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Engineering Notes

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Effects of Payload Heat Flux on Space Radiator Area

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Introduction

TWO-PHASE thermal management systems are essential in meeting the future military requirements for various space missions. These systems mainly consist of heat acquisition, heat transport, and heat rejection components. Figure 1 illustrates the basic concept of a generic thermal management system. The primary power source, energy storage, power conditioning, and electronic or sensor loads may require separate thermal control subsystems due to the large range of operational temperatures. Primary power sources may be nuclear, requiring a high temperature subsystem (850–1000 K), whereas the energy storage subsystems could be NaS batteries at 625 K. Power conditioning can be done typically in the 280–380 K range with sensor loads requiring thermal control for 10–100 K operations.

Over the past decade, spacecraft power requirements have increased by an order of magnitude; this trend will continue into the future. Current power systems are typically in the 1–5 kWe range, and future power systems are expected to reach the 50–100 kWe levels with peak to average pulsed loads of 10 to 1. As a result of the increasing power demands, heat acquisition, transport, and rejection schemes will require significant advancements when compared with the capabilities of today's low power technologies. Detailed discussions of these thermal systems are given by Chang and Hager.[1]

Although thermal control needs range from cryogenic to liquid metal temperatures, this Note is limited to a discussion of heat acquisition and rejection in the 280–380 K range. Specifically, this Note presents a simple analysis on how the performance of a high-flux heat acquisition device can have an impact on the required heat rejection area.

Heat Acquisition and Rejection

A heat acquisition device or heat-exchange cold plate must accept the waste heat generated by an electronics payload or power conditioning component as a result of device inefficiencies. The heat acquisition device is also expected to have a nearly isothermal mounting surface with minimal temperature gradients and thermal stresses between the device and a thermal bus or heat transport system.

Localized heat fluxes for today's electronics are normally less than 1 W/cm^2. As power demands increase, however, the heat acquisition system will likely require capabilities of up to 100 W/cm^2 with pulse mode operation of 10 to 1, peak to average. Current efforts are focusing on increasing the heat flux capabilities up to approximately 35 W/cm^2 at the component mounting surface. The internal heat fluxes of these devices may reach 250 W/cm^2 or higher if switching losses are considered.

In addition to cold plate heat exchangers for indirect payload thermal control, direct contact heat exchange techniques are also under investigation. Direct contact type systems include transpiration, immersion, and spray cooling concepts. At the device level, present heat fluxes are less than 15 W/cm^2 with envisioned needs of 1 kW/cm^2 capability. The future high heat-flux/large area requirements and applicable technologies are well described by Mahefkey.[2]

Current spacecraft designs utilize exposed surfaces to reject heat to space. As available rejection areas become limited and thermal requirements increase, spacecraft in the near-future will require separate appendages primarily designed for heat rejection. These appendages may be fixed or deployable structures with two-sided rejection capabilities. The total area A required to reject a given thermal load q is given by

$$q = \sigma \epsilon A (T_r^4 - T_s^4) \tag{1}$$

where σ is the Stefan-Boltzmann constant, ϵ the emissivity, T_r the radiator temperature, and T_s the equivalent sink temperature. The equivalent sink temperature includes the effects of solar and albedo loads, Earth infrared loads, and infrared exchange between spacecraft surfaces. As seen from this expression, the area required for heat rejection is directly related to the fourth power relationship between radiator and sink temperatures. Minimizing the temperature drops within the thermal control system allows the radiator to operate at a higher temperature, thus requiring a smaller area for heat rejection. To achieve significant area and weight savings of the

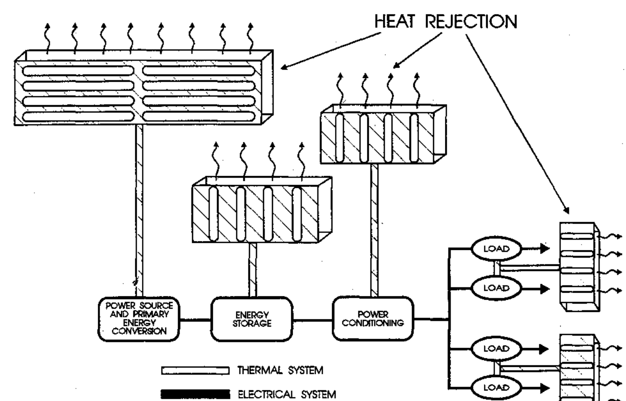

Fig. 1 Generic thermal management system.

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radiator, a more efficient cold plate, which provides an excellent heat transfer path, needs to be developed.

Radiator Area Analysis

A simple analysis can be performed to determine the savings of the radiator area for the new cold plate compared with the current ones. For most cases, the area savings is nearly directly proportional to the weight savings. The saving of the radiator area S in percent is defined as $S = [1 - (\text{area of new radiator}/\text{area of current radiator})] \times 100$. With the same emissivity and rejected heat for both the current and new radiators, S becomes

$$S = [1 - (T_{r,c}^4 - T_s^4)/(T_{r,n}^4 - T_s^4)] \times 100 \tag{2}$$

where $T_{r,c}$ and $T_{r,n}$ are the temperature of the current and new radiators, respectively. The temperature of the sink or space is determined by orbit conditions.

