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Solar Dynamic Heat Rejection Technology Task 1 Final Report System Concept Development

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EXECUTIVE SUMMARY

This report presents the results of a concept development study of heat rejection systems for Space Station solar dynamic power systems based on the Closed Brayton Cycle (CBC) and the Organic Rankine Cycle (ORC). The heat rejection system concepts are based on recent developments in high thermal transport capacity heat pipe radiators. The thermal performance and weights of each of the heat rejection system elements have been addressed in detail, including the following items:

- Heat pipes
- Radiator panels
- Heat exchanger
- Radiator/heat exchanger interface
- Transport loop
- Radiator surface coating
- Assembly and maintenance.

The monogroove heat pipe, which is currently under development by Grumman under contract to NASA-JSC for the Space Station central radiator system, has been shown to be applicable to the heat rejection system for a Rankine cycle. The heat pipe size and weight must be increased, however, in order to meet the higher operating temperature requirements for the ORC. The monogroove heat pipe is not feasible in the temperature range specified for the CBC heat rejection system without incurring a penalty for operating the heat pipes at a lower temperature. The dual-slot heat pipe is a derivative of the monogroove heat pipe which has the potential for lower weight and higher performance, and can be used in the temperature range required for both the ORC and CBC heat rejection systems. A development program for the dual-slot heat pipe is being conducted by Grumman under a separate Task Order as part of this contract.

A monocoque radiator panel construction is recommended for the ORC system in order to use long radiator panels to minimize weight and meet the natural frequency requirements. Wing panel construction is recommended for the CBC system (except in a hybrid radiator design) to minimize weight because the panel length is limited by heat pipe transport capacity.

An ORC heat exchanger based on the Grumman Space Station Thermal Bus condenser design which directly condenses the cycle fluid is recommended. This design allows for parallel flow through many individual condensers which minimizes pressure drop and results in a high radiating temperature, thus minimizing the amount of radiator required. An intermediate cooling loop and a standard plate-fin heat exchanger are recommended for the CBC system.

An anodized aluminum radiator coating is recommended for both the ORC and CBC systems to meet the 30-year life requirement. Work is currently being conducted by Acurex Corp. under contract to NASA-JSC to develop this type of coating for the Space Station central radiator system. The scope of this program should be extended to address the requirements for the solar dynamic radiator system.

Three different radiator system configurations were studied for their effects on assembly and maintenance requirements; a mechanical radiator/heat exchanger clamping mechanism, a heat pipe disconnect, and a heat exchanger disconnect. Each configuration has some advantages over the others in terms of weight, reliability and ease of maintenance. In addition, three different assembly methods were examined; EVA assembly, IVA assembly, and deployable systems. Final selection of these items must be made based on an evaluation of the critical resource requirements and availability.

Baseline and several alternate heat rejection system configurations and optimum designs were developed for both ORC and CBC systems. The thermal performance, mass properties, assembly requirements, reliability, maintenance requirements and life cycle cost were determined for each configuration. For the ORC, configurations using mechanical radiator/heat exchanger clamping mechanisms, heat pipe disconnects, and integral radiator/heat exchangers were examined together with dual-slot and monogroove aluminum-ammonia heat pipes. For each configuration a combination of heat pipe wall thickness and redundant panels was selected which results in requiring a small number of maintenance sessions over a 30-year system life together with a near-minimal life cycle cost. The configuration with the integral radiator/heat exchanger system has the lowest life cycle cost and requires the least maintenance.

This results from segmenting the radiator into a large number of segments with a large number of redundant elements without the additional weight required to make the system easily maintainable. This configuration also can most readily be made deployable.

For the CBC, configurations using mechanical radiator/heat exchanger clamping mechanisms, heat pipe disconnects, and integral radiator/heat exchangers were examined together with titanium/methanol dual-slot heat pipes, Stainless steel/methanol dual-slot heat pipes, and a combination of titanium/methanol and aluminum/ammonia dual-slot heat pipes. The integral radiator/heat exchanger system also resulted in the lowest life cycle cost system which requires the least maintenance.

1 - INTRODUCTION

The heat rejection system for Space Station solar dynamic power systems must have the capability to reject large quantities of thermal energy and represents a critical part of the power system. The scope of Task I of the Solar Dynamic Heat Rejection Technology contract was to perform concept development, design, and analysis of the heat rejection system for both the Rankine and Brayton thermodynamic cycles. Critical technologies which require development tests and evaluation were assessed and identified.

High capacity heat pipe radiators offer a very effective means of meeting the Space Station's long life, high reliability, and maintainability requirements. The heat rejection temperature is an important parameter in determining the power system efficiency, radiator size and applicable radiator technology. While high capacity heat pipe radiators are under development for the Space Station central radiator system, the higher operating temperature requirements for the solar dynamic system (280 K to 450 K) require an assessment of the feasibility of using those heat pipes for this system as well as alternate high capacity heat pipe designs.

Figure 1-1 shows a typical solar dynamic power module design which incorporates a heat pipe radiator system. The system consists of heat pipe radiator panels, a fluid transport loop, and heat exchangers to transfer heat from the loop to the radiator panels. The radiator system is segmented into many small panels so that maintenance can be performed by replacing individual panels. Usually, a small number of excess panels are incorporated into the design to provide redundancy and to limit the frequency of maintenance which will be required.

Task I of the Solar Dynamic Heat Rejection Technology contract consists of a study of the heat pipe requirements and design, the radiator panel construction, the heat transport loop and interface between it and the radiators, radiator surface coatings, reliability and maintainability of the system, and on-orbit assembly of the system. This report includes descriptions of the critical components in the heat rejection system and the results of trade studies which led to the selection of the recommended configurations and designs. The trade studies analyzed component and system designs with respect to system weight, radiator area, reliability, maintenance requirements, and life cycle costs.

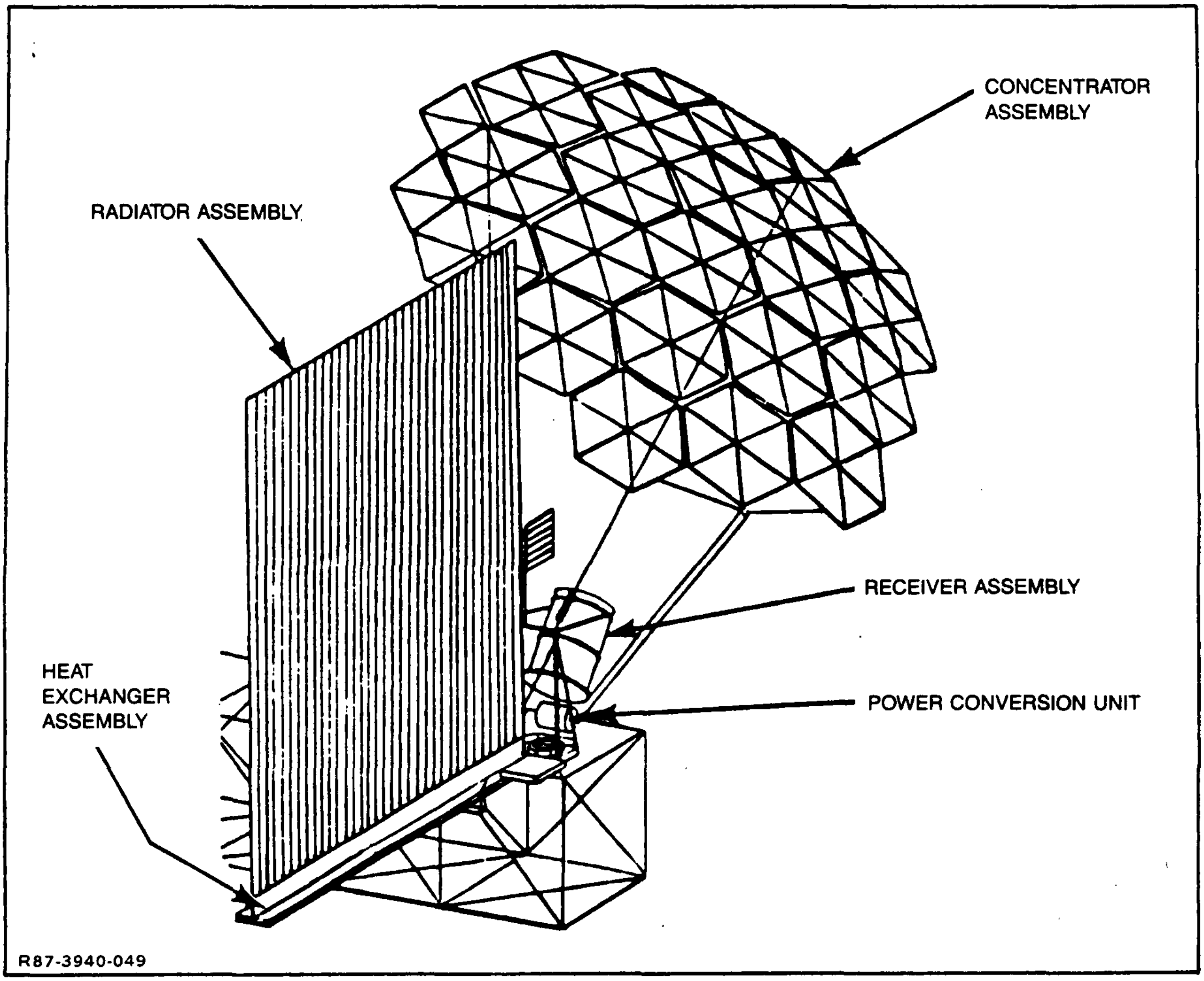


Figure 1-1 Solar Dynamic Power Module

2 - COMPONENT DESIGN & ANALYSIS

2.1 HEAT PIPE DESIGN

2.1.1 Configuration

The two types of high capacity heat pipes which were selected for evaluation to determine their applicability to the solar dynamic heat rejection system are the monogroove heat pipe and the dual-slot heat pipe. Typical cross-sections of these two configurations are shown in Figure 2-1. The monogroove heat pipe is currently under development by Grumman for the Space Station central radiator system under the Space Constructible Radiator (SCR) and Space Erectable Radiator System (SERS) contracts to NASA-Johnson Space Center (JSC). It is constructed from an aluminum extrusion. This limits its use to fluids which are compatible with aluminum. The dual-slot heat pipe is a variation of the monogroove heat pipe which has the potential for lower weight and higher performance, and can be made from materials other than aluminum. This allows for construction with a wide variety of fluids and envelope materials. Both of these configurations are included in a Grumman patent entitled "Dual Axial Channel Heat Pipe" (Reference 1).

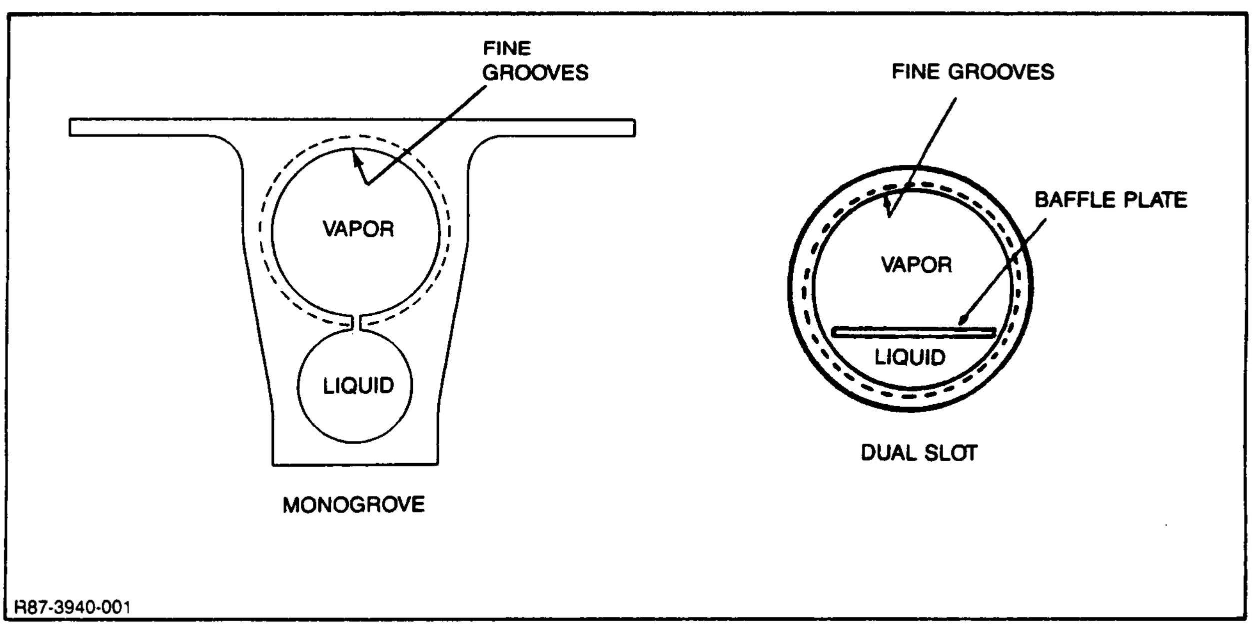
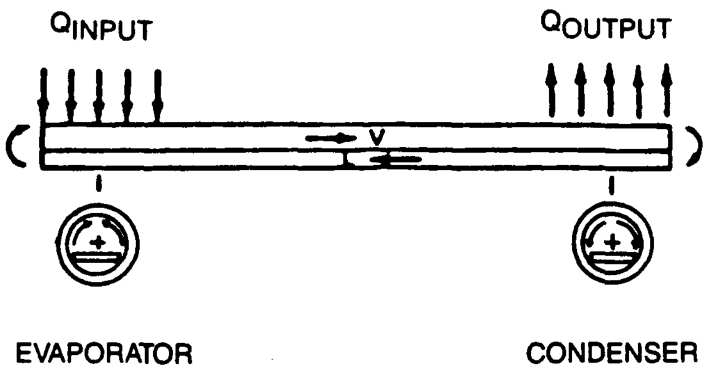


Figure 2-1 Heat Pipe Cross-Section

Both the monogroove heat pipe and the dual-slot heat pipe permit high heat transport capacity through large liquid and vapor flow areas and high heat transfer coefficients through fine circumferential wall grooves. The small slots separating the liquid and vapor channels support a high capillary pressure difference. This, coupled with the minimized flow resistance of the two separate channels, results in the high axial heat transfer capacity. The high evaporation and condensation film coefficients are provided by fine circumferential grooves in the walls of the vapor channel without interfering with the overall transport capability of the axial channels. In these heat pipe designs, evaporation takes place primarily at the meniscus contact lines in the circumferential grooves and hence is directly proportional to the number of grooves. Also, since the condensation liquid layer thickness is related to the spacing between grooves, the condensation film coefficient also depends on the number of grooves.

The operating principle of the heat pipes is characterized by two differential pressure balance relationships which must be satisfied simultaneously. As illustrated in Figure 2-2, the primary relationship requires the wall wick capillary pressure rise



1. $\Delta P_{\substack{\text{WALL}\\\text{CAP}}} = \Delta P_{\substack{\text{VAPOR}\\\text{CHANNEL}}} + \Delta P_{\substack{\text{LIQUID}\\\text{CHANNEL}}} + \Delta P_{\substack{\text{WALL}\\\text{WICK}}} + \Delta P_{\substack{\text{HEAD}\\\text{TILT}}} + \Delta P_{\substack{\text{HEAD}\\\text{DIA}}}$

2. $\Delta P_{\substack{\text{SLOT}\\\text{CAP}}} > \Delta P_{\substack{\text{VAPOR}\\\text{CHANNEL}}} + \Delta P_{\substack{\text{LIQUID}\\\text{CHANNEL}}} + \Delta P_{\substack{\text{HEAD}\\\text{TILT}}}$

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Figure 2-2 Monogroove Heat Pipe Operating Principle to offset the cumulative viscous pressure losses in the vapor channel, liquid channel, and circumferential wall grooves plus the gravity head losses associated with the height of the vapor channel and any elevation difference between the evaporator and condenser sections. In addition, the slots must develop enough capillary rise to overcome the vapor and liquid viscous losses plus the gravity head loss due to adverse tilt.

2.1.2 Fluid Selection

Fluid selection is based on an evaluation of three primary factors. The first is the variation of heat pipe transport capacity over the temperature range of interest. This was evaluated using our heat pipe design computer programs for the monogroove and dual-slot heat pipes. The second factor is the compatibility of the fluid with a suitable envelope material. The third is the minimum operating temperature for the heat pipe since this will affect the startup procedure.

The temperature range of interest for this study is 283 K to 367 K (50°F to 200°F) for the Organic Rankine Cycle (ORC) application and 283 K to 450 K (50°F to 350°F) for the Closed Brayton Cycle (CBC) application.

Following a preliminary screening, the fluids selected for further evaluation were ammonia, benzene, methanol (methyl alcohol) and water. Ammonia and benzene were evaluated for use in both monogroove and dual-slot heat pipes since both fluids are compatible with aluminum. Methanol and water were evaluated in only the dual-slot heat pipe since they are not compatible with aluminum. The transport capacity of each fluid was evaluated for the optimum heat pipe design for each particular fluid. The parameters which were varied to achieve the optimum design were the condenser and evaporator diameters and lengths, the number of parallel evaporator and condenser legs, the baffle-plate location and the circumferential groove dimensions. Figures 2-3 and 2-4 show the performance of the monogroove heat pipe using ammonia and benzene in near-optimal configurations for various vapor and liquid channel diameters over the applicable temperature range. Figures 2-5 through 2-8 show the performance of the dual-slot heat pipe for the four candidate fluids for various pipe diameters over their applicable temperature ranges.

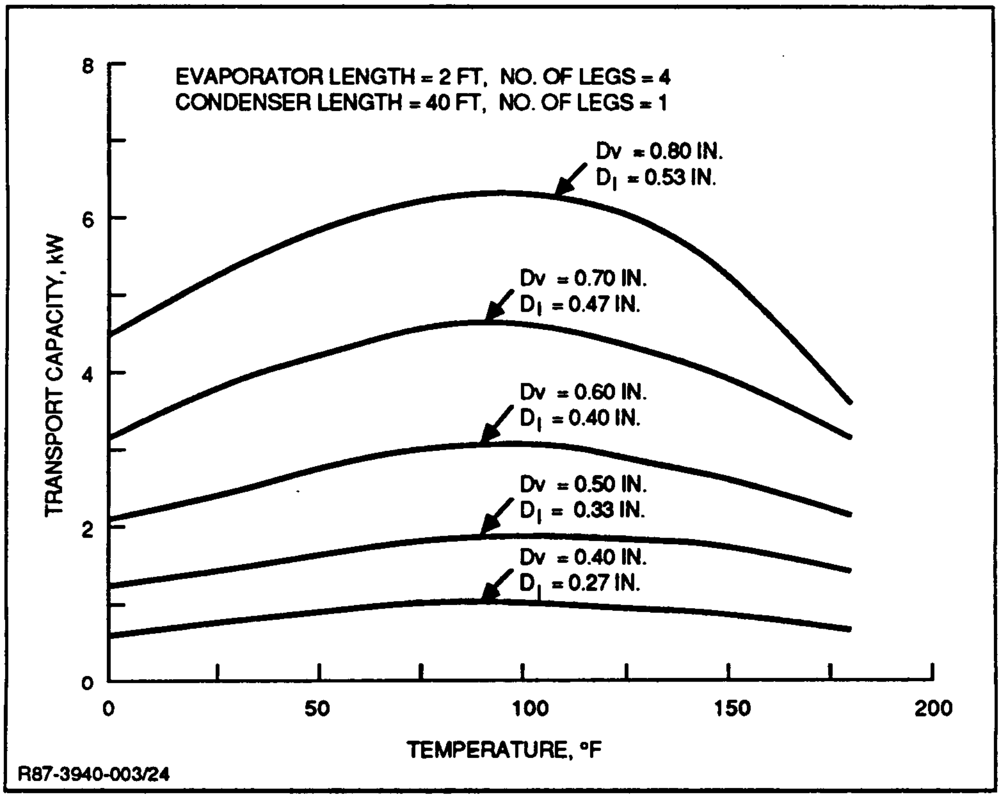

Figure 2-3 Monogroove Heat Pipe Performance with Ammonia

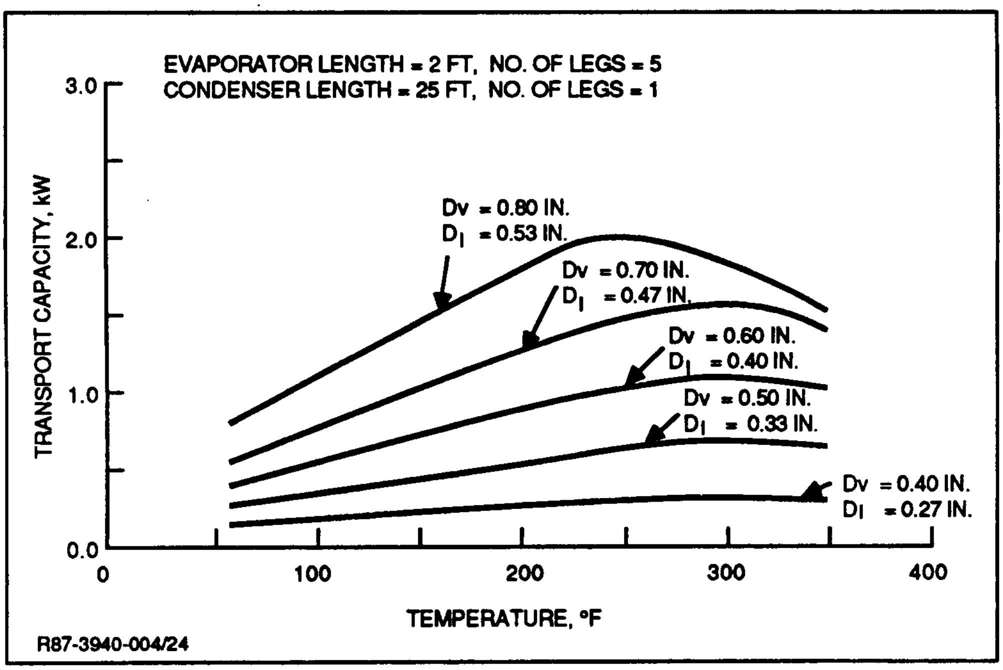

Figure 2-4 Monogroove Heat Pipe Performance with Benzene

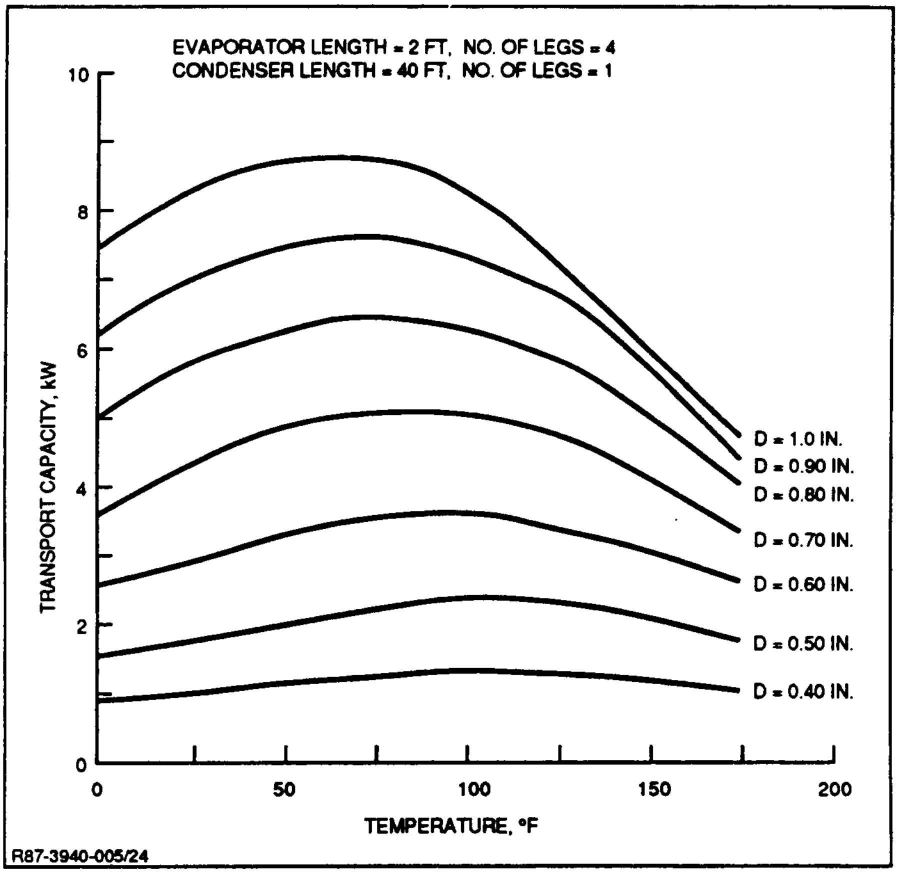

Figure 2-5 Dual-Slot Heat Pipe Performance with Ammonia

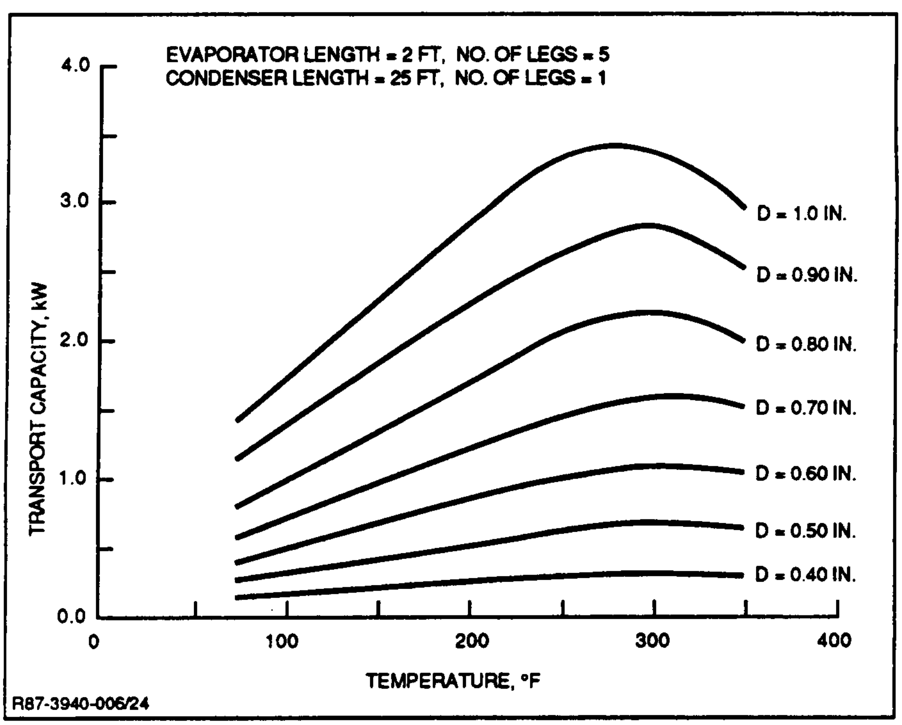

Figure 2-6 Dual-Slot Heat Pipe Performance with Benzene

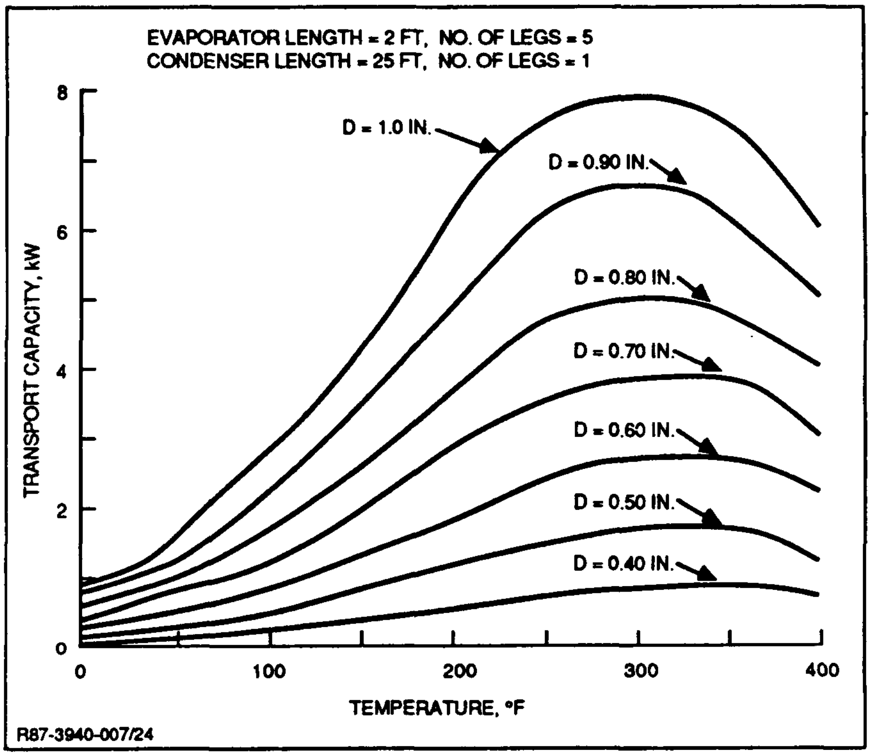

Figure 2-7 Dual-Slot Heat Pipe Performance with Methanol

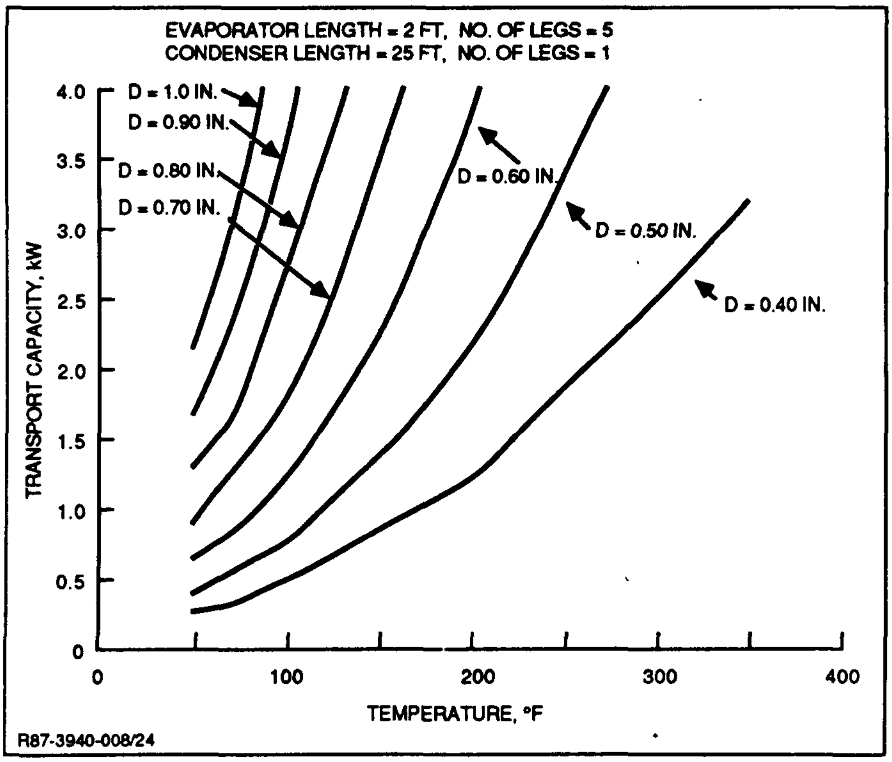

Figure 2-8 Dual-Slot Heat Pipe Performance with Water

Ammonia has a useful operating temperature range from approximately 206 K to 356 K (-90°F to 180°F). Its high transport capacity allows for very long radiator panels, thus minimizing the size of the heat exchanger and the number of coupling devices required. Its very good performance at low operating temperature minimizes startup problems. Comparison of the heat pipe performance with ammonia and with the other fluids, together with its compatibility with aluminum, clearly makes ammonia the fluid of choice in the temperature range required for the ORC.

