Fabrication and Testing of a Grid-Stiffened Panel with Integrated Thermal Control

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This paper presents the fabrication process and testing of a multifunctional thermal/structural panel that is intended to reduce the thermal limitations of spacecraft performance. The panel, inspired by the circulatory system of biological organisms, integrates cooling channels into a rib-stiffened structure in such a way as to maintain the stiffness-to-mass ratio of the original panel. The integration of these channels necessitates dramatic changes to the traditional approach of fabricating grid-stiffened structures, and a novel method of fabrication is discussed. A prototype panel has been fabricated using rapid-prototyping methods, and its thermal performance evaluated. Preliminary results of this testing are presented, including the pump power required to supply various flow rates through the panel and the temperature distribution of the panel in response to an applied heat load.

Introduction

THERMAL effects are beginning to limit the performance of today’s satellites, and it is anticipated that future satellites will face even more difficult thermal challenges. It is key for expanding the capability of future satellites that new thermal control concepts be developed to address these challenges in an efficient, affordable, and modular manner. One promising concept to advance thermal control systems is the use of multifunctional structures, in which a single component may perform many system functions. This paper presents a method of fabricating a grid-stiffened composite panel in which fluid channels are integrated. Thus, the panel is symbiotic in that it not only provides structure for the satellite, but also provides a mounting surface for electronic components and a powerful, modular thermal control system utilizing convection. This thermal control system is integrated in such a way as to preserve the original stiffness-mass ratio of the panel, and is designed for high reliability. The objective of the current study is to fabricate the symbiotic panel and determine its thermal performance.

The grid-stiffened symbiotic panel described in this paper was inspired by the circulatory system of biological organisms. The circulatory system provides thermal regulation, distributes oxygen, promotes self-healing, and improves the physical properties of structural tissue, all of which may be of interest in space vehicles. Incorporating an analogous system into the structure of a satellite could provide significant performance gains if it can be done without significantly increasing system mass. For the panel described in this paper, the geometry of a conventional grid-stiffened panel† is modified to incorporate supply channels into the panel ribs (analogous to arteries and veins in a circulatory system). These supply channels can transport high flow rates of fluid to a network of distribution channels embedded below the face sheet of the panel.

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(analogous to capillaries) at a relatively low pressure drop. The distribution channels provide efficient heat transfer away from (or in some cases, toward) electronic components mounted on the face sheet. The control of fluid through the supply and distribution channels is managed by a network of pumps and valves (analogous to the heart and valves found in biological organisms). The key elements of the panel are illustrated in Fig. 1, along with some possible distribution channel configurations.

The design and structure of a representative symbiotic panel was first presented by Williams, Arritt, and Lyall. Williams et al. designed an aluminum version of the panel and showed through a simplified beam analysis and finite-element modeling that it is possible to integrate fluid channels into the ribs of a grid-stiffened structure while maintaining the original stiffness-mass ratio of the panel in bending and torsion. This was achieved by converting the solid ribs used on a conventional panel to wider, taller hollow ribs, while maintaining the rib’s original material cross-sectional area. This increased the bending and torsional moments of inertia and at the same time created an area in the rib through which fluid can flow.

Lyall et al. conducted thermal performance testing on the first prototype of this aluminum symbiotic panel, and found that under the conditions investigated with water as the working fluid, the panel experienced a temperature rise of about 7 K over the heated area under a 70 W heat load and required an input power of only 0.02 W. Obviously, water is in general unsuitable for space applications, and the performance of the panel using a space-rated working fluid may

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**Fig. 1** Schematic of the symbiotic panel concept illustrating the key components and some possible distribution channel configurations. Views of (a) back of the panel and (b) front of the panel with the face sheet removed.
Regardless, these results are encouraging and suggest that the panel may be a very effective thermal control solution.

