Fabric Space Radiators

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SUMMARY

Future Air Force space missions will require thermal radiators that both survive in the hostile space environment and stow away for minimal bulk during launch. Advances in all aspects of radiator design, construction, and analysis will be necessary to enable such future missions.

Currently, the best means for obtaining high strength along with flexibility is through structures known as fabrics. Granted that these belong to some of the most ancient constructs, the development of new materials and bonding techniques has extended the application range of fabrics into areas traditionally dominated by monolithic and/or metallic structures. Possibly the most graphic example of this displacement of traditional materials, structures, and fabrication methods by novel techniques employing fabrics is seen in the use of composites for high-performance aircraft components and even sports equipment (e.g., tennis racquets). The extension of such applications into space is, therefore, viewed as a natural progression in the development of fabric structures. Given that even current spacecraft heat rejection considerations tend to dominate spacecraft design and mass, the larger and more complex designs of the future face daunting challenges in thermal control. Ceramic fabrics bonded to ultra-thin metal liners (foils) have the potential of achieving radiator performance levels heretofore unattainable, and of readily matching the advances made in other branches of spacecraft design.

In this study, we have examined some of the options presented by fabrics incorporated into space radiators. For the baseline helium/xenon working fluid system, preliminary analyses indicate that either direct or secondary radiator systems should be feasible. In the direct system, the helium/xenon from the heat engine is itself the coolant, transferring heat to the fabric wall. The secondary system has another fluid, optimized for the radiator, with which the helium/xenon exchanges heat. Only the first system has been analyzed in detail here and appears to offer great promise. It would be flexible, have a low mass (about 1.0 kg/m^2 of projected radiator surface area), and would be inherently somewhat hardened against laser threats. Survivability against micrometeoroid punctures is a principal concern and is not easily

achieved with a system that is subject to single point failure. Therefore, additional work is needed to incorporate system options that will harden the system. In any case, micrometeoroid armoring/redundancy would require approximately 2.4 kg/m^2 more mass than an unprotected radiator, for a total radiator mass of approximately 3.4 kg/m^2 . Of itself, this is probably an acceptable mass penalty; the true problem that needs resolution is one of effectively isolating damaged panels.

Even if a pumped-loop radiator system cannot be protected against micrometeoroid punctures, it might still have a niche. For large power systems, heat pipes are hardly appropriate. Moreover, such large power systems generally have only burst power capability. For this type of power system, a lightweight, pumped-loop system that could be stowed when not operating could well become the system of choice.

Heat pipes offer many attractive features in a space radiator role, of which the chief one is the redundancy resulting from each heat pipe being an isolated unit whose destruction need hardly affect the operation of the rest. But metal heat pipes are relatively heavy, and their fabrication is still more art than science. This study shows that both these drawbacks could be significantly decreased or entirely eliminated. Through the use of ceramic fabrics in the wick and outer (pressure containment) wall, mass can be reduced to about one-half that of metal heat pipes. An additional 2 kg/m^2 of armoring would provide adequate micrometeoroid puncture survivability, so that a fabric heat pipe radiator system with a potassium working fluid would have a total mass of about 8.2 kg/m^2 of projected area; a comparable titanium heat pipe would have a mass of about 32.0 kg/m^2 , which is in the 20- to 40-kg/m^2 range of current heat pipe designs (Dearien and Whitbeck 1987). Note that this mass is for a nonoptimized design; significant mass reduction could result from a decrease or elimination of the circumferential distributive wick. Currently, only a relative comparison of fabric versus metal pipes is meaningful. Absolute mass values can be obtained only from an experimental effort.

Although considerably heavier than a pumped-loop system, a fabric heat pipe system could probably be implemented in the near term and would be adequate for the multikilowatt power levels anticipated for the next spacecraft

generation. Regarding fabrication, the usual difficulties associated with wick and artery construction, and of having these components perform in a reproducible manner, can potentially be eased with the proposed ceramic wick structure bonded directly to a metallic foil liner.

In a parallel effort, various ceramic fabrics have been bonded to various metal foil liners. The results to date are encouraging, and suggest that the approach is sound. We have achieved good bonds, while retaining the desired flexibility in the sandwich. Development of optimal bonding techniques and material combinations requires additional work, but the current study provides good insight into the paths future research should take. The measured masses of fabric/braze/foil samples ranged from 0.6 to 1.3 kg/m². No attempt was made for these to be particularly lightweight, so the potential exists for additional mass reductions.

In summary, the research effort documented here indicates that both pumped loops and heat pipes constructed in ceramic fabrics stand to benefit in multiple ways. Flexibility and low mass are the main advantages exhibited by fabric radiators over conventional metal ones; other benefits are described in the body of the report. We feel that fabric radiators have intrinsic merits not possessed by any other radiator design and need to be researched further.

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CONTENTS

SUMMARY	iii
ACKNOWLEDGMENTS	vii
1.0 INTRODUCTION	1
2.0 MATERIALS STUDY	7
2.1. BACKGROUND	7
2.2. METALLIZATION/FABRICATION EXPERIMENTS	9
2.2.1. Common Adhesive Bonding	9
2.2.2. Braze Bonding	9
2.2.3. Ceramic Adhesive Bonding	11
2.2.4. Discussion	12
2.2.5. Conclusions	13
2.3. RELATED EXPERIMENTS	13
2.3.1. Permeability	15
2.3.2. Wetting/Wicking	16
3.0 SPACE RADIATOR CONCEPTUAL DESIGN AND HEAT TRANSFER ANALYSIS	19
3.1. HELIUM/XENON DIRECT PUMPED-LOOP RADIATOR	19
3.2. LITHIUM SECONDARY-SYSTEM PUMPED-LOOP RADIATOR	27
3.3. HEAT PIPE RADIATOR	27
4.0 MICROMETEOROID ANALYSIS	33
4.1. APPROACH	33
4.2. PUMPED-LOOP SURVIVABILITY CALCULATIONS	39
4.3. HEAT PIPE SURVIVABILITY CALCULATIONS	45
5.0 RADIATOR OPERATION	47
5.1. STARTUP/TRANSIENTS/SHUTDOWN	47

5.2. SHAPE ALTERATION	48
6.0 FEASIBILITY ISSUES	49
6.1. KEY TECHNICAL ISSUES IDENTIFIED	49
6.2. TECHNICAL ISSUES OUTSTANDING	49
6.3 RECOMMENDED APPROACH	50
7.0 CONCLUSIONS	53
8.0 REFERENCES	55
APPENDIX A - SAMPLE HEAT PIPE CODE RESULTS	A.1
APPENDIX B - SURVIVABILITY PROGRAM LISTING	B.1

FIGURES

1.1	Reference Brayton-Cycle Space Power System	3
1.2	Direct Radiator	4
1.3	Radiator System with Intermediate Heat Exchanger	4
1.4	Cross Section of a Fabric Heat Pipe	5
2.1	Fabric Coating/Bonding Techniques	10
2.2	Sample Leak Test Apparatus	16
2.3	Sample Leak Test Fixture	17
2.4	Wicking Heights of Standard Fabric Samples	18
3.1	Circular Direct Radiator Design	20
3.2	Thermal Resistance Model for Radiator Surface	21
3.3	Channel Exit Temperature For Helium/Xenon Flow	22
3.4	Radiator Heat Removal Design Curve For Helium/Xenon Flow	23
3.5	Radiator Pressure Drop Design Curve For Helium/Xenon Flow	24
4.1	Average Cumulative Meteoroid Mass Flux for Near-Earth Orbits	34
4.2	Product of Earth's Gravitational Defocusing and Body Shielding Factors, GeE, as a Function of Distance from Earth's Center	36
4.3	Probability Distribution of Observed Meteoroid Velocities	38

TABLES

2.1	Candidate Metal Sealant Layers for Various Working Fluids	7
2.2	Ultimate Tensile Strengths of Tungsten-Coated and Uncoated Silicon Carbide Filaments at Various Temperatures	8
2.3	Summary of Radiator Skin Fabrication Experiments	14
3.1	Calculated Radiative Heat Transfer Coefficients with Aluminum Surface Treatment	22
3.2	Design Conditions for Helium/Xenon Cases Studied	26
3.3	Heat Pipe Performance Summary	29
4.1	Constants for Threshold Penetration for a Monolithic Barrier at 298 K	37
4.2	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 1 [helium/xenon pumped loop, flow area $1.0 \times 10^{-3} \text{ m}^2$, channel length 10 m, required number of loops at end-of-life (EOL) 93, unarmored mass = 35.9 kg]	41
4.3	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 2 (helium/xenon pumped loop, flow area $1.0 \times 10^{-3} \text{ m}^2$, channel length 20 m, required number of loops at EOL 43, unarmored mass = 33.2 kg)	41
4.4	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 3 (helium/xenon pumped loop, flow area $4.91 \times 10^{-4} \text{ m}^2$, channel length 10 m, required number of loops at EOL 125, unarmored mass = 32.9 kg)	41
4.5	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 4 (helium/xenon pumped loop, flow area $4.91 \times 10^{-3} \text{ m}^2$, channel length 20 m, required number of loops at EOL 59, unarmored mass = 31.1 kg)	42
4.6	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 5 (helium/xenon pumped loop, flow area $7.85 \times 10^{-5} \text{ m}^2$, channel length 10 m, required number of loops at EOL 291, unarmored mass = 29.9 kg)	42
4.7	Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 6 (helium/xenon pumped loop, flow area $7.85 \times 10^{-5} \text{ m}^2$, channel length 20 m, required number of loops at EOL 134 unarmored mass = 27.6 kg)	42

4.8	System Mass Estimate of Helium/Xenon Pumped Loop with Three Pumped Loops Sharing a Common Isolation Valve (system reliability 0.999)	43
4.9	System Mass Estimate for Postulated Heat Pipe Systems as a Function of Structural Material (system reliability 0.99, operational lifetime 10 years)	45
6.1	Key Technical Issues	49
6.2	Approaches for Enhancing Survivability Against Micrometeoroid Punctures	51

FABRIC SPACE RADIATORS

1.0 INTRODUCTION

Numerous radiator types have been proposed for rejecting heat from the large power plants envisioned for future spacecraft. Although these radiators are in various stages of study and development, some of which are well along in providing definitive data, currently no radiator design exists that is capable of simultaneously meeting all design criteria. For Air Force missions, the most stringent criteria relate to size, survivability, and weight. Preliminary information indicates that these may be met by a novel class of radiators, based on fabric materials that have been treated for compatibility with the working fluid and for enhanced radiative characteristics (Webb and Antoniak 1986).

A study was recently initiated at the Pacific Northwest Laboratory (PNL) to analytically and experimentally consider design and construction of fabric material radiators. The full range of radiator types, from thin panels to single-phase pumped loops with extended surfaces (fins) to heat pipes, is under investigation. The approach taken is straightforward: an experimental effort seeks to find suitable fabric materials and coating/sealing techniques that will prevent working fluid loss while maintaining structural integrity and flexibility.