Methanol, benzene, and water can all be used in the temperature range required for the CBC. Benzene has a relatively low transport capacity, a very high freezing point of 279 K (42°F), and a high minimum practical operating temperature of approximately 286 K (55°F) which could make startup difficult. Water also has a high freezing point (273 K, 32°F) and high minimum practical operating temperature (approximately 300 K, 80°F) which makes it unsuitable for the low temperature portion of the CBC system. Its performance in the high temperature portion of the CBC system makes it attractive there, except that the long-life compatibility with a suitable envelope material is questionable. Methanol has a very low freezing point (177 K, -142°F) and is suitable over the entire range of CBC operating temperatures.

In all of the heat pipe sizing calculations performed in this study, a performance factor of 1.5 was used. This means that the design heat pipe capacity is 50% higher that the maximum which would ever be required. This safety margin is included primarily to assure that the heat pipes will not be adversely affected by small accelerations induced in rotating the power module or maintaining the Space Station orbit.

The conclusion of this analysis was that ammonia is the best fluid to be used for the ORC heat rejection system. For the CBC, two options were selected to be carried forward to the systems study. The first is an all methanol system, while the second is a hybrid system utilizing ammonia heat pipes for the low temperature portion of the cycle and methanol heat pipes for the high temperature portion.

2.1.3 Envelope Materials

A preliminary screening was made to determine materials which would be compatible with the candidate fluids. Both aluminum and stainless steel were found to be compatible with ammonia and benzene. Stainless steel is compatible with methanol. In addition, titanium is likely to be compatible with methanol, although confirmation of this assertion must be obtained by testing. Copper, titanium and stainless steel were found to be materials which are compatible with water, although the long life compatibility with stainless steel and titanium is not certain.

An analysis was made to determine the unit weight of the various heat pipe configuration/fluid/envelope material combinations under consideration. The results for a radiator with a 7.6 meter (25 ft) long x 0.30 meter (1 ft) wide condenser section, a 0.61 meter (2 ft) long x 0.30 meter (1 ft) wide evaporator section and both 2 and 3 parallel condenser legs are shown in Figure 2-9. The ammonia-aluminum dual-slot heat pipe has a large weight advantage over all of the other combinations including the ammonia-aluminum monogroove heat pipe.

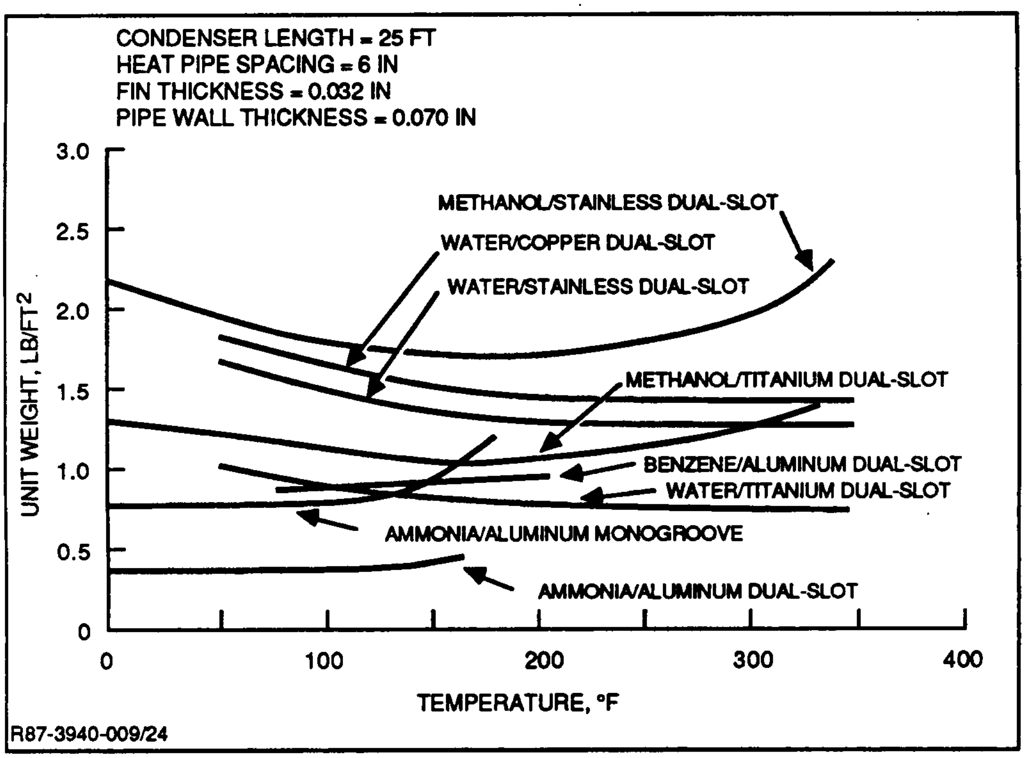

Figure 2-9 Heat Pipe Unit Weights

For a methanol heat pipe, titanium offers a large weight savings over stainless steel as the envelope material, but additional testing is necessary to determine the compatibility of this combination. Both the titanium and stainless steel options were carried forward into the systems investigation.

2.2 PANEL CONSTRUCTION

2.2.1 Fin Construction

Several types of fin construction were identified and are shown in Figure 2-10. They are referred to as monocoque construction and wing construction. The monocoque configuration is used in the Grumman SCR and SERS radiator panels. This type of fin construction results in a relatively high natural frequency for the panel. It requires a flat surface on the top and bottom of the heat pipe, however, so that

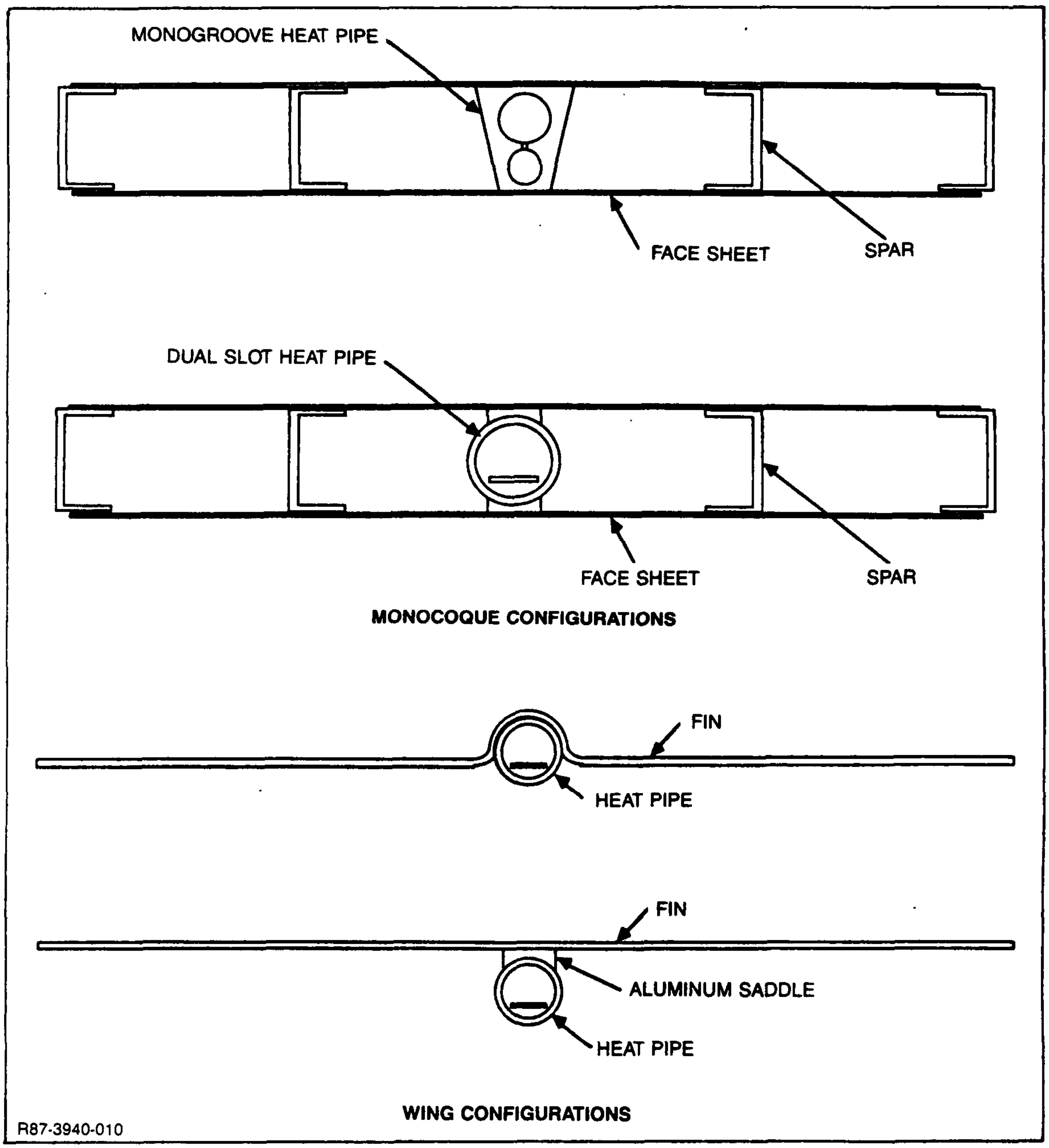

Figure 2-10 Radiator Panel Fin Construction Options

if used with the dual-slot heat pipe, a saddle must be added to the heat pipe, resulting in additional weight. The wing configuration can be used with either a dual-slot heat pipe or a monogroove heat pipe. No extraneous weight for stiffening the panel structure is required, but the panel length is limited to approximately 7.6 meters (25 ft) in order to meet natural frequency requirements. A disadvantage of using the basic wing configuration with a round dual-slot heat pipe is that the fin must be bent and wrapped around the pipe. An alternative is to add a saddle to the pipe so that bending the fin is not required. If dissimilar metals are used for the fin and pipe, differential thermal expansion between the two must be considered. Whichever configuration is chosen, a technique must be demonstrated for attaching the fin to the heat pipe.

The conclusion from comparing the two construction techniques was that the wing configuration is best for panels which were limited in length by heat pipe performance. Although additional weight is required for monocoque construction, it is more than offset by savings in heat exchanger and coupling weight if the panels can be made relatively long. For this reason, wing construction was selected for the methanol dual-slot heat pipes which are limited by the transport capacity to about 7.6 meters (25 ft) in length, while monocoque construction was selected for the ammonia monogroove and dual-slot heat pipe panels which can be made in lengths of 12.2 to 15.2 meters (40 to 50 ft).

Several different materials were evaluated for use as the radiator fin. In selecting a fin material, the primary selection criteria are the ratio of thermal conductivity to density and the ability to attach the fin to the heat pipe. For an aluminum heat pipe, an aluminum fin is the logical choice since it has the highest ratio of thermal conductivity to density and is easily attached to the heat pipe. For a stainless steel or titanium heat pipe, an aluminum fin is the best choice among common materials if it can be attached to the heat pipe. Both brazing and adhesive bonding processes can be used to attach the fin to the pipe. A development program is required to determine the suitability of these processes for this application, however. Also, if a low thermal conductivity adhesive is used, the thermal resistance through the adhesive could degrade the system performance.

In addition to fins made from conventional materials, several more advanced concepts were also identified. The first is an enhanced fin shown in Figure 2-11. This concept, which is being developed by Grumman in the SCR contract, utilized mini-heat pipes embedded in the fin oriented transverse to the primary heat pipe. The primary advantage of this is that a very high efficiency, wide fin can be constructed with a minimal increase in the fin weight. A wider fin requires a higher capacity heat pipe, however, which will increase the weight. Because of the additional development effort and the increased manufacturing complexity required, this concept has been eliminated from further consideration.

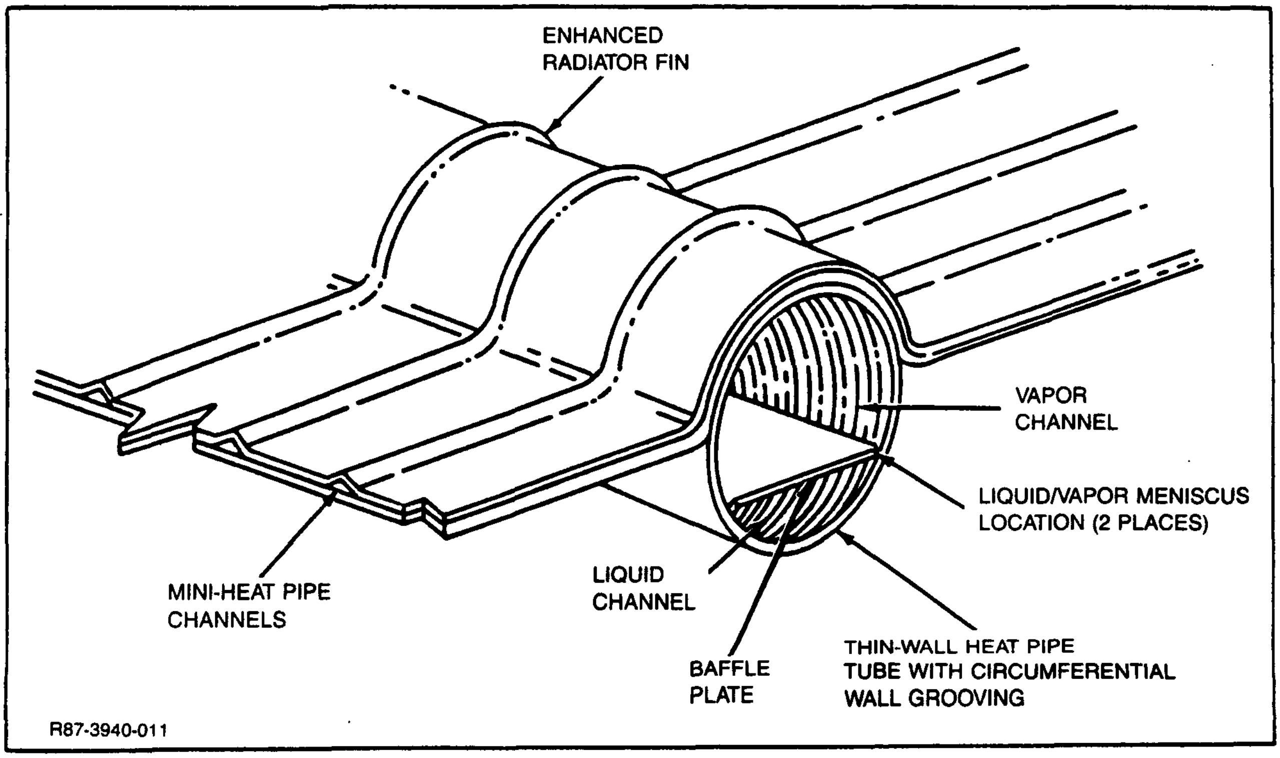

Figure 2-11 Enhanced Fin Configuration

Other advanced concepts are fins made from carbon-carbon or metal-matrix composites. Both of these types of materials could result in a thermal conductivity of up to two times that of aluminum, but require additional development. The advantages would be a reduction in fin weight and/or a higher fin effectiveness resulting in less required radiating area. They are both candidates for future development.

Aluminum was selected as the fin material for use with all of the candidate heat pipes. To verify that an aluminum fin can be attached to stainless steel and titanium, both brazing and adhesive bonding should be examined further.

2.2.2 Fin Thickness/Heat Pipe Spacing

Selection of fin thickness and heat pipe spacing is made by varying these parameters and analyzing the effects on the radiator system weight and area. A complete optimization would require a determination of the penalties resulting from an increase in weight or area, and then finding the combination which minimizes the total penalty. One approach to the problem is to attempt to minimize the life cycle cost for the radiator system. The life cycle cost, however, consists of many factors which complicate the relationship between life cycle cost and weight or area. A detailed discussion of system optimization with respect to life cycle cost can be found in Section 3. The approach which was used was to determine the relative trade-offs involved between weight and area, and to find a combination of fin thickness and heat pipe spacing which appears to be near optimal.

Figures 2-12 through 2-14 show the effect of varying fin thickness on the radiative fin effectiveness as a function of temperature with an aluminum fin for heat pipe spacings of 30, 15 and 10 cm (12, 6 and 4 inches). The heat pipe spacing, fin thickness and radiator panel length determine the amount of heat which will be radiated from the panel and hence the required heat transport capacity of the heat pipes. As the heat pipe spacing is decreased, the required transport capacity of the heat pipes is decreased as well, so that a smaller, lighter weight heat pipe can be used. Figures 2-15 through 2-17 show the effect of varying the fin thickness and heat pipe spacing on the required heat pipe transport capacity.

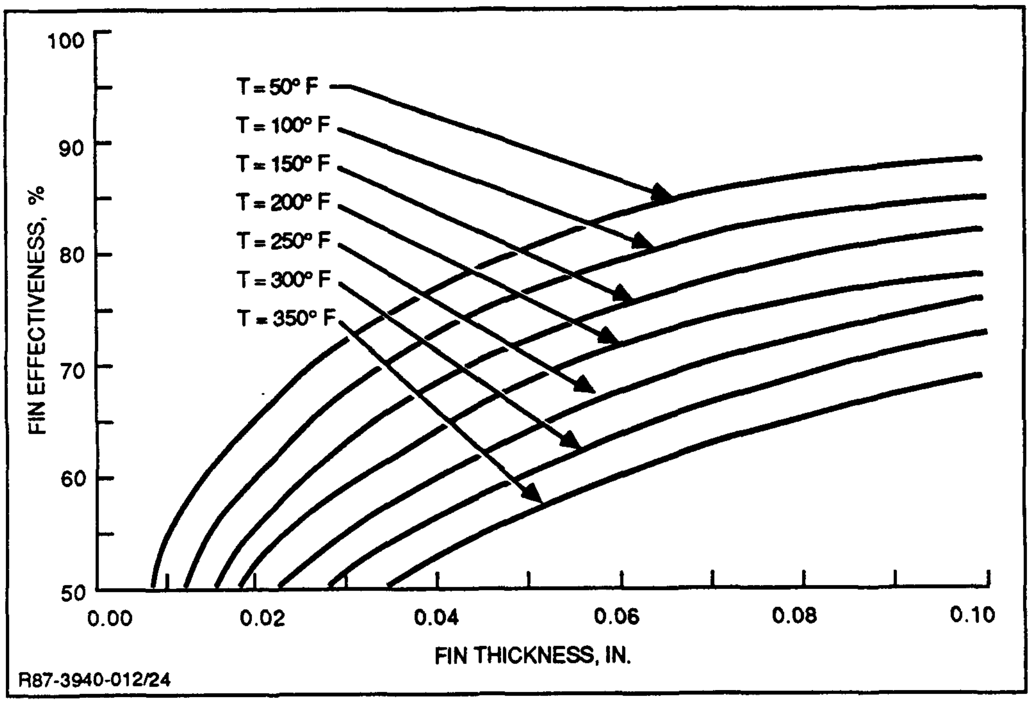

Figure 2-12 Fin Effectiveness for 12-inch Heat Pipe Spacing

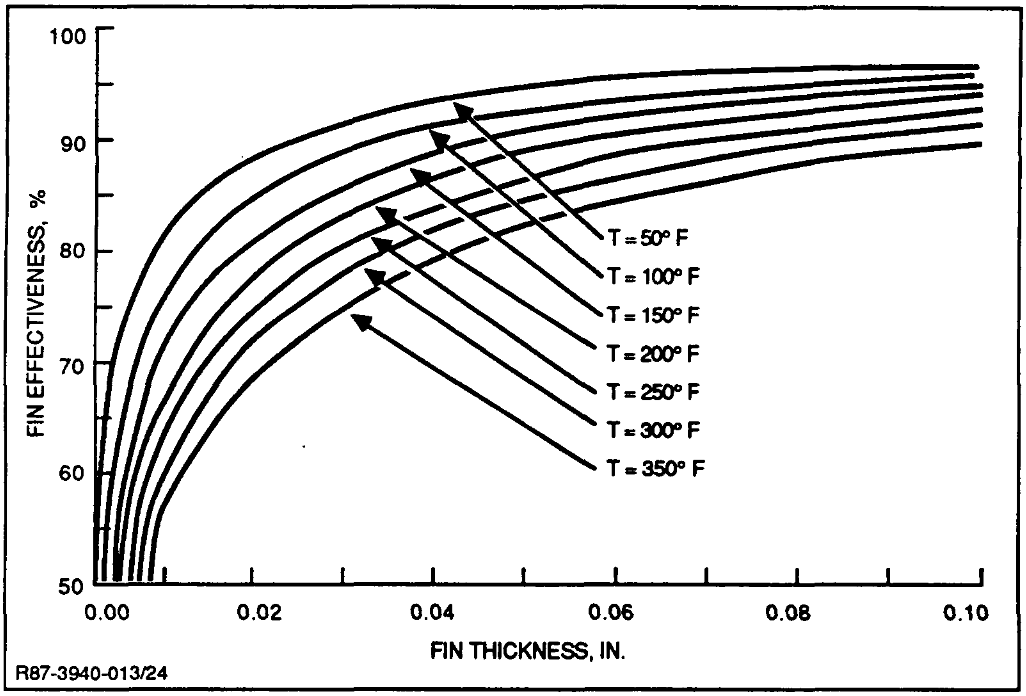

Figure 2-13 Fin Effectiveness for 6-inch Heat Pipe Spacing

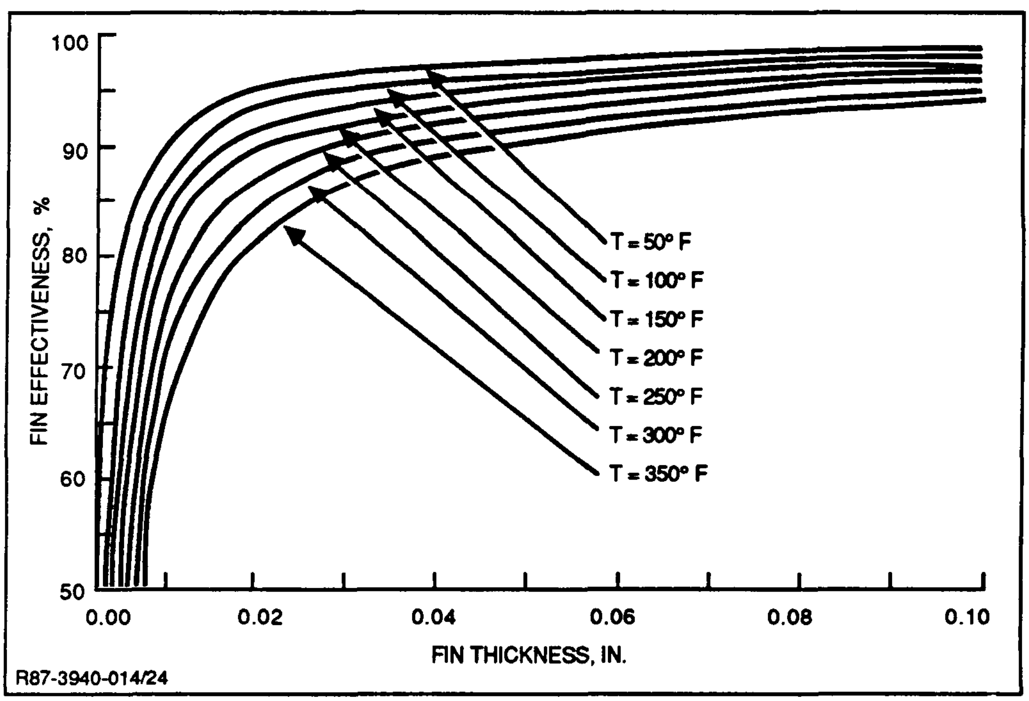

Figure 2-14 Fin Effectiveness for 4-inch Heat Pipe Spacing

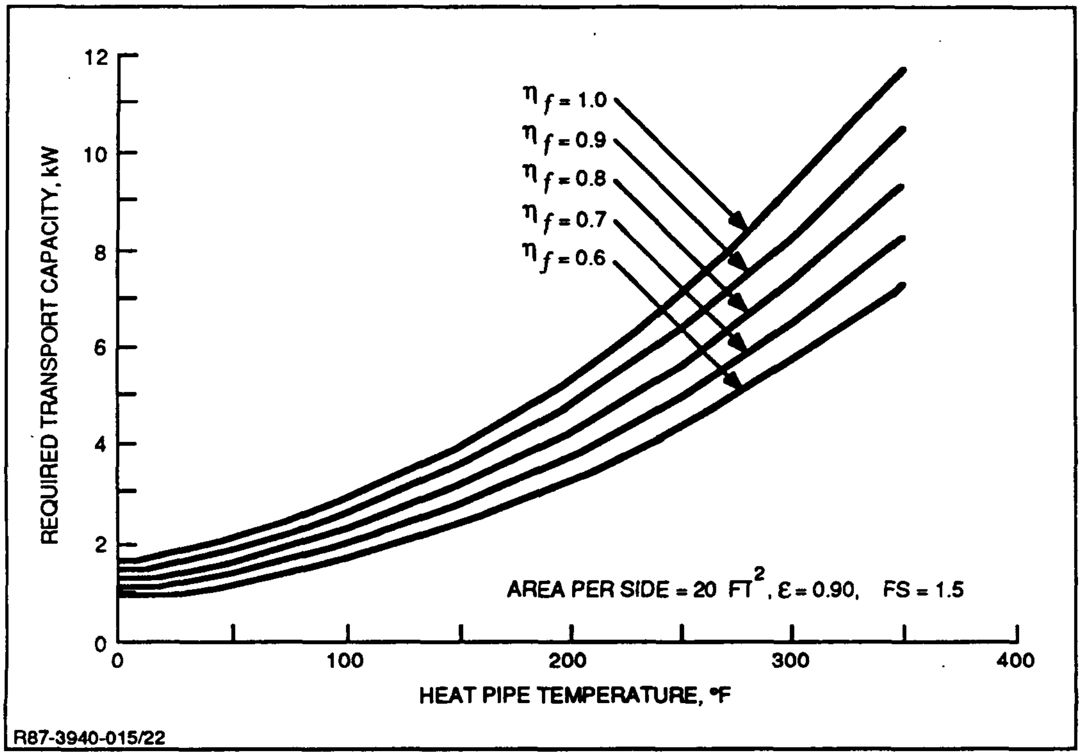

Figure 2-15 Required Heat Pipe Transport Capacity for 12-inch Heat Pipe Spacing

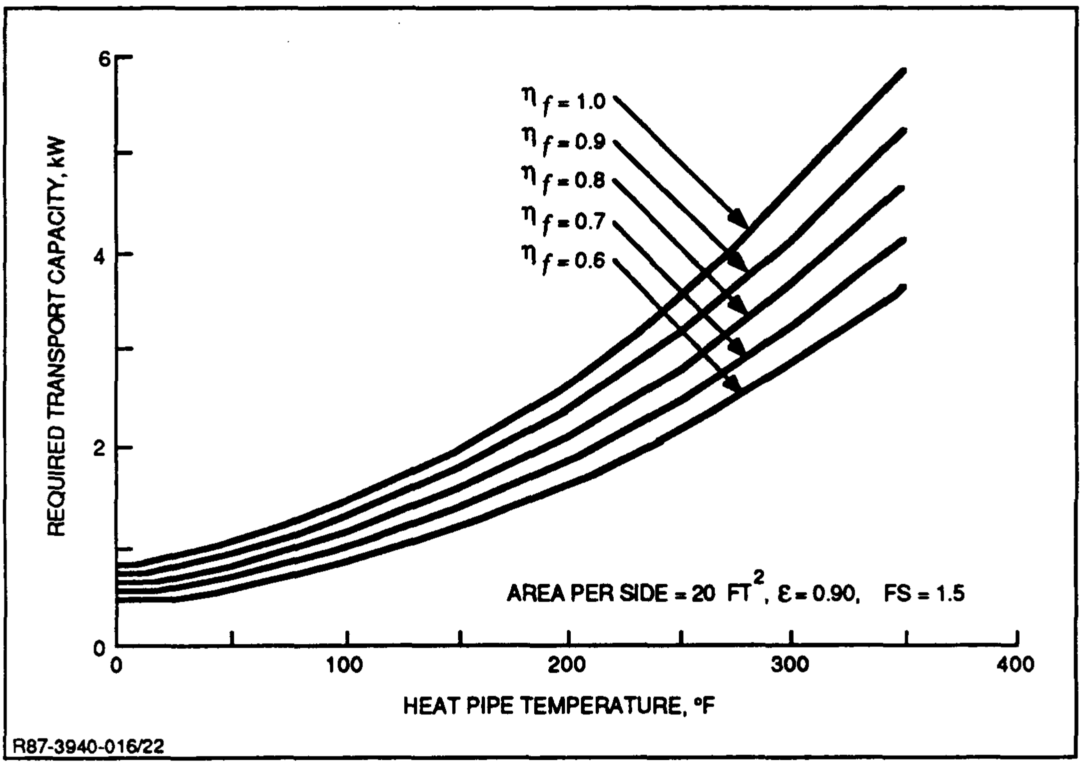

Figure 2-16 Required Heat Pipe Transport Capacity for 6-inch Heat Pipe Spacing

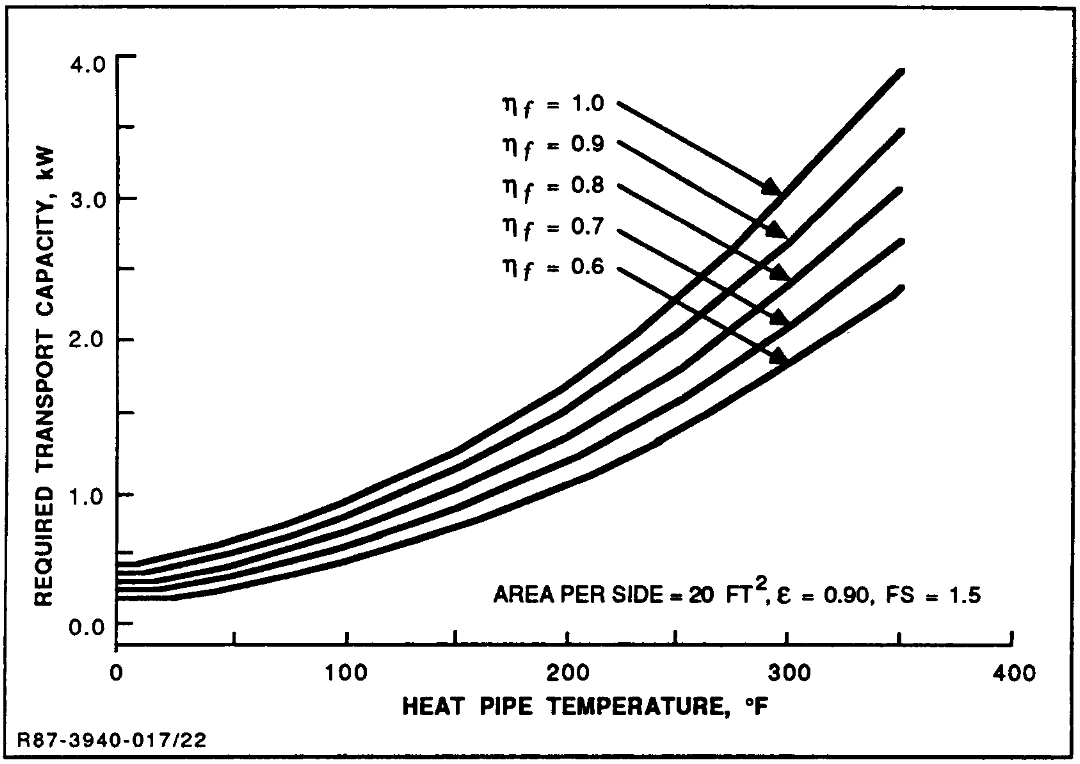

Figure 2-17 Required heat Pipe Transport Capacity for 4-inch Heat Pipe Spacing

One way of analyzing the possible designs is to determine the ratio of total weight to total area required for the possible combinations. The total system weight will be directly related to weight per unit area divided by the fin effectiveness, while the total area will be directly related to the reciprocal of the fin effectiveness. Figures 2-18 through 2-20 show this comparison for a 12.2 meter (40 ft) ammonia-aluminum monogroove heat pipe radiator panel, a 12.2 meter (40 ft) ammonia-aluminum dual-slot heat pipe panel and a 7.6 meter (25 ft) methanol-stainless steel dual-slot heat pipe panel. Each curve corresponds to a constant heat pipe spacing and varying fin thickness, resulting in a varying fin effectiveness. The weights shown include the total radiator panel weight plus heat exchanger and coupling device weight. For each case, the required heat pipe size was determined and its unit weight calculated. The results show that in each case, there is a large advantage in having a heat pipe spacing of 15 cm (6 in.) as opposed to 30 cm (12 inches). Comparison of a 15 cm (6 in.) spacing to a 10 cm (4 in.) spacing shows little difference in the weight vs. area curves. Thus, 15 cm (6 in.) spacing was selected as the baseline configuration because of the lower manufacturing cost which would be associated with requiring fewer heat pipes.