A composite version of the symbiotic panel was described by Williams et al.\textsuperscript{5} Diagrams of this panel are shown in Fig. 2. While the general design of the composite panel is similar to that of the aluminum panel, some key changes were made to strengthen the panel and adapt it for composite fabrication. The most notable of these is in the design of the ribs and the junction between the ribs and the face sheet. For the aluminum panel and conventional composite isogrid panels, this junction forms a sharp corner, as illustrated in Fig. 3(a). However, the sharp corner creates a stress concentration, shown by Higgins et al.\textsuperscript{6} to be the weak point in these structures. Filleting this junction, as shown in Fig. 3(b), has two effects: it eliminates this stress concentration and it creates a void through which fluid can flow. The optimal radius of curvature of this fillet was shown to be 7.6 mm, based on a balance of the thermal performance, power required to pump the working fluid, and deflection of the face sheet due to pressurization of the working fluid.\textsuperscript{5}

Using this panel design, Williams et al.\textsuperscript{7} conducted a computational thermal analysis on the composite symbiotic panel. Both the flow through the supply channels and the temperature distribution of the panel in response to an applied heat load of 70 W were studied. Applying the heat load over a 152-mm x 152-mm area in the center of the panel resulted in a temperature rise of about 13 K, compared to a 7 K temperature rise measured by Lyall et al.\textsuperscript{4} for the aluminum version of the symbiotic panel for the same flow rate and heat input. It was expected that the predicted temperature rise of the composite panel would be greater than that measured for the aluminum panel, as aluminum has a higher thermal conductivity than carbon fiber and the distribution channels were not modeled for the composite panel (whereas a serpentine distribution channel was embedded directly under the heater on the aluminum panel). Regardless, the predicted temperature rise across the composite symbiotic panel was considerably less than predicted for the panel without the integrated cooling channels.

Given the promising results of both Lyall et al.\textsuperscript{4} and Williams et al.\textsuperscript{7}, additional development work has been conducted. This paper presents the techniques used to fabricate flight-like

\textbf{Fig. 2 Three views of the design of the composite symbiotic panel: (a) front view, (b) front view with the face sheet removed, and (c) back of the panel.}
versions of the panel, as well as the results of thermal performance conducted on a prototype panel. The paper is divided into 4 additional sections. The next section, entitled Composite Symbiotic Panel Fabrication, discusses the details of the procedure used to fabricate a composite multi-functional panel. The third section, Thermal Testing Experimental Methods, explains the experimental setup and the methods used to determine the thermal response of a prototype version of the panel. The fourth section, Thermal Testing Results, presents some preliminary results of the thermal testing. The final Conclusions and Future Work section reviews the important results obtained from the thermal testing of the prototype panel and discusses the future work being planned for the development of the panel.

Composite Symbiotic Panel Fabrication

As the incorporation of fluid passageways into the ribs of a composite grid-stiffened panel complicates the geometry considerably, especially with the use of filleted rib-face sheet junctions, the techniques used in the fabrication of traditional composite grid-stiffened structures must be heavily modified. This section presents the techniques used to construct the composite symbiotic panel.\textsuperscript{§}

The fabrication tooling consists of three assemblies: a base tooling fixture, a set of expansion tooling blocks, and a cover plate. The panel itself is made of a composite prepreg and a series of aluminum plugs (the function of these plugs will be described later in this section). Each of these components is illustrated in Fig. 5. The general assembly process is to drape the prepreg over the expansion tooling blocks, assemble the prepreg/block assembly into the base assembly, and then bolt the cover plate to the base assembly for curing. This process will now be discussed in additional detail.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig3.png}
\caption{Schematic of the junction between the ribs and face sheet of (a) a traditional grid-stiffened panel and (b) the composite symbiotic panel discussed in the current study.}
\end{figure}

The expansion tooling consists primarily of a set of solid silicon rubber blocks, most of which are triangular in shape (Fig. 4). Composite prepreg is draped over each of these blocks, so that it covers all but one of the sides (the bottom side as shown in Fig. 4). The prepreg is then debulked

to get it as close to the final thickness as possible. The expansion tooling blocks (with the prepreg) are then assembled such that the prepreg has the desired geometry and placed into the base tooling fixture. A planform view of the assembly is shown in Fig. 6. Aluminum plugs (also shown in Fig. 6) are used to seal voids created by the filleted corners of the expansion tooling blocks, and may be used as mounting locations for valves or pumps that control the flow of fluid through the supply channels.