The latter feature is especially desirable, as payload volume on launch vehicles is always restricted. A flexible radiator also offers benefits such as adaptability to a variable heat load, and could be retracted for enhanced survivability. Such features have already been incorporated into flexible metal heat pipes (Merrigan et al. 1984; Ponnappan, Beam, and Mahefkey 1986) and other radiators (Chow and Mahefkey 1986), but it is not clear at this time if these are truly viable. A fabric/metal-foil radiator construction is inherently highly compliant; it is felt that the compliance can largely be retained in layering and/or metallizing the base fabric during radiator construction. Depending on the fabric selected, optical properties may or may not require enhancement (Covington and Sawko 1986).

The second part of this study is concerned with resolving design issues and comparing fabric and other radiator types. To this end, the ATHENA code developed at Idaho National Engineering Laboratory is being used in the design, optimization, and system analysis of various space radiators. Systems with and without heat exchangers are being considered. Meanwhile, the performance of fabric heat pipes is being studied with a small code created at Los Alamos National Laboratory (LANL).

The Air Force Weapons Laboratory (AFWL) selected a closed Brayton-cycle space power system for analysis (see Figure 1.1). The working fluid is a helium/xenon mixture, requiring approximately 80 kWt to be rejected to space to produce 30 kWe net power. With inlet and outlet temperatures of 575 K (575°F) and 367 K (200°F) respectively, the temperature difference across the radiator becomes a fairly severe 208 K. This Brayton system is similar to ones proposed earlier, and the usual considerations regarding the optimal characteristics of a working fluid mixture apply (Mock 1977).

Hand calculations were performed here to scope the radiator size and performance. Assuming a linear axial temperature profile, the effective radiator temperature is 452 K. If the emittance-transmittance-view factor product is taken as 1.0, the minimal radiator surface area needed is 33.9 m^2 . A more realistic radiator size of 50 m^2 might consist of 10 thin rectangular panels (see Figure 1.2).

Additional calculations indicated that there was no clear-cut advantage evident for a radiator system directly coupled to a Brayton heat engine with a helium/xenon working fluid, compared to the same engine rejecting heat across an intermediate heat exchanger and, thence, to a radiator with a liquid metal fluid. The simplicity of the first system (see Figure 1.2) is obtained at the price of relatively poor convective heat transfer and attendant high temperature gradients.

Survivability (Tilton and Chow 1986), including leak detection and isolation of damaged radiator sections, is also a major issue. The liquid-metal radiator system (see Figure 1.3) ameliorates such concerns to an extent.

FIGURE 1.1. Reference Brayton-Cycle Space Power System

FIGURE 1.2. Direct Radiator

FIGURE 1.3. Radiator System with Intermediate Heat Exchanger

However, it raises new ones, including compact heat exchanger design, single-point failure in the pumped loop, and materials compatibility. Both systems require detailed study and comparison.

One more radiator design was considered. A fabric heat pipe radiator has been conceived that uses technology similar to that of the pumped loops, in a sandwich construction (see Figure 1.4). Although ceramic fiber wicks have received consideration in the past (Dunn and Reay 1982, pp. 95, 144-146; Brennan and Kroliczek 1979, Vol. 1, p. 283), their flexibility was a problem. Here, we propose to take full advantage of this flexibility and ensure good contact with the liner by bonding. Again, detailed study and comparison against existing heat pipe designs are necessary to ascertain the value of this novel design.

FIGURE 1.4. Cross Section of a Fabric Heat Pipe

2.0 MATERIALS STUDY

In this section, the experiments and tests conducted on several alternative fibers are described. A literature search disclosed that ceramic fibers offered the best combination of high-temperature properties suitable for a fabric radiator (Tai-il Mah et al. 1987; Weeton, Peters, and Thomas 1987).

2.1 BACKGROUND

For this study, candidate ceramic fabrics were selected based on commercial availability and fiber flexibility. Four fabric compositions were chosen for study: silica, silicon carbide, aluminoborosilicate, and graphite. To date, no consideration of the type of weave has been attempted, although the selection is almost limitless. It is felt that this selection would best be made later in conjunction with overall design criteria.

The selection of the metal layer was dominated by the need for chemical compatibility with the working fluid. Weight was a secondary consideration. Three working fluids were considered: helium/xenon, sodium/potassium, and lithium. The respective candidate materials chosen for the different working fluids are listed in Table 2.1.

Compatibility with the space environment, notably atomic oxygen, is also required of the different materials. However, thin coatings of silicon, aluminum, or chromium could be applied to vulnerable materials (e.g., graphite fibers) to provide adequate protection without significantly degrading their mechanical properties.

It is envisioned that the fabric radiator will consist of two primary constituents: an outer ceramic fabric to provide structural strength and

TABLE 2.1. Candidate Metal Sealant Layers for Various Working Fluids

<u>Working Fluid</u>	<u>Candidate Metal Layer</u>
Helium/xenon	Aluminum, titanium, nickel
Sodium/potassium	Austenitic steel, molybdenum
Lithium	Titanium, molybdenum

integrity (see Table 2.2 for representative strength values, which are considerably higher than those for monolithic metals, especially when the comparison includes density), and an inner metallic liner to contain the working fluid. The most feasible approach identified to date appears to be the separate fabrication of the fabric skeleton and the metal foil liner, followed by an insertion and joining process. However, the direct application of the metal layer to a preformed fabric shell is also being considered.

Thermal properties of the fabric material samples were not measured in this study. However, thermal conductivity values are frequently available from the manufacturer. Some difficulty arises in interpreting such values; often it is unclear if the reported thermal conductivity is a true conductivity along the fibers, or if it also includes thermal radiation (preliminarily, the latter conclusion appears the more likely).

Radiative properties of various fabrics reported recently (Covington and Sawko 1986) are consistent with some unreported measurements made at PNL several years ago. In general, one can conclude that the intrinsic properties are not optimal. Two methods of enhancing the radiative heat transfer to space suggest themselves: coating the fabric fibers, or opening the weave to permit

TABLE 2.2. Ultimate Tensile Strengths of Tungsten-Coated and Uncoated Silicon Carbide Filaments at Various Temperatures (from Weeton, Peters, and Thomas 1987, p. 5-12)

Temperature,		UTS ^(a) (b)		UTS		σ/σ_0
°C	°F	(coated)		(uncoated)		
		GPa	ksi	GPa	ksi	
R.T.	R.T.	1.835	266	2.828	410	0.649
800	1470	1.648	239	2.359	342	0.699
1000	1830	1.531	222	2.083	302	0.735
1200	2190	1.414	205	1.883	273	0.751
1400	2550	1.083	157	1.359	197	0.797
1600	2910	0.855	124	1.062	154	0.805

(a) Mean ultimate tensile strength

(b) Coating thickness, 12.7 μm (0.0005 in.)

the impermeable liner to directly view space plus radiate via several bounces from the very reflective fabric. We believe that, alone or in combination, these methods will achieve an effective fabric radiator emittance rivaling that of surface-treated metal heat pipes, i.e., about 0.6 to 0.8.

2.2 METALLIZATION/FABRICATION EXPERIMENTS

Extensive bonding of the metal layer to the fabric does not appear to be a critical concern. In fact, for a radiator application the bond can be quite tenuous, or even nonexistent, because the internal pressure of the system will press the metal layer firmly against the surrounding fabric upon deployment. In terms of bonding, there appears to be a tradeoff between an intimate bonding, which tends to reduce overall flexibility of the sandwich, and a looser attachment, such as one that provides only a few contact points, which permits the formation of fewer but sharper creases in the metallic layer when the composite is flexed. Sharp creasing of the metal layer is felt to be detrimental because highly worked regions would be expected to develop pinhole leaks first.

2.2.1 Common Adhesive Bonding

A number of sandwiching experiments have been performed (see Figure 2.1). Adhesive bonding of thin metal foils to ceramic fabrics has been found to work extremely well; this technique maintains flexibility while providing an impermeable barrier to the working fluid. The best results are obtained when care is taken to minimize the filling of the interstices in the fabric weave by the adhesive. Silicone adhesive appears to represent a plausible adhesive for use in the helium/xenon system where operating temperatures will be below 600 K. Silicone compounds are stable in space environments (National Aeronautics and Space Administration 1985) and can be used continuously at temperatures up to 600 K. An abundance of other bonding agents (for example, paraffin) could be used to provide an interim bond that would be present for deployment but then rapidly disappear.

2.2.2 Braze Bonding

For liquid-metal cooling systems where high temperatures are required, other approaches have been explored, including brazing of foils to metallized fabrics and using ceramic adhesives. Preliminary brazing experiments have

FIGURE 2.1. Fabric Coating/Bonding Techniques

been encouraging. Fabrics have been successfully metallized using both electroless deposition (nickel) techniques and physical vapor deposition (titanium). In both cases the metallization left the fabric with its original flexibility. Following metallization, several brazing experiments were performed to try to bond the fabrics to steel, titanium, and molybdenum foils. In general, significant reductions in flexibility of the sandwiches were experienced. In cases where the fabric was completely coated with metal (e.g., electroless deposited nickel), the braze sometimes wetted the entire fabric, which produced a very rigid composite sheet. However, if this could be avoided,

some flexibility could be retained. For example, a nickel-coated silicon-carbide fabric was successfully brazed to a 25- μm molybdenum foil such that it could be wrapped around a 3-cm mandrel with no visible distress.

By far, the best brazing results were obtained when 225- μm iron wires were woven into nonmetallized ceramic fabrics to act as the points of attachment between the fabric and the foil. In this manner an excellent bond could be obtained between the fabric and the foil, and the intermittent nature of the bonding permitted excellent flexibility. For example, using wires spaced about 2 cm apart, an aluminoborosilicate fabric was bonded to a 25- μm steel foil and then repeatedly bent over a 1-cm mandrel. This is roughly equivalent to the performance of the adhesively-bonded sandwiches and yet would be applicable to much higher temperatures (about 1200 K).

2.2.3 Ceramic Adhesive Bonding

As an alternative to brazing, several commercially-available alumina- and silica-based ceramic adhesives, rated for temperatures up to about 2000 K, were also examined. When these were applied as a continuous bond layer, extremely rigid composites were produced. Although a strong bond developed between the fabric and the foil, the bond layer prevented any sliding between threads in the fabric. Thus, the fabric lost its original flexibility. The flexibility of the ceramically-bonded sandwiches was improved markedly when the bonding was reduced to a set of discrete points or islands. This provided a connected network of unbonded fabric between the joined areas, which accommodated the strains needed to provide maximum flexibility. Unfortunately, the ceramic adhesive was found to be very brittle, and the islands of adhesive gradually disintegrated with continued flexing of the fabric. Thus, the ceramic adhesives would be appropriate only when a limited amount of redeployment was anticipated.