Each of the curves shown in Figures 2-18 through 2-20 shows that there is a value of fin effectiveness (corresponding to a particular fin thickness) which minimizes the radiator system weight. There would be no advantage to making the fin thinner because in addition to increasing the system weight, the area would also increase. Increasing the fin thickness results in a heavier system, but one which would require less radiator area. The lower practical limit for fin thickness is approximately 0.04 cm (0.016 in.) sheet thickness (0.08 cm, 0.032 in. total for two sheets) for the monocoque configuration and 0.08 cm (0.032 in.) for the wing configuration based on manufacturing capability. The dashed part of the curves correspond to fin thickness less than the minimum practical fin thickness. The minimum thickness is close to the thickness for minimum system weight and results in a fin effectiveness close to 90%. These are the thicknesses selected for the baseline configuration.

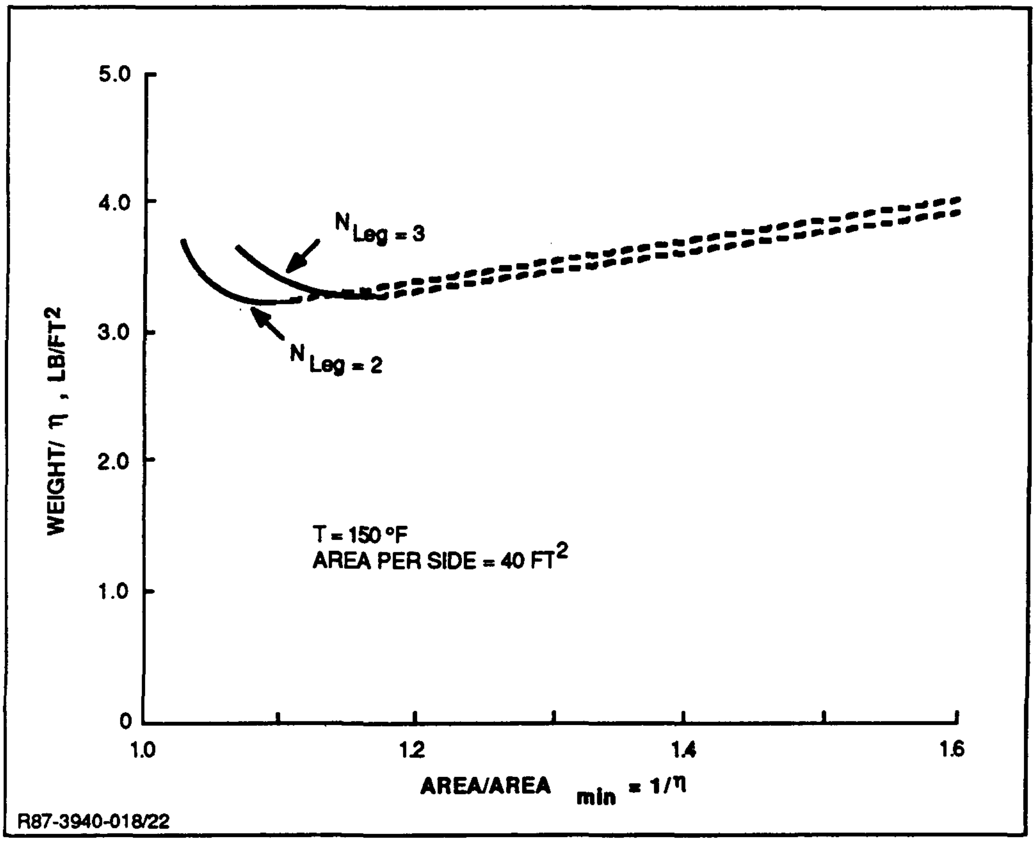

Figure 2-18 Heat Pipe Unit Weight to Fin Effectiveness for Ammonia-Aluminum Monogroove Heat Pipe Panel

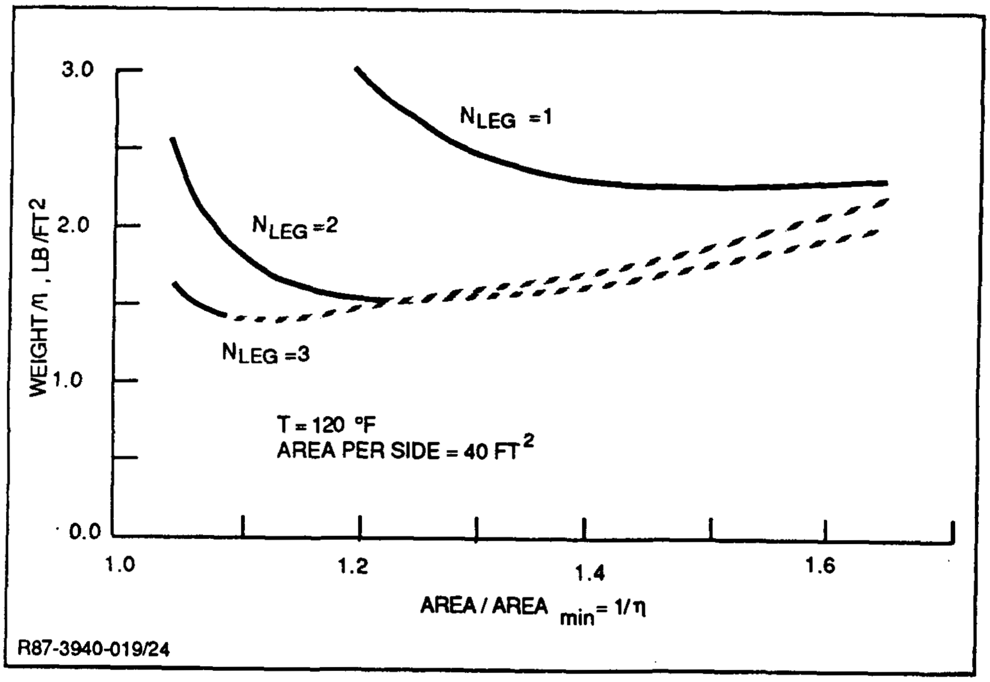

Figure 2-19 Heat Pipe Unit Weight to Fin Effectiveness for Ammonia-Aluminum Dual-Slot Heat Pipe Panel

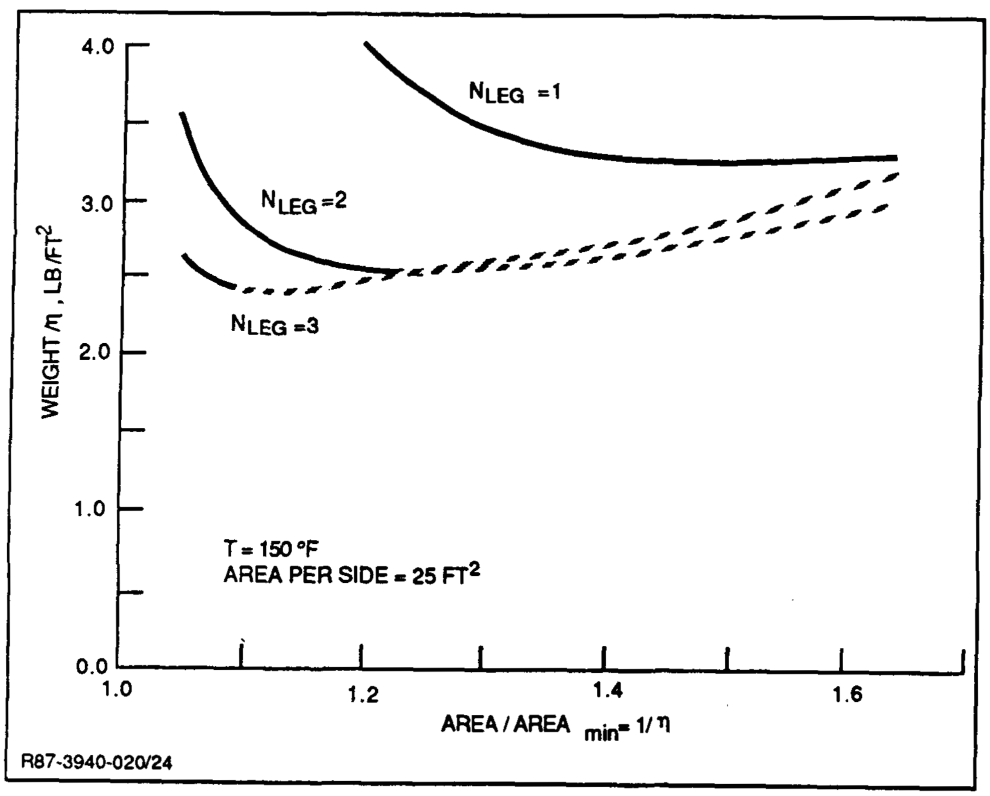

Figure 2-20 Heat Pipe Unit Weight to Fin Effectiveness for Methanol-Stainless Dual-Slot Heat Pipe Panel

2.2.3 Panel Length

The optimum panel length is determined by a procedure similar to that used for selecting the optimum fin thickness and heat pipe spacing. The system weight is determined as a function of the panel length, taking into account the difference in heat pipe size required. This procedure was carried out for the same three heat pipe configurations as considered for the fin thickness/heat pipe spacing analysis using a total fin thickness of 0.05 cm (0.020 inch). The results are shown in Figure 2-21 for a design which includes a relatively heavy heat exchanger (11 kg

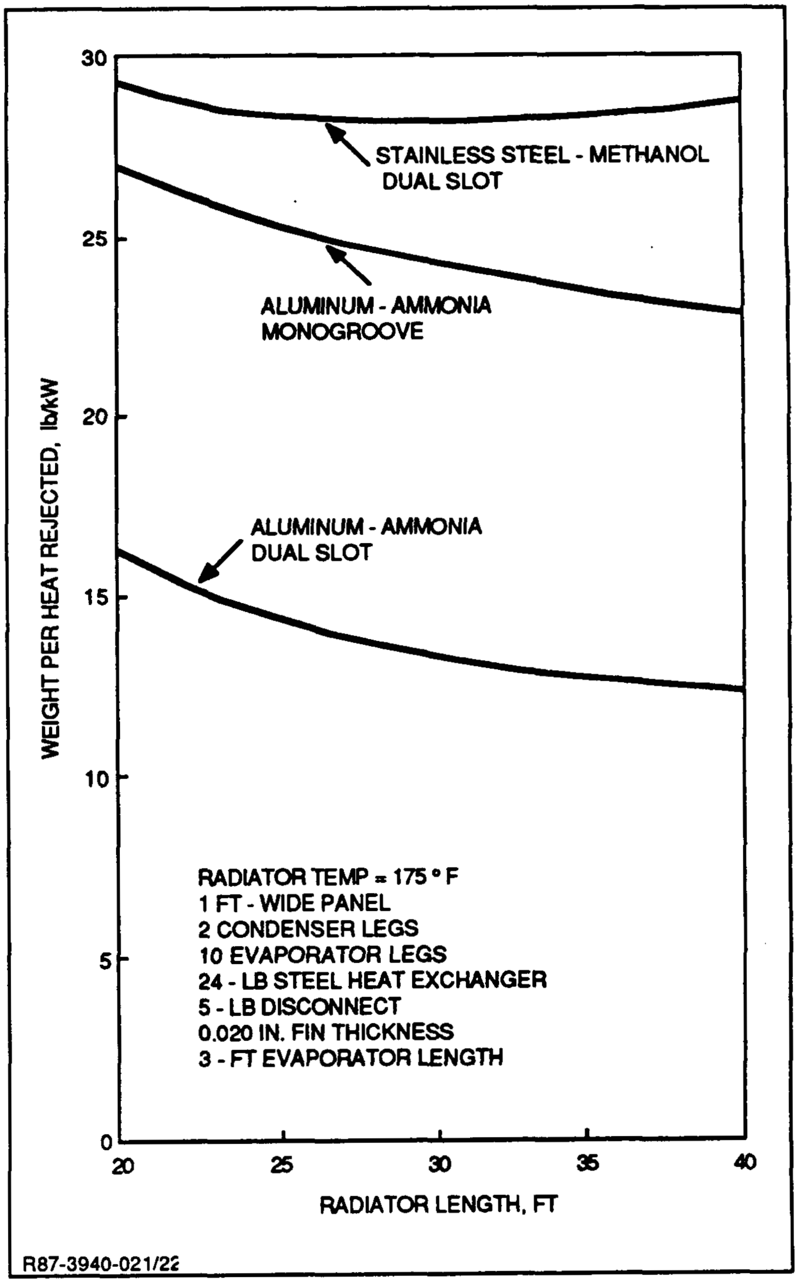

Figure 2-21 Radiator Panel Unit Weight Variation with Panel Length

per heat exchanger segment) and a lightweight heat pipe disconnect (2.3 kg). From this analysis, it can be seen that longer aluminum-ammonia panels generally result in the lowest unit weight, while for the stainless steel-methanol panels there is a length which minimizes the system weight. The actual optimum length depends upon the heat exchanger and interface mechanism weight for that particular design. This is because while longer panels require larger and heavier heat pipes, the number and weight of the heat exchangers and interface mechanisms is reduced. Another factor which must be considered is the cost per panel. Since the panel cost may not increase significantly with weight, reducing the number of panels by increasing the length may reduce the total cost significantly.

The final step in this procedure is to verify that the design selected meets the natural frequency requirements. A minimum radiator panel natural frequency requirement of 0.15 Hz was assumed. This is the requirement which was initially set for the central radiator system in the SCR program. Figure 2-22 shows how the natural frequency typically varies with panel length for the monocoque configuration. The selected length must also fit within the Space Shuttle launch volume. The maximum allowable total length is approximately 14.6 meters (48 ft). Total panel lengths of 14.6 meters (48 ft.) were selected for the ammonia-aluminum monogroove and dual-slot panels, while a length of 8.5 meters (28 ft.) was selected for the methanol-stainless steel (or titanium) panels. This length includes the evaporator, transport and condenser lengths.

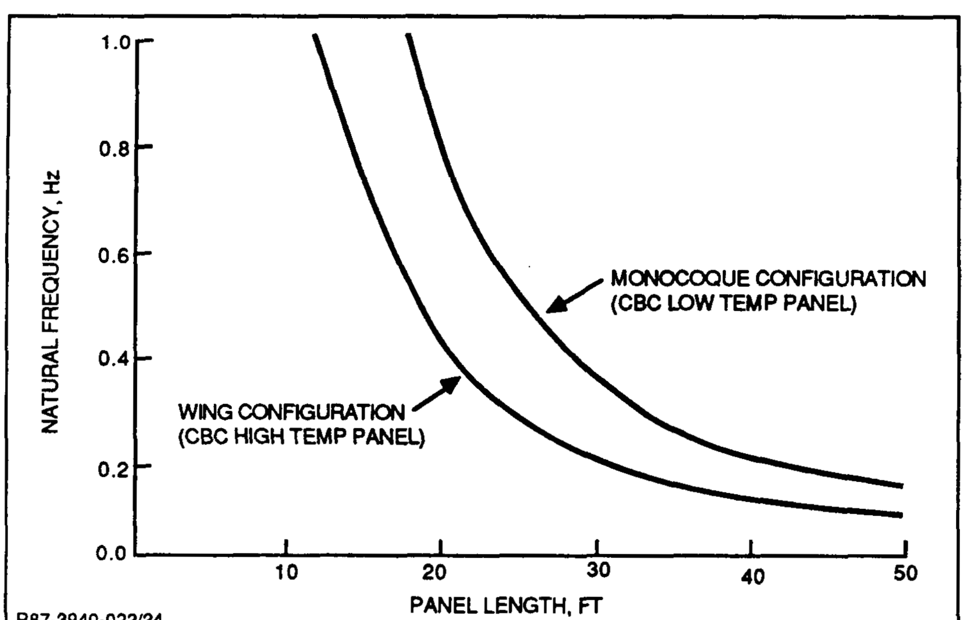

Figure 2-22 Radiator Panel Natural Frequency

2.3 RADIATOR/HEAT EXCHANGER INTERFACE

The radiators must be attached to a heat exchanger in such a way that it can be both assembled and maintained on-orbit. Three different configurations were analyzed for the radiator/heat exchanger interface; the wiffletree clamping mechanism, the heat pipe disconnect, and a heat exchanger disconnect. The wiffletree clamping mechanism is shown in Figure 2-23. This concept is being developed by Grumman under the SERS contract. The wiffletree maintains nearly uniform contact pressure through a large number of equally distributed pressure pads. A network of

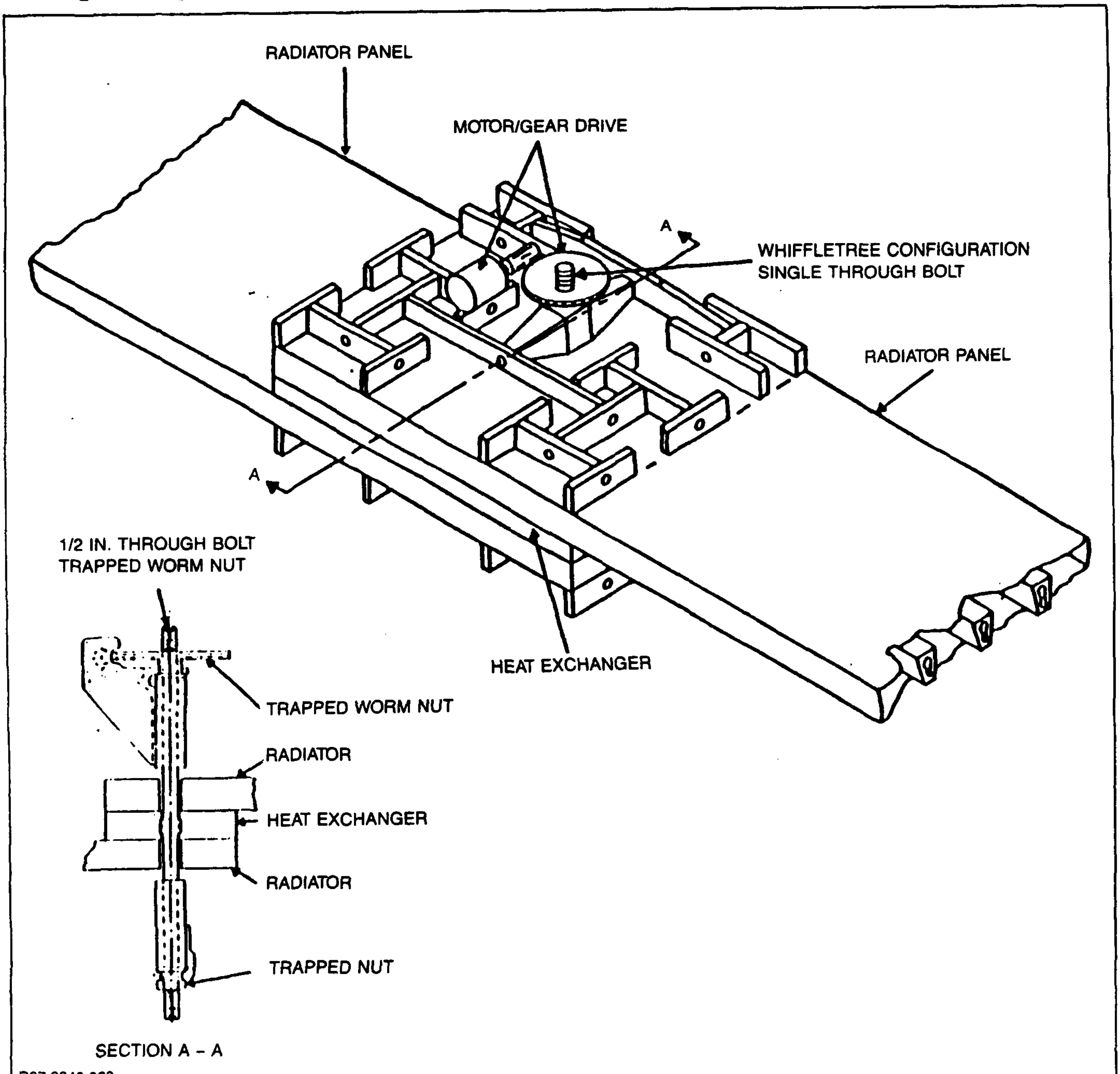

Figure 2-23 Wiffletree Clamping Mechanism

machined beam elements progressively spreads a central load to a large number of points of application to achieve uniform loading. In addition to the beam links which spread the load, the concept features a motorized worm gear drive to engage the central high-strength steel bolt. It combines the separation function and the load application function in a single device. The design is also amenable to a manual backup by tightening or loosening the nut at the opposite end. The wiffletree is expected to have an extremely low failure rate once it is clamped into position. The disadvantage of this concept is the fairly high weight of the wiffletree, estimated to be approximately 13.6 kg (30 lbs), and the high thermal interface resistance between the heat exchanger and the radiator panel.

The thermal interface resistance depends on the contact pressure between the heat exchanger and heat pipe evaporator and on the surface finishes. The SERS wiffletree is designed to provide a contact pressure of 6.9×10^5 N/m^2 (100 psi) at the interface, which is expected to yield a contact conductance of 2835 W/m^2-°K (500 Btu/hr-ft^2-°F). A higher conductance can be achieved by increasing the contact pressure. This requires increasing the weight of the wiffletree, however, for small gains in system performance. Various methods to enhance the conductance are also being evaluated. These include surface machining and polishing, and vapor deposition of aluminum on the surfaces. Testing of these concepts is being performed at Grumman under an IR&D program.

In a heat pipe disconnect design, the heat exchanger and the evaporator section of the radiator would be brazed together, providing a very thermally efficient interface. The evaporator and condenser sections of the radiator would be connected by a unique two-fluid (liquid/vapor) channel disconnect coupling as shown in Figure 2-24. The disconnect is designed to keep the liquid and vapor streams separated while maintaining continuity between similar channels. Under contract to NASA-JSC, Grumman has successfully demonstrated proof-of-concept hardware for a disconnect coupling configured for use with a monogroove heat pipe. Prototype units have also been procured and are currently undergoing testing in the SERS program. Each half of the disconnect contains two separate internal channels which can be joined to their corresponding mate in a single operation. The complete assembly consists of the external housings, and an adapter block which is bolted to each disconnect housing and also contains a short piece of the monogroove extrusion. The latter is the part that is subsequently butt-welded to the primary heat pipe extrusion.

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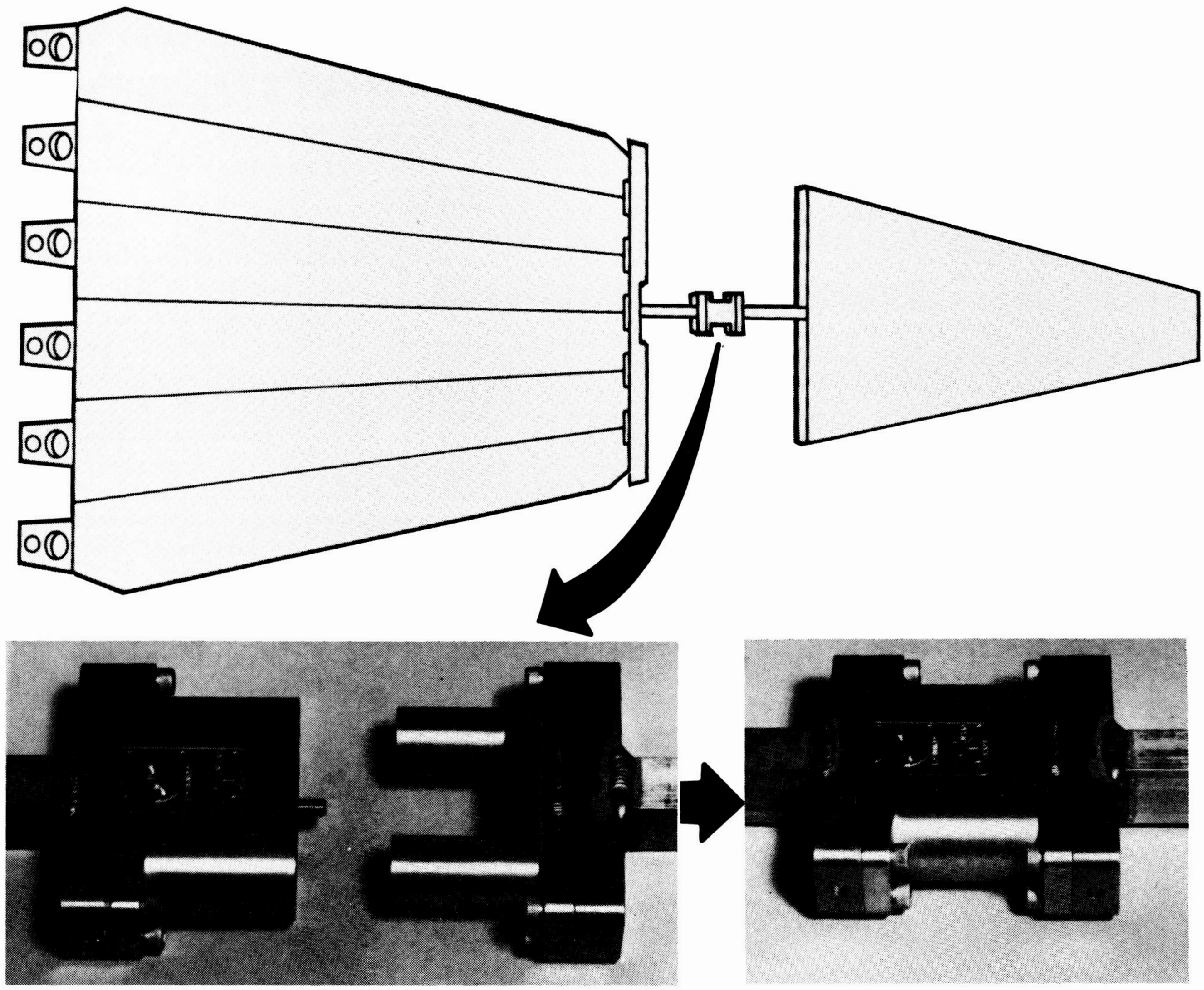

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Figure 2-24 Heat Pipe Disconnect

The weight of the prototype unit is approximately 5.9 kg (13 lbs), which represents a substantial weight savings in additional to the improved thermal interface conductance over the wiffletree concept. Use of the quick disconnect with methanol would require changing the fluid passage materials, which may slightly increase the weight. The housing itself would still be aluminum. Some disadvantages of the heat pipe disconnect are lower reliability, an increased difficulty in replacing a failed heat pipe evaporator, and a slightly lower heat pipe performance resulting from the pressure drops in the disconnect. Maintainability is also affected since radiator failure may arise from several possible causes. Likely failure mechanisms are a micrometeoroid puncture of condenser section or the failure of a disconnect seal. It may not be known which part of the radiator needs to be replaced. The savings in weight and interface conductance gained from the heat pipe disconnect must be compared to the effects of lower reliability and maintainability in order to choose between the heat pipe disconnect and the wiffletree.

A third possibility in interfacing the radiator panel and heat exchanger is to modularize the heat exchanger and radiator panel by brazing the heat exchanger to the heat pipe evaporator. If there is a failure, the entire unit is replaced. Disconnects could be used for the connections between the heat exchanger and the transport loop as shown schematically in Figure 2-25. Alternately, hard connections could be made which would require the lines to be cut and then repaired whenever a replacement must be made. The technology for these coupling concepts is being developed for use in making repairs to the central thermal bus by Grumman under the Space Station Work Package 2 contract. The advantage of the hard connection over the disconnect for this application is that one failure mechanism (the disconnect) is eliminated.

