendicular to the base, connect the points on the intermediate plane to the base plane. The geometry can be fully defined by specifying the length of one hexagon side of the base, $h$, and the apex angle, $\varphi$, defined as the angle between two edges connected to the apex point. The area of each quadrilateral surface is defined by:

$$A = \frac{3h^2}{2\tan\left(\frac{\varphi}{2}\right)}$$ (1)

To create the radiator panel in this study, the trihedral cells are arrayed in a circular pattern, with one central element surrounded by six others, as shown in Figure 2. This generates a number of cavities between cells which can be different depths depending on the value of $\varphi$. 

![Figure 1. Trihedral Radiator Geometry](image)
Figure 2. Trihedral radiator geometry with seven cells for (a) $\phi = 60^\circ$, (b) $\phi = 90^\circ$, (c) $\phi = 110^\circ$.

MODEL DESCRIPTION

As outlined in the introduction section, this study focuses on modelling the thermal performance of the trihedral radiator geometry when it is placed on the lunar surface. In this study, the landing site of the Apollo 17 mission has been selected. The lunar surface is represented in the model as a simplified horizontal flat disc with diameter $D$, the value of which will be discussed later. All faces in the model are discretised into a surface triangular mesh by the software TetGen v1.5\textsuperscript{13}. Note that only the quadrilateral surfaces participate in the radiative exchange, with the small triangular surfaces considered to be perfectly transparent. The thermal model is developed using Python v3.5.

Thermal Model

In order to determine the thermal heat transfer performance of the radiator geometry, a radiative heat balance is performed between all the participating surfaces with the surrounding environment. The radiator must dissipate all heat sources in order to keep the spacecraft within its thermal control limits. These include direct solar irradiation, indirect scattered and reflected solar radiation (albedo), internal heat generated by electronics, and other sources of IR thermal radiation, such as the lunar surface\textsuperscript{2}. Treating the solar and IR sources independently, the net radiation leaving a participating surface $i$ is the difference between outgoing and incoming radiation, given by:

$$ q_{net,i} = \left(q_{solar_{out,i}} + q_{IR_{out,i}}\right) - \left(q_{solar_{in,i}} + q_{IR_{in,i}}\right) \quad (2) $$

where $q$ is the rate of radiative heat transfer. Conduction between surfaces is not included in the current model.

The net solar radiation is the total absorbed from both direct and indirect solar sources,
where $\alpha$ is the solar absorptivity of the surface. The radiator surface coating values of $\alpha$ and $\varepsilon$, the emissivity, are given values consistent with optical solar reflectors (OSR) equal to 0.12 and 0.8 respectively.\(^\text{14}\) The intensity of direct solar irradiation is a function of Sun's angular position above the horizon, and the orientation of that surface with respect to the ground. In the horizon reference frame,\(^\text{15}\) the angular position of the Sun can be determined from the location on the lunar surface in terms of latitude, longitude and declination of the Moon relative to the celestial equator. For a given solar ray, the direct incident solar heat flux is calculated from:

$$q_{solar_{direct}} = G_{SC} \cos \theta$$

(4)

where $q''$ is the heat flux, $G_{SC}$ is the solar constant, and $\theta$ is the angle of incidence of the solar ray onto the surface. At the Earth's mean distance from the Sun, $G_{SC}$ has a value equal to 1367 W/m\(^2\). To account for solar shading between the radiator surfaces, eq. (4) is applied to each face in the mesh. Using ray tracing, if it is found that a meshed faces view to the Sun is blocked by another face, $q''_{solar_{direct}}$ is set to 0. The total direct incident radiation on a radiator face is then:

$$q''_{solar_{direct,i}} = \frac{1}{A_i} \sum_{k} q''_{solar_{direct,k}} a_k$$

(5)

where $k$ is the number of mesh faces on a radiator surface $i$, and $a$ is the area of a single face in the mesh.