A gas permeability experiment was performed to test whether these sandwich materials could remain gas impermeable following modest flexing. A titanium foil, which was adhesively bonded to an aluminoborosilicate fabric, was tested before and after being randomly wrapped around a 1.3-cm-diameter mandrel 50 times. Leak rates below the sensitivity limits of the apparatus (10 to 11 std cc/s cm^2) were observed in both cases for a pressure drop of

3 atm across the sample, indicating that the sandwich materials tested are capable of maintaining their integrity after modest flexing.

2.2.4 Discussion

The intent of this study was to demonstrate the feasibility of incorporating ceramic fabrics into space-based radiators as the structural component and, thus, present the option for providing future radiator designs with exceptional flexibility and weight savings. A preliminary review identified combinations of commercially-available materials that appear to satisfy the major criteria, notably chemical compatibility with the working fluids and low mass. By using thin metal foils backed with ceramic fabrics, an extremely durable, gas impermeable skin appears possible. Fabrication of several different candidate materials proceeded without difficulty. The relative merits of the various schemes are now being evaluated in terms of mechanical durability and thermal transport behavior.

Preliminary experiments indicate that foil thicknesses will have to be kept below about 50 μm to retain good flexibility of the sandwich material, e.g., to permit wrapping around a 1-cm mandrel. This foil thickness limitation does pose some concern for the liquid-metal radiators where surface corrosion rates of as much as 5 $\mu\text{m}/\text{yr}$ may be experienced. Over the proposed 7- to 10-year life of the power plant, a single foil layer may prove to be too thin. However, a multilayered ply containing several 50- μm foil layers should be able to provide adequate thickness, strength, and flexibility.

Metallization of the fabrics has proven to be quite simple, effective, and innocuous. This not only increases the number of bonding options available, but it also suggests that thermal transport properties of the fabrics could easily be modified with surface coatings. There is clearly a penalty to be paid in a space-based radiator system that incorporates a thermal insulator in its outer skin. However, radiation out through the open weave of the fabric and conductivity through the fabric, which could be enhanced by metallization of the ceramic fabric, will decrease this penalty.

Many practical matters associated with the design and assembly of a space-based fabric radiator have not yet been addressed. To date, no materials issues have emerged that would appear insurmountable. For example, a major

concern might be whether the consolidation techniques used in the current study might be adapted to the manufacture of a complex system such as a radiator. The concept of fabricating the fabric skeleton and the metal foil liner separately and then bonding the two to form the final structure could prove to be extremely difficult in practice. Ultimately, it might be necessary to develop methods for directly depositing the metal layer onto the preformed fabric shell. The challenge would be to develop techniques that would permit the deposition of a thin, defect-free coating over the rough, discontinuous fabric surfaces. This could probably be accomplished using techniques such as bias sputtering that are capable of preparing defect-free coatings. By temporarily filling the interstitial gaps of the fabric with a material that could later be removed, it should be possible to develop an in situ deposition process. Thus, preliminary results to date suggest that ceramic fabrics could be effectively incorporated into space-based radiators to permit the construction of an extremely lightweight, deployable system.

2.2.5 Conclusions

Preliminary studies suggest that ceramic fabrics can be combined with thin metal layers to produce gas impermeable sandwiches with exceptional flexibility and durability. Use of this material as the skin of a space-based radiator would permit the construction of extremely lightweight systems that could be easily stowed for launch. Candidate material combinations have been identified that could be used in conjunction with different working fluids (helium/xenon, sodium/potassium, lithium) at temperatures up to at least 1200 K (see Table 2.3).

2.3 RELATED EXPERIMENTS

Initially, a test series for all developed samples was planned that could incorporate tests for permeability, leak detection and survivability, thermal properties, and wetting/wicking characterization. Because of various constraints for the current year's effort, sample testing was limited to permeability and simple flexibility tests on a limited number of samples. Test conditions were simplified to ambient temperature, as compared to a

TABLE 2.3. Summary of Radiator Skin Fabrication Experiments

Process	Materials	Total Skin	Maximum	Comments
	Fabric/Bond/Foil (Thickness, μm)	Mass (a), kg/m^2	Temperature, K	
Adhesive bonding	$\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /paraffin/Al(25)	0.6	400	Very good flexibility and bonding
	$\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /RTV/Al(25) SiC graphite	---	600	Very good flexibility and bonding
	SiC	0.4		
	graphite	0.4		
	$\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /RTV/Ti(25)	---	600	Some flexibility, good bond
Braze bonding	Ni metallized $\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /quick silver/steel(25)	---	800	Some flexibility, good bond
	Ni metallized $\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /cusil(63)/Ti	1.3 ^(b)	900	Limited flexibility, good bond
	Ti metallized $\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /Pacusil(25)/Mo(25)	---	1200	Poorly bonded
	$\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ with steel wires/Pacusil(25)/steel(25)	1.1	1200	Excellent flexibility and bonding
Ceramic adhesive	$\text{Al}_{2/3}\text{O}_{3/2}\text{B}_{2/3}\text{O}_{3/2}\text{SiO}_2$ /silica- or alumina-based adhesive/Mo(25)	---	1500	Limited to good flexibility with poor bonding
Metallization close-out	Ti(76)sputtered onto SiO_2 fabric	---	1500	Some pin holes present, some flexibility

(a) Not optimized in any manner.

(b) Braze material mass was 0.6 kg/m² of this total.

radiator design figure of 575 K (575°F). However, the radiator design condition was used to design and fabricate the leak test fixture such that should the test program be extended, the existing test fixture is usable. Some wicking tests were also performed.

2.3.1 Permeability

As a first test of radiator fabric sample permeability, the leak rate of helium through a small sample subjected to an approximate 345 kPa (50 psia) pressure differential was measured. Leak rates of helium were measured using a helium "sniffer" unit mounted as part of vacuum pump system. The test fixture (see Figures 2.2 and 2.3) provided a small helium reservoir on the metallized side of the sample. The opposite side of the sample holder was connected to the vacuum system and subjected to "hard" vacuum conditions. Any leakage across the metal foil/fabric sample would be noted by the helium sniffer unit in the vacuum system exhaust.

The test fixture was originally designed to make use of a standard large male, large female tongue-and-groove flange set, although the fixture was ultimately fabricated from carbon-steel round bar stock. The 3.51-cm (1.38-in.) inside diameter of the 1-1/4-in. sch. 40 pipe selected yields a minimum 9.65 cm^2 (1.50 in.²) of exposed sample surface when the test sample is mounted between the mating flange surfaces. The room-temperature vulcanizing (RTV) sealant was used to form leak-tight seals of the sample on both the male and female flange faces. A soft rubber O-ring was fitted on the outside of the tongue-and-groove flange seal to preclude leakage of outside air into the test fixture assembly. The threaded pipe construction of the helium reservoir allowed access to the assembled sample/flange set such that vacuum grease could be applied at the sample/flange interfaces in the event additional seal leaks were found.

Initial leak tests were performed on the test fixture using a "dummy" sample fabricated from 0.127-mm (0.005-in.) brass shim stock to ensure that no unexpected leakage paths were present. The test fixture was found to be leak-tight. Further testing was performed on metal foil samples before and after several cycles of bending the samples around a 1.3-cm (0.5-in.) diameter

FIGURE 2.2. Sample Leak Test Apparatus

mandrel. The bending was intended as a simple simulation of the flexing that would be experienced by a fabric radiator during packaging and deployment/retraction operations.

2.3.2 Wetting/Wicking

For a heat pipe to perform well, the working fluid must wet the wick and container wall (Dunn and Reay 1982, p. 92; Chi 1976, p. 36). A simple wicking test was performed using water at standard temperature and pressure (STP) and three arbitrarily-selected wick materials. The results, shown on Figure 2.4, are consistent with those in the literature (Dunn and Reay 1982, p. 96).

FIGURE 2.3. Sample Leak Test Fixture

These tests indicate that ceramic fiber fabrics should serve as acceptable heat pipe wicks. Ceramic wicks have been considered in the past (Dunn and Reay 1982, pp. 95, 144-146), although we have not seen heat pipe performance figures with such wicks.

FIGURE 2.4. Wicking Heights of Standard Fabric Samples

3.0 SPACE RADIATOR CONCEPTUAL DESIGN AND HEAT TRANSFER ANALYSIS

As previously mentioned, the primary purpose of this study was to design and assess radiator systems that are coupled to a closed Brayton-cycle nuclear heat engine. A helium/xenon gas mixture is assumed to be the reactor coolant fluid. The radiator designs that were the subject of this study are discussed below; any number of panel configurations could be used (Josloff, Kirpich, and Biddiscombe 1987).

Analysis is required to properly design and optimize a radiator system. Analysis is also required to permit comparison of the three studied fabric radiator designs. Computer models of the direct and secondary system radiators have been prepared for use by the ATHENA computer code; heat pipes have been examined with a code originally created at LANL and modified by PNL.

3.1 HELIUM/XENON DIRECT PUMPED-LOOP RADIATOR

This system is the simplest. The helium/xenon fluid is pumped directly through one or more flexible radiator panels. A panel consists of a series of parallel flow channels. Heat is transferred from the gas to the radiator wall by forced convection. The radiator walls reject heat by radiation to space. The primary feasibility issue is survivability, including leak detection and isolation.

A base model of one channel of the direct radiator was constructed for input into ATHENA (Carlson, Roth, and Ransom 1986). Modifications were then made to this model to perform sensitivity studies. The direct radiator base model (Case 1) consists of a hot entrance channel 5 m long and a cooler 5-m return channel. Because one of the beneficial properties of employing a fabric radiator is its flexibility, it is difficult to guarantee a noncircular flow channel. Thus, circular flow areas were assumed. The dimensions for the geometries analyzed are indicated in Figure 3.1. The analysis model neglects heat transfer between adjacent channels. A 50- μ m-thick (2-mil) aluminum layer is assumed present on the interior of the 1/10-mm-thick fabric wall. One side of the radiator is assumed to be facing space; the other side faces earth.