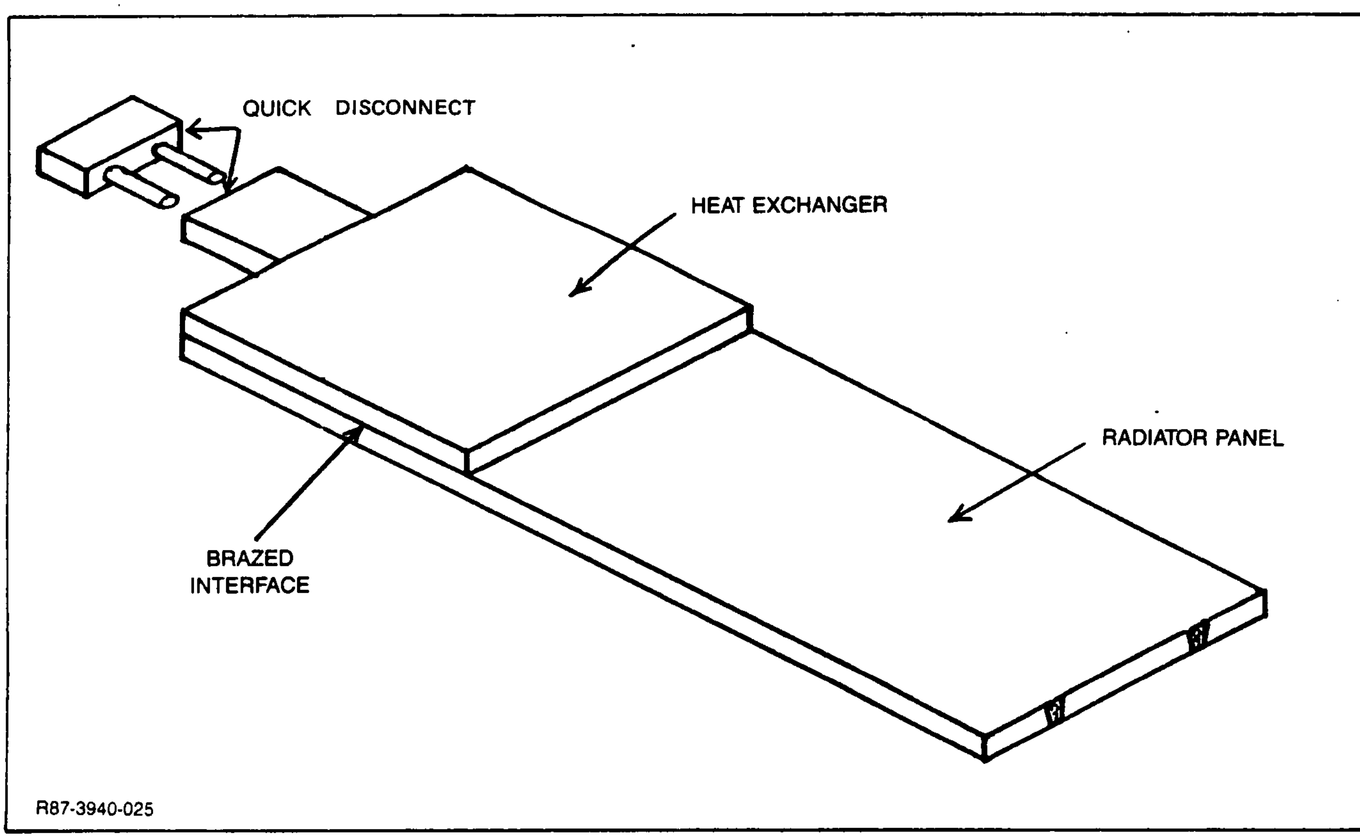


Figure 2-25 Modularized Radiator/Heat Exchanger Concept

2.4 HEAT EXCHANGER & TRANSPORT LOOP

Several options are available for the configuration of the heat rejection system with respect to whether or not an intermediate loop is employed between the cycle fluid and the radiator system, and whether or not to segment the heat exchangers. The issues which must be addressed are the effects on system performance, reliability, weight, and cost.

Figure 2-26 shows schematically two possible configurations: a direct loop, and an intermediate loop which could utilize either a single or a two phase fluid. With respect to thermal performance of the heat rejection system, a direct loop is the most efficient. Disadvantages of the direct loop are that it would result in a higher pressure drop for the cycle fluid, which will have a negative impact on system performance, and it also exposes the cycle fluid loop to the possibility of a micrometeoroid puncture as it passes through the heat exchangers. A high pressure drop penalizes the power cycle performance by increasing the pump (ORC) or compressor (CBC) power requirement and also by lowering the condensing and heat rejection temperature for the ORC system.

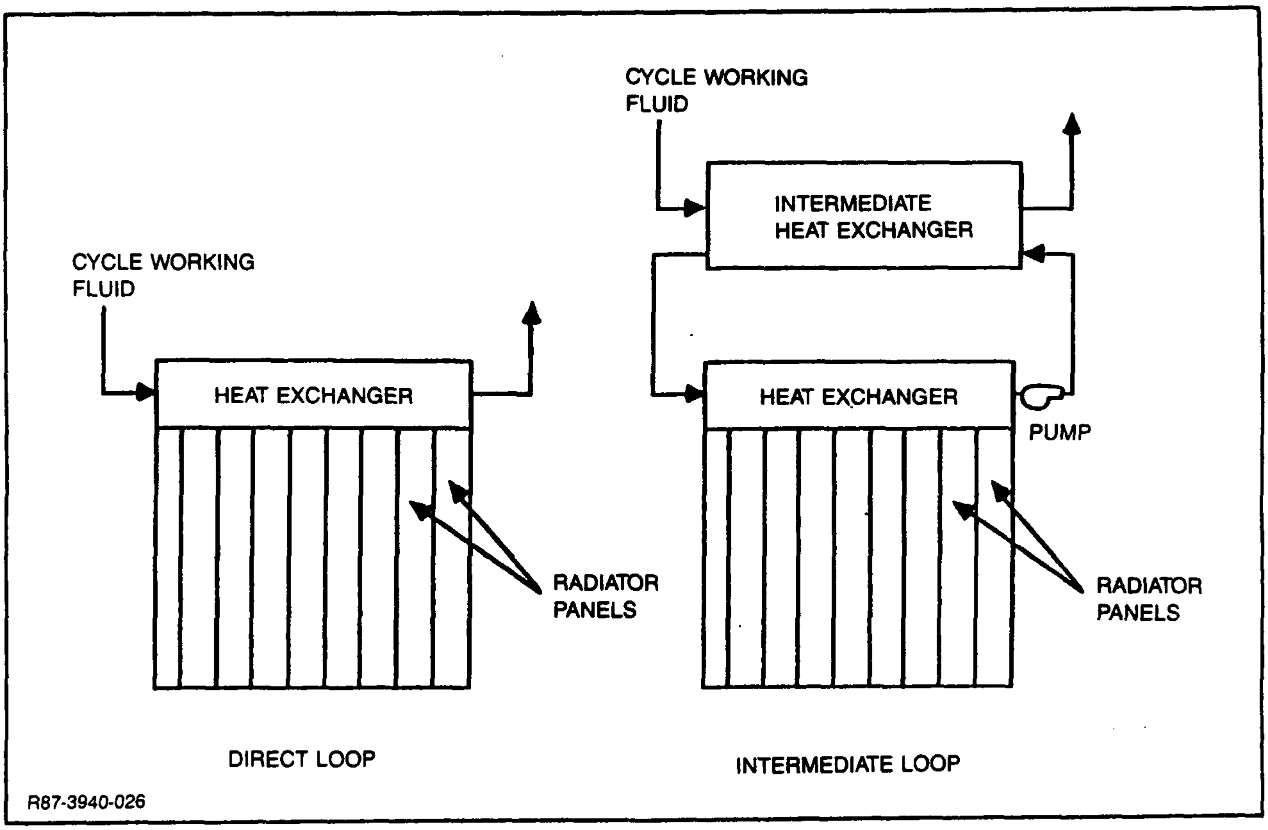


Figure 2-26 Heat Transport Loop Configurations

2.4.1 ORC Application

A single phase intermediate loop is not suited for use with the ORC system, since it would require heat rejection over a range of temperatures with the maximum temperature near the condensing temperature and the minimum temperature much lower. Use of a single phase intermediate loop would make the system more complex and also require much more radiating area. Since there were no advantages identified to using a single phase intermediate loop, this concept was dropped from further consideration.

A two-phase intermediate loop is better suited to use with the ORC than a single phase intermediate loop since it would operate at a nearly constant temperature close to the condensing temperature. A two-phase system would be more complex than a direct loop and also require more radiating areas. Since no advantages were identified for this concept, it also was dropped from further consideration.

Two condensing heat exchanger concepts were analyzed. The first is the Sundstrand shear flow condenser, which is shown in Figure 2-27. In this concept,

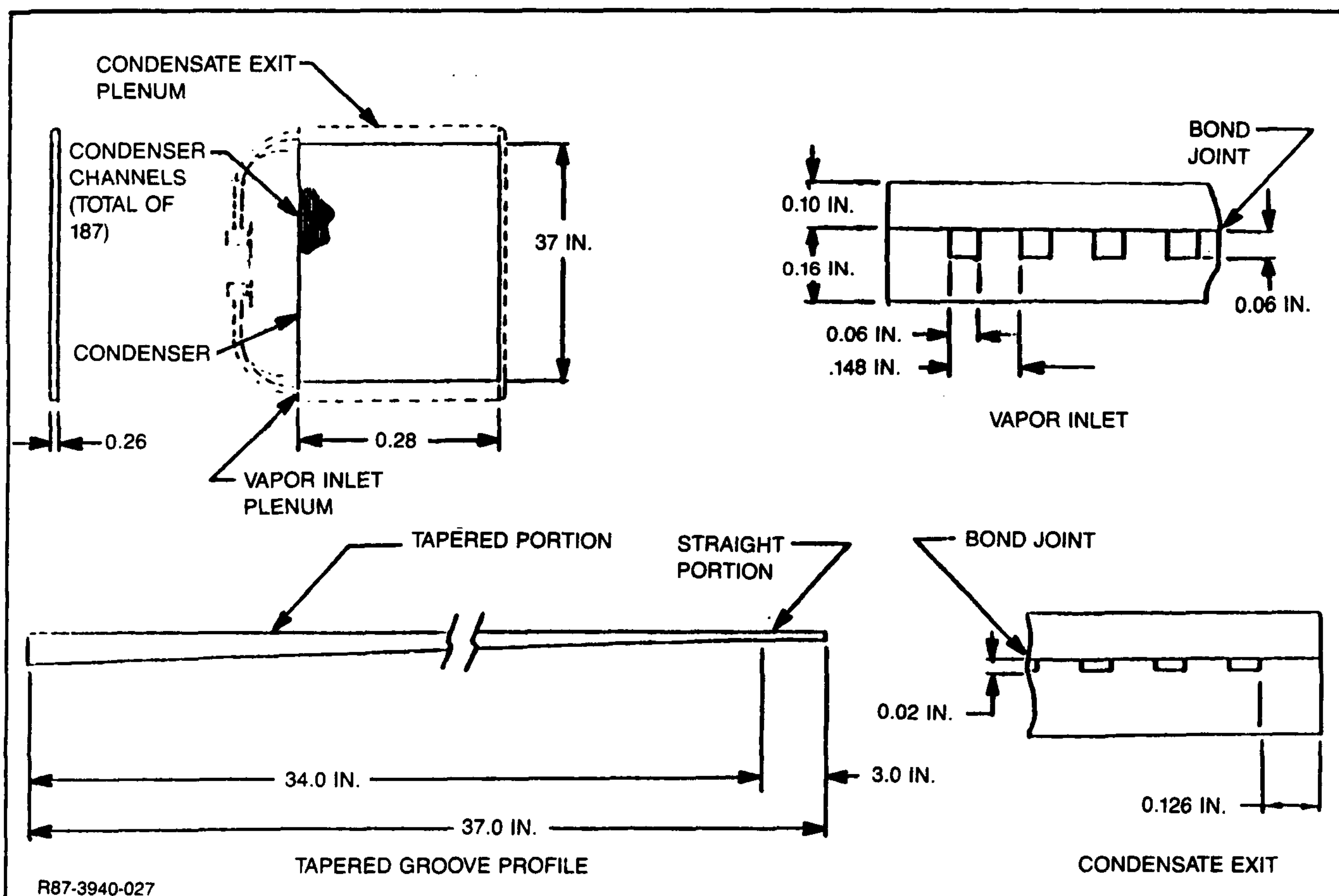


Figure 2-27 Shear-Flow Condenser

the fluid is passed straight through small passages where the vapor is condensed. A section at the inlet is used for bringing the fluid temperature down to the saturation temperature. The center section is used for condensing the fluid and a section near the outlet is used for subcooling the liquid. The fluid temperature typically varies as shown in Figure 2-28. In the central section, there is two-phase flow as the fluid is condensed. The condensing temperature drops along the length as the pressure drops. One of the disadvantages of this system is that either the radiators must be sized to operate over a range of temperatures between the superheated vapor inlet temperature and the subcooled liquid outlet temperature or the interface between the heat exchanger and radiator must be "spoiled" to effectively reduce the temperature of the radiators at the inlet. For a system using ammonia heat pipes with an inlet temperature over about 339 K (150°F), this presents problems because the performance of ammonia heat pipes decreases greatly at temperatures above this point. A larger and heavier heat pipe is needed to operate at a maximum temperature of 367 K (200°F) than is needed at 339 K (150°F).

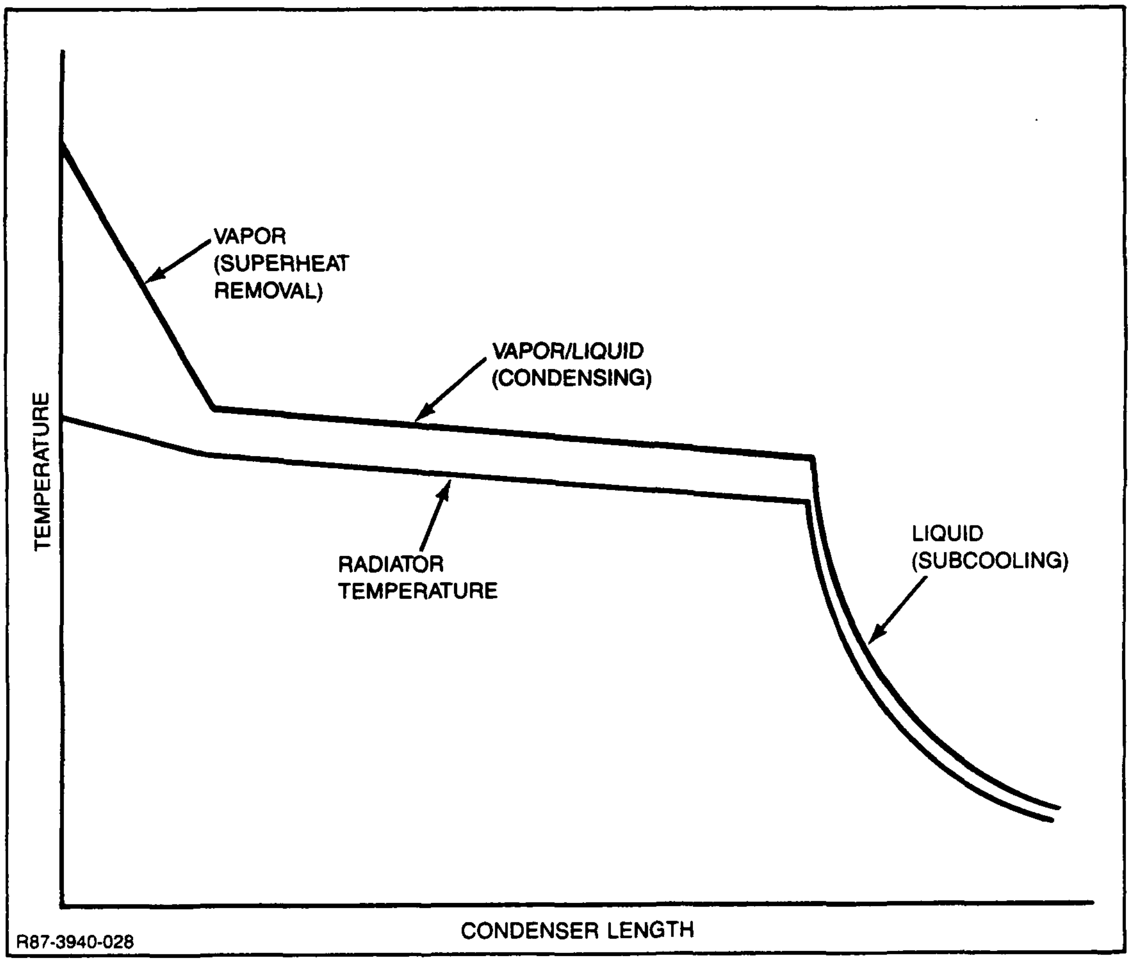


Figure 2-28 Thermal Performance of Shear-Flow Condenser

Also, at operating conditions where there is excess heat rejection capacity due to factors such as excess radiator panels, beginning-of-life radiator surface coating properties, power turndown to recover from peaking or a low insolation orbit, the liquid outlet temperature could become very low. This could result in not only degraded system performance but also risk freezing the radiator heat pipes at the outlet in a cold environment. Testing of the monogroove heat pipe in the SCR program demonstrated that a frozen radiator would not function again when the load is increased without first gimballing the radiator system into the sunlight to thaw the heat pipe fluid. This is undesirable for the solar dynamic radiators. Further analysis which includes the variation in the cycle operating conditions is required in order to fully quantify this potential problem.

An alternate concept is the condenser which was developed by Grumman for the Space Station thermal bus. This is shown in Figure 2-29. In this design, many in-

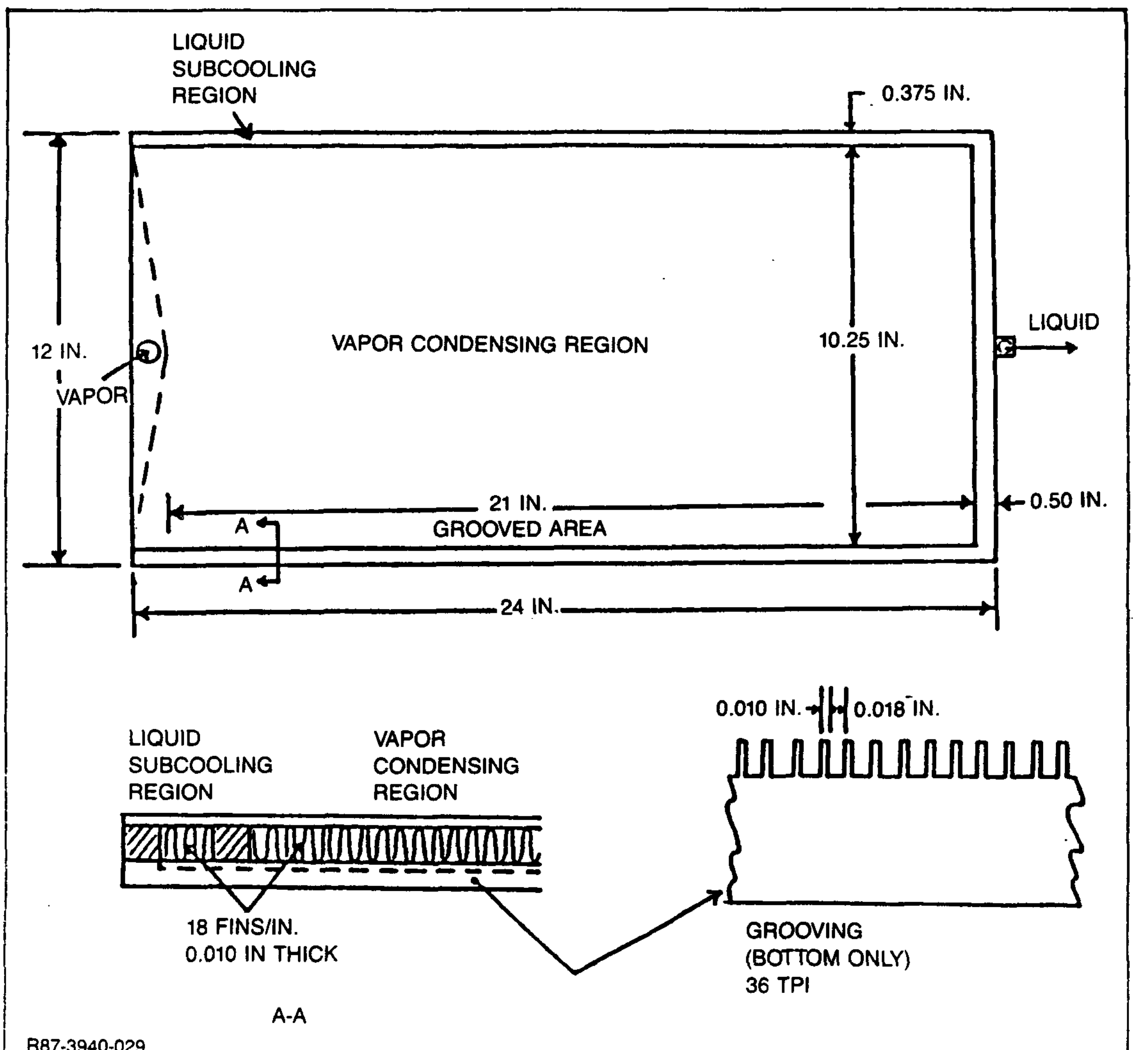


Figure 2-29 Grumman Thermal Bus Condenser

dividual condenser units are plumbed in parallel. The condenser is essentially a standard finned heat exchanger, except that small grooves are machined into one inside surface for liquid flow between a condensing section and a subcooling section. The subcooling section is nearly 100% efficient at cooling the outgoing liquid down to the radiator temperature. An advantage to this configuration is that the pressure drop is much lower than in the shear flow condenser. A disadvantage is that each radiator panel will operate at a temperature which is slightly less than the condensing temperature. In the comparison between condensers, however, this disadvantage may be more than offset by the higher condensing temperature resulting from a lower pressure drop in the Grumman condenser. The heat pipes can be sized to operate at the subcooled liquid outlet temperature and there is no chance of freezing any operating radiator heat pipes.

The concept which is recommended for the ORC heat rejection system is a direct loop using many individual condensers plumbed in parallel. The Grumman thermal bus condenser is ideally suited to this configuration. The Sundstrand shear flow condenser may be suitable if it is configured into individual condenser units.

2.4.2 CBC Application

The same three options for the heat rejection loop are also available for the CBC system: the direct loop, the single phase intermediate loop, and the two phase intermediate loop. The same basis of comparison is also applicable: impact on system performance, reliability, weight and cost.

As in the ORC system, a direct loop offers the best thermal performance. The pressure drop is more of a problem for the CBC system, however, because a much higher penalty is incurred for the heat rejection system pressure drop in the CBC system than in the ORC system. The pressure drop can be made small by having a large flow area for the gas through the heat exchanger, but this diminishes the thermal performance and increases the system weight. As the heat rejection system capacity is increased, both the flow rate and the length of heat exchanger are also increased, resulting in an increased pressure drop. The operating pressure also effects the pressure drop, since at lower pressures, the fluid density is lower, resulting in higher pressure drop. At the baseline CBC cycle operating conditions specified by NASA (shown in Part III), a direct loop becomes impractical because the pressure drop is too large. In addition, the baseline CBC system also requires alternator cooling which cannot be easily accommodated with a direct loop configuration.

A single phase liquid intermediate loop is suited to the CBC system since the heat rejection must be done over a range of temperatures. An intermediate heat exchanger between the system fluid loop and the intermediate loop transfers the heat between loops. The outlet temperature on the intermediate loop side can be close to the inlet temperature on the cycle fluid side, causing only a small system penalty. The pressure drop through the intermediate heat exchanger can be made small on the cycle fluid side, while the pressure drop in the intermediate loop is also small so that the pumping power requirement is not large. There is some increase in complexity and decrease in reliability in having an intermediate loop, but this is offset by the lower cycle loop pressure drop, resulting in a smaller, lower weight radiator-heat exchanger. The lower weight radiator-heat exchanger also results from being able to make the heat exchanger from aluminum instead of stainless steel which would be required with a direct loop heat exchanger. The lower reliability of an intermediate loop compared to a direct loop is only due to the intermediate heat exchanger, the pump, and the accumulator for the loop. Addition of a redundant pump would not significantly increase the system weight, and the heat exchanger and accumulator should have high reliabilities. An entire redundant intermediate loop and heat exchangers could also be added to increase reliability, but the additional weight and complexity is probably not justified. The pressure drop in the intermediate loop is typically very low (on the order of several psi) which results in very low pumping power requirements.

A two phase intermediate loop is not compatible with the CBC system since the heat would have to be rejected at a very low temperature and much more radiating area would be required. Also, the complexity is greater for a two phase system.

A single flow-through radiator-heat exchanger offers the best thermal performance for the CBC since each radiator rejects heat at the highest possible temperature. As an alternate, several heat exchangers could be plumbed in parallel. Each heat exchanger would be attached to a number of heat pipes which would operate at different temperatures. This alternative is attractive when looking at deployable systems, since the heat exchanger could be folded into several sections making deployment easy without affecting the thermal performance very much.

The concept selected for the CBC system is a single-phase intermediate loop. The heat exchanger can be either a single flow-through design or several parallel units.

2.5 RADIATOR SURFACE COATINGS

The subject of radiator surface coating selection was not examined in detail since this subject is being studied in detail by the Acurex Corporation under contract to NASA-JSC (Long Life Durable Radiator Coatings NAS 9-17430). The performance objective for this coating is a 10-year life in a low earth orbit (LEO) environment (ultra-violet radiation and atomic oxygen) with end-of-life (EOL) optical properties of solar absorptance (alpha) = 0.2 and emissivity greater than 0.75.

The initial portion of the Acurex study eliminated paints and silver teflon. Paints are susceptible to microcracking caused by thermal cycling, which would expose the substrate to erosion by atomic oxygen impingement. Silver teflon is susceptible to delamination at the adhesive/substrate interface. The coating selected by Acurex was anodized aluminum because of its likely low rate of degradation. In this study, we have examined the impact of using this coating for the solar dynamic radiators and how it may be optimized for this application.

In the Acurex study, final consideration is being given to three different aluminum alloys as the substrate materials (6061-T6, 3002 and 5252) with various emittance coatings. The 6061-T6 alloy has been the baseline material for the heat pipe radiator panels. It has a good combination of high thermal conductivity and strength, but does not have good optical properties when anodized. Both 3002 and 5252 alloys have lower strength but better optical properties. One combination that was not considered by Acurex, but which may offer the best combined properties, is a core material of 6061-T6 clad with 3002 aluminum, then anodized. This gives good strength with high thermal conductivity and a surface that can be anodized to achieve the desired optical properties.

Another issue to be addressed is the proper coating thickness. Generally, as the thickness is increased, both the emittance and solar absorptance are increased. For the solar dynamic radiators, which will always be oriented edge-on to the sun, achieving a low solar absorptance is less important than a high emittance. A trade study was conducted so that the optimum combination of emissivity and solar absorptance can be selected. For this analysis, the orbital heat loads to which the radiators will be exposed was obtained from NASA. The effective maximum sink temperature and heat rejection rate per unit area were then determined as a function of emissivity and solar absorptance. This data is presented in Figures 2-30 and 2-31.

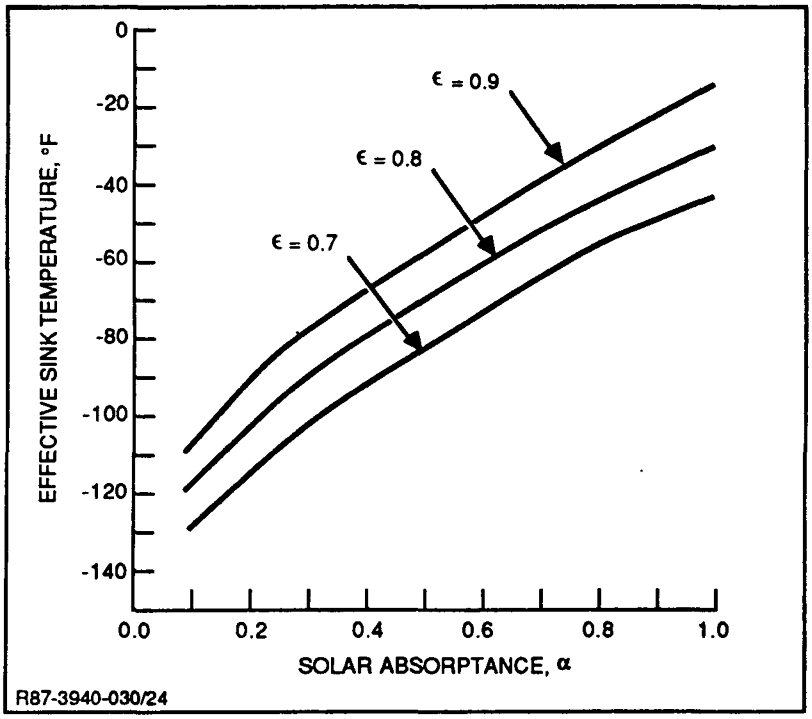


Figure 2-30 Effective Sink Temperature

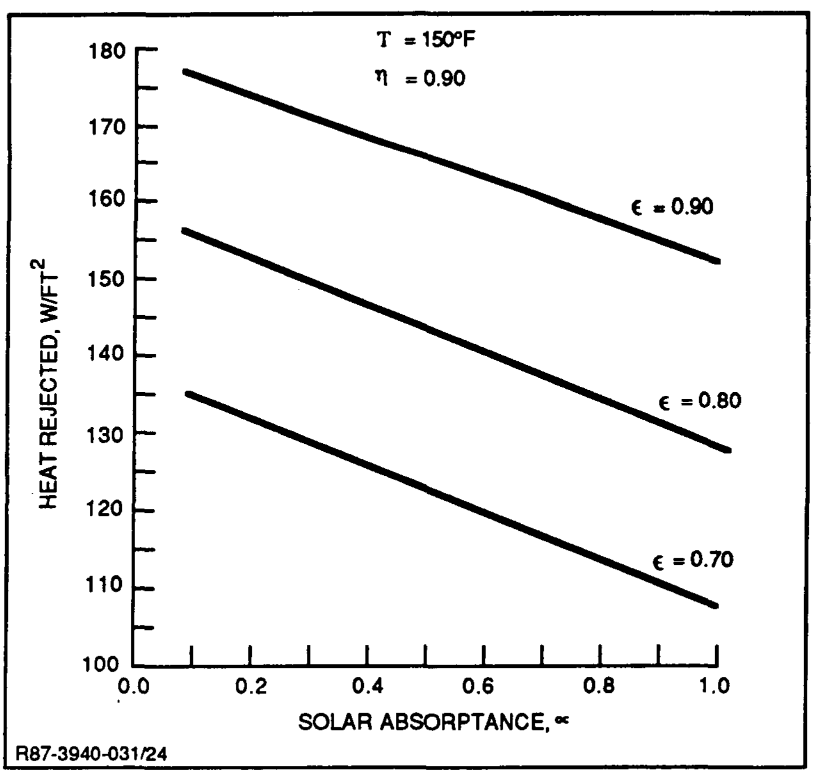


Figure 2-31 Heat Rejection Variation with Surface Properties

An examination was made of the effect of obtaining an emissivity higher than 0.75 while also increasing the solar absorptance above 0.20. If the emissivity is increased to 0.90, the thermal performance is increased regardless of the increase in the solar absorptance. This comparison shows that, generally, the emissivity should be maximized as much as possible, and the solar absorptance does not have a large effect. A 30-year life should be attainable with a higher solar absorptance. Development of relatively thick anodic coatings should be pursued for the solar dynamic radiators to achieve a high emissivity coating. The method of attaching a heat pipe to a fin with an anodized coating also needs further investigation.