The indirect solar radiation reflected and absorbed by surfaces is assumed to be diffuse and to follow Lambert's Law. Furthermore, the surfaces are considered to be ideal grey bodies. Then, applying the radiation network methodology,\(^\text{16}\) the net reflected exchange between the $i^{th}$ surface and all other $j$ surfaces can be determined from:

$$J_i (1 - F_{ii}) - \sum_{i \neq j} F_{ij} J_j = (1 - \alpha_i) q''_{solar_{direct,i}}$$

(6)

where $J$ is the radiosity, $F$ is the view factor, and $\sigma$ is the Stefan-Boltzmann constant. For the lunar surface, the value of $\alpha$ is calculated from the average albedo reflectivity value of 0.12.\(^\text{17}\) Eq. (6) is applied to each surface, and then solved simultaneously for the values of $J$. The total indirect solar heat flux arriving at surface $i$ can then be calculated from:

$$q''_{solar_{indirect,i}} = \sum_{i \neq j} F_{ij} J_j$$

(7)

The IR heat radiated between solid surfaces, both emitted and reflected, follows the same assumptions given above for the solar reflections. The net IR heat transfer is calculated from:
where $\varepsilon$ is the emissivity and $T$ is the temperature. As before, applying the radiation network methodology, the resulting equation takes a different form to that of eq. (6) and is given by:

$$J_i[1 - F_{ii}(1 - \varepsilon_i)] - (1 - \varepsilon_i) \sum_{l \neq j} F_{ij}J_j = \varepsilon_i \sigma T_i^4$$

(9)

For the radiator surfaces, the value of $T$ in eq. (9) is defined by a maximum allowable temperature, which in this case is set to 313 K. For the lunar surface, the value of $T$ varies as a function of location on the surface and lunar hour\textsuperscript{17}. The surface temperature variation for the coordinates of the Apollo 17 landing site is plotted in Figure 3.

Solving eq. (9) for the values of $J$, the value $q_{IR_{net,i}}$ in eq. (8) can be calculated. Finally, the net radiation leaving a surface in eq. (2) can be determined from:

$$q_{net,i} = q_{solar_{net,i}} + q_{IR_{net,i}}$$

(10)

Figure 3. Lunar surface temperature at the Apollo 17 landing site for one lunar day.
Radiation View Factors

A Monte Carlo method is applied in order to calculate the view factors $F_{ij}$ between participating surfaces in eq.s (6), (7) and (9). This method relies on random repeated experiments in order to estimate the real solution\textsuperscript{18}. The ray tracing algorithm used here generates $N$ rays originating from the centroid of each mesh face, the hemispherical distribution of which is defined by Lambert's cosine law. The directions of the rays relative to the surface normal are defined by:

$$\gamma_n = \sin^{-1} \sqrt{R \gamma_n}$$  \hspace{1cm} (11)

$$\phi_n = 2\pi R \phi_n$$  \hspace{1cm} (12)

where $\gamma_n$ and $\phi_n$ are the polar and azimuth angles, and $R$ is a random number in the range $[0,1]$. The algorithm then traces each ray to find the closest surfaces that they hit. The view factor is therefore determined by:

$$F_{kj} = \frac{n_{kj}}{N_k}$$  \hspace{1cm} (13)

where $n$ is the number of hits from surface $k$ to $j$. If a ray hits no surface, it is included in the determination of the view factor to deep space. For a single radiator surface,

$$F_{ij} = \frac{1}{A_i} \sum_{k}^k F_{kj} a_j$$  \hspace{1cm} (14)

RESULTS AND DISCUSSIONS

In the following section, results will firstly be presented for a number of verifications and validations of the modelling approach. Then, some initial results of the trihedral design radiator will be presented, benchmarked against a flat plate radiator.

Model Validation

A number of different checks were performed during model development. They are outlined in the following sections.

**Radiation View Factors**

The methodology described previously was validated against two cases for which analytical solutions exist for determination of view factors. They are:

- Perpendicular rectangles with a common edge.
- Parallel, equal, directly opposed rectangles.
The analytical solutions can be found in the relevant textbooks\textsuperscript{19}. A comparison between the Monte Carlo method and the analytical solution for the two cases are shown in Figure 4 and Figure 5. A value of $N = 1000$ was used for different aspect ratios. There is excellent agreement in both cases, with an average difference of 0.5\% and 0.2\% respectively.

**Figure 4.** Comparison of view factor results for perpendicular rectangles with a common edge for four different aspect ratios. Results shown are for view factor from horizontal to vertical face.