Case	Diameter, cm	Length, m	Flow Area, m ²	Channel Projected Area, 2Ld _t , m ²
1	3.57	5	1.0×10^{-3}	0.378
2	1.0	5	7.85×10^{-5}	0.121
3	2.5	5	4.91×10^{-4}	0.271
4	3.57	10	1.0×10^{-3}	0.756
5	1.0	10	7.85×10^{-5}	0.242
6	2.5	10	4.91×10^{-4}	0.542

FIGURE 3.1. Circular Direct Radiator Design

The radiation boundary temperature of space was assumed to be 4 K and for earth, 260 K. Radiation heat transfer at the surfaces was calculated using an effective temperature-dependent, heat transfer coefficient at the aluminum liner outer surface. The heat transfer coefficients for the sides facing space and earth are different. Both heat transfer coefficient values were calculated accounting for the fabric material transmissivity, reflectance, and absorption. Heat conduction through the fabric was assumed small and negligible. The thermal resistance diagram used to determine the radiative-heat-transfer coefficients is shown in Figure 3.2. The relationships

FIGURE 3.2. Thermal Resistance Model for Radiator Surface

indicated in this figure were simplified by assuming that space or earth are black bodies ($\epsilon_2 = 1.0$), that all viewfactors are equal to 1 ($F_{12} = F_{1f} = F_{2f} = 1.0$), and A_2 is much larger than A_1 . Assuming that treated aluminum has an emissivity of 0.6 and that the fabric is made of SiO_2 with an areal density of 120 g/m^2 ($\rho = 0.46$, $\tau = 0.5$, $\alpha = \epsilon = 0.04$), the radiative-heat-transfer coefficients indicated on Table 3.1 are calculated using

$$h = \frac{\sigma(T_1^4 - T_2^4)/(T_1 - T_2)}{\frac{1}{\frac{\epsilon_1}{1 - \epsilon_1} + \rho_f} + \frac{1}{\tau_f + \epsilon_f}}$$

Using the cycle values supplied for the 30-kW closed Brayton cycle, the mass fraction of the helium was calculated to be 0.072 and the xenon mass fraction was calculated to be 0.928. These values translate to a helium volume fraction of 0.716 and a xenon fraction of 0.284. Assuming all helium/xenon flow in the channels, the results of the ATHENA parametric study are indicated on Figures 3.3, 3.4, and 3.5. These figures reveal that larger flow

TABLE 3.1. Calculated Radiative Heat Transfer Coefficients with Aluminum Surface Treatment

Temperature, K	Space, ($T_2 = 4$ K) $h(\text{W/m}^2 \text{ K})$	Earth, ($T_2 = 260$ K) $h(\text{W/m}^2 \text{ K})$
300	0.6568	2.1184
350	1.0410	2.7834
400	1.5517	3.6056
450	2.2069	4.6031
500	3.0246	5.7937
550	4.0227	7.1956
600	5.2194	8.8267

FIGURE 3.3. Channel Exit Temperature For Helium/Xenon Flow

areas remove more heat for an equivalent length. However, the temperature drop along the channel is larger for the smaller areas. The design conditions of the assumed Brayton cycle require a radiator outlet temperature of 366.5 K. The required mass flow to meet this criterion for a given flow area

FIGURE 3.4. Radiator Heat Removal Design Curve For Helium/Xenon Flow

and total length are indicated on Figure 3.3. Using this information, the design curves of Figure 3.4 and 3.5 were developed. Figure 3.4 could be used to obtain the required size of the radiator. For instance, if 93 channels are desired for a 79.9-kW, 10-m-long (5-m inlet and 5-m return) radiator, with a $1 \times 10^{-3} \text{ m}^2$ flow area, the required channel mass flux is about $8.5 \text{ kg/m}^2/\text{s}$ with 860 W heat removal per channel. An equivalent 20-m-long radiator would require a mass flux of $18 \text{ kg/m}^2/\text{s}$ and supply a heat removal of 1900 W per channel. The total number of channels required would be 43. Both of these designs would supply a radiator outlet temperature of 366.5 K. The resultant channel pressure drop can be obtained from Figure 3.5. The pressure drop would be about 105 Pa for the 10-m-long design, and 820 Pa for the 20-m design.

Using Figures 3.4 and 3.5, the conditions specified by the designed Brayton cycle can be used to determine the design points of a number of radiators that can meet the desired operating requirements. For the six sample design points, Table 3.2 lists the required projected radiator surface area for one side. The total projected surface area is twice this value. The required projected areas vary between 32 and 35 m^2 . Because the radiator inlet and outlet temperatures are fixed, the temperature gradient along a channel is approximately the same for all the radiator design points. Because the gradients are about equal, the radiator surface area required for rejecting 79.9 kWt at the Brayton-cycle design operating conditions are about equal for all the radiator design points. This occurs because heat transfer from the radiator is dominated by radiation heat transfer from the outer surface. Therefore, it can be concluded that the required radiator surface area is 35 m^2 or slightly less for all radiator design points irrespective of channel flow area or channel mass flow.

The weight of the radiator system for all six sample design points ranges between 28 and 36 kg. Therefore, a slight weight savings exists by using smaller-diameter channels. However, when the projected area is divided by the weight, the resulting values indicated on Table 3.2 fall between 0.85 and 1.02. It should be noted the calculated radiator weight does not include weights for the radiator inlet or outlet headers, any failure isolation equipment, or redundant channels.

TABLE 3.2. Design Conditions for Helium/Xenon Cases Studied

Flow Area, m ²	Channel Length m (a)	Heat Removal per Channel, W	No. Channels (for 79.9 kW removal)	Projected Area, m ² (b)	Weight, (c) kg	Weight Projected Area, kg/m ²	Total Flow Rate, kg/s	Pressure Drop, Pa	Required Pumping Power, W (d)
1x10 ⁻³	10	860 (872) (e)	93 (92)	35.2 (34.8)	35.9 (35.5)	1.02 (1.02)	0.753 (0.745)	105 (103)	17.4 (17.0)
1x10 ⁻³	20	1900	43	32.5	33.2	1.02	0.753	820	135.6
4.91x10 ⁻⁴	10	640	125	33.9	32.9	0.97	0.756	380	62.9
4.91x10 ⁻⁴	20	1370 (1362)	59 (59)	32.0 (32.0)	31.1 (31.1)	0.97 (0.97)	0.743 (0.743)	2450 (2513)	405.3 (416.7)
7.05x10 ⁻⁵	10	275	291	35.2	29.9	0.85	0.736	5850	967.7
7.05x10 ⁻⁵	20	600	134	32.4	27.6	0.85	0.673	40000	6617.0

(a) Total length from radiator inlet to outlet

(b) For one side. Total projected area is twice this value.

(c) Includes liner, fabric, braze, and gas only. Does not include inlet or outlet headers, isolation equipment, or redundant channels.

(d) Assuming pump efficiency of 1.0

(e) Numbers in parentheses represent values obtained from an actual ATHENA run.

The proposed direct radiator design provides a promising method for rejecting heat. The design of a direct radiator for a nuclear helium/xenon cooled Brayton cycle using a metal-lined fabric material can result in lightweight radiator designs with area to weight ratios below 1.0 kg/m^2 . The actual design will be dictated by pumping power, survivability (such as channel redundancy and isolation designs), and fabricability concerns. Therefore, the exact choice of a design is not possible at this early stage. However, the results, curves, and methods presented in this section can be used for scoping studies for future design efforts, which would serve as a basis for more exact design calculations.

3.2 LITHIUM SECONDARY-SYSTEM PUMPED-LOOP RADIATOR

This design removes heat from the helium/xenon primary system through an intermediate heat exchanger. The heat is transferred to a secondary system consisting of a one-phase, liquid-metal working fluid such as lithium or sodium/potassium. Heat is removed from the secondary system through a liquid-metal radiator consisting of one or more flexible radiator panels. These panels would consist of circular channels connected by beryllium spacers that are bonded to the channel liner and act as fins. This radiator system has better heat transfer and survivability characteristics than the direct radiator, but raises other concerns such as compact heat exchanger design, material compatibility, and liquid-metal thaw problems. A detailed analysis of this design was not performed here but is recommended as a future activity.

3.3 HEAT PIPE RADIATOR

An alternative to the pumped loops described above is a variation on the familiar heat pipe technology. Rather than constructing a heat pipe in the traditional manner, with a rigid outer wall, we conceive an advanced heat pipe fabricated in a triple-layered, fully-flexible configuration (see Figure 1.4). The benefits of such an approach are manifold: construction and reproducibility problems are eased; low mass is readily attained; and, as a final boon, flexibility is retained not only during launch/startup, but potentially during the entire lifetime of the radiator system.

Heat pipe analysis is more complex than that of pumped loops because of the larger number of variables, many of which, lacking specific test data, can only be estimated. The following analysis is by no means optimized, as only a few options have been examined and these not fully. We believe that optimized fabric heat pipes could be constructed significantly lighter than the current analysis shows.

The LANL code employed is not a systems code. It treats the heat pipe as a one-dimensional, compressible-flow, thermodynamic system, consisting solely of the working fluid (Prenger 1979). The base code has been modified somewhat by incorporating many other fluid property tables (Brennan and Kroliczek 1979) and by calculating the mass of the fluid and heat pipe components. A number of code runs have been made; sample output is provided in Appendix A. Note that the mass estimate shown (Table 3.3) does include an assumed 0.05-kg/m^2 fabric of braze/bonding material for the metallic liner. (The mass estimates for the pumped-loop designs also included such mass). Because optimal bonding techniques have yet to be developed, it would be premature to include the braze mass used in the current test samples (which ranged up to 0.6 kg/m^2 of fabric). One additional comment: the heat pipe mass/area requirement calculation does not include the mass of the arteries. The reasons are discussed below.

Analysis was begun by examining the simplest wick structure, an annular homogeneous one consisting of one or more layers of ceramic wick bonded to an impermeable metallic foil. In a 5-m condenser length, this design consistently was a poor performer, with an unacceptably low capillary pumping limit. This forced us to examine the currently preferred space heat pipe wick concept, which employs arteries for efficient liquid return to the evaporator (Dunn and Reay 1982, p. 95). Although flexible heat pipe designs employing arteries have been studied and constructed (Ernst 1981; Brennan and Kroliczek 1979, Vol. 1, p. 341), they can be problematic and are usually to be avoided (Josloff, Kirpich, and Biddiscombe 1987).

The design we have selected is assumed to contain artery-type flow passages woven into the wick structure; no separate arteries exist. These artery-type

TABLE 3.3. Heat Pipe Performance Summary

I.D., m	Fluid	Temp., K	Limiting Q, kW/pipe(c)	Mesh(d)	Thick., m	Material	Liner		Pressure Wall		No. (a) Arteries	Mass, (b) kg/pipe	Specific Mass kg/m ² (e)
							Thick., m	Material	Thick., m	Material			
0.01	Hg	575	2.3 C	400	5×10^{-4}	Ceramic	2×10^{-5}	St. Stl.	$8 \times 10^{-5(f)}$	Ceramic	3	0.70	14.3
						St. Stl.				St. Stl.		0.98	19.9
	Dowtherm-A		0.15C			Ceramic		Ti		Ceramic		0.18	3.0
0.025			0 C	100	1×10^{-3}						0	0.80	8.7
			1.2 C	400							8		
	K	810	0.4 C	100							0	0.75	6.3
			0 C	400									
			8.9 C								10		
			7.0 C								5		
						Ti	2.5×10^{-4}		2.5×10^{-4}	Ti		2.1	16.8

(a) Assumed to be part of wick structure and not included separately in mass calculation

(b) For all heat pipes, lengths were: evaporator section, 0.4 m; adiabatic, 0.1 m; condenser, 5.0 m.