2.6 ASSEMBLY & MAINTENANCE

A key trade in optimizing the solar dynamic heat rejection system is comparing a deployable radiator with a constructible radiator. From the point of view of both speed and ease of achieving an operational configuration after transport to orbit, the following hierarchy of methods may be presented in order of preference:

- System launched pre-assembled in operational configuration
- System launched pre-assembled but requiring on-orbit deployment
- System constructible on-orbit.

In addition to the speed and ease of achieving the initial configuration, critical evaluation factors include life-cycle costs, reliability, maintainability, technology readiness and critical resource usage. The latter include Remote Manipulator System (RMS) time, astronaut extra-vehicular activity (EVA) time, launch weight and launch volume. These factors are also important in establishing the overall system configuration.

The first option of launching the system pre-assembled in an operational configuration was immediately eliminated because the heat rejection system would not fit into the Shuttle cargo bay. The second two options were analyzed in greater detail.

The deployable heat pipe radiator system design can draw heavily on the technology which has been developed for deployable pumped fluid loop radiator systems. Several of these systems have been built and demonstrated for NASA (References 2 and 3). Figure 2-32 shows a conceptual design of a deployable system which utilizes a scissors-type deployment mechanism. A gear drive at the evaporator end of the radiator panels rotates the scissor arms to deploy the entire array. The mechanism could be driven by either a motor attached to the radiator, a motor attached to the

RMS, or an EVA astronaut using a motor manually. The recommended approach is to use a motor attached to the radiator in order to minimize astronaut involvement, with backup provisions for RMS or EVA driven deployment.

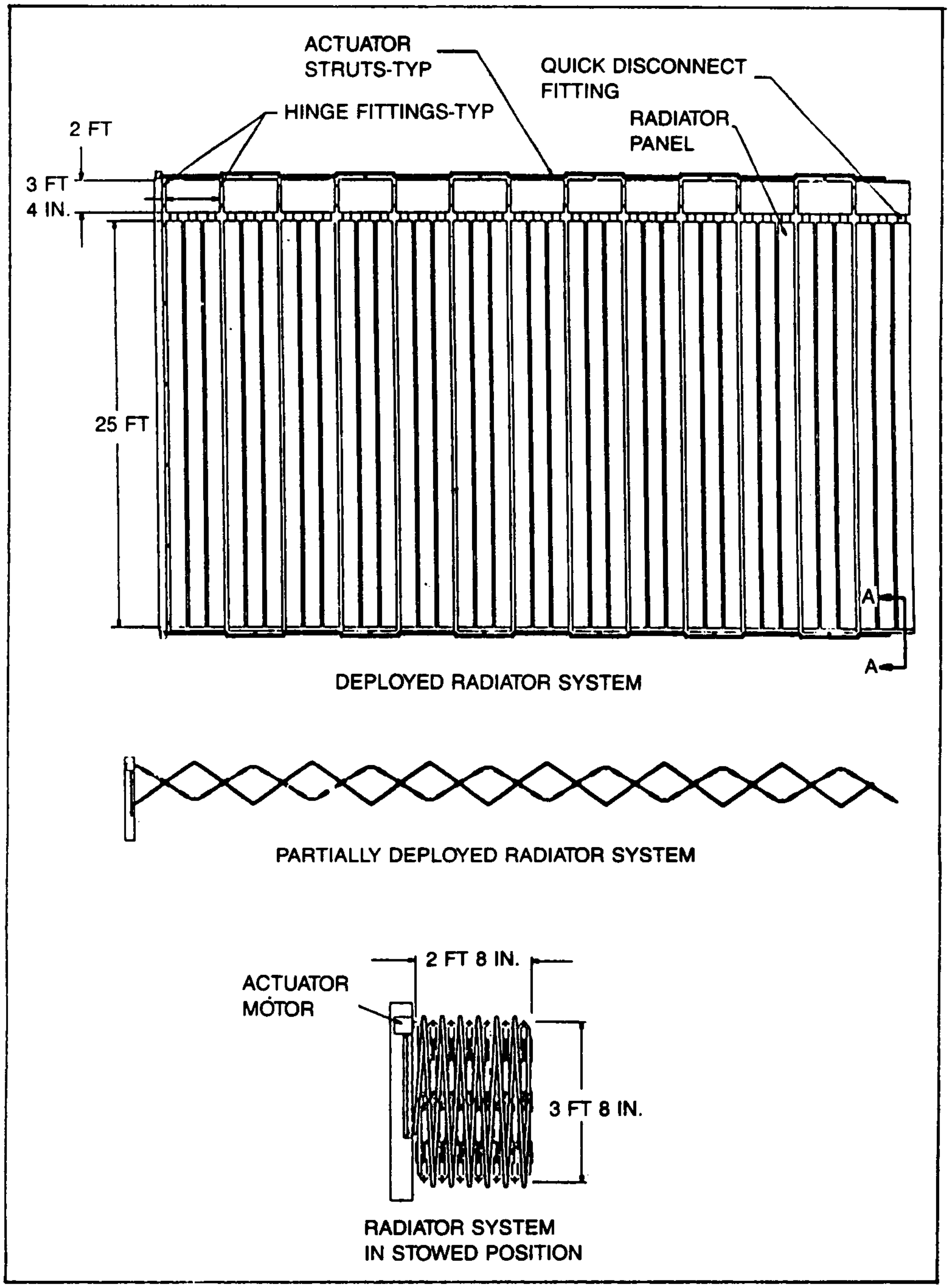


Figure 2-32 Deployable Heat Pipe Radiator System Concept

The heat exchanger interface must be compatible with the deployment system. The wiffletree mechanism is not feasible, because it would not allow the radiator panels to be folded compactly. Both the heat pipe disconnect and the heat exchanger disconnect options are possible. The use of wide panels is particularly advantageous with this concept to minimize the complexity of the deployment mechanism. The use of a single heat pipe disconnect with many condenser legs is not feasible for wide panels because all of the legs would have to be connected to the same disconnect, thus reducing the heat pipe performance. Multiple heat pipe disconnects per panel would likely cause problems with alignment during assembly because of manufacturing tolerances and thermal expansion or contraction of the condenser part of the radiator panel relative to the evaporator section. The modularized radiator/heat exchanger or the heat pipe disconnect is the recommended configuration with a deployable system. Flex hoses shielded for micrometeoroid protection must be used between heat exchanger segments.

On-orbit assembly can be done using either RMS, EVA or a combination of the two. The details of assembly using these methods have been studied under the Grumman SCR contract, Space Station Work Package 2, Grumman IR&D programs, and by NASA. Under the solar dynamic heat rejection contract, the information gained from the other studies has been used to examine the effect on the overall system of RMS or EVA assembly.

The availability of EVA time and an RMS system are primary drivers in examining the assembly options. NASA has specified that the use of EVA for radiator assembly should not exceed 2 man-hours (2 EVA astronauts for 1 hour) per module. The primary Space Station mobile remote manipulator (MRMS) and a dedicated power system remote manipulator (PRMS) will be available for remote assembly of the radiator. Since the use of EVA should be eliminated if at all possible, the radiator system assembly should be done with the RMS systems. This appears feasible with all of the system configuration options considered. The radiator system can be transported from the Shuttle to the power module location with the MRMS and then assembled using the PRMS. A detailed assembly scenario for RMS assembly of the ORC and CBC heat rejection systems is shown in Part III.

On-orbit construction using RMS or EVA both appear feasible, as does a deployable system. Further comparison of the three options and a comparison of the critical resource usage requirements of each is presented in Part III. Radiator panel replacement can also be done using either RMS or EVA for both the wiffletree and heat pipe disconnect concepts. Further study is required to examine the feasibility of replacing radiator panels with the RMS for the modularized radiator/heat exchanger concept.

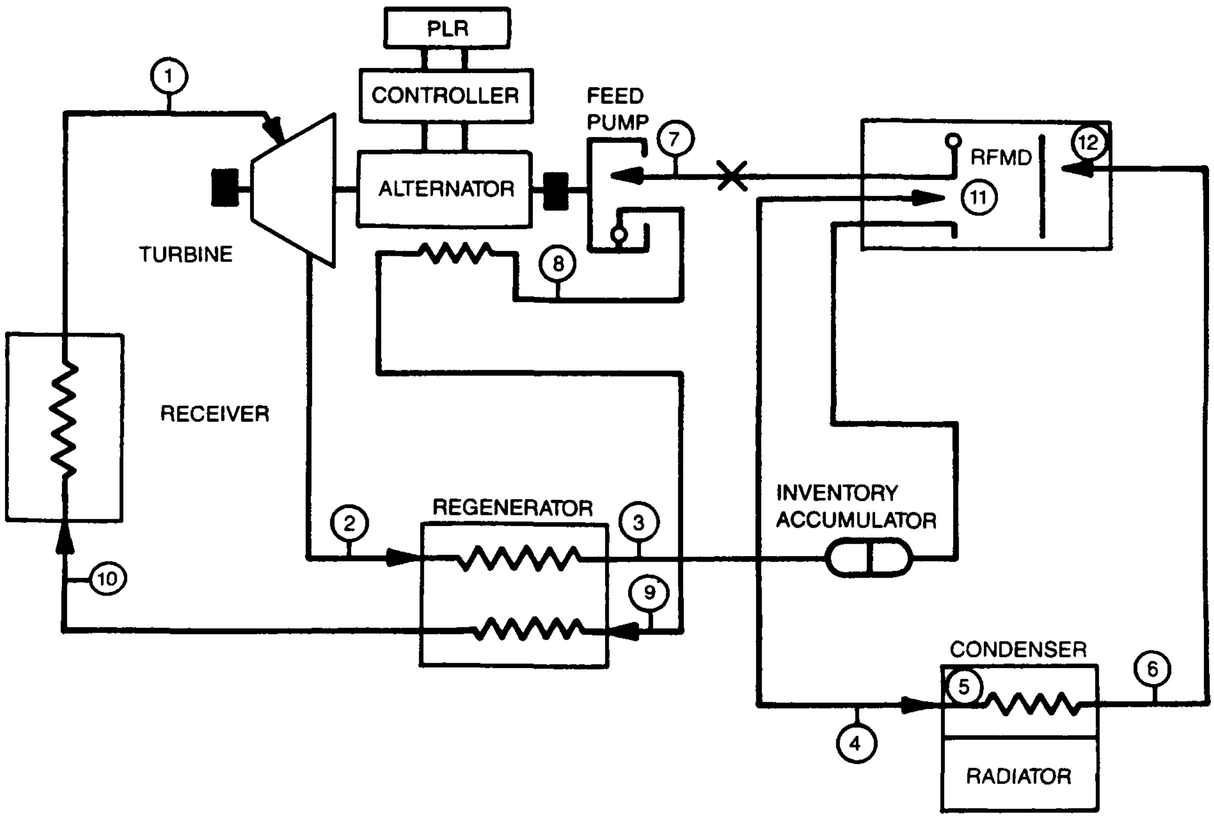
3 - SYSTEM DESIGN & ANALYSIS

3.1 SYSTEM CONFIGURATIONS

3.1.1 ORC System Design

The ORC cycle which was specified by NASA to be used for the system design studies was obtained from Reference 4 and is shown in Figure 3-1. The total heat rejection requirement is 113.3 kW with a condenser inlet pressure of 3.45×10^4 N/m^2 (5.0 psi). The heat rejection system on which this is based utilizes a flow-through shear flow condenser with a relatively high pressure drop. Since it was recommended in Section II that a parallel condenser arrangement be used, the cycle was modified by decreasing the condenser pressure drop, thus raising the heat rejection temperature. The predicted condenser pressure drop for a parallel arrangement was 344 N/m^2 (0.05 psi), which reduces the condensing temperature by only 0.3 K (0.5°F). The remainder of the cycle was unchanged. In reality, the pumping power requirement would also be decreased and the cycle efficiency increased, but this was not included in the analysis.

Based on Section 2 studies, a baseline and several alternate system configurations were selected for further analysis. The baseline system is not necessarily the recommended configuration, but merely the basis by which comparison between systems is done. Figure 3-2 shows schematically the baseline heat rejection system which utilizes ammonia-aluminum dual-slot heat pipes, the wiffletree clamping mechanism to join the heat exchanger and radiator panel, and the Grumman thermal bus condenser. The radiator panels have external dimensions of 14.6 meters (48 ft) length by 0.30 meters (1 ft) wide. Each panel has a 0.6 meter (2 ft) long evaporator section, a 0.3 meter (1 ft) long transport section, and a 13.7 meter (45 ft) long condenser section. Each panel has two independent heat pipes, each of which has a single condenser leg and four parallel evaporator legs. For the reliability and maintainability calculations, the two heat pipes are not assumed to function independently because they are sized to equally share the heat transport and thus would not maintain the required performance factor of safety with one heat pipe failed. The aluminum fin is configured in a monocoque structure with the two



STATE POINTS

LOCATION	DESCRIPTION	TEMPERATURE, °F	PRESSURE, PSI	MASS FLOW, LB/SEC
1	TURBINE INLET	676.0	731.	0.604
2	REGENERATOR VAPOR IN	442.9	5.70	0.604
3	REGENERATOR VAPOR OUT	203.3	5.56	0.600
4	CONDENSER VAPOR IN	203.3	5.00	0.576
5	CONDENSER SATURATED VAPOR	170.2	5.00	-
6	CONDENSER EXIT	137.7	3.85	0.576
7	FEED PUMP EXIT	168.5	4.86	0.604
8	FEED PUMP EXIT	178.0	937.	0.604
9	REGENERATOR LIQUID IN	187.0	927.	0.604
10	FEED PUMP EXIT	372.0	867.	0.604
11	RFMD VAPOR (WARM END)	168.2	4.86	0.604
12	RFMD VAPOR (COLD END)	128.1	3.51	0.576

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Figure 3-1 ORC Cycle Diagram

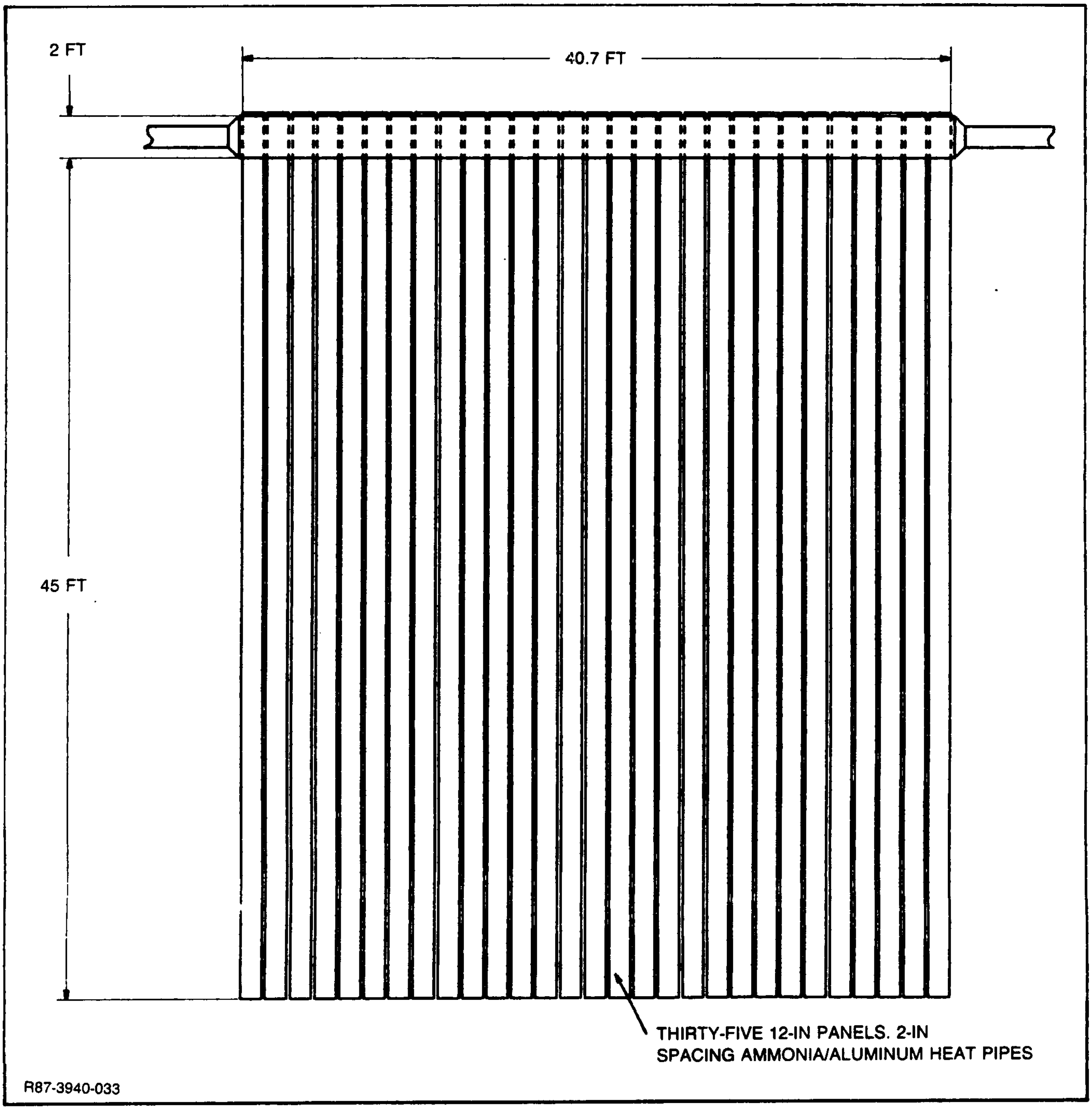


Figure 3-2 ORC Baseline Radiator System Configuration

face sheets each 0.041 cm (0.016 in.) thick for a total fin thickness of 0.81 cm (0.032 in.). Figures 3-3 through 3-4 show details of the radiator panel design. This design can be constructed on-orbit using either RMS or EVA.

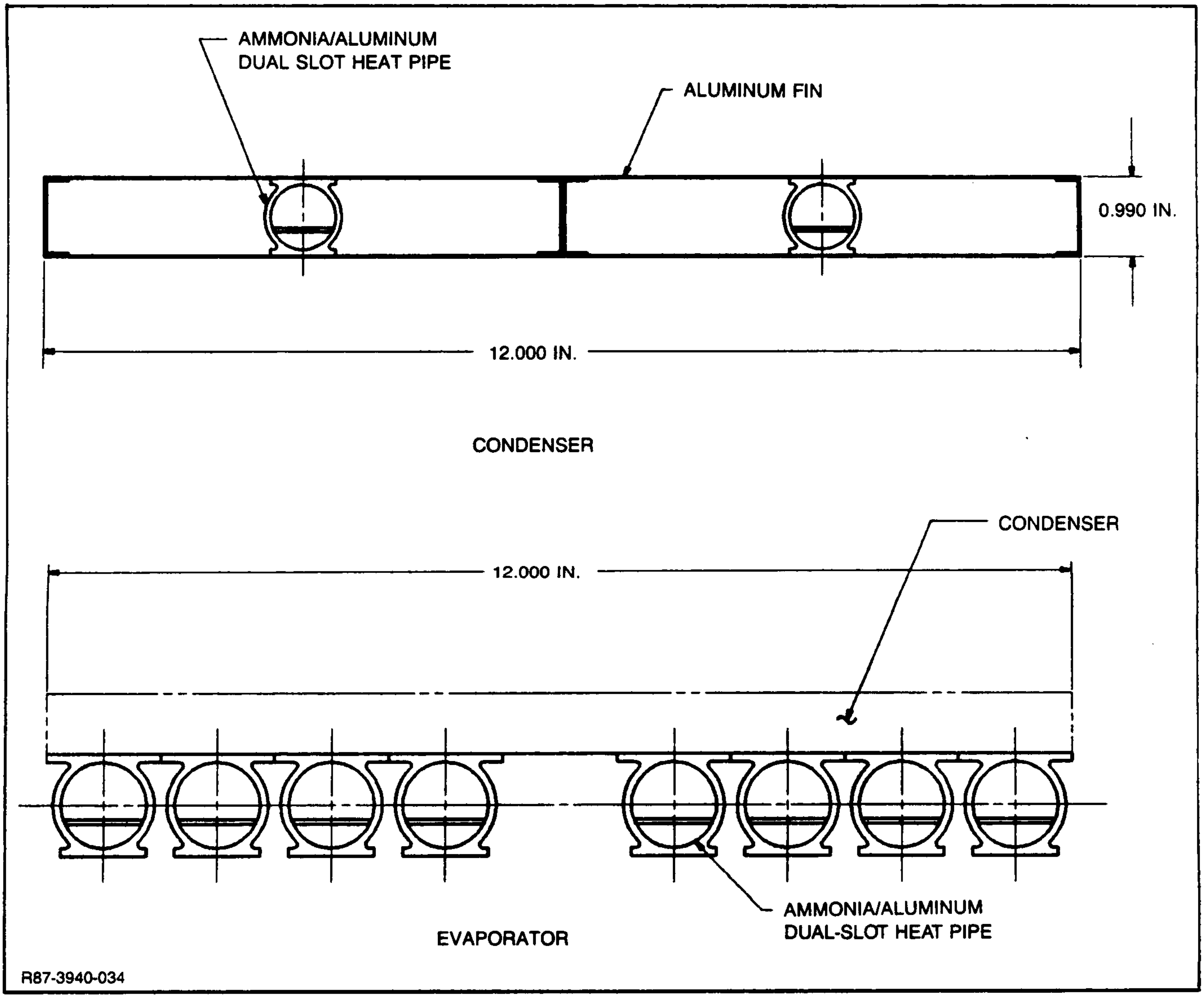


Figure 3-3 ORC Baseline (Dual-Slot) Radiator Panel Cross Section

Alternate configuration A is similar to the baseline configuration, except that it uses the monogroove heat pipe instead of the dual-slot heat pipe design. The monogroove heat pipe is currently further along in development than the dual-slot heat pipe and savings could be realized in development cost by its selection as well as eliminating some of the risk involved in the dual-slot heat pipe development program. While there are advantages to maintaining commonality between the central radiator system and the solar dynamic heat rejection system, there are also several problems involved because of the different operating temperature requirements. The maximum operating temperature requirement for the central radiator system is 294 K (70°F), as opposed to the ORC heat rejection system requirement of approximately 339 K

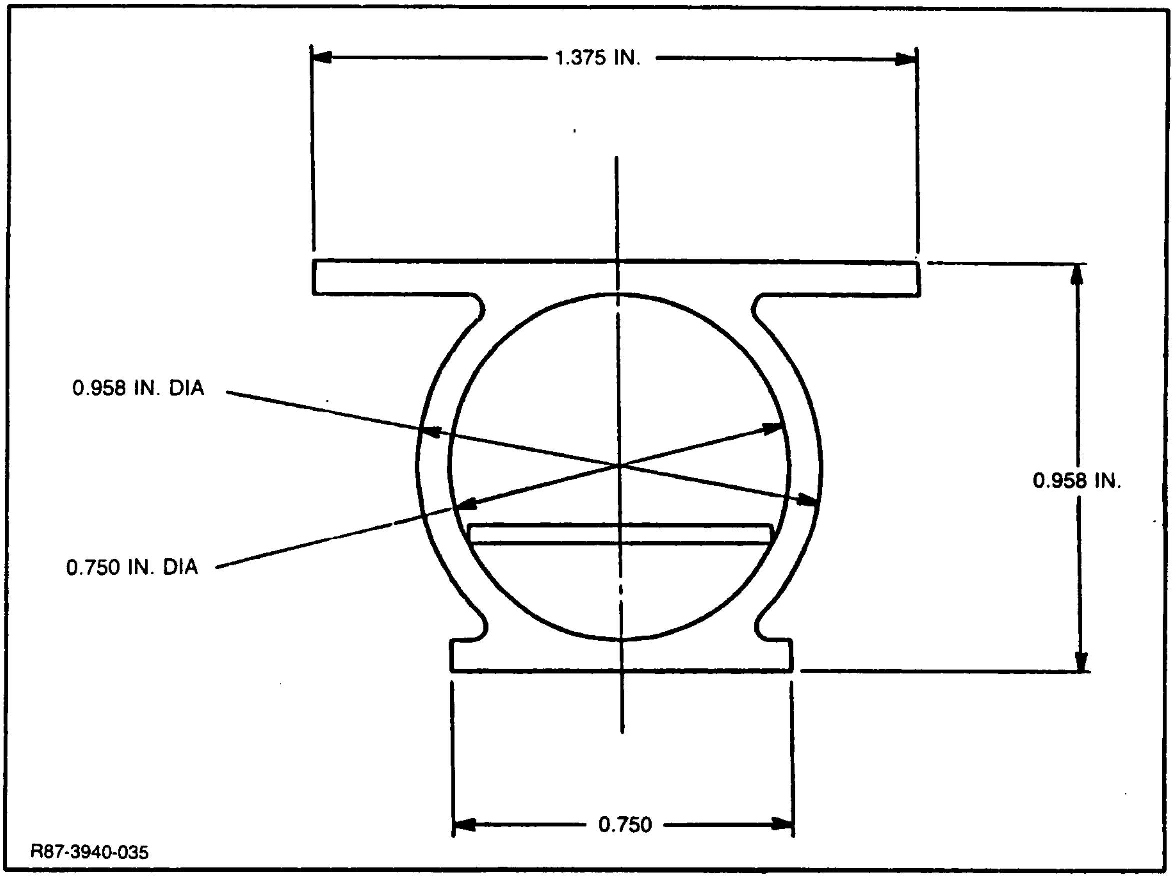


Figure 3-4 ORC Dual-Slot Heat Pipe Detail

(150°F). For the higher operating temperatures, larger heat pipes are required, which means that either the central radiator system heat pipes must be oversized, or different radiator panels must be used for the ORC heat rejection system. Of course, the dual-slot heat pipe could be used in both the ORC heat rejection system and the central radiators. A smaller weight penalty would be involved in maintaining commonality with the dual-slot, since it is inherently lower in weight than the monogroove. Figures 3-5 and 3-6 show the details of a radiator panel with the monogroove heat pipe. The remainder of the system is similar to the baseline system.

Alternate configuration B differs from the baseline configuration in that a heat pipe disconnect is used in place of the wiffletree clamping mechanism. This results in a weight savings per panel compared to the baseline configuration. Also, fewer panels are required to meet the heat rejection requirements. The interface thermal conductance is higher with the disconnect configuration and each panel will operate at a slightly higher temperature and maintain a higher heat rejection rate. Because

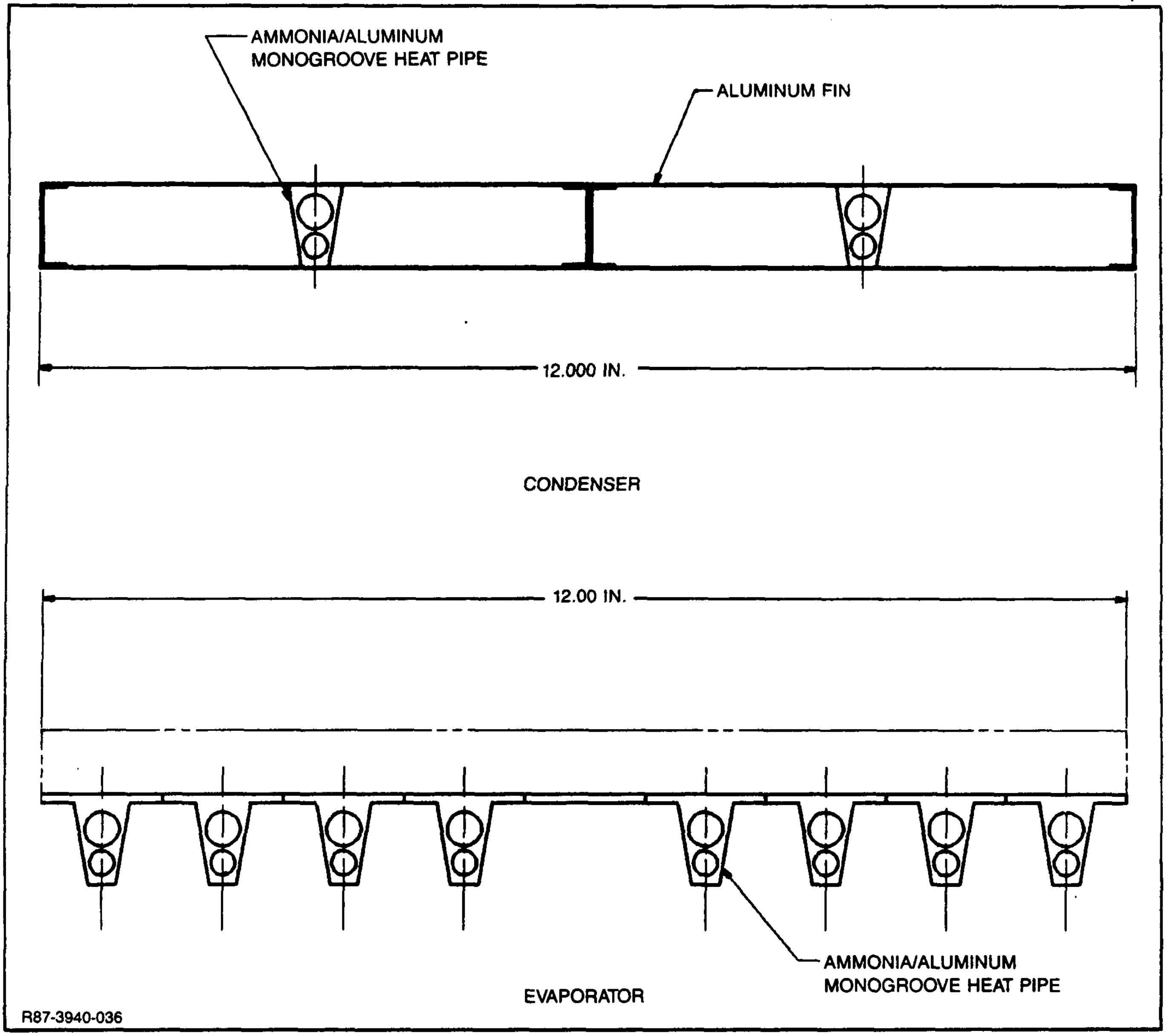


Figure 3-5 ORC Alternate (Monogroove) Radiator Panel Cross Section

the heat pipe disconnect has a higher failure rate more redundancy may be required to achieve a similar system reliability. Alternate configuration C uses both the monogroove heat pipe in place of the dual-slot heat pipe and the heat pipe disconnect in place of the wiffletree. The overall system configurations for alternates B and C are similar to the baseline configuration in that the panels are the same size and the same number of panels are required.