The base tooling fixture is made of aluminum, which has a low coefficient of expansion compared to the silicon rubber expansion tooling blocks. When the cover plate is bolted to the base tooling fixture (Fig. 5) and the assembly is heated for curing, the expansion tooling tries to expand much more than allowed by the base tooling, compressing the composite prepreg. This ensures good compaction of the ribs and a geometry reasonable close to that which is desired. Unfortunately, thermal expansion of the base tooling fixture makes it difficult to exactly obtain the desired geometry, so a very low coefficient of thermal expansion material should be used. To fabricate the prototype panel, aluminum is used for its good machineability; this is considered to be an acceptable tradeoff for its higher-than-desired coefficient of thermal expansion.

![Fig. 5 Exploded view of tooling used to fabricate a prototype of the composite symbiotic panel. The panel is upside-down in this view.](image)

![Fig. 4 Illustration of some of the expansion tooling blocks used in the fabrication of the prototype composite panel.](image)
Another issue with the use of aluminum as the base tooling material is that it has a higher coefficient of thermal expansion than does carbon fiber. During cure, the base fixture expands and the composite prepreg cures in this slightly deformed geometry. During the post-cure cooldown, the base fixture contracts back to its original geometry, but the cured composite panel contracts only slightly. This difference in thermal expansion between the cured panel and the base fixture causes thermal stresses on the panel that may cause it to crack. To account for this, the expansion tooling blocks are designed to completely envelop the panel (shown in Fig. 6), providing a buffer zone that cushions the panel, substantially reducing the thermal stress that the panel experiences and preventing cracking.

![Fig. 6 Planform view of a corner of the expansion tooling blocks and composite prepreg installed in the base tooling fixture.](image)

The final stage in the fabrication of the panel is assembly of the face sheet. The face sheet consists of multiple layers. Distribution channels are machined into the top of the lower layers and sealed with an upper layer. The upper layer also provides a smooth surface to which electronic components can be mounted. The entire face sheet assembly is then bonded to the rib structure of the panel. For purposes of testing the thermal performance of the panel, an adapter plate was machined so that a conventional pump could be used to control the flow of working fluid through the panel. In later versions of the panel, electrohydrodynamic (EHD) pumps will be integrated directly into the supply channels. EHD pumps have potentially increased reliability over conventional mechanical pumps, as they have no moving parts.

**Thermal Testing Experimental Methods**

This section describes the experimental methods used to determine the thermal and fluid properties of a prototype of the multi-functional panel. The prototype was fabricated using the Stereolithography (SLA) rapid-prototyping process (Fig. 7). The material used, Accura 55, possesses neither the structural nor the thermal properties of the flight-like composite material. However, thermal testing on this panel is still useful for three reasons. First, fluid flow through the panels will be nearly identical for any material used, so this testing provides valuable results regarding the power required to operate the panel at a given flow rate. Second, both materials
have sufficiently low conductivity that the primary mode of heat transfer through the system will be from fluid convection, rather than solid conduction, so relatively small variations in the conductivity of the material should have only a small effect on the thermal performance of the panel. Third, the nearly isotropic properties of the prototype panel will simplify thermal analysis and provide a good starting point from which to validate a thermal/fluid model of the system. Note that the panel prototype has only primary-supply and sub-supply channels. Distribution channels were not included as a simplified model was desired for thermal model validation and a process for producing small diameter channels using SLA is still in development.