(c) C = capillary; E = entrainment; S = sonic (limits)

(d) Per inch.

(e) Assumes heat rejection from both sides of radiator panels consisting solely of contiguous heat pipes and mass based on projected area.

(f) Based on a yarn strength of 1550 MPa (2.25×10^6 psi) (Fareed et al. 1987) and assuming little degradation in strength to about 700 K (Sawko and Tran 1985). If the loss in strength incurred by the twisting of fibers into yarn could be reduced, even greater strength might be feasible. Also, 0.7 MPa (100 psia) pressure and a safety factor of 2 were assumed.

passages should be achievable in a variety of ways and should serve the same function of lessened return liquid flow resistance achieved by true arteries. Given their small diameter, such "arteries" would minimally affect the flexibility of the wick structure. To ensure proper heat pipe performance after prolonged stowage, multiple redundant "arteries" would be required. The proposed design incorporating these "arteries" is currently assumed to consist of a variable-mesh structure, to attain artery pumping power without the associated complexity and fragility. The results of the heat pipe analyses are shown in Table 3.3. All the designs assume radiators constructed of parallel, contiguous heat pipes. For the liquid-metal heat pipes, reductions in mass and improved survivability might result from a finned radiator. The working fluids possessing low thermal conductivities probably would have unacceptable temperature gradients with fins. Sample heat pipe code runs are included in Appendix A. The input values shown were generated using standard equations (Chi 1976, pp. 35-42), and assume the wick pore size and fiber diameters to be equal.

The calculated results indicate that significant mass savings can be realized with the proposed fabric heat pipes. Further optimization would likely improve this mass advantage over conventional heat pipes even further. For example, as illustrated in Table 3.3, a potassium heat pipe operating at 810 K (SP-100 environment) constructed in fabric with a titanium liner is estimated to weigh 10.3 kg/m^2 , while a state-of-the-art, all-titanium heat pipe weighs 26.8 kg/m^2 . To repeat, this 10.3 kg/m^2 is far from fully optimized, although material dimensions are of the same order as current, highly-optimized designs (McNamara et al. 1987), which weigh approximately 20 to 40 kg/m^2 (Dearien and Whitbeck 1987). However, other considerations such as launch or survivability criteria may well have the contrary tendency and increase mass. Only additional analyses and, even more important, experiments, can improve the accuracy of the mass predictions presented here.

Note that a major portion of the mass resides in the wick, as shown in Appendix A. It is entirely possible that the wick thickness may be reduced considerably, or even that the wick may be entirely eliminated for heat pipes

smaller and shorter than the 5-m lengths analyzed here. The corrugated surface of the thin metal liner, naturally formed from internal pressure forces pressing it against the weave of the fabric outer wall, may provide sufficient wicking to feed several return arteries without the need for a conventional wick. This approach has been studied in some detail by LANL (Meier, Girrens, and Dickinson 1980).

smaller and shorter than the 5-6 bands analyzed here. The concept of surface of the thin metal film, actually formed from thermal processes and pressure against the weave of the fabric outward, may provide an indirect working to find several other artifacts without the need for a conventional work. This approach has been studied in some detail by L. H. (1960) and

(continued)

4.0 MICROMETEOROID ANALYSIS

To properly assess the operational lifetime survivability of the heat rejection system, a realistic design will need to address system threats posed by space and man. System threats posed by man include not only passive orbiting space debris, but also active threats, such as directed and kinetic energy weapons.

An assessment of the threat of impact from space orbital debris will not be attempted here because of its complexity and the variability of the threat due to orbital direction, altitude, and the orientation of the heat rejection system to the debris path. However, preliminary calculations by PNL and experiments by the National Aeronautics and Space Administration (NASA) indicate that the threat posed by orbital debris is comparable to that posed by micrometeoroids.

4.1 APPROACH

Meteoroids can be characterized as small solid particles ranging in size from 10^{-12} to 1 gm in mass, traveling through interplanetary space with an average velocity of 20 km/s and a density of 0.5 g/cm^3 . While the masses of micrometeoroids are generally small, the impact of these high-velocity particles on unprotected spacecraft and equipment can result in significant damage. As a consequence, extensive investigations have been and are being made to further characterize their masses, velocities, and quantities, so future spacecraft and equipment designs can be improved for added protection against them. As a result of these studies, the meteoroid mass flux has been characterized as comprising two components--an average sporadic background flux and a stream flux associated with annual meteor showers.

The Near Earth to Lunar Surface Meteoroid Environmental model was developed by NASA to approximate the average sporadic mass flux for use in spacecraft design. To obtain a total meteoroid threat assessment for both the average sporadic and stream meteoroid components, this model has been modified to account for the stream meteoroid component. Figure 4.1 provides an estimate

FIGURE 4.1. Average Cumulative Meteoroid Mass Flux for Near-Earth Orbits

of the average cumulative micrometeoroid mass flux as a function of micrometeoroid mass (Cour-Palais 1969). The corrected total meteoroid mass flux for meteoroid masses of interest, 10^{-6} to 1 gm, is given by

$$\text{Log}_{10} N_t = -14.37 - 1.213 \text{ Log}_{10} m$$

where N_t is the number of meteoroids of mass m or greater, $\text{m}^2\text{-s}$, and m is the meteoroid mass, g.

The total meteoroid mass flux is then corrected for differences between near earth and deep space operations, primarily the shielding effects by the earth, E , and deviations in the mass flux due to gravitational influence, G_e . These, in turn, are calculated as follows:

$$E = 0.5 \{1 + \cos[\arcsin 6378/(6378 + H)]\} \quad (1)$$

where H is the orbital altitude of the spacecraft, km, and

$$G_e = 0.568\{1 + 0.761[6378/(6378 + H)]\}$$

Figure 4.2 shows the product of $G_e E$ as a function of spacecraft orbital altitude.

In assessing the probability of survival for various system configurations, the thickness of component walls necessary to prevent incipient penetration by an impacting particle of specified mass and velocity is given by

$$t = k_1 m_p^{0.352} \rho_p^{0.167} v^{0.875} \quad (2)$$

where t = monolithic barrier thickness, cm

m_p = particle mass, g

ρ_p = particle density, g/cm³

v = particle velocity, km/s

k_1 = constant determined experimentally for materials of specific interest and listed in Table 4.1.

FIGURE 4.2. Product of Earth's Gravitational Defocusing and Body Shielding Factors, $G_e E$, as a Function of Distance from Earth's Center

Because the metallic-ceramic fabric materials in this study are not homogeneous structures, and because of their intrinsically higher yield strengths, it has been suggested(a) that the incipient penetration equation be referenced to aluminum alloys and modified as follows:

$$t = k_1 m_p^{0.352} \rho_p^{0.1667} v^{0.875} \left(\frac{483}{\sigma_{ys}} \right)^{\frac{1}{2}} \quad (3)$$

where k_1 is specified as 0.57 and σ_{ys} is the yield strength for the ceramic fabric fibers (MPa) at operating temperature, assumed to be 3,000 MPa for SiO_2 at 452 K. While it is recognized that fabric materials in general do not demonstrate yield strengths as high as the individual fibers, it is assumed that, because the hypervelocity impact is essentially instantaneous, the

(a) Cour-Palais, B. September 1987. Personal communication, NASA Johnson Space Center, Houston, Texas.

TABLE 4.1. Constants for Threshold Penetration for a Monolithic Barrier at 298 K (Fraas 1986; Lundberg et al. 1982)

Material	k_1
Aluminum alloys	0.54 to 0.57
Stainless steel	0.32 to 0.38
Magnesium-lithium	0.80
Niobium alloys	0.34
Titanium and titanium alloys	0.57

material does not have time to react; the result is ultimate yield strengths that approach the strengths of the fibers themselves. This relationship for incipient barrier thickness can then be transposed to solve the particle mass capable of penetrating a specified wall thickness at a given velocity:

$$m_1 = t^{2.84} k_1^{-2.84} \rho_p^{-0.474} v^{-2.485} \left(\frac{483}{\sigma_{ys}} \right)^{-1.422} \quad (4)$$

Equation (4) can be substituted into Equation (1) and transposed to obtain a direct relation for the micrometeoroid mass flux capable of incipient penetration of a monolithic barrier of a specific thickness.

$$N_{tt} = \frac{4.266 \times 10^{-15} t^{-3.445} k_1^{3.445} \rho_p^{0.575} v^{3.014} \left(\frac{483}{\sigma_{ys}} \right)^{1.725}}{G_e E} \quad (5)$$

Values of N_{tt} are then calculated over the range of velocity intervals and a net flux increment developed. The net flux increment is then multiplied by its associated probability and the results summed to develop a total lethal micrometeoroid mass flux, $N_{\ell t}$. Figure 4.3 provides an estimate of the probability distribution for velocities of observed meteoroids (Cour-Palais

FIGURE 4.3. Probability Distribution of Observed Meteoroid Velocities

1969). In doing so, the mean probability velocity for an impacting particle can be shown to be 25.21 km/s (Fraas 1986).

With the micrometeoroid mass flux so defined, the probability of impact on a specified heat rejection system with known areas and operational lifetime by n or less particles is determined by the Poisson formula,

$$P(h \leq n) = \exp(-N_{lt}At) \sum_{r=0}^{n} \frac{(N_{lt}At)^r}{r!} \quad (6)$$

where $P(h \leq n)$ = probability of impact by n or fewer lethal particles

N_{lt} = total lethal particle flux, particles/m²-s

A = total exposed surface area, m² (2x projected area for flat panels)

t = operational lifetime, s

r = specified number of impacts.

Because the heat rejection systems to be studied will include redundancy, Equation (6) can be used to assess the probability that less than a specific number of penetrations will occur for a specific design. Conversely, it then holds that if various system characteristics are known (i.e., material composition, surface area, number of heat rejection loops, number of allowable penetrations) and system reliability is specified, a required wall thickness can be calculated that will prevent incipient penetration by impacting particles. This, in turn, can be used to calculate corresponding heat rejection system masses.

4.2 PUMPED-LOOP SURVIVABILITY CALCULATIONS

Based on the equations and methodology described, a computer program was written to perform survivability calculations for a space-based heat rejection system. The program assumes that the fabric materials proposed behave as monolith barriers ratioed to a reference yield strength of 483 MPa (70 k psia) (equivalent to Al-6061T at room temperature). In addition, the code assumes that the radiator surfaces are comprised of tubes only, without the benefit of radiating fins, arranged in a flat panel configuration. Based on these assumptions, the total projected surface area of the heat rejection system is calculated for use in the survivability calculations.