**Figure 5.** Comparison of view factor results for the parallel, equal, directly opposed rectangles for four different aspect ratios. Results shown are for view factor from bottom to top face.
**Lunar Disc Diameter**

As mentioned earlier, the lunar surface is included in the model as a simplified flat horizontal disc with diameter $D$. The size of $D$ will have an effect on the view factors from the radiator surfaces to the lunar surface.

The influence of $D$ on the view factors for surfaces 1, 2 and 3 is shown in Figure 6. The value for $D$ is increased up to the ideal limit for the distance to the horizon, calculated using the height of the geometry from the lunar surface and the lunar equatorial radius of 1738 km. It can be seen that the values of $F$ remain approximately constant when $D \geq 200$ m.

![Figure 6. Effect of lunar disc diameter on view factor $F_{l,lunar}$ for surfaces 1-3.](image-url)
**Thermal Model Comparison with Commercial Software**

Two pieces of commercial software which can model radiative heat transfer between surfaces, as well as direct solar radiation, were used to generate results for the trihedral radiator with $\varphi = 90^\circ$ (as shown in figure Figure 2(b)). The software used were:

- **SYSTEMA/THERMICA v4** by Airbus. This is a thermal analysis package for space missions and computes external fluxes, radiative and conductive dependencies.
- **FloTHERM XT v3.3** by Mentor. This is a CAD-centric thermal solution tool intended for use during all design stages of electronics.

The geometry was sitting horizontally on the lunar surface, with surfaces 18 and 21 facing south as shown in Figure 7.

![Meshed radiator geometry](image)

**Figure 7.** Meshed radiator geometry for $\varphi = 90^\circ$ and $h = 9.5$ mm sitting on lunar surface. Labels highlight the surface numbering convention used.

Figure 8 and Figure 9 show the results of the direct incident solar heat flux for the 21 individual radiator surfaces for three different times during the lunar day: 06:30, 09:00 and 12:00. Figure 8 shows the comparison of the three different models, while Figure 9 visualises the results of the developed model on the radiator surfaces.
Figure 8. Comparison of direct incident solar heat flux for radiator surfaces for the 3 models at 3 different times: (a) 06:30, (b) 09:00, (c) 12:00.

Figure 9. Surface plots of direct incident solar heat flux for radiator surfaces at 3 different times: (a) 06:30, (b) 09:00, (c) 12:00.

At the earliest time (Figure 8(a) and Figure 9(a)), surfaces facing the rising Sun receive the highest $q''_{solar/direct}$, i.e., surfaces 5, 14 and 20. A large proportion of faces are in shade due to their orientation away from the Sun (surfaces 1, 4, 7, 10, 16 and 19), while the rest receive a lower value of $q''_{solar/direct}$ due to their orientation and shading from other surfaces. As the sun continues to rise into the lunar morning (Figure 8(b) and Figure 9(b)), the partially shaded faces receive more $q''_{solar/direct}$, however surfaces 1, 4, 7, 10, 16 and 19 remain in shadow. Finally at noon, where the Sun reaches its highest point overhead (Figure 8(c) and Figure 9(c)), all surfaces are now illuminated. The surfaces facing south (3, 6, 9, 12, 15, 18 and 21) receive the highest $q''_{solar/direct}$, whereas all remaining surfaces receive a lower, equal, amount. It is important to note that the different models in Figure 8 all predict very similar values of $q''_{solar/direct}$ across the different times. The root-mean-square-errors (RMSE) between THERMICA and the model developed here are 7.96, 7.27 and 8.44 W/m$^2$ for 06:30, 09:00 and 12:00 respectively.
Figure 10. Comparison of direct and indirect absorbed solar heat flux for radiator surfaces for the 3 models at 3 different times: (a) 06:30, (b) 09:00, (c) 12:00.

Figure 11. Surface plots of direct and indirect absorbed solar heat flux for radiator surfaces at 3 different times: (a) 06:30, (b) 09:00, (c) 12:00.