The radiator temperature T_r in Eq. (2) may be found from

$$T_r = T_p - (T_p - T_c) - (T_c - T_r) \tag{3}$$

where T_p is the payload temperature and T_c the condenser temperature of the cold plate. The temperature drop $T_p - T_c$ is composed of the temperature drops across the interface between the payload and the cold plate and the temperature drop through the cold plate to the thermal bus vapor. With known interface heat transfer coefficients between the payload and the cold plate h_i and the overall heat transfer coefficient across the cold plate h_c and using the applied heat flux q'', this temperature drop can be found from

$$T_p - T_c = [(1/h_i) + (1/h_c)]q'' \tag{4}$$

Substituting Eq. (4) into Eq. (3) yields

$$T_r = T_p - [(1/h_i) + (1/h_c)]q'' - (T_c - T_r) \tag{5}$$

Here, the temperature drop across the thermal bus to the radiator $T_c - T_r$ is given as a design parameter. With a fixed payload temperature, the radiator temperature can thus be calculated from Eq. (5). Equation (5) can be used with Eq. (2) to determine radiator savings by developing new cold plate designs.

As an example, consider a problem related to a payload in a low Earth orbit with an assumed sink temperature of 200 K. Let h_i be 0.4 W/cm^2-K and $T_c - T_r$ be 10 K. Tables 1 and 2 show results of radiator temperatures and area savings for payload temperatures of 293 and 323 K, respectively. Figure 2 shows the results graphically. In these tables the overall heat transfer coefficients used are 0.25 W/cm^2-K for a current cold plate and 4 W/cm^2-K for a cold plate that is being developed by the Air Force under the Two-Phase Survivable Thermal Management (TSTM) Components Demonstration program. The maximum heat fluxes listed in the tables, 12 and 17 W/cm^2, are above the limits of current cold plates and therefore the new cold plate design represents an enabling technology allowing higher payload fluxes. At the upper flux levels, the calculated savings are not realistic since current cold plate designs cannot handle these fluxes and cannot be used as a basis for comparison.

The tables indicate that radiator temperature decreases with increasing payload flux. The radiator savings with the new cold plate is due to the higher heat transfer coefficient h_c. The increase in h_c from current values of 0.25 to 4 W/cm^2-K is an enabling technology that also allows significant radiator savings as shown in Fig. 2. Increasing h_c beyond 4 W/cm^2-K has marginal benefit in radiator savings since the primary thermal resistance is due to the interface. A value of 4 W/cm^2-K represents the best detachable interface available. For example, if h_c were 10 W/cm^2-K, then the radiator savings over current designs for a 4 W/cm^2-K payload, operating at 293 K, would be 29.6%. This value can be compared to the value of 28.7% shown in Table 1 for a TSTM cold plate.

Table 1 Radiator with a payload temperature of 293 K

Heat flux, W/cm^2	Radiator temperature, K		Savings, %
	Current	TSTM	
0.1	282.3	282.7	0.7
1.0	276.5	280.2	7.1
4.0	257.0	272.0	28.7
8.0	231.0	261.0	59.0
12.0	205.0	250.0	92.8

Table 2 Radiator with a payload temperature of 323 K

Heat flux, W/cm^2	Radiator temperature, K		Savings, %
	Current	TSTM	
0.1	312.3	312.7	0.6
1.0	306.5	310.2	5.7
4.0	287.0	302.0	22.8
8.0	261.0	291.0	45.4
12.0	235.0	280.0	68.1
17.0	202.5	266.2	97.6

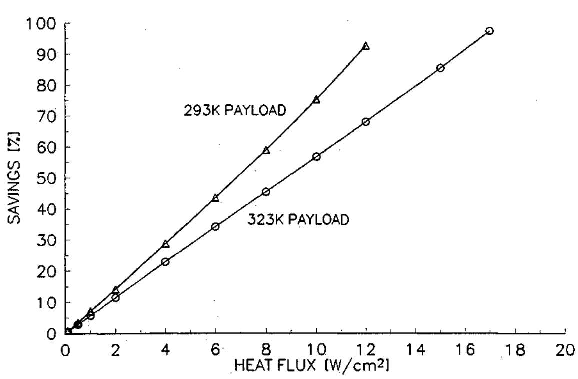

Fig. 2 Radiator savings vs heat flux.

Conclusions

To ensure reliable operation of future military spacecraft, the thermal control system needs to maintain the electronics and the payload equipment at acceptable temperatures under normal and adverse conditions. The temperature drop between the payload and the radiator must be minimized to reduce the area and weight of the radiator. A simple analysis demonstrates that radiator area savings increase as the applied heat flux increases. The analysis also indicates that significant benefits can be achieved by increasing the cold plate heat transfer coefficient beyond the current cold plate capabilities.

References

[1]Chang, W. S., and Hager, B. G., "Advanced Two-Phase Thermal Management in Spacecraft," *Proceedings of the 25th Intersociety Energy Conversion Engineering Conference,* American Institute of Chemical Engineers, Vol. 2, 1990, pp. 121–129.

[2]Mahefkey, E. T., "Military Spacecraft Thermal Management: The Evolving Requirements and Challenges," AIAA Paper 82-0827, June 1982.