The following items must be specified:

- the minimum required number of loops that the system must contain at end of life
- number of additional loops to consider
- the diameter and length of the exposed loops
- density of the fabric and metallic liner materials
- strength of the fabric materials
- operational characteristics of the heat rejection system (i.e., reliability, operational lifetime, and orbital factors).

The micrometeoroid lethal mass flux is then found by using a Newton-Raphson convergence method, and the associated system armor mass is calculated. The number of total system loops is incremented by 1, and the process is repeated. A listing of the survivability assessment program and calculation runs are found in Appendix B.

Using this program, a series of parametric cases was run to determine the probability of system survivability and the effect of system redundancy on total radiator mass for the pumped-loop systems described in Table 3.2. In these calculations, it is assumed that each radiator system has an operational lifetime of 10 yr and is in near-earth orbit such that the orbital correction factor is 0.7. The fabric composite material is composed of a 0.002-cm-thick titanium alloy inner liner, for which no credit is taken in the survivability calculations. The outer structural material is assumed to be a silicon dioxide fabric material with a density of 1.2 g/cm^3 (assuming a 50% porosity) and an ultimate fiber yield strength of 3,000 MPa at 452 K. In addition, it is assumed that each pumped loop can be isolated as it fails. The reader should note that the intent of these calculations is to assist in defining the cost of system survivability. System masses reported here are for fabric/ foil materials alone and do not include working fluids, valves, and similar items, in the radiator system mass estimates. Tables 4.2 through 4.7 provide the results of these calculations.

Based on these studies, it can be seen that significant increases in system reliability can be achieved with relatively small increases in the total number of pumped loops and masses. Furthermore, it is observed that the benefit of adding redundant pumped loops quickly reaches an asymptotic value. A beneficial consequence of this is that the eventual system designer can choose from a broad range of fabric material thickness without significantly affecting the heat rejection system mass. It is believed that such choices will then translate into reduced development and construction costs, as the designer is allowed to choose from existing commercial stocks. Table 4.8 provides a mass estimate for a pumped-loop system in which it is assumed that three individual pumped loops share a common isolation valve, and a loss of any one loop results in the loss of the others. The coupling of three

TABLE 4.2. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 1 [helium/xenon pumped loop, flow area $1.0 \times 10^{-3} \text{ m}^2$, channel length 10 m, required number of loops at end-of-life (EOL) 93, unarmored mass = 35.9 kg^(a)]

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	123	162.9	88.3
0.98	124	166.5	89.0
0.99	125	169.0	89.7
0.999	128	176.0	91.9

TABLE 4.3. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 2 (helium/xenon pumped loop, flow area $1.0 \times 10^{-3} \text{ m}^2$, channel length 20 m, required number of loops at EOL 43, unarmored mass = 33.2 kg^(a))

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	58	189.6	84.0
0.98	59	195.9	85.4
0.99	60	200.1	86.9
0.999	62	212.5	89.7

TABLE 4.4. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 3 (helium/xenon pumped loop, flow area $4.91 \times 10^{-4} \text{ m}^2$, channel length 10 m, required number of loops at EOL 125, unarmored mass = 32.9 kg^(a))

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	164	138.2	82.5
0.98	165	140.8	83.0
0.99	166	142.6	83.5
0.999	170	147.5	85.5

(a) From Table 3.2

(b) Radiator mass includes liner and fabric structure/armor only. Does not include inlet or outlet headers or isolation equipment.

TABLE 4.5. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 4 (helium/xenon pumped loop, flow area $4.91 \times 10^{-3} \text{ m}^2$, channel length 20 m, required number of loops at EOL 59, unarmored mass = 31.1 kg^(a))

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	79	162.8	80.0
0.98	80	167.3	81.0
0.99	81	170.4	82.0
0.999	83	179.4	84.0

TABLE 4.6. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 5 (helium/xenon pumped loop, flow area $7.85 \times 10^{-3} \text{ m}^2$, channel length 10 m, required number of loops at EOL 291, unarmored mass = 29.9 kg^(a))

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	372	98.6	74.6
0.98	375	99.8	75.2
0.99	377	100.6	75.6
0.999	383	102.9	76.8

TABLE 4.7. Effects of System Reliability on Pumped-Loop Redundancy and System Mass, Case 6 (helium/xenon pumped loop, flow area $7.85 \times 10^{-3} \text{ m}^2$, channel length 20 m, required number of loops at EOL 134 unarmored mass = 27.6 kg^(a))

Probability of Survival	Total Number of Loops	Total Radiator Mass, kg ^(b)	2x Projected Surface Area, m ²
0.95	175	111.4	70.4
0.98	177	113.5	71.2
0.99	178	114.8	71.6
0.999	180	118.8	72.4

(a) From Table 3.2

(b) Radiator mass includes liner and fabric structure/armor only. Does not include inlet or outlet headers or isolation equipment.

TABLE 4.8. System Mass Estimate of Helium/Xenon Pumped Loop with Three Pumped Loops Sharing a Common Isolation Valve (system reliability 0.999)

Case	Cross Section, m^2	No. of Loops at EOL	No. of Loops at BOL ^(a)	Radiator Mass, kg ^(b)	2x Projected Area, m^2
1	1×10^{-3}	31	46	204.9	100.4
2	1×10^{-3}	14	22	257.45	98.3
3	4.91×10^{-4}	42	60	170.2	91.5
4	4.91×10^{-4}	20	30	218.7	93.0
5	7.85×10^{-5}	97	132	113.4	79.8
6	7.85×10^{-5}	45	64	136.3	78.0

(a) Beginning-of-life

(b) Radiator mass includes metallic liner and fabric structure/armor only.
Does not include inlet or outlet headers or isolation equipment.

individual pumped loops into a common isolation valve then results in a 40 to 60% increase in the mass of the pumped loop panels to ensure an adequate degree of survivability.

4.3 HEAT PIPE SURVIVABILITY CALCULATIONS

During this study, it was postulated that the development of fabric composite materials might also be applicable to conventional heat pipe systems by reducing the weight penalty associated with traditional monolithic metal armors, such as titanium. To study the effects of this substitution of armor materials, two heat pipe systems were examined.

In Case 1, a heat pipe system was chosen based on a Dowtherm-A^R working fluid operating at 452 K, with a heat rejection load of 79.9 kW. Each heat pipe is assumed to be 2.5 cm in diameter and 5 m long, and to have a surface emissivity of 0.6. In Calculation A, the heat pipe is made of titanium. In Calculation B, the heat pipe is constructed of a fabric material with a titanium liner and fabric wick. In each calculation, the internal configuration of the heat pipe includes a 400-mesh wick, 0.1 cm thick, with eight arteries 0.1 cm

^RDowtherm-A is a heat transfer agent manufactured by Dow Chemical Co., Midland, Michigan.

in diameter. Each heat pipe is estimated to carry 0.353 kW, which, in turn, requires the system to contain 226 working heat pipes at EOL. To ensure an overall system reliability of 0.99, the minimum mass system will contain approximately 7% redundant heat pipes.

In Case 2, a heat pipe system was chosen based on current SP-100 requirements. For this system, the working fluid is potassium operating at 800 K. Total heat rejection load requirements are estimated to be approximately 2.2 MW. Each heat pipe is assumed to be 2.5 cm in diameter and 5 m long, and to have a surface emissivity of 0.6. In Calculation A, the heat pipe is made of titanium. In Calculation B, the heat pipe is made of a fabric with a titanium liner and a metallic coated fiber wick. In each calculation, the internal configuration of the heat pipe includes a 400-mesh wick, 0.1 cm thick, with five arteries 0.1 cm in diameter. Each heat pipe is estimated to carry 3.49 kW, which then requires the heat rejection system to contain 630 working heat pipes at EOL. To ensure an overall system reliability of 0.99, the minimum mass system will contain approximately 202 redundant heat pipes.

The results of these calculations are provided in Table 4.9. Based on these, it is expected that the use of fabric composite materials for replacement of traditional metal components offers significant mass savings to existing heat pipes, namely, a mass reduction of about 75%. It should be noted that, although the heat pipe configurations studied here are not optimal, it is believed that a ratio of titanium to fabric radiator masses will not be greatly affected by optimization.

TABLE 4.9. System Mass Estimate for Postulated Heat Pipe Systems as a Function of Structural Material (system reliability 0.99, operational lifetime 10 years)

<u>Case 1</u>	Heat rejection load:	79.9 kW	Radiator temperature:	452 K
	Working fluid:	Dowtherm-A	Surface emissivity:	0.6
	Pipe length:	5.0 m	Pipe diameter:	2.5 cm

<u>Calculation</u>	<u>Structure</u>	<u>Heat Pipe Mass As Designed (a)</u>	<u>Armor Mass(b)</u>	<u>Total Mass(b)</u>	<u>Heat Pipe Mass/Projected Area</u>
A	Titanium	--	2.94 kg	3.66 kg	29.12 kg/m ²
B	Fabric- foil	0.80 kg	0.27 kg	1.07 kg(c)	8.56 kg/m ²

<u>Case 2</u>	Heat rejection load:	2.2 MW	Radiator temperature:	800 K
	Working fluid:	Potassium	Surface emissivity:	0.6
	Pipe length:	5.0 m	Pipe diameter:	2.5 cm

<u>Calculation</u>	<u>Structure</u>	<u>Heat Pipe Mass As Designed (a)</u>	<u>Armor Mass(b)</u>	<u>Total Mass(b)</u>	<u>Heat Pipe Mass/Projected Area</u>
A	Titanium	2.10 kg	1.90 kg	4.00 kg	32.00 kg/m ²
B	Fabric- foil	0.75 kg	0.27 kg	1.02 kg	8.16 kg/m ²

(a) As calculated in Table 3.3

(b) Mass per heat pipe

(c) Indicates over design

5.0 RADIATOR OPERATION

A successful radiator design must not only work well during steady-state operation, but must also be able to adjust to varying conditions. These include startup/shutdown, as well as transients of all degrees of severity.

5.1 STARTUP/TRANSIENTS/SHUTDOWN

Although each radiator concept would have some unique aspects, a general approach to transients can be developed. All radiators would be stowed in a compact configuration for launch. Deployment would occur after spacecraft insertion into orbit. Heat pipes would most likely deploy and expand solely from vapor pressure forces (Ponnappan, Beam, and Mahefkey 1986). The pumped loops might deploy solely from pump-generated pressure heads, or might require additional mechanical aids (e.g., such as currently used to deploy spacecraft booms and antennae). The presence of mechanical devices would serve the additional function of permitting radiator shape alteration at will.

Startup should be simplified by the flexible nature of the proposed radiator design. That is, the long heat paths present in more conventional radiator designs are inherently avoided by the stowed, flexible radiators. As for transients, these again are more readily surmounted by the proposed designs. Variations in heat load could be adjusted to in several ways: for the heat pipe radiator, a higher temperature/pressure operating point is readily achievable given the inherent tensile strength of the fabric containment at high temperature. Similar comments hold for the pumped loops. The most severe possible transient (due to a laser threat) would require a hardened design. Such is certainly within reason, with minimal mass penalty. Lower heat fluxes could be handled either by lowering the radiator temperature or by retracting a suitable portion of the radiator area.