Figure 10 and Figure 11 show the results for the combined direct and indirect solar flux that is absorbed by each radiator surface, again at three different times during the lunar day: 06:00, 09:30 and 12:00. At the earliest time (Figure 10(a) and Figure 11(a)), a similar trend is observed to that seen previously, whereby the faces orientated towards the rising Sun have the highest absorbed $q''_{solar}$. Surfaces 1, 4, 7, 10, 16 and 19 are no longer 0 W/m$^2$ because of both solar reflections between other radiator surfaces and the lunar albedo. Surfaces which have a view factor only to the lunar surface are affected the least by this (1 and 7) compared to others which have both view factors to the lunar surface and other radiator surfaces (4, 10, 13, 16 and 19). The effect of reflections is also observed at 09:00 (Figure 10(b) and Figure 11(b)). Surface 20 has the highest values of absorbed $q''_{solar}$ because it is orientated towards the Sun and has view factors to both the lunar surface and other radiator surfaces. Surfaces 5 and 14, which were equal to 20 before (Figure 8(b)), only have a view factor to the lunar surface. Similarly at 12:00, surfaces which are facing south with view factors to many radiator surfaces (3, 6 and 12) have higher $q''_{solar}$ compared to those with view factors to fewer radiator surfaces (9 and 15), or only the...
lunar surface (18 and 21). Again, in all cases, the models show very good agreement, with RMSE values between THERMICA and the model developed here equal to 2.87, 3.22 and 1.65 W/m² for 06:30, 09:00 and 12:00 respectively.

Radiative Heat Transfer of Trihedral Radiator

This section examines the radiative heat transfer of the trihedral design radiator when a number of parameters in the model are varied. These are namely the apex angle (see Figure 2), azimuth angle and surface inclination angle.

The azimuth angle defines the orientation of the panel with respect to true North, as shown in Figure 12. In effect, this is a rotation of the geometry about the z-axis, where \( \phi \) is positive in the clockwise direction. The inclination angle \( \beta \) of the panel is the angle its base makes with respect to the lunar surface (see Figure 12). This is equivalent to a rotation about the x-axis, where the values are positive in the counter-clockwise direction.

![Diagram of orientation of arbitrary radiator panel with respect to lunar surface.](image)

**Figure 12. Orientation of arbitrary radiator panel with respect to lunar surface.**

**Effect of Apex Angle**

The effect of increasing \( \phi \) results in increased \( A \), shading between surfaces and improved view factor to deep space. However, there must then be also an increased view factor to the lunar surface, so a trade-off is required in order to achieve optimal performance. Figure 13 shows the results for the total radiative heat transfer from the 21 radiator surfaces in this model with the panel sitting horizontally on the lunar surface. The results are compared to a flat plate, which is equivalent to \( \phi = 120^\circ \).

In the lunar morning and evening, it can be seen that the trihedral designs can dissipate more heat than the flat plate. However, there is an intersection point where this trend is reversed, and
the flat plate out-performs the trihedral radiators around noon at the worst case scenario. Even though the incident solar radiation is lower for the trihedral design at this time, the view factors and increasing lunar surface temperature results in this decreased performance.

By integrating to find the areas under the curves in Figure 13, the total energy radiated from the surfaces across the lunar day can be determined. These values are found to be 7.57, 7.13, 6.87 and 6.58 Wh for $\phi$ of 60°, 90° and 110°, and the flat plate, respectively.

The results for the effect of apex angle are summarised in Table 1. The percentage differences between the trihedral radiator for a given value of $\phi$ and the flat plate radiator are presented for the best and worst cases where they are subjected to the minimum and maximum environmental heating, as well as the comparison of the total energy radiated across the lunar day.

![Figure 13](image)

**Figure 13.** Total radiative heat transfer over lunar day for trihedral radiator compared to flat plate for different values of $\phi$. Panel base is sitting horizontally on lunar surface.

**Table 1. Summary of Apex Angle Results for Percentage Difference between Trihedral and Flat Plate Radiator**

<table>
<thead>
<tr>
<th>$\phi$</th>
<th>Best case (sunrise/sunset)</th>
<th>Worst case (noon)</th>
<th>Full day</th>
</tr>
</thead>
<tbody>
<tr>
<td>60°</td>
<td>+84.2%</td>
<td>-40.2%</td>
<td>+15.0%</td>
</tr>
<tr>
<td>90°</td>
<td>+31.7%</td>
<td>-11.8%</td>
<td>+8.4%</td>
</tr>
<tr>
<td>110°</td>
<td>+8.8%</td>
<td>-0.6%</td>
<td>+4.4%</td>
</tr>
</tbody>
</table>
**Effect of Azimuth Angle**

Figure 14 shows the results of varying the angle of $\phi$ from 0° to 180°. It can be seen that there is little variation in the total radiative power for different values of $\phi$. While local variations exist across the surfaces, the total radiative power for all 21 surfaces remains approximately the same at each time. The flat plate is unaffected by $\phi$.