![Prototype panel](image)

**Fig. 7 (a) Top and (b) bottom views of a prototype of the multi-functional panel. In the top view, a heater is shown installed in the center of the panel. For clarity, the panel is shown disconnected from the fluid loop and prior to the application of black spray paint.**

A schematic of the test setup is shown in Fig. 8. The working fluid in this experiment was Novec 7600. It was selected for its large liquid range and favorable dielectric properties (with respect to EHD conduction pumping). It is also a low global warming, zero-ozone-depletion potential refrigerant, and so is not as affected by some of the issues that may make many other refrigerants unavailable for use in future systems. Flow through the panel was supplied from a Polyscience model 9106 refrigerated bath with a built-in mechanical pump. The pump fed two loops: a bypass loop and a loop containing the panel and associated instrumentation. The bypass loop was used during installation to avoid fluid hammer in the instrumentation during startup, and during operation to allow very low flow rates through the panel without damaging the pump. The relative flow rates through the bypass and panel loops were controlled using a bypass valve in the bypass loop and a control valve in the line feeding the panel. The rate of flow through the panel was measured using an Omega FTB-1312 turbine flow meter, and the pressure drop across the panel was recorded with an Omega PX2300 series pressure transducer. The inlet temperature of the working fluid ranged from 5 deg. C at higher flow rates to 10 deg. C for lower flow rates. A heat load was applied to the center 152-mm x 152-mm of the panel using a Kapton resistive heater powered by a Lambda GEN150-10 DC power supply. The temperature of the flow into and out of the panel was measured using thermocouples installed on the outer walls of the stainless steel supply tubing. The error introduced using this method rather than submerging instrumentation directly in the supply tubing was small, as the stainless steel tubing had a wall thickness of only 0.005-inches. Six RTDs were installed on the surface of the panel; the locations of these RTDs is shown in Fig. 8. During testing, the bottom and top surfaces of the heater were insulated using plywood and polyethylene foam insulation to limit the effects of natural convection.
In addition to the instrumentation described above, a FLIR systems model SC6000 infrared camera was used to provide improved spatial resolution of the surface temperature distribution of the panel when subjected to a heat load. While IR thermography provides primarily a qualitative measurement of temperature, attempts were made to provide more quantitative data. The panel was painted with Krylon 1602 Ultra-Flat Black spray paint, which helps to minimize background reflections and has a known, well-characterized emissivity. To obtain IR images, the system was brought to steady state under an applied heat load, flow rate, and inlet temperature, the top surface insulation was removed, and the image was quickly taken before the temperatures changed significantly. After the final image was acquired, the top layer of insulation was left off and the system was allowed to come to room temperature, at which point a background image was taken. By subtracting this background image from the original image, any background radiation that was reflected in the original image was removed. The images were then calibrated using known temperatures from thermocouples installed in the center of the panel and near the panel inlet.

![Fig. 8 Schematic of the thermal performance test setup. The panel is shown without the top face sheet for reference.](image)

**Thermal Testing Results**

This section presents the performance of the multi-functional panel from a thermal/fluid perspective. The range of flow rates selected for these tests was based on a thermal model of the panel constructed in Cullimore and Ring Technology’s Thermal Desktop and solved with Sinda/Fluint, which uses finite difference methods to solve the governing thermal/fluid differential equations. Although this model was more representative of the composite version of the panel than the prototype panel, the trends should hold for the prototype panel. The plot in Fig. 9 shows that the temperature difference between the hottest and coldest points on the panel...
decreases rapidly with increasing flow rate for low flow rates, but is nearly constant at higher flow rates. Thus, a point of diminishing returns is reached around a flow rate of 1 L/min. However, power increases rapidly with flow rate at all flow rates, suggesting reduced efficiency (in terms of thermal performance per unit of power expended) at very high flow rates. Thus, flow rates near 1 L/min were selected for the experimental testing.