Shutdown, if it becomes necessary, could be the reverse of the startup scenario. Mechanical reels or other retraction devices could stow nonoperating radiators, reducing drag on the spacecraft while simultaneously making it more difficult to track. The details of a planned shutdown operation have not been addressed here.

5.2 SHAPE ALTERATION

A unique feature of flexible radiators is the inherent ability to alter the shape of the radiator in radical ways. This freedom opens up many possibilities. The signature of the spacecraft could be altered to make identification more difficult. Threats could be countered by stowing, retracting, or otherwise moving the threatened radiator panels; new panels could be unfurled to replace damaged ones and/or change the radiator shape.

6.0 FEASIBILITY ISSUES

In this section, the key technical issues are described. As with any new technology, numerous issues have been identified here that inhibit the immediate application of fabric radiators.

6.1 KEY TECHNICAL ISSUES IDENTIFIED

These issues are listed in Table 6.1, as well as relevant comments regarding their current status and anticipated resolution.

6.2 TECHNICAL ISSUES OUTSTANDING

Among the issues shown in Table 6.1, several are crucial to development of a fabric radiator technology base. Survivability, retention of long-term flexibility, and long-term compatibility with the working fluid are the chief issues that must be resolved to establish the validity of the proposed radiator

TABLE 6.1. Key Technical Issues

<u>Issue</u>	<u>Status</u>	<u>Comments</u>
Flexibility	Demonstrated	Needs to be explored more
Survivability	Not demonstrated	Must be demonstrated for pumped loops
Mass	Low mass potential demonstrated analytically	Low mass appears feasible
Compatibility	Not demonstrated	Requires selection of proper materials
Bonding	Demonstrated	Needs to be explored more and optimized
Startup	Not demonstrated	Only experiment can verify startup/deployment features
Performance	Demonstrated analytically	Only experiment can verify
Emittance/transmittance	Measured	Need to be explored more and optimized

concept. Although we have explored these issues, definitive data are lacking. The next step in a fabric radiator development program would be to acquire such data, which would serve both to confirm the claimed advantages of fabric radiators over conventional approaches and provide design and system optimization inputs. If a viable fabric radiator design is achievable, system optimization may show that the entire spacecraft is beneficially affected through incorporation of this radiator. One can envision that a power system could readily benefit from the lighter mass of a fabric radiator, permitting operation at lower heat rejection temperature and increasing power system reliability through a decreased reactor temperature. Conversely, the high-temperature potential of ceramic materials may be exploited to reduce radiator area, assuming the rest of the spacecraft's power system were not the limiting factor on the temperature.

6.3 RECOMMENDED APPROACH

Based on the results from this research effort, we feel that the earliest application of fabric technology could occur in heat pipes. These largely avoid the survivability issue; until that is resolved, pumped loops cannot be recommended.

It is proposed that an experimental fabric heat pipe development program be initiated. The work would encompass a number of activities: study of ceramic wick/working fluid compatibility, bonding techniques, and optimization of the ceramic wick structure and weave. If a ceramic wick is found to be incompatible for long-term use with desirable working fluids, currently available (individual) fiber coating techniques could be used to provide compatibility while retaining most of the all-ceramic wick's mass advantage (Research and Development 1987). In conjunction with these material studies, a proof-of-concept low-temperature heat pipe should be fabricated and tested. We project a 1-year program, at the end of which fabric heat pipe technology should be firmly established.

A parallel effort, in pumped-loop fabric radiator survivability, would serve to determine if such is realistically attainable. All possible means of increasing survivability need to be explored; we have thought of many approaches, but have been unable to explore them in any detail. Some of the candidate methods are shown in Table 6.2. Note that survivability consists of hardening as well as improved methods of failure detection. If a design cannot be hardened sufficiently to be survivable, it may still be possible to generate an acceptable design. Accurate detection of the location and size of a failure point allows one to take active countermeasures to limit or repair the damage inflicted.

TABLE 6.2. Approaches for Enhancing Survivability Against Micrometeoroid Punctures

Method	Comments
Sealing	
Pinch-off	Would take advantage of fabric's compliance
Sealant	Problematic
Self-healing	Eutectic braze development required
Redundancy	Of benefit only if isolation of punctured panels can be ensured
Detection	
Infrared	Very sensitive; could triangulate on leak location
Sonic	Might work for gases
ΔP	Probably not feasible
Fluid inventory	Gross measure only
Temperature change	Gross measure only

7.0 CONCLUSIONS

The design of large, high-temperature space radiators is still an evolving discipline. We have expanded the data base available to a designer by studying a novel class of radiators, those constructed of ceramic fabrics lined with a metal foil. It appears that fabric radiators have features not available in more conventional radiator types, while offering the light weight and stowability of advanced radiators, such as liquid droplet. Experiments performed to date show that fabrics can be treated to become impermeable to the working fluid while retaining flexibility. This inherent feature of fabrics, when allied to their low density, provides a material well-adapted to radiator construction. As study proceeds, the advantages of fabric over other radiator types will become even more evident.

A direct pumped-loop fabric radiator system offers significant mass savings over conventional radiators and even over fabric heat pipes. An indirect pumped-loop system has not been analyzed completely, and probably would not offer as great a mass saving. Both pumped-loop systems are viable only if some means is developed to ensure a high survivability. Barring that, the heat pipe system is the obvious choice, given its inherent redundancy. But a heat pipe radiator system exacts a price in terms of added complexity and mass. Determination and design of an optimized radiator system remains as a task for future endeavors.

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APPENDIX A

SAMPLE HEAT PIPE CODE RESULTS

APPENDIX A

SAMPLE HEAT PIPE CODE RESULTS

Heat Pipe Analysis: Power Vs Tee

Total Number of Arteries	=	5
Artery Inside Radius	=	.50000E-01
Artery Pumping Radius	=	.31800E-02
Artery Geometry	=	1 Circular
No. of Art. Not Operating	=	0
Artery Screen Thickness	=	.30000E-01
Wick Porosity	=	.50000E+00
Wick Permeability	=	.99000E-07
Distrib. Screen Thickness	=	.50000E-01
Evaporator Length	=	.40000E+02
Adiabatic Length	=	.10000E+02
Condenser Length	=	.50000E+03
Angle From Horizontal	=	.00000E+00
Pipe Inside Radius	=	.12500E+01
Working Fluid No.	=	21 Potassium
Pipe Effective Length	=	.28000E+03
Vapor Passage Area	=	.40542E+01
Total Liquid Flow Area	=	.39270E-01
Wicking Screen Thickness	=	.10000E+00
Wicking Density	=	.24000E+01
Metallic Liner Thickness	=	.20000E-02
Metallic Liner Density	=	.45000E+01
Pressure Wall Thickness	=	.40000E-02
Pressure Wall Density	=	.24000E+01

Tee Power	-----Pressure Drops-----				Temperature		Limits Qsonic Qentrn
	Liquid Hydrostatic	Vapor(L) Vapor(T)	Inertial Capillary	Evaporator Adiabatic Condenser	Tbe Tbeson	Tbc Tbcson	
650. -2687.	8603. 0.	0. 27530.	47639. 55579.	49679. -2024. -680.	795. 677.	674. 637.	602. 4925.
670. -3264.	10115. 0.	0. 24979.	45476. 54748.	47326. -2165. -528.	795. 701.	690. 659.	959. 6077.
690. -3910.	11767. 0.	0. 22771.	43452. 53918.	45138. -2570. -417.	796. 724.	709. 680.	1482. 7389.

Tee Power	Liquid Hydrostatic	-----Pressure Drops-----			Temperature		Limits Qsonic Qcntrn
		Vapor(L) Vapor(T)	Inertial Capillary	Evaporator Adiabatic Condenser	Tbe Tbeson	Tbc Tbcson	
710.	13613.	0.	41936.	43492.	799.	730.	2225.
-4644.	0.	21017.	53088.	-3681. -337.	747.	701.	8865.

Flow is choked, Mach Number exceeds Unity at Exit

730.	16227.	0.	43862.	45408.	807.	758.	3255.
-5670.	0.	20882.	52258.	-9089. -288.	769.	722.	10505.

Flow is choked, Mach Number exceeds Unity at Exit

750.	16329.	0.	33114.	34273.	804.	755.	4651.
-5816.	0.	15647.	51428.	1034. -208.	791.	742.	12306.

770.	17269.	0.	28133.	29098.	810.	770.	6501.
6263.	0.	13038.	50597.	4392. -163.	813.	761.	14267.

790.	19360.	0.	27319.	28226.	823.	794.	8909.
7142.	0.	12244.	49767.	2320. -139.	834.	780.	16272.

810.	21137.	0.	25503.	26328.	835.	814.	11989.
7916.	0.	11136.	48937.	1588. -117.	855.	799.	18554.

830.	22775.	0.	23481.	24224.	849.	834.	15869.
8643.	0.	10030.	48107.	1206. -98.	877.	818.	20982.

850.	24291.	0.	21432.	22097.	864.	853.	20689.
9329.	0.	8983.	47277.	971. -83.	898.	836.	24615.

Vapor Mass(kg)	Liquid Mass(kg)	Wick Mass(kg)	Metallic Liner Mass(kg)	Pressure Wall Mass(kg)	Total System Mass(kg)	Density 2-Sided, e=.6 (kg/m ²)
.0002	.1529	.5184	.0389	.0415	.7518	4.9884

Heat Pipe Analysis: Power Vs Tee

```

Total Number of Arteries      = 5
Artery Inside Radius          = .50000E-01
Artery Pumping Radius         = .31800E-02
Artery Geometry               = 1 Circular
No. of Art. Not Operating     = 0
Artery Screen Thickness       = .30000E-01
Wick Porosity                 = .50000E+00
Wick Permeability             = .99000E-07
Distrib. Screen Thickness     = .50000E-01
Evaporator Length            = .40000E+02
Adiabatic Length              = .10000E+02
Condenser Length              = .50000E+03
Angle From Horizontal         = .00000E+00
Pipe Inside Radius            = .12500E+01
Working Fluid No.             = 21 Potassium
Pipe Effective Length         = .28000E+03
Vapor Passage Area            = .40542E+01
Total Liquid Flow Area        = .39270E-01
Wicking Screen Thickness      = .10000E+00
Wicking Density               = .45000E+01
Metallic Liner Thickness      = .25000E-01
Metallic Liner Density        = .45000E+01
Pressure Wall Thickness       = .25000E-01
Pressure Wall Density         = .45000E+01

```

Tee Power	-----Pressure Drops-----				Temperature		Limits Qsonic Qcntrn
	Liquid Hydrostatic	Vapor(L) Vapor(T)	Inertial Capillary	Evaporator Adiabatic Condenser	Tbe Tbeson	Tbc Tbcson	
650. -2687.	8603. 0.	0. 27530.	47639. 55579.	49679. -2024. -680.	795. 677.	674. 637.	602. 4925.
670. -3264.	10115. 0.	0. 24979.	45476. 54748.	47326. -2165. -528.	795. 701.	690. 659.	959. 6077.
690. -3910.	11767. 0.	0. 22771.	43452. 53918.	45138. -2570. -417.	796. 724.	709. 680.	1482. 7389.
710. -4644.	13613. 0.	0. 21017.	41936. 53088.	43492. -3681. -337.	799. 747.	730. 701.	2225. 8865.