Alternate configuration D is a deployable heat pipe radiator design which utilizes wide panels in which the heat exchanger segments are brazed to the heat

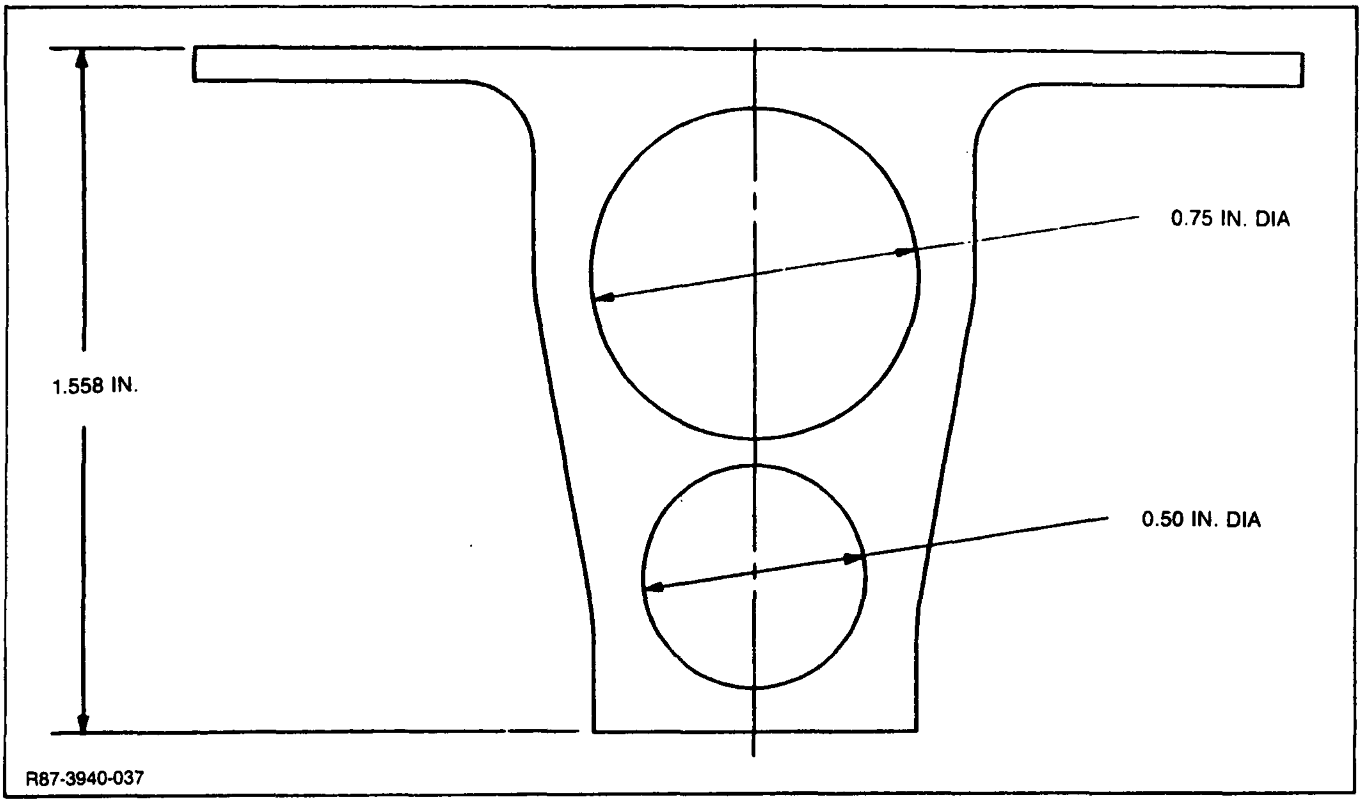


Figure 3-6 ORC Monogroove Heat Pipe Detail

pipe evaporators and there is no disconnect in the heat pipe. This configuration retains the advantage of the heat pipe radiator compared to the pumped fluid loop radiator (i.e., the system is segmented to improve reliability and allow the system to operate with multiple failures due to meteoroid failure). Flex hoses are used in the piping between radiator segments. Maintenance in this configuration would require cutting the piping that supplies fluid to the heat exchanger. An advantage of this design is that a primary source of failure, the heat exchanger - radiator panel interface, is eliminated. Because radiator panel replacement is made more difficult, a larger number of redundant heat pipe elements is incorporated into the design so that there is a low probability of ever requiring radiator panel replacement over the 30 year life of the system. Figure 3-7 shows this system schematically. This configuration was made deployable to simplify on-orbit assembly. Alternately, the system could be constructed on-orbit using the technology being developed for the Space Station thermal bus. While this would be more time consuming, it would result in significant weight savings and possibly reduce the development and manufacturing costs. Alternate configuration E is also a deployable configuration, but uses the monogroove heat pipe instead of the dual-slot heat pipe. The radiator panel dimensions for this configuration are the same as those for configuration D.

The radiator panels for alternate configuration D have external dimensions of 13.8 meters (45.3 ft) long by 1.2 meters (4.0 ft) wide and utilize aluminum-ammonia dual-slot heat pipes. The panels have a 0.6 meter (1.9 ft) long evaporator section, a 0.3 meter (0.9 ft) adiabatic section, and a 12.9 meter (42.5 ft) condenser section. Each panel has eight heat pipes, each of which is composed of a single condenser leg and four parallel evaporator legs. The fin is configured in a monocoque structure, the same as in the baseline configuration.

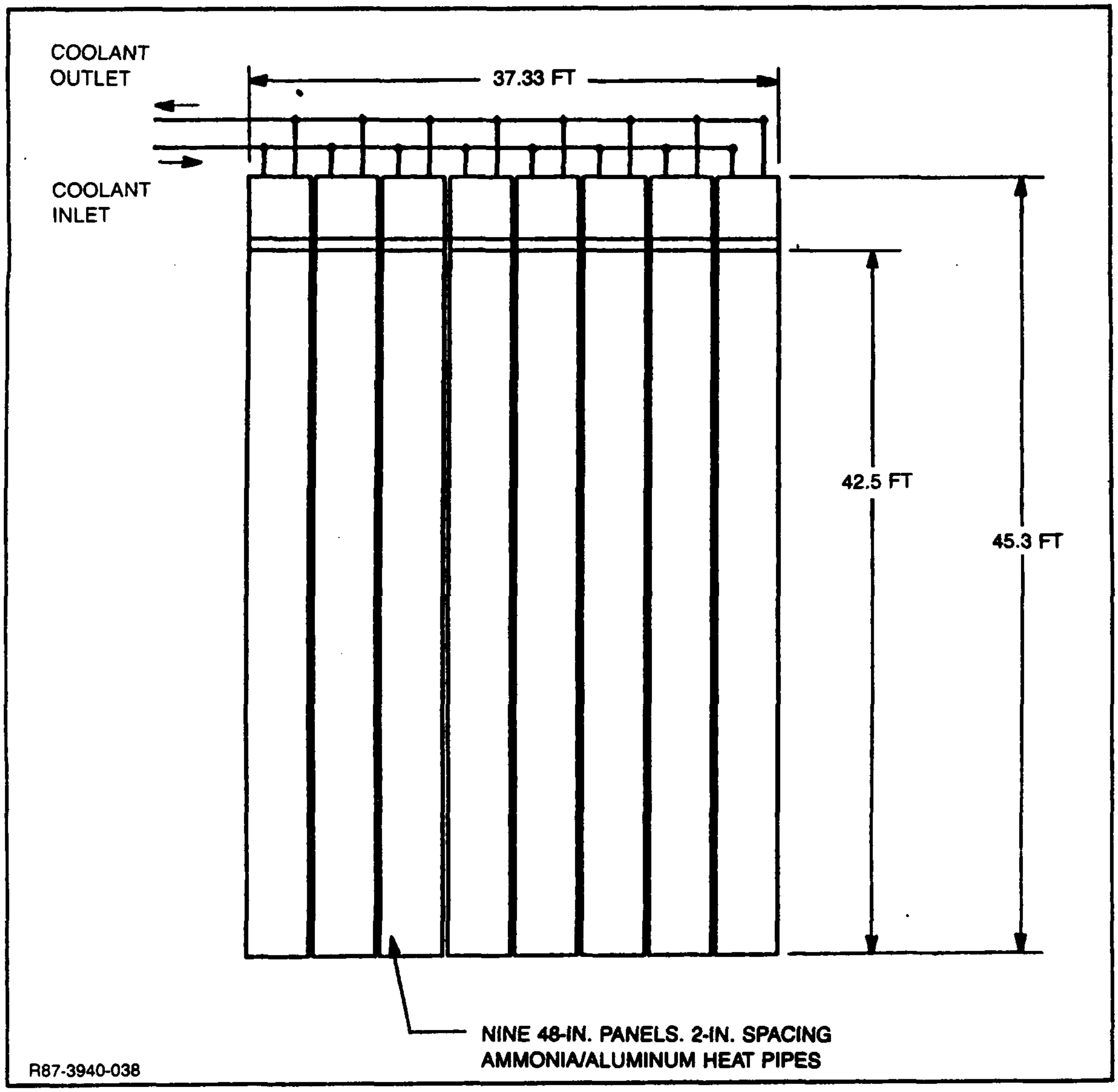


Figure 3-7 ORC (Alternate) Deployable Radiator System Configuration

3.1.1.1 <u>Thermal Performance</u> - The radiator system is designed to reject 113.3 kW at a saturation pressure of 3.45×10^4 N/m^2 (5.0 psi), which corresponds to a condensing temperature of 346 K (162°F). The interface contact conductance between the radiator and the condenser is assumed to be 2835 W/m^2-K (500 Btu/hr-ft^2-°F). The total interface conductance which includes the contact conductance, the effective conductance of the condensing toluene on the condenser side and the evaporating ammonia on the heat pipe evaporator side is 163 W/K (309 Btu/hr-°F). An emissivity of 0.90 and a solar absorptance of 0.50 were used in the calculations. These values are representative of projected end-of-life surface properties for a thick anodized coating. The radiator root temperature is 323 K (122°F). The amount of active radiator area required is 276 m^2 (2970 ft^2), which is achieved with 33 radiator panels. The system design calls for 35 radiator panels, which provides for 2 redundant panels in the event of heat pipe failures. Redundancy and reliability are discussed in Section 3.3.

In the alternate configurations with a brazed heat exchanger interface, an infinite interface conductance is assumed, which results in a total conductance from the condensing toluene to the evaporating ammonia in the heat pipe of 236 W/K (447 Btu/hr-°F). The radiator root temperature is 329 K (132°F). The amount of active radiating area required is 850 m^2 (2790 ft^2) which is achieved with 31 radiator panels in alternate configurations B and C and 7 radiator panels in configurations D and E. An additional 2 panels in alternates B and C and one additional panel in alternates D and E are provided for redundancy.

3.1.1.2 Mass Properties - The weights of each of the candidate ORC heat rejection systems are shown in Table 3-1. Only the radiator, heat exchanger, interface and deployment system (if applicable) weights are included. Not included are the piping or structural support weights. Table 3-2 shows a breakdown of the various items of the radiator weight.

Table 3-1 ORC Heat Rejection System Weights

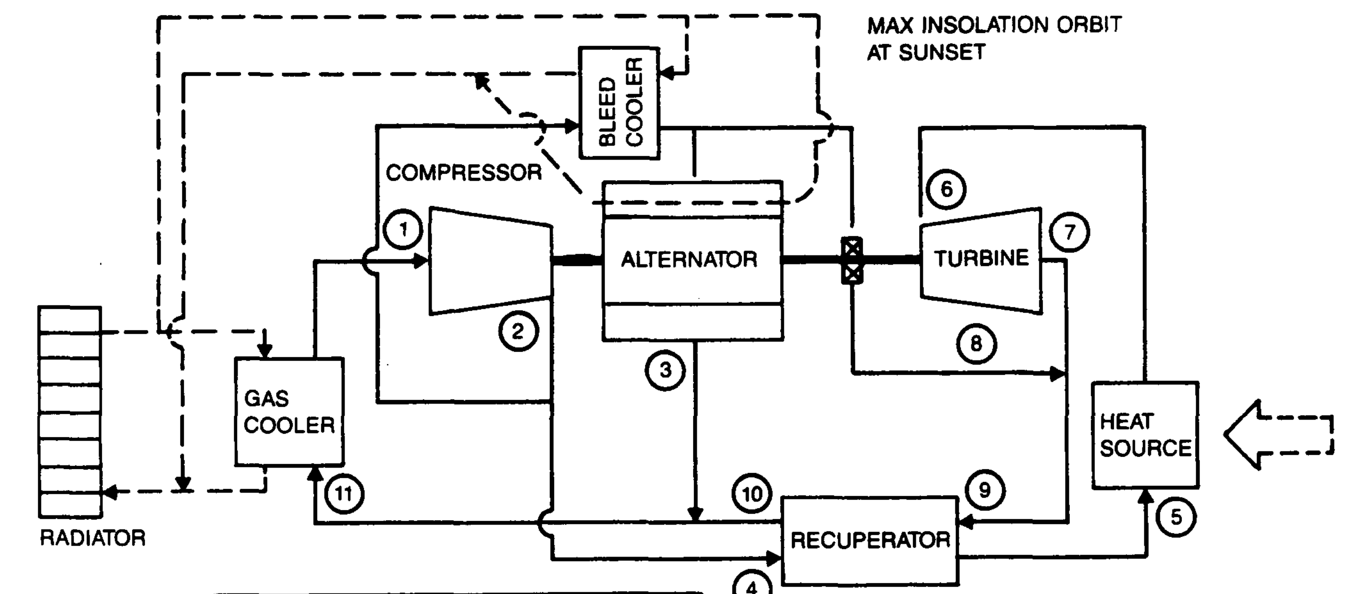
	Configuration					
Component	Baseline	Alt A	Alt B	Alt C	Alt D	Alt E
Heat Pipe Interface No. Panels	Dual-Slot Wiffletree 35	Monogroove Wiffletree 35	Dual-Slot Disconnect 33	Monogroove Disconnect 33	Dual-Slot Brazed 9	Monogroove Brazed 9
Panel Wt (lb)	3045	4256	2871	4013	2942	4112
Interface Wt (lb)	1050	1050	429	429	500	500
HX Wt (lb)	350	350	330	330	338	338
Total Wt (lb)	4445	5656	3630	4471	3780	4950

Table 3-2 ORC Radiator Panel Weights

Item	Dual-Slot Panel	Monogroove Panel
Heat Pipe (lb)	55.2	89.8
Radiator Fin (lb)	21.0	21.0
Fin Stiffeners (lb)	8.1	8.1
Radiator Coating (lb)	2.7	2.7
Total (lb)	87.0	121.6

3.1.2 CBC System Design

The CBC cycle which was specified by NASA to be used for the system design studies was obtained from Reference 5 and is shown in Figure 3-8. The total heat rejection requirement is 85.1 kW.



	TEMPERATURE °F	PRESSURE PSIA	FLOW RATE LB/SEC
1	90.3	37.19	2.49
2	270.7	68.49	2.49
3			.050
4	271.0	68.40	2.43
5	1035.5	67.96	2.43
6	1429.8	68.86	2.43
7	1095.8	38.10	2.43
8			.012
9	1019.8	38.05	2.44
10	33.3	37.53	2.44
11	338.1	37.53	2.49

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Figure 3-8 CBC Cycle Diagram

The baseline heat rejection system design for the CBC cycle uses methanol-titanium dual-slot heat pipes in the radiator panels and wiffletree clamping mechanisms to attach the radiator panels to the heat exchanger. Figure 3-9 presents a schematic of this configuration. An intermediate transport loop with FC-75 coolant is used between the cycle fluid and the radiator system. The heat exchanger is a single flow-through unit with a standard plate-fin design.

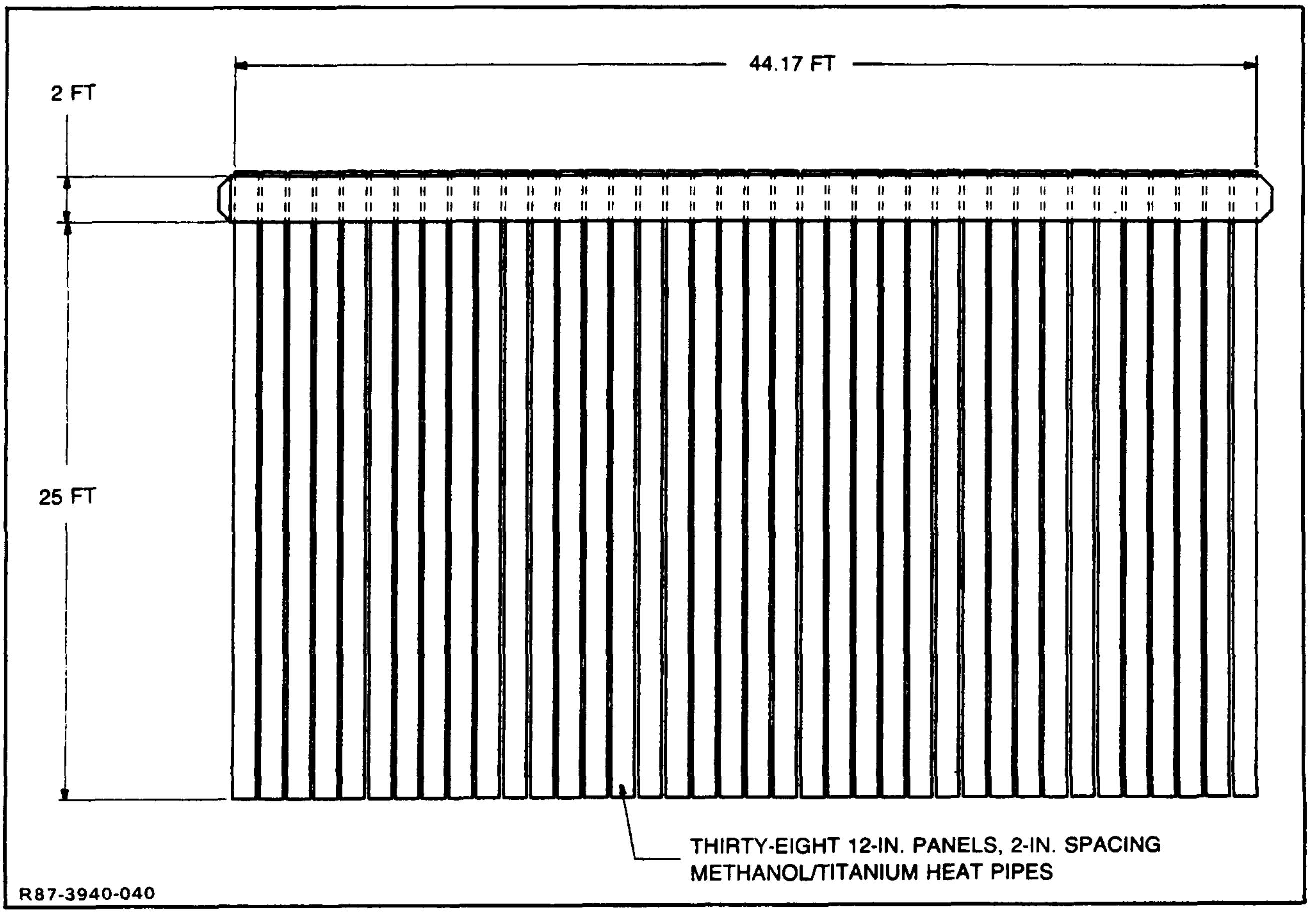

Figure 3-9 CBC Baseline Radiator System Configuration

The radiator panels have external dimensions of 8.5 meters (28 ft) length by 0.3 meters (1 ft) wide. Each panel has a 0.6 meter (2 ft) long evaporator section, a 0.3 meter (1 ft) long transport section, and a 7.6 meter (25 ft) long condenser section. Each panel has two independent heat pipes, each of which has a single condenser leg and 5 parallel evaporator legs. The fin is configured in a wing structure and is 0.081 cm (0.032 in.) thick. Figures 3-10 through 3-11 show details of the radiator panel design. This system can be constructed on-orbit using either RMS or EVA.

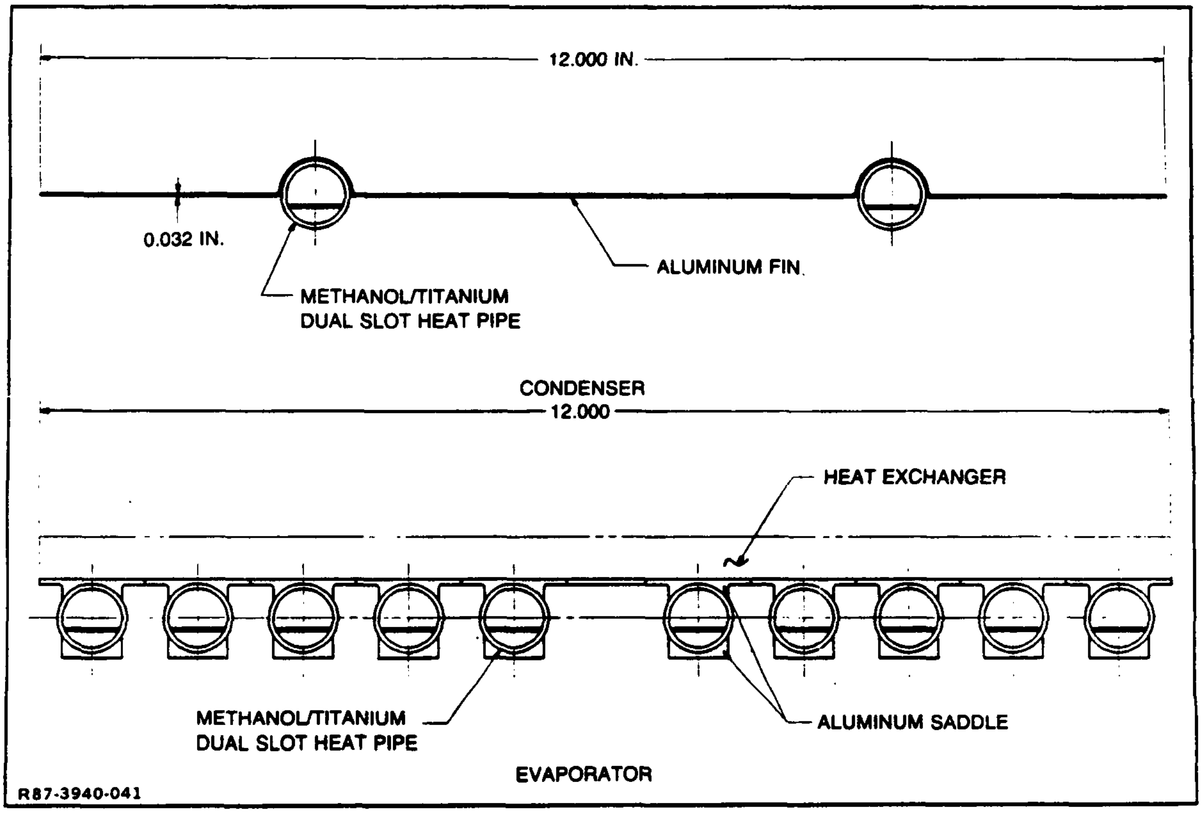

Figure 3-10 CBC Baseline Radiator Panel Cross Section

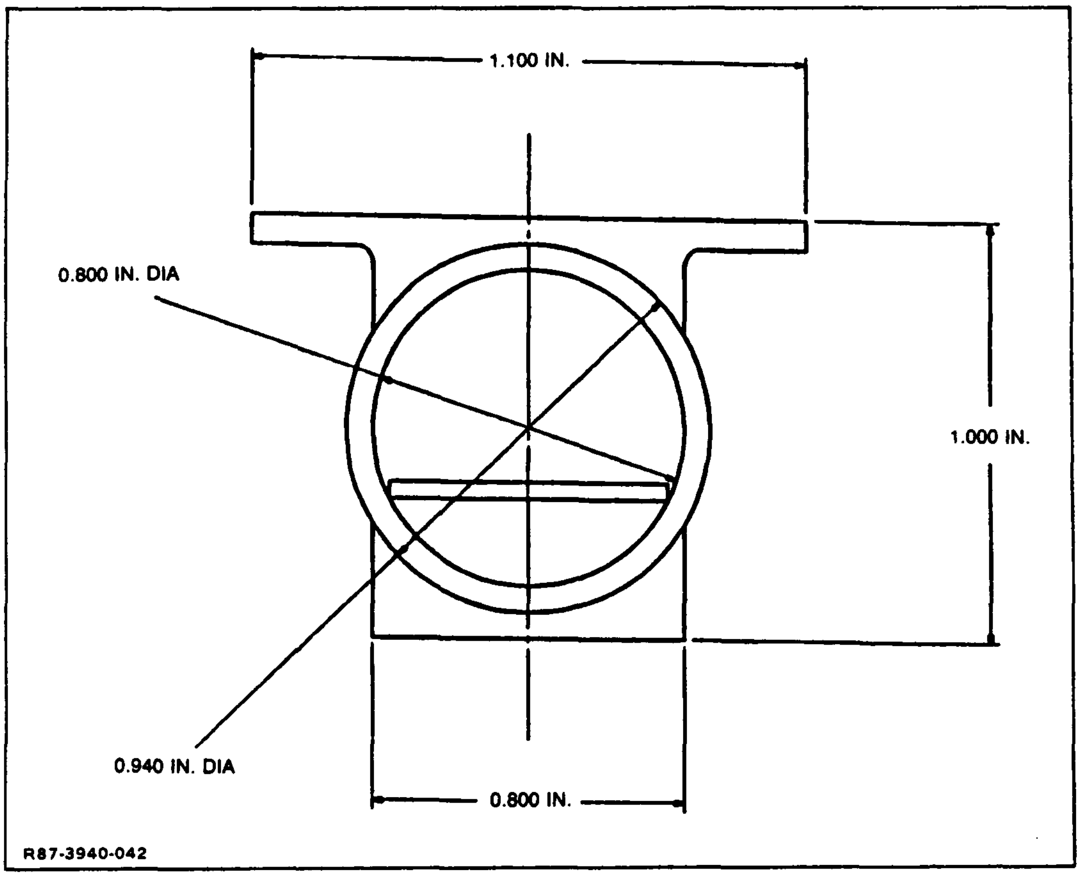

Figure 3-11 CBC Dual-Slot Heat Pipe Detail

Alternate configuration A uses stainless steel for the heat pipe envelope material instead of titanium. This heavier design is included for comparison in the event that titanium does not prove to be compatible with methanol.

Alternate configuration B is a hybrid heat pipe system, utilizing methanol-titanium dual-slot heat pipes for the higher temperature heat rejection and ammonia/aluminum dual-slot heat pipes for the lower temperature heat rejection. Figure 3-12 shows the system layout. The low temperature panels have the same configuration as in the ORC baseline design. The high temperature panels are the same as in the CBC baseline configuration. The wiffletree designs are not deployable, but could be assembled using either RMS or EVA.

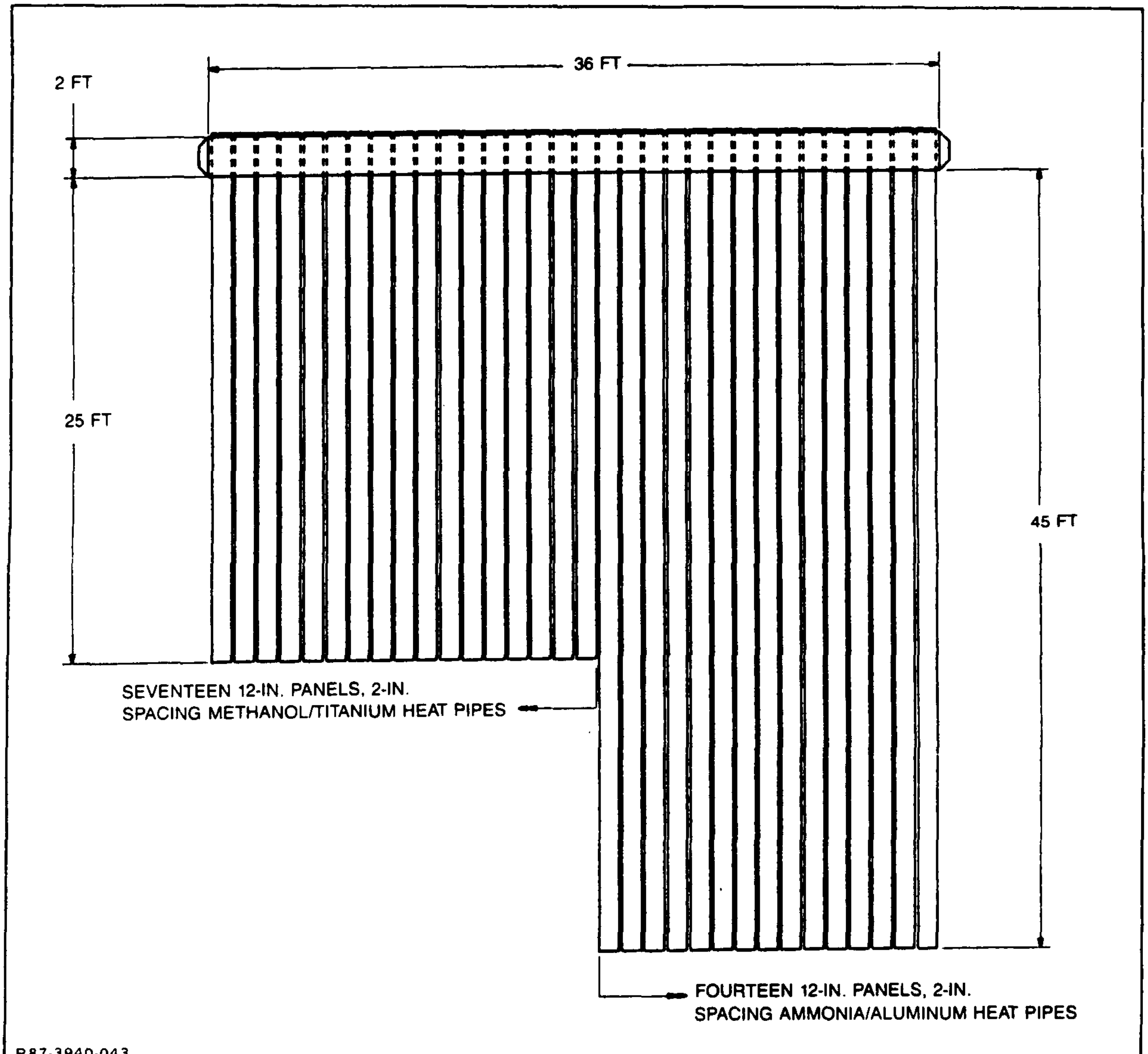

Figure 3-12 CBC Alternate (Hybrid) Radiator System Configuration

Use of two different types of radiator panel could increase development costs and spares requirements, although if the low temperature ammonia/aluminum heat pipe radiator panels are being developed for the central radiator system, the increase may be minimal. The design incorporating the wiffletree uses a single heat exchanger with all of the radiator panels in series. If the wiffletree interface is used for the central radiator system, then the additional development cost and spares requirement for that would also be minimal.

Alternate configuration C differs from the baseline configuration in that the heat pipe disconnect is used in place of the wiffletree clamping mechanism. The radiator panels are the same size as in the baseline configuration.