![Graph showing power required and temperature difference against flow rate.](image)

**Fig. 9** Thermal model results for the composite version of the multi-functional panel showing the estimated pump power required to drive flow through the panel and the resulting difference in temperature between the hottest and coldest points on the panel surface.

The power required to drive flow through the prototype panel was calculated from the product of the volumetric flow rate and the pressure drop through the panel, both of which are directly measured quantities, and is presented in Fig. 10. Note that these data are presented using linearly-scaled axes rather than logarithmic, as the range of flow rates tested was smaller than that modeled in Fig. 9. At very low flow rates, the panel requires only minimal power, on the order of 1/20th of a milliwatt. Pump power required increases rapidly with increasing flow rate, but is still only about 10 mW at a flow rate of 1 L/min. Even at a typical EHD pump efficiency of 5%, only 0.2 W of power would be required to drive the panel with this flow rate in this configuration. Results obtained using a fluid model of the system are also presented in Fig. 10. This model was constructed using Thermal Desktop’s FloCAD add-on software package, which allows for a geometric representation of the channel network to be constructed using a lumped-fluid model, and solved using Sinda/Fluint. Agreement between the experiment and model results was good, as it was within the experimental uncertainty of the measurements.
In addition to the pressure loss associated with the flow of fluid through the panel, the temperature rise of the fluid between the panel inlet and outlet was measured when subjected to varying heat loads. The plot shown in Fig. 11 shows this temperature rise for varying flow rates and an input heat load of 25 W. As would be expected, at low flow rates the fluid temperature rise is relatively large, but diminishes rapidly as the flow rate increases. This high temperature rise is due to the decreased time each unit volume of fluid spends in contact with the panel. As the flow rate increases, fluid moves through the panel more quickly and each unit volume of fluid removes less heat energy. At higher flow rates the temperature rise decreases much more gradually as flow rate is increased. Two other curves are also shown in Fig. 11: results from the fluid model and the analytically-calculated fluid temperature rise. The fluid model used to obtain these data was the same as that used to determine required pump power in Fig. 10. The apparent discrepancy in the model and experiment results is likely due primarily to uncertainty in the thermocouple measurements, which was approximately 1 deg. C. per thermocouple for a total uncertainty of 1.4 deg. C in the temperature rise data. Work is currently underway to reduce this uncertainty and obtain a better comparison. The final curve in Fig. 11 shows the analytically-calculated fluid temperature rise, obtained from eq. 1 below, and agrees well with the fluid model results.

\[ q = \dot{m} C_p \Delta T \]  

*Fig. 10 Pump power required to drive flow through the panel at various flow rates.*
IR thermography was used to determine the temperature distribution of the panel face sheet at relatively high spatial resolution, as explained in the Thermal Testing Experimental Methods section. A representative image taken at a flow rate of 1.5 L/min under an applied heat load of 25 W and an inlet fluid temperature of 5.8 deg. is shown in Fig. 12. The paths of the primary-supply and sub-supply channels are clearly visible from the face sheet temperature distribution, due to the low thermal conductivity of the prototype panel material. Similarly, the heated location of the panel is significantly hotter than the unheated portion, and the left and right side edges of the heater are located directly over sub-supply channels. This results in extremely high thermal gradients at the heater side edges, in which the face sheet surface temperature drops from 65 C to 15 C over a distance of about 20 mm. It is anticipated that the incorporation of a well-designed network of face sheet distribution channels will drastically mitigate the temperature gradients near the heater edges. The center of the heater is cooler than most other portions of the heater, as this is where eight supply channels converge.

A comparison of surface temperatures measured using thermocouples or RTDs with the calibrated IR image is shown in Table 1. Again, thermocouples located at positions 1 and 2 in Fig. 12 were used to calibrate the IR image, which accounts for the good temperature agreement at these locations. In general, at the other four positions where RTDs were mounted, agreement between the two temperature measurements was fair, with a maximum temperature difference between RTD data and IR thermography data of 3.8 deg. C.