-----Pressure Drops-----					Temperature		Limits
Tee	Liquid	Vapor(L)	Inertial	Evaporator	Tbe	Tbc	Qsonic
Power	Hydrostatic	Vapor(T)	Capillary	Adiabatic	Tbeson	Tbcson	Qentrn
Flow is choked, Mach Number exceeds Unity at Exit							
730.	16227.	0.	43862.	45408.	807.	758.	3255.
-5670.	0.	20882.	52258.	-9089.	769.	722.	10505.
				-288.			
Flow is choked, Mach Number exceeds Unity at Exit							
750.	16329.	0.	33114.	34273.	804.	755.	4651.
-5816.	0.	15647.	51428.	1034.	791.	742.	12306.
				-208.			
770.	17269.	0.	28133.	29098.	810.	770.	6501.
6263.	0.	13038.	50597.	4392.	813.	761.	14267.
				-163.			
790.	19360.	0.	27319.	28226.	823.	794.	8909.
7142.	0.	12244.	49767.	2320.	834.	780.	16272.
				-139.			
810.	21137.	0.	25503.	26328.	835.	814.	11989.
7916.	0.	11136.	48937.	1588.	855.	799.	18554.
				-117.			
830.	22775.	0.	23481.	24224.	849.	834.	15869.
8643.	0.	10030.	48107.	1206.	877.	818.	20982.
				-98.			
850.	24291.	0.	21432.	22097.	864.	853.	20689.
9329.	0.	8983.	47277.	971.	898.	836.	24615.
				-83.			

Vapor	Liquid	Wick	Metallic	Pressure	Total	Density
Mass(kg)	Mass(kg)	Mass(kg)	Liner	Wall	System	2-Sided, e=.6
			Mass(kg)	Mass(kg)	Mass(kg)	(kg/m ²)
.0002	.1529	.9719	.4860	.4860	2.0970	13.4423

APPENDIX B

SURVIVABILITY PROGRAM LISTING

APPENDIX B

SURVIVABILITY PROGRAM LISTING

```
c Radiator Survivability Program
c Microsoft FORTRAN77
c 15 July 1987
c
c Program divided into Subroutines Setup, Output, Film,
c Threat, and Mass
c
c Common As,Esp,Ireg,Ys,C1,Rho(2),PI,Vmass,Tmass,Pmass,Bmass,To,Np,
+R,GeE,D,T1,Ast,Tn(101),Apm(101),Sth(101),Tm,Apert
c
c Real*8 Rp,dRp,G,Tn,Xmp,Apm,Sth,Sum,Sum1,X,Factr
c
c Open(7,file='output.txt',status='new')
c
c Call Setup
c Call Threat
c Call Output
c Stop
c End
c
c Subroutine Setup
c Initialization of Parameters
c
c Input System Parameters
c
c Common As,Esp,Ireg,Ys,C1,Rho(2),PI,Vmass,Tmass,Pmass,Bmass,To,Np,
+R,GeE,D,T1,Ast,Tn(101),Apm(101),Sth(101),Tm,Apert
c
c Real*8 Rp,dRp,G,Tn,Xmp,Apm,Sth,Sum,Sum1,X,Factr
c
c Pi=3.1415
c Esp=.00005
c Write (*,101)
c Write (*,109)
c Write (*,001)
c Write (*,002)
c Write (*,109)
c Write (*,024)
c Read (*,102) D
c Write (*,025)
c Read (*,102) T1
c Write (*,D16)
```

```

Read (*,104) Ireg
Write (*,020)
Read (*,104) Np
Write (*,021)
Read (*,102) To
Write (*,022)
Read (*,102) R
Write (*,023)
Read (*,102) GeE
Write (*,026)
Read (*,102) Tm
Write (*,030)
Read (*,102) Rho(1)
Write (*,031)
Read (*,102) Rho(2)
Write (*,028)
Read (*,102) Ys
Write (*,029)
Read (*,102) C1
Write (*,101)

c
c Calculate System Parameters
c
c Calculate Surface Area and Operational Lifetime (s)
c
    To = To*24.*365.*3600.
    As=(Pi*D*.01*Tl*(Ireg))
    Apert = As/Ireg
    Ast=(D*.02*Tl*Ireg)
    Return

c
c Format Statements
c
001 Format(1X,'Radiator Survivability Program.')
002 Format(1X,'All values will be expressed Metric Units, (unless othe
+wise noted).')
016 Format(/1X,'What is the Required Number of Radiator Tubes to be us
+ed? ')
020 Format(/1X,'What is the Allowable Number of Penetrations?')
021 Format(/1X,'What is the Design Life Time (years)? ')
022 Format(/1X,'What is the System Reliability?')
023 Format(/1X,'What is the Orbital Characteristic Factor? ')
024 Format(/1X,'What is the Tube Diameter (cm)?')
025 Format(/1X,'What is the Radiator Length? ')
026 Format(/1X,'What is the Inner Membrane Thickness (cm)? ')
027 Format(/1X,'What is the System Pressure (psia)? ')
028 Format(/1X,'What is the Material Yield Strength (kpsi)? (Reference
+ Material 70 kpsi)')
029 Format(/1X,'What is the Material Constant K1? ')
030 Format(/1X,'What is the Inner Membrane Density (g/cm^3)? ')

```

```

031 Format(/1X,'What is the Sheathing Membrane Density (g/cm^3)? ')
101 Format(1X)
102 Format(F8.2)
103 Format(2F8.2)
104 Format(I5)
107 Format(6X,I5,4(4X,F8.4))
109 Format(1X,70('*'))
117 Format(6X,I5,4X,F8.6,4X,F8.4,2(4X,F8.2))
    Stop
    End

c
    Subroutine Threat
c
c Calculate Poisson Distribution Values and Perform Newton-Raphson
c Method to converge on solution
c
    Common As,Esp,Ireg,Ys,C1,Rho(2),PI,Vmass,Tmass,Pmass,Bmass,To,Np,
    +R,GeE,D,Tl,Ast,Tn(101),Apm(101),Sth(101),Tm,Apert
c
    Real*8 Rp,dRp,G,Tn,Xmp,Apm,Sth,Sum,Sum1,X,Factr
c
c Function of Micrometeoroid Particle Size
c
    XMP(X) = 10**(-(DLOG10(X)+14.37)/1.213)
c
c Operational Lifetime, Reliability, and Area Constant
c
    G = GeE*To
    At= Apert*Ireg
c Calculate Lifetime limiting Particle Flux
c
    Tn(1) = (-Log(R)/(G*At))
c
    Do 5 I = 1,Np
        Factr=1.
        Sum=1.
        Sum1=1.
        Tn(I+1)=Tn(I)
        At=Apert*(Ireg+I)
007    Do 10 J=1,I
            Factr=Factr*Real(J)
            Sum=Sum+((G*At*Tn(I+1))**J)/Factr
010    Sum1=((G*At*Tn(I+1))**J)/Factr
        Rp=Dexp(-G*At*Tn(I+1))*Sum
        dRp=-G*At*Dexp(-G*At*Tn(I+1))*Sum1
        If (Abs(Rp-R).Le.Esp) Then
            Goto 5
        Else
            Tn(I+1)=Tn(I+1)-(Rp-R)/dRp
        Endif

```

```

        Factr=1.
        Sum=1.
        Sum1=1.
        Goto 7
005  Continue
c
c Required Monolithic Barrier Thickness
c
        Do 30 I=1,Np+1
            Apm(I) = Xmp(Tn(I))
            amm = (.5)**.1667
            ratstr = (70/Ys)**.5
030  Sth(I) = C1*16.83**.875*ratstr*amm*Apm(I)**.352
        Return
        Stop
        End
c
        Subroutine Output
c
c Print out System Results
c
        Common As,Esp,Ireg,Ys,C1,Rho(2),PI,Vmass,Tmass,Pmass,Bmass,To,Np,
+R,GeE,D,Tl,Ast,Tn(101),Apm(101),Sth(101),Tm,Apert
c
        Real*8 Rp,dRp,G,Tn,Xmp,Apm,Sth,Sum,Sum1,X,Factr
c
        Too=(To/(24.*365.*3600.))
c
        Write (7,101)
        Write (7,103)
        Write (7,101)
        Write (7,105)
        Write (7,107) As,Tl,D
        Write (7,109)
        Write (7,111) Ireg,R,Too
        Write (7,113)
        Write (7,115) GeE,Tm,Ys
        Write (7,117)
        Write (7,119) C1,Rho(2),Rho(1)
        Write (7,121)
        Write (7,123)
c
        Do 880 J=1,Np
            Bm = (Sth(J)*Rho(2)+Tm*Rho(1))*PI*D*Tl*(Ireg+J)/(10.)
            Area=(D*(Ireg+J)*.02*Tl)
880  Write (7,125) J-1,Tn(J),Apm(J),Sth(J),Bm,Area
        Return

```



```

C
C Format Statements
C
101 Format(1X,70('*'))
103 Format (1X,'Radiator Survivability Calculation Results ')
105 Format (/1X,'Required Area      Radiator Length      Tube D
+iameter')
107 Format (3X,F8.2,' m^2',14X,F8.2,' m',13X,F8.2,' cm')
109 Format (/1X,'Min. No. Tubes      Reliability      Lifeti
+me')
111 Format (5X,I5,21X,F6.4,15X,F8.2,' years')
113 Format (/1X,'Orbital Factor      Membrane Thickness      Sheath
+YS ')
115 Format (2X,F8.2,18X,F8.4,' cm',12X,F8.2,' kpsi')
117 Format (/1X,'K1 Factor      Sheath Density      Membra
+ne Density ')
119 Format (2X,F8.4,18X,F8.2,' g/cm^3',8X,F8.2,' g/cm^3 ')
121 Format (/1X,' Allowable      Particle Flux      Particle Mass      Wal
+l Thickness      Panel Mass      Projected Area')
123 Format (1X,'Penetrations      (part/m^2-s)      (grams)
+ (cm)      (kg)      (m^2) '/')
125 Format (3X,I5,9X,E10.4,10X,F8.6,8X,F8.4,6X,F8.2,6X,F8.2)
Return
Stop
End

```


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