![Figure 14. Total radiative heat transfer over lunar day for trihedral radiator ($\varphi = 90^\circ$) compared to flat plate for different values of $\phi$. Panel base is sitting horizontally on lunar surface.](image)

**Effect of Inclination Angle**

The value of $\beta$ is varied from 0°, i.e. horizontal, up to a value of 90°, i.e. vertical. This results in surfaces 18 and 21 closest to the ground (see Figure 7). The panels are also raised by 1 m from the lunar surface to allow for these rotations, and to emulate the panel placed on the top/side of a lunar lander. The results are shown in Figure 15 for a fixed apex angle of $\varphi = 90^\circ$.

For $\beta = 0^\circ$, the results are identical to those presented in Figure 13. Note the similar trend compared to the flat plate, where the trihedral design outperforms it during the morning and evening, with intersection points occurring around noon. This trend is repeated for the other two values of $\beta$, however the total radiative power is negatively impacted as the panel rotates towards vertical. In these cases, the view factor to the lunar surface increases resulting in an overall reduction in performance. In the vertical case ($\beta = 90^\circ$), both radiators can no longer dissipate any heat around noon, indicated by net negative radiative power values.

As before, through integration, the total energy dissipated for the trihedral radiator across the lunar day is equal to 7.14, 5.39 and 2.22 Wh for 0°, 45° and 90° respectively.
The results for the effect of inclination angle are summarised in Table 2. The percentage differences between the trihedral and flat plate radiators for a given value of $\beta$ are presented for the best and worst cases where they are subjected to the minimum and maximum environmental heating, as well as the comparison of the total energy radiated across the lunar day.

![Graph showing total radiative heat transfer over lunar day for trihedral radiator ($\varphi = 90^\circ$) compared to flat plate for different values of $\beta$ (horizontal = 0°, vertical = 90°).]

Table 2. Summary of Surface Inclination Angle Results for Percentage Difference between Trihedral and Flat Plate Radiator

<table>
<thead>
<tr>
<th>$\beta$</th>
<th>Best case (sunrise/sunset)</th>
<th>Worst case (noon)</th>
<th>Full day</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>+31.5%</td>
<td>-11.8%</td>
<td>+8.4%</td>
</tr>
<tr>
<td>45°</td>
<td></td>
<td>-28.3%</td>
<td>+8.8%</td>
</tr>
<tr>
<td>90°</td>
<td></td>
<td>-96.2%</td>
<td>+2.7%</td>
</tr>
</tbody>
</table>

CONCLUSIONS

A modelling approach for studying the radiative heat transfer from panels on the lunar surface has been presented. This model includes all sources of irradiation which a spacecraft would experience in this environment, including direct and indirect solar radiation, lunar albedo, and IR thermal radiation. Reflections between surfaces are captured using the radiation network methodology. The developed model shows excellent agreement to commercially available software.
This model was applied to a trihedral design radiator, similar in shape to retro-reflectors used in optical and radar industries. It is built from 7 elemental cells, with a total of 21 radiating surfaces. The trihedral design aims to minimise the effects of environmental radiation sources and increase the surface area visible to deep space for cooling. It has been shown that, in the conditions studied here, this design offers good potential as an effective radiator panel for lunar applications. The effect of decreasing the vertex angle to 60° results in the maximum amount of energy dissipated across the lunar day, a potential improvement of 15.0% compared to a flat plate, but it also suffers the largest heat transfer performance reduction at noon. The surface azimuth angle was found to have little impact on performance. When changing the inclination angle, a reduction in radiative dissipation was observed as the panel rotates from horizontal to vertical position. However, it was still found to dissipate more energy across the lunar day when compared to the flat plate, with a maximum heat transfer enhancement of 8.8%.

Further work is required to fully characterise the trihedral design. This includes studying the effect of an increased number of cells used in the panel -- this may further reduce lunar albedo and IR radiation for central elements in an array. The addition of thermal conduction in future models would also significantly improve prediction of real world behaviour. It is expected that conduction will result in significant smoothing of discontinuities between connecting faces and lead to some spatial integration across the entire array.

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