Alternate configuration D is a deployable heat pipe radiator design, with the radiator panels brazed to the heat exchanger segments and flex joints used in the coolant supply lines. The ten radiator panels have external dimensions of 8.0 meters (26.4 ft) length by 1.3 meters (4 ft) wide. Each panel has a 0.6 meter (1.9 ft) long evaporator section, a 0.3 meter (0.9 ft) long transport section, and a 7.2 meter (23.6 ft) long condenser section. Each panel has eight heat pipes, each of which is composed of a single condenser leg and five parallel evaporator legs. The coolant is pumped in parallel through the radiator panels, but in series over the eight heat pipes within each panel, so that the eight heat pipes will each operate at successively lower temperatures. Figure 3-13 shows this system design schematically.

3.1.2.1 <u>Thermal Performance</u> - The radiator system is designed to reject 85.1 kW by cooling the FC-75 coolant from 423K (301°F) to 296K (73°F) at a flow rate of 0.59 kg/sec (1.31 lb/sec). The interface conductance between the radiator and the heat exchanger is assumed to be 2835 W/m^2-°K (500 Btu/hr-ft-°F). The total interface conductance which also includes the effective conductance of the FC-75 heat exchanger fluid and the evaporating methanol on the heat pipe evaporator side is 201 W/°K (382 Btu/hr-°F). This is higher than in the ORC heat exchanger design because of the differences in heat exchanger design and condensing and convective heat transfer coefficients. The heat exchanger pressure drop on the FC-75 side is 1.72×10^4 N/m^2 (2.5 psi). An emissivity of 0.90 and a solar absorptance of 0.50 were used as the end-of-life surface coating properties for a thick anodized coating.

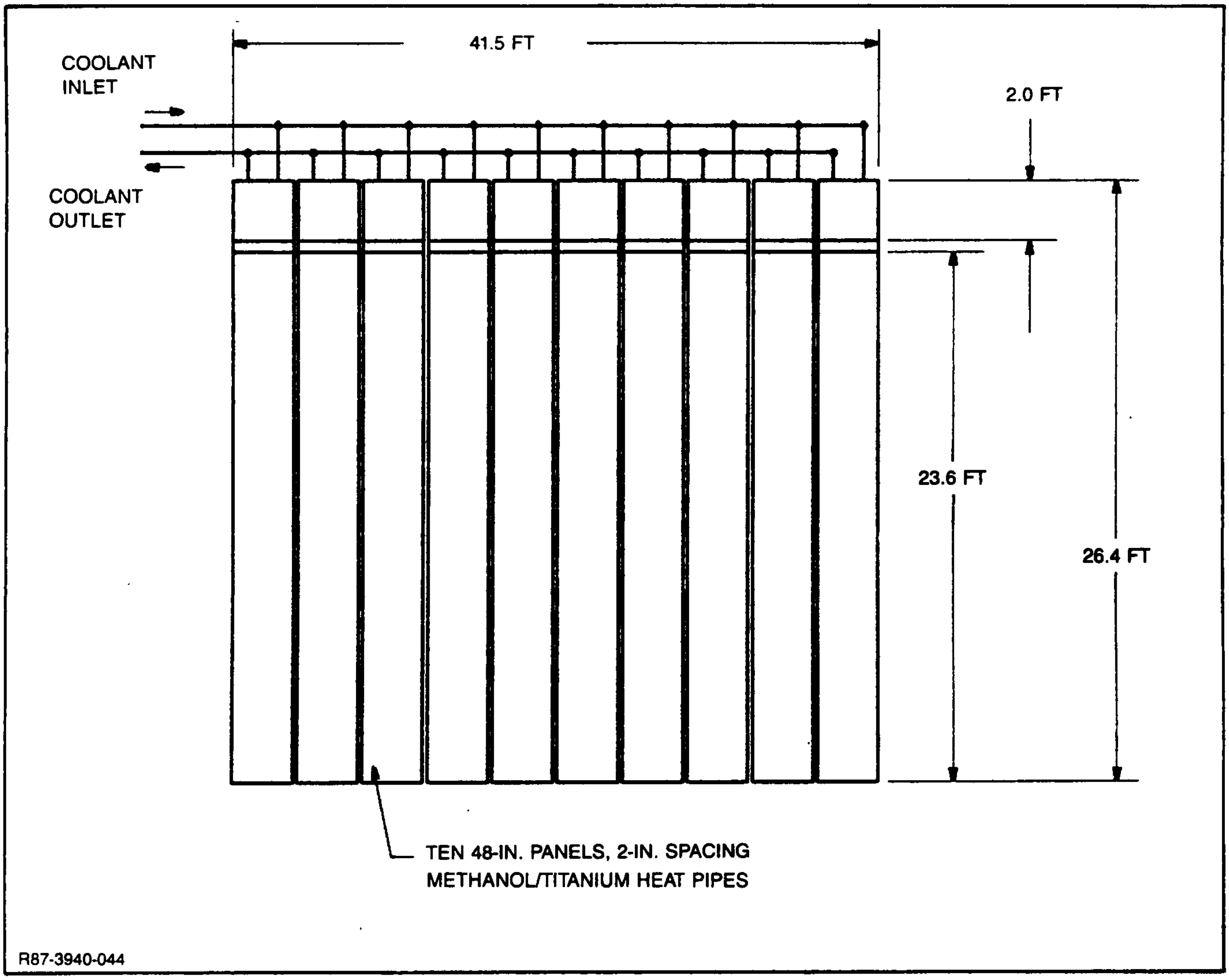

Figure 3-13 CBC Alternate (Deployable) Radiator System Configuration

Figure 3-14 shows the temperature profiles in the radiator panels for the baseline design without the redundant panels included. Figures 3-15 through 3-17 show the radiator temperature profiles for alternate configurations B, C, and D. In alternates C and D, an infinite interface conductance is assumed for the brazed joint between the heat exchanger and radiator panel. For this case the total interface conductance is 326 W/°K (618 Btu/hr-°F).

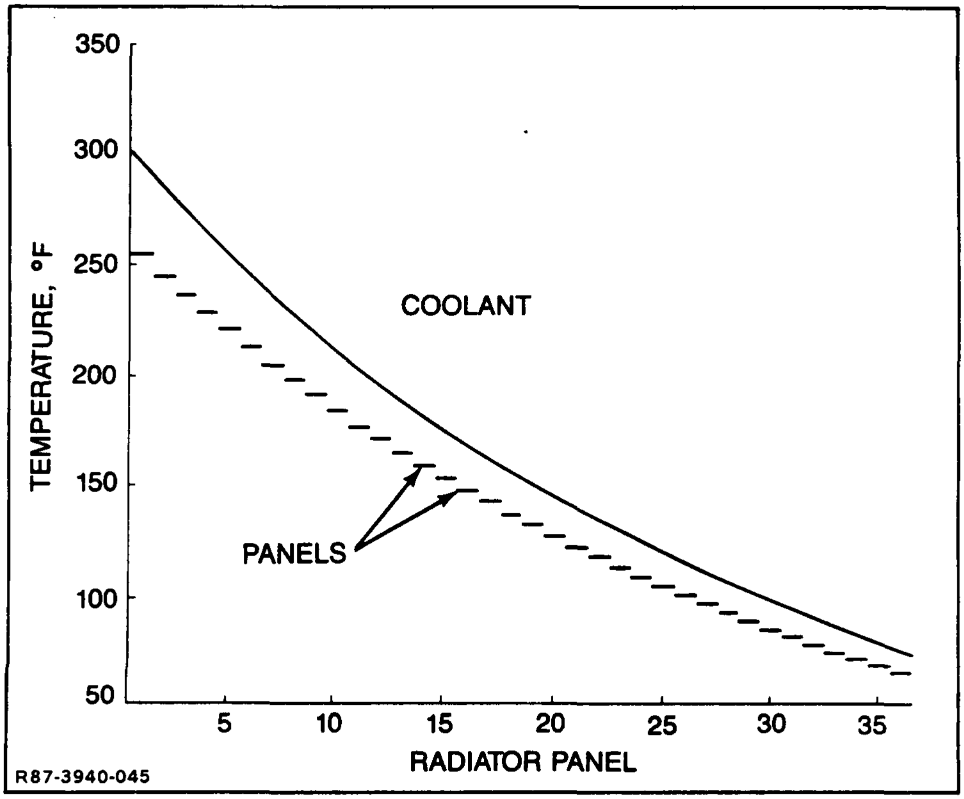

Figure 3-14 CBC Baseline Radiator System Temperature Profile

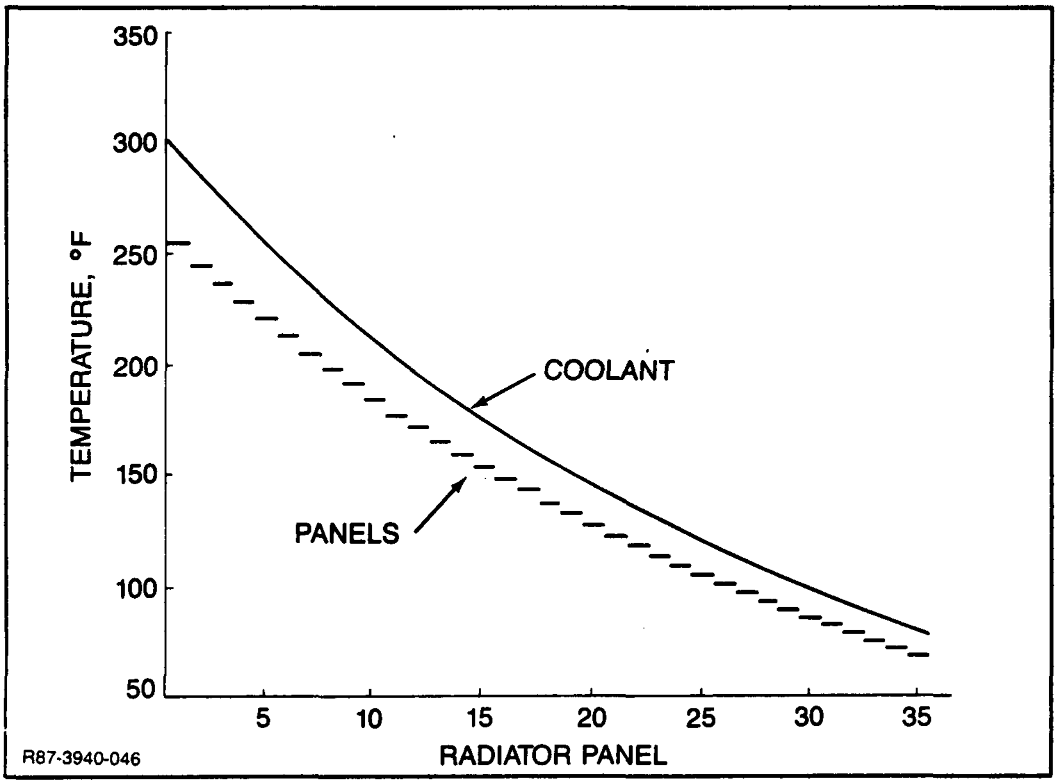

Figure 3-15 CBC Alternate (Disconnect) Radiator System Temperature Profile

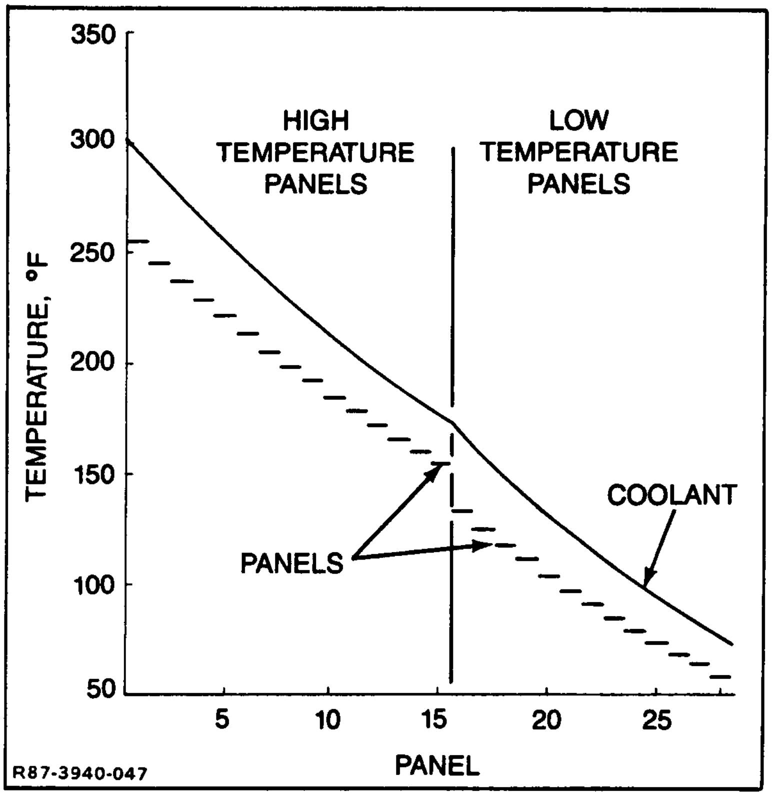

Figure 3-16 CBC Alternate (Hybrid) Radiator System Temperature Profile

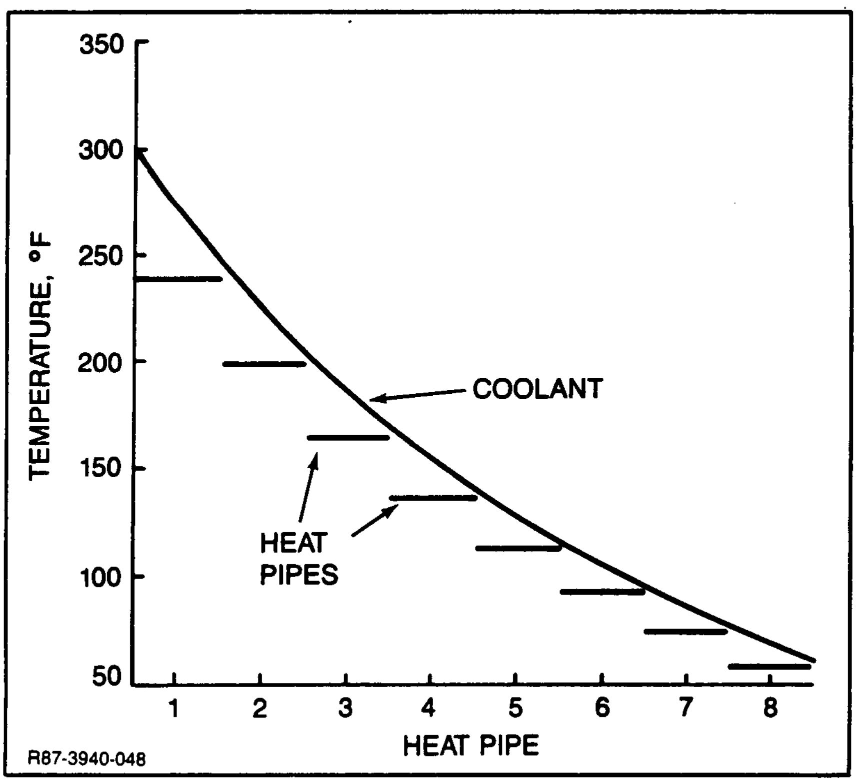

Figure 3-17 CBC Alternate (Deployable) Radiator System Temperature Profile

3.1.2.2 Mass Properties - The weights of each of the candidate CBC heat rejection system designs are shown in Table 3-3. Only the radiator, heat exchanger, interface and deployment system (if applicable) weights are included. Not included are the weights associated with the intermediate loop or structural support weight. Table 3-4 shows the breakdown of the various items of the radiator weight.

Table 3-3 CBC Heat Rejection System Weights

	Configuration				
Component	**Baseline**	**Alt A**	**Alt B**	**Alt C**	**Alt D**
Heat Pipe Fluid	**Methanol**	**Methanol**	**Hybrid**	**Methanol**	**Methanol**
Heat Pipe Material	**Titanium**	**Stainless**	**Hybrid**	**Titanium**	**Titanium**
Interface	**Wiffletree**	**Wiffletree**	**Wiffletree**	**Disconnect**	**Brazed**
No. Panels	**38**	**38**	**17/14**	**37**	**10**
Panel Wt (lb)	1701	2449	1923	1662	1700
Interface Wt (lb)	1140	1140	930	481	500
HX Wt (lb)	380	380	310	370	378
Total (lb)	3227	3969	3163	2513	2578

Table 3-4 CBC Radiator Panel Weights

Item	Dual-Slot Methanol Titanium Panel	Dual-Slot Methanol Stainless Panel	Dual-Slot Ammonia Aluminum Panel
Heat Pipe (lb)	31.4	50.9	55.2
Radiator Fin (lb)	12.0	12.0	21.0
Fin Stiffening (lb)	0.0	0.0	8.1
Radiator Coating (lb)	1.5	1.5	2.7
Total (lb)	44.9	64.4	87.0

3.2 ON-ORBIT ASSEMBLY

The method of assembly should be selected on the basis of life cycle cost and critical resource usage. The critical resource factors are launch weight and volume, EVA time, and RMS time. All of the system configurations which have been presented can be assembled entirely using the RMS facilities available on the Space Station. A detailed assembly sequence and time estimate were generated for assembling the radiator systems (except for the deployable systems) and is shown in Table 3-5. On this basis, assembly time estimates were made for each of the system designs shown previously.

Table 3-5 IVA-RMS Radiator Assembly Sequence

SOLAR DYNAMIC RADIATOR ASSEMBLY

IVA SCENARIO USING MOBILE RMS (MRMS) AND POWER SYSTEM RMS (PSRMS)
(2 power modules, N panels per module)

TASK #	TASK DEFINITION	DURATION (min.)
1	MRMS moves into position to reach radiator panels in STS cargo bay	5
2	MRMS locates and unlatches a bundle of 2N radiator panels	
2a	- Engages Tool #1 (unlatcher)	5
2b	- Unlatches bundle	20
3	MRMS grapples bundle	
3a	- Releases and stows Tool #1	5
3b	- Engages Tool # 2 (grappler/inserter)	5
3c	- Locates grapple adapter	5
3d	- Engages bundle	10
4	MRMS removes bundle from cargo bay	5
5	MRMS positions bundle for transport	5
6	MRMS traverses to first alpha joint	15
7	Bundle #1 transfered to PSRMS #1	10
8	MRMS traverses to second alpha joint	15
9	Bundle #2 transfered to PSRMS #2	10
10	MRMS returns to STS cargo bay	10
11	PSRMS #1 traverses to 1st power module	5
12	PSRMS #1 docks and releases bundle	5
13	PSRMS #1 inserts N radiator panels on first module	
13a	- Removes panel from bundle (N times)	2N
13b	- moves into glide path (N times)	2N
13c	- inserts panel (N times)	5N
13d	- clamping mechanism engaged	5N
13e	- returns to bundle location (N times)	2N
14	PSRMS #1 returns to alpha joint	5
15	PSRMS #2 traverses to 2nd power module	5
16	PSRMS #2 docks and releases bundle	5
17	PSRMS #2 inserts N radiator panels on second module	
17a	- Removes panel from bundle (N times)	2N
17b	- moves into glide path (N times)	2N
17c	- inserts panel (N times)	5N
17d	- clamping mechanism engaged	5N
17e	- returns to bundle location (N times)	2N
18	PSRMS #2 returns to alpha joint	5
		155 + 32N per two modules
		(77.5 + 16N per module)

3.2.1 ORC System Assembly

Table 3-6 presents estimates of the critical resource usage for each of the six system configurations presented in Section 3.1.1 for RMS assembly, EVA assembly or a deployable system design.

Table 3-6 ORC Radiator System Critical Resource Requirements

	Deployable System	RMS Assy	EVA Assy
Wiffletree Design			
Launch Weight (lb)	N/A	4445	4445
Launch Volume (ft^3)	N/A	218	218
EVA Time (Man-hr)	N/A	0.0	7.1
RMS Time (Man-hr)	N/A	10.6	7.1
Heat Pipe Disconnect Design			
Launch Weight (lb)	4130	3630	3630
Launch Volume (ft^3)	146	140	140
EVA Time (Man-hr)	0.0	0.0	6.8
RMS Time (Man-hr)	1.5	10.1	6.8
Integral HX/Radiator Design			
Launch Weight (lb)	3780	3370	3370
Launch Volume (ft^3)	150	144	144
EVA Time (Man-hr)	0.0	0.0	3.0
RMS Time (Man-hr)	1.5	3.4	3.0

3.2.2 CBC System Assembly

Table 3-7 presents estimates of the critical resource usage for each of the five system configurations presented in Section 3.1.2 for RMS assembly, EVA assembly or a deployable system design.

3.3 RELIABILITY & MAINTAINABILITY

Heat rejection system failures which are considered in the system reliability study are heat pipe puncture due to micrometeoroid or space debris, heat exchanger failure due to leakage, and wiffletree or heat pipe disconnect failure. Failure of a heat pipe due to leakage or buildup of non-condensible gas are not considered be-

Table 3-7 CBC Radiator System Critical Resource Requirements

	Deployable System	RMS Assy	EVA Assy
Wiffletree Design			
Launch Weight (lb)	N/A	3227	3227
Launch Volume (ft^3)	N/A	174	174
EVA Time (Man-hr)	N/A	0.0	7.6
RMS Time (Man-hr)	N/A	11.4	7.6
Heat Pipe Disconnect Design			
Launch Weight (lb)	3013	2513	2513
Launch Volume (ft^3)	104	98	98
EVA Time (Man-hr)	0.0	0.0	7.5
RMS Time (Man-hr)	1.5	11.2	7.5
Integral HX/Radiator Design			
Launch Weight (lb)	2578	2178	2178
Launch Volume (ft^3)	106	100	100
EVA Time (Man-hr)	0.0	0.0	2.6
RMS Time (Man-hr)	1.5	3.4	2.6

cause no data are available on failure rates by these mechanisms and these events are considered to be very unlikely. Since there exist essentially no historical data for failure rates on many of the heat rejection system components in spacecraft environments, the failure rate estimates used in this study are based on existing data for similar equipment designs.

The main parameters involved in optimizing the system reliability are the number of excess heat pipe elements for redundancy and the heat pipe wall thickness. Given the exposed area per pipe, the exposure time, and a valid flux model, the Poisson distribution can be used to find the probability of failure of a single pipe due to puncture. NASA report SP-8013 (Reference 6) is used to determine micrometeoroid flux and the Kessler Space Debris model (Reference 7) is used to determine the debris flux. Appropriate factors for defocusing and shielding are also included. The flux predictions are a function of altitude and particle mass. The minimum particle mass able to penetrate the heat pipe wall is a function of the wall material. The appropriate factors for aluminum and stainless steel were obtained from NASA report SP-8013, while that for titanium was estimated by using a correlation based on material properties. Once the failure rate of a single heat pipe is known, the heat exchanger and wiffletree or disconnect failure rates are added and the radiator system reliability is calculated using the binomial distribution. In calculating the failure rate, the exposed area per heat pipe is assumed to be the product of the heat pipe diameter and length. For monocoque construction aluminum heat pipe panels, the effective thickness is assumed to be the sum of the heat pipe wall thickness and the fin face sheet thickness. The effective thickness of the wing construction titanium and stainless steel heat pipe panels is assumed to be just the heat pipe wall thickness, which is conservative.

Different scenarios can be used to determine the number of maintenance sessions which would be required, depending on what the requirement is for the module capacity. A primary advantage of the heat pipe radiator system over a pumped fluid loop radiator system is that the heat pipe radiator system will normally degrade in small increments instead of failing completely. When the number of radiator panel failures exceeds the number of redundant panels, the first effect is that the peak power capacity of the module is decreased below the design point. As additional radiator panel failures occur, the peak power capacity is further decreased until the steady state power requirement cannot be met as well. In a power system composed of many power modules, the different modules will normally have varying numbers of failed panels, resulting in varying capacities. A total system power system requirement could then be met with one or more modules having less than the individual module required capacity.

Radiator panel replacement must be performed when the module is shut down. Since modules will likely need to be shut down much more often for maintenance on other solar dynamic power module subsystems (e.g., engine, generator, receiver or concentrator) than for radiator maintenance, one scenario for radiator maintenance is to replace all of the failed panels in a module whenever the module is shut down for maintenance on another subsystem, whether or not the number of failed panels is greater than or equal to the number of redundant panels. With this scenario, it is very unlikely that the total power system capacity will ever be reduced to less than the design capacity because of radiator failures. Alternately, the radiator maintenance could be done only when the number of failed panels is equal to the number of redundant panels. In this study, the second scenario was assumed.

It has been assumed for this study that all maintenance will be performed using IVA. While this may not prove feasible for the integral radiator/heat exchanger concept, the average number of maintenance sessions required over a 30-year life is very low and use of EVA instead of IVA would have a very small effect on the life cycle cost.

3.3.1 ORC System

Table 3-8 shows the failure rate per panel and radiator panel weights (excluding disconnect) as a function of heat pipe wall thickness for dual-slot and monogroove heat pipes. Since the radiator system reliability and maintenance requirements are determined by the amount of micrometeoroid shielding (pipe thickness) and the number of redundant panels, a parametric analysis was done to determine the effects. Table 3-9 shows the results of the analyses of the system reliability for the baseline ORC configuration. The reliability quantity shown is the probability that the number of failures in a 2-year period will not exceed the number of redundant radiator panels. Similar tables are shown in Appendix A for the alternate configurations. For Alternates D and E the reliability was calculated for varying numbers of heat pipe elements instead of radiator panels. For these configurations, the actual panel design would be selected after the number of redundant elements was determined in order to arrive at a design with an integral number of panels and heat pipes per panel. Table 3-10 shows the average of the number of maintenance sessions which will be required for the baseline radiator system configuration under the scenario described previously. Comparable tables for the alternate configurations are shown in Appendix A.

Table 3-8 ORC Radiator Panel Weight and Reliability

Heat Pipe Wall Thickness (in.)	Dual-Slot Heat Pipe Panel Wt (lb)	Monogroove Heat Pipe Panel Wt (lb)	Panel Failure Rate ($year^{-1}$)
0.070	67.3	109.6	0.0252
0.080	71.1	112.0	0.0159
0.090	74.9	114.4	0.0106
0.100	78.8	116.8	0.0074
0.110	82.8	119.2	0.0053
0.120	87.0	121.6	0.0039
0.130	91.2	124.0	0.0030
0.140	95.5	126.4	0.0023
0.150	99.9	128.8	0.0018

These results were then factored into the life cycle cost analysis to determine the optimum amount of micrometeoroid shielding and number of redundant panels to minimize the life cycle cost as described in Section 3.4.

Table 3-9 ORC Baseline Radiator System Reliability Trade

Required No. of Radiator Panels	=	33
Pipe Material	=	Aluminum
Pipe Diameter	=	0.75 in
Total Pipe Length	=	90.0 ft
Clamp Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	48.0 ft^2

	2-YEAR RELIABILITY No. of Redundant Panels				
Pipe Wall Thickness	**1**	**2**	**3**	**4**	**5**
0.070	0.45642	0.71762	0.87894	0.95579	0.98590
0.080	0.66264	0.87564	0.96310	0.99080	0.99801
0.090	0.79258	0.94399	0.98799	0.99785	0.99967
0.100	0.86851	0.97295	0.99562	0.99941	0.99993
0.110	0.91262	0.98577	0.99818	0.99981	0.99998
0.120	0.93884	$\boxed{0.99184}$	0.99915	0.99993	0.99999
0.130	0.95498	0.99491	0.99955	0.99997	1.00000
0.140	0.96528	0.99659	0.99974	0.99998	1.00000
0.150	0.97211	0.99756	0.99983	0.99999	1.00000

3.3.2 CBC System

Table 3-11 shows the failure rate per panel and radiator panel weight (excluding disconnect) as a function of heat pipe wall thickness for dual-slot titanium and stainless steel heat pipes. Tables 3-12 and 3-13 show the results of the analyses on the system reliability and the required number of maintenance sessions for the baseline ORC configuration. Similar tables for alternate configurations are shown in Appendix A.

These results have been factored into the life cycle cost analysis to determine the optimum heat pipe wall thickness and number of redundant panels to minimize the life cycle cost as described in Section 3.4.

Table 3-10 ORC Baseline Radiator System Maintenance Requirement Trade

Required No. of Radiator Panels	=	33
Pipe Material	=	Aluminum
Pipe Diameter	=	0.75 in
Total Pipe Length	=	90.0 ft
Clamp Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	48.0 ft^2

	AVERAGE NO. OF MAINTENANCE SESSIONS No. of Redundant Panels				
Pipe Wall Thickness	**1**	**2**	**3**	**4**	**5**
0.070	27.336	13.645	9.057	6.772	5.398
0.080	18.111	8.938	5.878	4.353	3.433
0.090	12.783	6.241	4.047	2.961	2.311
0.100	9.536	4.596	2.934	2.112	1.625
0.110	7.448	3.530	2.220	1.564	1.179
0.120	6.064	$\boxed{2.815}$	1.742	1.204	0.886
0.130	5.088	2.329	1.411	0.951	0.674
0.140	4.396	1.983	1.177	0.771	0.524
0.150	3.895	1.732	1.007	0.641	0.420

Table 3-11 CBC Radiator Panel Weight and Reliability

Heat Pipe Wall Thickness (in.)	Titanium Heat Pipe Panel Wt (lb)	Stainless Steel Heat Pipe Panel Wt (lb)	Panel Failure Rate (year)
0.030	30.1	38.1	0.0456
0.040	33.7	44.5	0.0169
0.050	37.3	51.0	0.0079
0.060	41.1	57.6	0.0042
0.070	44.9	64.4	0.0025
0.080	48.8	71.4	0.0016
0.090	52.8	78.6	0.0010
0.100	56.8	85.8	0.0007
0.110	61.0	93.3	0.0005

Table 3-12 CBC Baseline Radiator System Reliability Trade

Required No. of Radiator Panels	=	36
Pipe Material	=	Titanium
Pipe Diameter	=	0.80 in.
Total Pipe Length	=	50.0 ft
Waffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	28.0 ft^2

	2-YEAR RELIABILITY No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.030	0.13899	0.31760	0.52287	0.70446	0.83626
0.040	0.59842	0.83359	0.94420	0.98425	0.99614
0.050	0.83754	0.96226	0.99308	0.99895	0.99986
0.060	0.92394	0.98858	0.99866	0.99987	0.99999
0.070	0.95705	$\boxed{0.99529}$	0.99960	0.99997	1.00000
0.080	0.97152	0.99749	0.99983	0.99999	1.00000
0.090	0.97869	0.99839	0.99991	1.00000	1.00000
0.100	0.98262	0.99882	0.99994	1.00000	1.00000
0.110	0.98496	0.99905	0.99995	1.00000	1.00000

Table 3-13 CBC Baseline Radiator System Maintenance Requirement Trade

Required No. of Radiator Panels	=	36
Pipe Material	=	Titanium
Pipe Diameter	=	0.80 in.
Total Pipe Length	=	50.0 ft
Wiffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	28.0 ft^2

	AVERAGE NO. OF MAINTENANCE SESSIONS No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.030	51.551	24.873	17.325	13.050	10.490
0.040	20.805	10.289	6.785	5.041	3.983
0.050	10.884	5.271	3.389	2.464	1.901
0.060	6.867	3.220	2.014	1.405	1.050
0.070	4.947	$\boxed{2.258}$	1.359	0.910	0.636
0.080	3.940	1.752	1.016	0.651	0.430
0.090	3.367	1.463	0.822	0.504	0.319
0.100	3.019	1.288	0.704	0.415	0.252
0.110	2.796	1.175	0.628	0.357	0.209

3.4 LIFE CYCLE COST

The life cycle cost analysis was undertaken to evaluate and optimize heat rejection system designs with respect to the life cycle cost. The following factors were considered in the analysis:

- Development Cost
- Initial Manufacturing Cost
- Launch Cost
- Assembly Cost
- Drag Penalty Cost
- Replacement Manufacturing Cost
- Maintenance Cost
- Total Life Cycle Cost.