Fig. 11 Temperature rise of fluid through the panel when subjected to a heat load of 25 W.
Fig. 12 IR image of the top surface of the prototype multi-functional panel at a flow rate of 1.5 L/min and an applied heat load of 25 W. The approximate locations of surface-mounted RTDs are indicated with numbered circles. Thermocouples located at positions 1 and 2 were used for calibration of the IR image.

Table 1 Comparison of surface-mounted thermocouple and RTD data with IR thermography data. Thermocouples located at positions 1 and 2 were used to calibrate the IR image.

<table>
<thead>
<tr>
<th>Location</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>TC/RTD values</td>
<td>6.8</td>
<td>25.5</td>
<td>52.1</td>
<td>31.0</td>
<td>11.0</td>
<td>5.7</td>
</tr>
<tr>
<td>IR image values</td>
<td>6.9</td>
<td>26.3</td>
<td>52.7</td>
<td>34.7</td>
<td>7.2</td>
<td>7.8</td>
</tr>
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</table>

To show the variation of the prototype panel’s thermal performance with flow rate, the temperature difference between thermocouples mounted at positions 1 (panel inlet) and 2 (center of panel) is shown in Fig. 13. This temperature difference provides a measure of the degree of the panel’s isothermality. Data are shown for two different heat loads: 25 W and 50 W. Low flow rate data was not obtained for the 50 W case to limit the maximum temperature to which the panel material was exposed, as its heat deflection temperature is relatively low. Consistent with
the thermal model data shown in Fig. 9, the temperature difference decreases more rapidly with increasing flow rate at low flow rates, and appears to level out at higher flow rates. Thus, for a heat load of 25 W, the desired operating point appears to be in the range of 1 L/min, as the fluid model indicated. However, the data obtained with a 50 W heat load show a substantial decrease in temperature difference of 8 deg. C as the flow rate increases from 0.7 to 1.5 L/min, suggesting that the optimum operating point is dependent on the applied heat load.

Fig. 13 Representative temperature differences indicating the degree of panel isothermality across the panel face sheet for applied heat loads of 25 W and 50 W.

Conclusions and Future Work

This paper discussed the development of a biologically-inspired composite thermal/structural panel. The need to preserve the stiffness-to-mass ratio of a structural panel while incorporating a high-performance thermal management system creates a unique geometry that is not easily created using traditional composite processing techniques. A novel process was discussed which allows for complex internal passageways to be formed during the fabrication of an isogrid panel, and does not significantly affect the panel’s mass. The paper also presented preliminary thermal performance test results obtained from a prototype panel constructed using rapid-prototyping techniques. For the configuration tested, it was shown that the power requirements of the panel were in the tens of milliwatts for relevant flow rates, and that even with the relatively low 5% efficiency of the planned EHD conduction-pumping system, power requirements would still only be a fraction of a watt. While this power requirement will increase if the allowable temperature difference across the panel is decreased or the heat load is increased, it suggests that the final power requirements of the panel will be quite low. Face sheet temperature distribution data were also obtained using IR thermography and surface-mounted RTDs, and it was found that the
desired flow rate through the panel to maximize thermal performance while minimizing power requirements will likely be dependent on the applied heat load.

Future work will include additional thermal and structural testing of the multi-functional panel. Small diameter face sheet distribution channels will be added to the channel network, which will substantially enhance surface heat transfer, but will likely drive up the panel’s power requirements. Both thermal/fluid modeling and testing will be conducted with various configurations of face sheet distribution channels to determine the tradeoffs between enhanced thermal performance and increased power requirements, and the experimental test setup will be modified to allow rapid panel changeout to facilitate these trade studies using different panel materials (e.g., SLA, composite, and aluminum). Additionally, the setup will be refined to provide reduced uncertainty values, especially for the measurements of fluid temperature rise across the panel. Finally, structural testing and modeling will be conducted on flight-like units to verify that the panel’s structural performance is not compromised by incorporation of the fluid network.

References