A large number of assumptions must be made in estimating these costs and, to a large extent, the optimized design depends on the assumptions made. Each of the factors and the assumptions made in determining the cost is discussed in detail below.

- Development Cost - Some of the factors which affect the full-scale development cost are the degree of development accomplished under this contract, the amount of commonality with other space station systems such as the central radiator system, and the full-scale test requirements. It is assumed that a single full-size radiator panel and heat exchanger will have been designed, built and ground tested in a vacuum chamber under the NASA Advanced Development Program which is not included as part of the development cost. No commonality with any other radiator panels is assumed, but it is assumed that the heat exchanger and wiffletree will be very similar to that used for the central radiator system and that development costs will be reduced accordingly. At some time in the development program, 0-g testing will be required. A full-scale test article similar to SHARE (Space Station Heat Pipe Advanced Radiator Experiment), which will be performed for the central radiator system should be performed for the Solar dynamic radiator system. The cost of building such a flight-test article is included in the development cost as well as several additional ground test radiator panels, heat exchangers, and wiffletree clamp mechanisms (if required). It should be realized that the development cost estimates shown are very preliminary at this time, but in any event, the development cost is only a small portion of the total life cycle cost
- Manufacturing Cost - Preliminary manufacturing cost estimates are typically based on the estimated weight and include a complexity factor to account for the difficulty of manufacturing. In optimizing a design with respect to life cycle cost, this can lead to significant errors because in most cases, the actual manufacturing cost is not related to weight. Examples of this are in optimizing the fin and heat pipe wall thickness. In each case, increasing the thickness increases the weight, but does not increase the manufacturing complexity and may actually reduce it. Another example is in evaluating different heat pipe materials, where lighter materials may cost more and be more difficult to manufacture than heavier materials. In this study, cost estimates are based on the actual materials and manufacturing process which would be used. The total number of items which are to be produced greatly

affects manufacturing cost since significant cost-savings can be realized by production in large quantities. For this study it was assumed that radiator systems for twelve power modules would be produced

- Launch Cost - The launch cost calculation is typically based on the system weight only, and usually represents a large portion of the total life cycle cost. The cost factor specified by NASA for this study was \$1587/kg (\$3500/lb). Using this method of cost calculation will usually result in selection of a near-minimum weight system, which could actually be a less than optimal design under many conditions, such as when the launch capacity is limited by volume rather than weight. If the launch capacity is volume limited, then a design which minimizes the launch volume rather than weight is probably a better design. Determining the total power module weight and whether the launch is weight limited or volume limited is beyond the scope of this study, however. For this study, the launch cost was based on weight, but additional analyses were made to determine the life cycle costs excluding the launch cost. This shows the variation in direct costs (manufacturing, assembly and maintenance, and drag penalty) which will result from variations in heat pipe wall thickness and the number of redundant radiator panels. The results show primarily how the difference in wall thickness and redundancy changes the maintenance costs
- Assembly Cost - The cost of assembly is calculated based on the amount of RMS and EVA time required and factors for the cost per hour of each. The cost of EVA time specified by NASA was \$118,200/hr for a two-person EVA crew plus an IVA operator, while the cost for RMS time was specified as \$15,200/hr for an IVA operator. Estimates of the amount of time required for assembly are very preliminary at this time and are based primarily on studies performed under Work Package 2 Phase B studies. The assembly costs are a very small portion of the life cycle cost, however, and are primarily useful in evaluating the critical resource usage required
- Drag Penalty - A drag penalty associated with propellant resupply to support station orbit maintenance is calculated based on the propellant cost due to the aerodynamic drag of the components and the tankage associated with launch and storage of propellant. The drag penalty used was \$5275/m^2-yr (\$490/ft^2-yr), and the tankage factor multiplier was 1.2. The area used to determine the drag cost is based on the effective drag area which is projected perpendicular to the station velocity. For the solar dynamic power

modules, the total radiator system area is multiplied by 0.23 to get the effective average drag area

- Replacement Manufacturing Cost - The manufacturing cost of replacement units is calculated in the same manner as for the initial units
- Maintenance Cost - The maintenance cost is calculated in the same manner as the assembly. Although the radiator system is designed to be maintainable with by either RMS or EVA, the use of RMS has been assumed, to be consistent with the method of assembly
- Total Life Cycle Cost - For each of the system designs to be evaluated, the total life cycle cost of a single power module heat rejection system over a 30-year life was determined parametrically as a function of the number of redundant radiator panels and the heat pipe wall thickness (for micrometeoroid shielding). An optimum design for each configuration was then selected based on these results and on the previous reliability and maintenance requirements analysis. This results in both a low life cycle cost and low critical resource usage requirements.

The total life cycle cost for all of the solar dynamic power modules is calculated for a 30-year station life. The projected launch sequence for the modules is shown in Table 3-14. A total of 12 modules will be placed in operation over a period of 8 years, so that at the end of 30 years, each pair of modules will have a different life. The cost of maintaining each module is determined for that specific life and the total costs for all 12 modules are added to give the total system life cycle cost. In each analysis, RMS assembly and maintenance is assumed.

Table 3-14 Solar Dynamic Power Module Launch Timetable

Launch Time (yr)	No. of Modules	Life (yr)
IOC	2	30
IOC + 3	2	27
IOC + 5	2	25
IOC + 6	2	24
IOC + 7	2	23
IOC + 8	2	22

3.4.1 ORC System

Table 3-15 shows the results of the parametric life cycle cost estimates of the baseline radiator system for a single power module with respect to heat pipe wall thickness and the number of redundant radiator panels. Similar tables for the alternate configurations are shown in Appendix A. These estimates do not include the development cost (which is assumed to be independent of these parameters) since the development cost is applied only once for all twelve power modules. Table 3-16 shows the results for the baseline configuration where the launch cost is not included. In each table, the cost of the optimum design is circled. The optimum designs generally have a life cycle cost very near the minimum, but may not be the actual minimum because limiting the number of maintenance sessions which would be required was also included in the selection procedure. For example, in Table 3-15 which shows the results for the baseline ORC configuration, the minimum life cycle

Table 3-15 ORC Baseline Radiator System Life Cycle Cost Trade

Required No. of Radiator Panels	=	33
Pipe Material	=	Aluminum
Pipe Diameter	=	0.75 in.
Total Pipe Length	=	90.0 ft
Clamp Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	48.0 ft^2

	LIFE CYCLE COST (MILLIONS OF DOLLARS) No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.070	34.728	35.187	35.802	36.475	37.159
0.080	32.076	32.576	33.193	33.848	34.507
0.090	30.685	31.228	31.844	32.496	33.158
0.100	29.997	30.572	31.196	31.849	32.518
0.110	29.714	30.311	30.954	31.611	32.285
0.120	29.699	30.312	30.977	31.651	32.335
0.130	29.827	30.469	31.149	31.836	32.527
0.140	30.069	30.737	31.436	32.138	32.839
0.150	30.392	31.085	31.803	32.525	33.247

cost per module is $29.7 million per module, while the life cycle cost for the selected design is $30.3 million per module. This is because the 2-year reliability is increased from 0.94 to 0.99 in going to the selected design, and the maintenance requirement is reduced from 6.1 to 2.8 sessions over a 30-year life. Table 3-16 shows that neglecting the launch cost results in the selection of a much different design than when it is included. When the launch cost is not included, the best design is heavier, more reliable, and has lower maintenance requirements. Thus it is very important that the launch of the entire power module be considered and weight limits be established in order to select a true optimum design.

Table 3-16 ORC Baseline Radiator System Cost Trade Excluding Launch Cost

Required No. of Radiator Panels	=	33
Pipe Material	=	Aluminum
Pipe Diameter	=	0.75 in.
Total Pipe Length	=	90.0 ft
Clamp Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	48.0 ft^2

	LIFE CYCLE COST (MILLIONS OF DOLLARS) No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.070	15.783	15.876	16.143	16.459	16.790
0.080	14.640	14.805	15.090	15.408	15.735
0.090	13.979	14.191	14.486	14.806	15.136
0.100	13.577	13.818	14.120	14.442	14.773
0.110	13.318	13.578	13.888	14.211	14.542
0.120	13.147	13.418	13.734	14.061	14.393
0.130	13.026	13.309	13.630	13.958	14.290
0.140	12.941	13.233	13.557	13.887	14.220
0.150	12.879	13.178	13.505	13.838	14.173

Table 3-17 shows the breakdown of the life cycle costs per module for a 30-year life (excluding development cost) for the baseline configuration system design. Similar tables for the alternate configuration designs are shown in Appendix A. Table 3-18 shows the total life cycle cost (including estimated development cost) for all twelve power modules over the 30 year life for the baseline system. Similar tables for the alternate configurations are shown in Appendix A. Table 3-19 is a summary of the life cycle costs per module for a 30-year life lifetime (excluding development cost) and the totals for all twelve power modules over a 30-year life for the baseline and alternate configurations.

Over the life of the system, the development, assembly and maintenance costs are all relatively small compared to the total life cycle cost. The minimum cost configuration is the deployable system with the dual-slot heat pipe, while the highest cost configuration is the wiffletree and monogroove heat pipe configuration. The highest cost design has the least technical risk associated with it, however, since it uses the same technology which has been baselined by NASA for the Space Station central radiator system.

3.4.2 CBC System

Table 3-20 shows the results of the parametric life cycle cost estimates (per module) of the baseline CBC radiator system configuration with respect to heat pipe wall thickness and the number of redundant radiator panels. Similar data for the alternate configurations are included in Appendix A. Table 3-21 shows the results for the baseline configuration where the launch cost is not included. In each table, the cost of the optimum design is circled. The optimum designs generally have a life cycle cost very near the minimum, but may not be the actual minimum because limiting the number of maintenance sessions which would be required was also included in the selection procedure. For example, in Table 3-20 which shows the results for the baseline CBC configuration, the minimum life cycle cost per module is $22.8 million per module, while the life cycle cost for the selected design is $23.2 million per module. This is because the 2-year reliability is increased from 0.96 to 0.995 in going to the selected design, and the maintenance requirement is reduced from 4.9 to 2.3 sessions over a 30-year life. Table 3-21 shows that neglecting the launch cost results in the selection of a much different design than when it is included. When the launch is not included, the best design is heavier, more reliable, and has lower maintenance requirements. Thus it is very important that the launch of the entire power module be considered and weight limits be established in order to select a true optimum design.

Table 3-17 ORC Baseline Radiator System Life Cycle Cost Breakdown

Required No. of Radiator Panels	=	33
Number of Redundant Panels	=	2
Pipe Material	=	Aluminum
Pipe Diameter	=	0.750 in.
Wall Thickness	=	0.120 in.
Total Pipe Length	=	90.0 ft
Panel Failure Rate	=	0.0039/yr
Wiffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Total Failure Rate	=	0.0059/yr
Design Life	=	30.0 yr
Area per Panel	=	48.00 ft^2
2-Year System Reliability	=	0.9918
Avg. No. of Maintenance Sessions	=	2.815

Life Cycle Cost Breakdown **Initial Manufacturing**				
Radiator Panels	35 at	$100000.0 each	3.500	
Wiffletrees	35 at	$ 20000.0 each	0.700	
Heat Exchangers	35 at	$ 50000.0 each	1.750	
				5.950
Initial Launch				
Radiator Panels	35 at	87.0 lb	3045.0 lb	
Wiffletrees	35 at	30.0 lb	1050.0 lb	
Heat Exchangers	35 at	10.0 lb	350.0 lb	
				15.557
Initial Assembly (IVA)				
IVA Usage of 10.6 hr				0.162
Replacement Manufacturing				
Radiator Panels	3.7 at	$100000.0 each	0.374	
Wiffletrees	1.9 at	$ 20000.0 each	0.038	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.412
Replacement Launch				
Radiator Panels	3.7 at	87.0 lb	325.1 lb	
Wiffletrees	1.9 at	30.0 lb	56.8 lb	
Heat Exchangers	0.0 at	10.0 lb	0.0 lb	
				1.337
Maintenance (IVA)				
IVA Usage of 1.6 hr				0.078
Reboost (Drag)				
Projected Area of 1680.0 ft^2				6.816
Total Cost per Module				30.312

Table 3-18 ORC Baseline Radiator System Total Life Cycle Cost

Module Life	No. of Modules	Cost per Module ($million)	Total ($million)
30	2	30.31	60.62
27	2	29.43	58.86
25	2	28.84	57.69
24	2	28.55	57.10
23	2	28.26	56.51
22	2	27.96	55.92
Sub-total	12		346.71
Development			20.00
Total			366.71

Table 3-19 Life Cycle Cost Summary for ORC Radiator Design Options

Configuration	Single Module Life Cycle Cost ($million)	Total System Life Cycle Cost ($million)
Baseline	30.3	366.7
Alternate A	35.0	422.1
Alternate B	28.9	345.9
Alternate C	33.9	403.7
Alternate D	25.9	315.5
Alternate E	30.1	365.5

Table 3-20 CBC Baseline Radiator System Life Cycle Cost Trade

Required No. of Radiator Panels	=	36
Pipe Material	=	Titanium
Pipe Diameter	=	0.80 in.
Total Pipe Length	=	50.0 ft
Wiffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 years
Area per Panel	=	28.0 ft^2

	LIFE CYCLE COST (MILLIONS OF DOLLARS) No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.030	31.396	31.462	31.873	32.368	32.902
0.040	25.033	25.322	25.752	26.222	26.693
0.050	23.212	23.586	24.027	24.501	24.974
0.060	22.738	23.148	23.612	24.084	24.571
0.070	22.758	$\boxed{23.202}$	23.681	24.168	24.655
0.080	23.003	23.475	23.971	24.476	24.982
0.090	23.366	23.860	24.375	24.898	25.430
0.100	23.795	24.309	24.843	25.384	25.938
0.110	24.274	24.808	25.360	25.919	26.493

Table 3-21 CBC Baseline Radiator System Life Cycle Cost Trade Excluding Launch Cost

Required No. of Radiator Panels	=	36
Pipe Material	=	Titanium
Pipe Diameter	=	0.80 in.
Total Pipe Length	=	50.0 ft
Wiffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Design Life	=	30.0 yr
Area per Panel	=	28.0 ft^2

	LIFE CYCLE COST (MILLIONS OF DOLLARS) No. of Redundant Panels				
Pipe Wall Thickness	1	2	3	4	5
0.030	16.872	16.672	16.814	17.040	17.302
0.040	13.063	13.121	13.318	13.553	13.796
0.050	11.834	11.981	12.198	12.440	12.686
0.060	11.336	11.518	11.751	11.993	12.245
0.070	11.098	11.303	11.543	11.792	12.043
0.080	10.974	11.192	11.439	11.693	11.949
0.090	10.903	11.130	11.381	11.640	11.902
0.100	10.860	11.093	11.348	11.609	11.876
0.110	10.832	$\boxed{11.070}$	11.327	11.591	11.860

Table 3-22 shows the breakdown of the life cycle cost per module for a 30-year life (excluding development cost) for the selected design of the baseline configuration. Similar tables for the alternate configurations are shown in Appendix A. Table 3-23 shows the total life cycle cost over a 30-year life for the baseline system. Similar tables are shown in Appendix A for the alternate configurations. Table 3-24 is a summary of the life cycle costs per module for a 30-year lifetime (excluding development cost) and the totals for all twelve power modules over a 30-year life for the baseline and alternate configurations.

Over the life of the system, the development, assembly and maintenance costs are relatively small compared to the total life cycle cost. The minimum cost configuration is the deployable system, while the highest cost configuration is the wiffletree and stainless steel heat pipe configuration. However, the highest cost design has the least technical risk associated with it.

Table 3-22 CBC Baseline Radiator System Life Cycle Cost Breakdown

Required No. of Radiator Panels	=	36
Number of Redundant Panels	=	2
Pipe Material	=	Titanium
Pipe Diameter	=	0.800 in.
Wall Thickness	=	0.070 in.
Total Pipe Length	=	50.0 ft
Panel Failure Rate	=	0.0025/yr
Wiffletree Failure Rate	=	0.0020/yr
Heat Exchanger Failure Rate	=	0.0000/yr
Total Failure Rate	=	0.0045/yr
Design Life	=	30.0 yr
Area per Panel	=	28.00 ft^2
2-Year System Reliability	=	0.9953
Avg No. of Maintenance Sessions	=	2.258

Life Cycle Cost Breakdown **Initial Manufacturing**				
Radiator Panels	38 at	$100000.0 each	3.800	
Wiffletrees	38 at	$ 20000.0 each	0.760	
Heat Exchangers	38 at	$ 50000.0 each	1.900	
				6.460
Initial Launch				
Radiator Panels	38 at	44.9 lb	1707.0 lb	
Wiffletrees	38 at	30.0 lb	1140.0 lb	
Heat Exchangers	38 at	10.0 lb	380.0 lb	
				11.294
Initial Assembly (IVA)				
IVA Usage of 11.4 hr				0.174
Replacement Manufacturing				
Radiator Panels	2.5 at	$100000.0 each	0.250	
Wiffletrees	2.0 at	$ 20000.0 each	0.040	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.290
Replacement Launch				
Radiator Panels	2.5 at	44.9 lb	112.1 lb	
Wiffletrees	2.0 at	30.0 lb	60.6 lb	
Heat Exchangers	0.0 at	10.0 lb	0.0 lb	
				0.604
Maintenance (IVA)				
IVA Usage of 1.3 hr				0.063
Reboost (Drag)				
Projected Area of 1064.0 ft^2				4.317
Total Cost per Module				23.202

Table 3-23 CBC Baseline Radiator System Total Life Cycle Cost

Module Life	No. of Modules	Cost per Module ($million)	Total ($million)
30	2	23.20	46.40
27	2	22.66	45.33
25	2	22.31	44.61
24	2	22.13	44.25
23	2	21.95	43.90
22	2	21.77	43.54
Sub-total	12		268.03
Development			20.00
Total			288.03

Table 3-24 Life Cycle Cost Summary for CBC Radiator Design Options

Configuration	Single Module Life Cycle Cost ($million)	Total System Life Cycle Cost ($million)
Baseline	23.2	288.0
Alternate A	26.0	320.9
Alternate B	22.1	281.5
Alternate C	22.0	271.0
Alternate D	19.6	246.6

4 - SUMMARY OF RESULTS

4.1 CRITICAL TECHNOLOGY ASSESSMENT

4.1.1 Background

The design concepts developed under this Task Order include several technologies that are beyond the current state of the art. These technologies offer potential advantages in performance, weight and cost over technologies already developed or currently undergoing development. However, in order to maintain an acceptable level of technological risk, consideration has been limited to concepts that require technologies that have a good chance of being developed in an acceptable time frame at acceptable cost. Considerable use has also been made of concepts that take advantage of commonality with similar technologies being developed under other NASA programs.

The heat rejection system design in this study is based on a similar heat rejection system currently under development for the Space Station central radiator system. Grumman has been conducting work on a heat pipe radiator system for the central radiator since 1979 under contract to NASA-Johnson Space center. As part of that program Grumman has developed the high capacity monogroove heat pipe radiator, which has been successfully demonstrated at full scale in thermal vacuum chamber ground tests and on a reduced scale in a flight test on the Shuttle Orbiter. Full scale flight tests on the Shuttle Orbiter are scheduled. For the solar dynamic heat rejection system, the dual-slot heat pipe is proposed. This is a derivative of the monogroove heat pipe, and its development can take advantage of the experience gained in the development of the monogroove heat pipe.

In addition to considerations of thermal performance, advantage can be taken of the assembly and maintenance approaches developed for the central radiator system. Grumman has demonstrated (in ground tests) the feasibility of assembly and radiator panel replacement using either EVA astronauts or a remote manipulator system guided by an IVA astronaut. Flight tests on the Shuttle Orbiter are scheduled. The hardware required to implement the assembly and maintenance operations for the solar dynamic radiator system is very similar to that required for the central radiator system, and advantage can thereby be taken of the commonality between the two systems.

4.1.2 Radiator Technologies for the Organic Rankine Cycle

The dual-slot heat pipe has been shown to offer a substantial weight advantage over the monogroove heat pipe and has therefore been recommended for development for the ORC solar dynamic heat rejection system. Such a development program is being conducted by Grumman under a separate Task Order as part of this contract. This advanced development program currently includes analysis, design, fabrication, ground testing and evaluation of ammonia-aluminum dual-slot heat pipes. The monogroove heat pipe is considered to be a viable backup to the dual-slot heat pipe for the ORC application.

The other components of the heat rejection system include the heat exchanger (toluene condenser) and the disconnect devices. Since the thermal bus condensing fluid in the central radiator system is ammonia, some modifications in the design details for the ORC condenser may be required to accommodate the difference in fluid. However, it does not appear to be necessary to initiate a development program for the condenser until further progress has been made in evaluating the central radiator system condensing heat exchanger. The wiffletree clamping mechanism and the heat pipe disconnect currently being evaluated under the central radiator system advanced development program can also be used for the ORC heat rejection system with minor modifications. There is no ongoing program to provide such modifications at this time.

Another area requiring development work is the radiator surface coating. Work is currently being conducted by Acurex Corp. under contract to NASA-JSC to develop a long-life radiator coating for the Space Station central radiator system. The solar dynamic radiator system will require a similar long-life coating, but since it will operate at higher temperatures, the design criteria should be modified. It is recommended that the NASA-JSC long-life radiator coating program be extended to address the requirements for the ORC solar dynamic radiator system.

4.1.3 Radiator Technologies for the Closed Brayton Cycle

The dual-slot heat pipe has been determined to be the best heat pipe configuration for use with methanol as the heat pipe fluid. Either titanium or stainless steel would be used for its construction. Therefore it has been recommended that the dual-slot heat pipe development program being conducted by Grumman (see Section 1.2 of Part IV.) include analysis, design, fabrication, ground testing and evaluation of methanol/stainless steel dual-slot heat pipes as well as ammonia/aluminum heat pipes. This recommendation is now being carried out under a separate Task Order as part of this contract. This task order also includes an evaluation of techniques for bonding aluminum fins to stainless steel heat pipes. Because of the lower weight achievable with titanium, it is also recommended that investigations be conducted into the compatibility between methanol and titanium as well as techniques for bonding aluminum fins to titanium heat pipes.

The other components of the heat rejection system include the heat exchanger and the disconnect devices. The heat exchanger for the CBC transfers heat between the intermediate liquid loop (FC-75) and the heat pipe evaporator. Since this is a conventional finned-passage heat exchanger on the FC-75 side, a development program is not required for this component. The wiffletree clamping mechanism and the heat pipe disconnect currently being evaluated under the central radiator system advanced development program can also be used for the CBC heat rejection system with minor modifications. There is no ongoing program to provide such modifications at this time.

As indicated previously for the ORC technology requirements, the solar dynamic radiator system for the CBC will require a long-life radiator coating. It is therefore also recommended that the NASA-JSC long-life radiator coating program be extended to address the requirements for the CBC solar dynamic radiator system.

4.2 SYSTEM DESIGN & ANALYSIS

4.2.1 System Configuration Options

Table 4-1 defines the baseline and alternate radiator system configurations included in the design optimization and life cycle cost analysis. This table also displays the calculated life cycle cost per module (excluding development costs) over a 30-year life for each configuration. The options labelled "baseline" are taken as

Table 4-1 Summary of ORC and CBC Radiator System Configurations and Life Cycle Cost Estimates

Organic Rankine Cycle						
Configuration	**Baseline**	**Alt A**	**Alt B**	**Alt C**	**Alt D**	**Alt E**
Heat Pipe Type						
• Design	Dual Slot	Monogroove	Dual Slot	Monogroove	Dual Slot	Monogroove
• Fluid/Containment	Ammonia/Al	Ammonia/Al	Ammonia/Al	Ammonia/Al	Ammonia/Al	Ammonia/Al
Radiator Construction						
• Interface Type	Wiffletree	Wiffletree	Heat Pipe Disconnect	Heat Pipe Disconnect	Brazed	Brazed
• Assembly	Erectable	Erectable	Erectable	Erectable	Deployable	Deployable
Life Cycle Cost* Estimate (10^6\$)	30.3	35.0	28.9	33.9	25.9	30.1

Closed Brayton Cycle						
Configuration	**Baseline**	**Alt A**	**Alt B**	**Alt C**	**Alt D**	
Heat Pipe Type						
• Design	Dual Slot	Dual Slot	Dual Slot	Dual Slot	Dual Slot	
• Fluid/Containment	Methanol/Ti	Methanol/SS	Hybrid Meth/Ti-Ammonia/Al	Methanol/Ti	Methanol/Ti	
Radiator Construction						
• Interface Type	Wiffletree	Wiffletree	Wiffletree	Heat Pipe Disconnect	Brazed	
• Assembly	Erectable	Erectable	Erectable	Erectable	Deployable	
Life Cycle Cost* Estimate (10^6\$)	23.2	26.0	22.1	22.0	19.6	

* Life Cycle Cost per module (excluding development cost) over 30-year life

points of reference and do not necessarily represent the optimum configurations. The various configurations were analysed to determine the optimum designs for each option. Of primary concern were the the overall heat rejection system reliability, maintenance requirements and life cycle costs. The components of the life cycle cost include

- Development cost
- Initial manufacturing cost
- Launch cost
- Assembly cost
- Drag penalty cost
- Replacement manufacturing cost
- Maintenance cost.

All of the candidate configurations utilize heat pipes of either the monogroove or dual-slot type. For the ORC system the heat pipes are ammonia/aluminum heat pipes. For the CBC system, the heat pipes are the dual-slot type and use methanol as the working fluid except for alternate B, which uses a hybrid system with methanol heat pipes for the higher temperature radiator panels and ammonia/aluminum heat pipes for the lower temperature panels. The heat pipes using methanol are constructed from stainless steel or titanium. The three types of radiator/heat exchanger interface methods considered were the wiffletree, heat pipe disconnect and brazed connections. The radiator systems for the configurations utilizing the wiffletree and heat pipe disconnect interfaces are space erectable, while the systems for the configurations with the brazed interface are deployable.

4.2.2 Reliability & Maintenance Analysis

The reliability and maintenance requirements are primarily dependent on the heat pipe wall thickness (barrier to micrometeoroid and space debris penetration) and the number of redundant panels. The life cycle costs are primarily dependent on system weight (launch costs) and radiator area (drag penalty costs) and are only secondarily affected by maintenance requirements since the maintenance-related costs are a small percentage of the total life cycle cost.

However, the optimum configurations were not established on the basis of life cycle cost alone. Because of the limitations on allocation of EVA, IVA and RMS resources for the solar dynamic subassemblies, a separate accounting was made of the required frequency of maintenance as a function of radiator design parameters. Trade offs were then made to determine the design parameters which result in a combination of low maintenance and near-minimum life cycle cost.

The reliability analysis was used only indirectly to determine maintenance frequency because the heat pipe radiator system does not have a direct impact on power system failure. Since redundant radiator panels are provided and each heat pipe failure (e.g., due to micrometeoroid penetration) results in a very small degradation in performance, it is unlikely that the module would ever have to be shut down due to heat rejection system failure. Damaged panels could be replaced when the modules are shut down for repairs to other subassemblies, thus postponing or eliminating the need to shut down just to repair the radiator.

4.3 LIFE CYCLE COST ANALYSIS

The life cycle cost estimates per module over a thirty-year life are summarized in Table 4.1 for the various configurations. Development costs are not included in this table since the development cost should be distributed over the full set of twelve modules.

4.3.1 Radiator Cost Comparisons for the Organic Rankine Cycle

Taking the baseline as a point of reference, Alternate A has a higher life cycle cost due primarily to greater launch weight because a heavier heat pipe design (monogroove) is utilized. However, this alternate has the lowest technical risk because of the substantial development work done on the monogroove heat pipe for the central radiator system. Alternates B and C have lower costs, respectively, than the Baseline and Alternate A primarily because of the lower weight of the heat pipe disconnect and elimination of the contact resistance of the wiffletree interface. However, ultimate selection of either the wiffletree or the heat pipe disconnect will depend on further evaluations which are being conducted under the central radiator system development program. Although the initial launch weights are higher for Alternates D and E than for Alternates B and C, respectively, the overall life cycle costs for Alternates D and E are lower because of lower replacement manufacturing and replacement launch costs. The modular radiator/heat exchanger configuration used in Alternates D and E allows for a high degree of redundancy and a low probability of requiring maintenance over a 30-year module life.

4.3.2 Radiator Cost Comparisons for the Closed Brayton Cycle

Taking the baseline as a point of reference, Alternate A has a higher life cycle cost due primarily to greater launch weight because of the use of stainless steel heat pipes in place of titanium. Alternate B is the hybrid system. The system weight for the hybrid system is nearly the same as the weight for the baseline system. However, the manufacturing cost is lower since fewer panels are required, resulting in a lower life cycle cost estimate for alternate B. Alternate C has a lower estimated life cycle cost than the Baseline primarily because of the lower weight of the heat pipe disconnect and elimination of the contact resistance of the wiffletree interface. Although the initial launch weight is higher for Alternate D than for the Baseline, the overall life cycle cost for Alternate D is lower because of lower replacement manufacturing and replacement launch costs. The modular radiator/heat exchanger configuration used in Alternate D allows for a high degree of redundancy and a low probability of requiring maintenance over a 30-year module life.

The final selection of the heat rejection system configuration to be recommended for either the ORC or CBC solar dynamic power systems is a function of the priorities established for the selection criteria. The emphasis in this study has been on life cycle costs and maintenance requirements. There is considerable flexibility in the radiator system design parameters, so that the designs can be adjusted to accommodate shifts in emphasis between initial costs, life cycle costs, reliability, maintainability and assembly time as well as a possible shift of dependency of launch cost on weight to dependency on launch packaging flexibility.

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APPENDIX A

APPENDIX

Tables

APPENDIX

Tables

Table A-1 ORC Radiator System Trade Summary — Baseline (Dual-Slot Heat Pipe and Wiffletree)

Required Number of Panels	33
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30

SYSTEM WEIGHT (KG)

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.070	1654.5	1703.2	1751.8	1800.5	1849.2
0.080	1713.1	1763.5	1813.9	1864.3	1914.6
0.090	1771.7	1823.8	1875.9	1928.0	1980.1
0.100	1831.8	1885.7	1939.6	1993.5	2047.3
0.110	1893.5	1949.2	2004.9	2060.6	2116.3
0.120	1958.3	2015.9	2073.5	2131.1	2188.7
0.130	2023.0	2082.5	2142.0	2201.5	2261.0
0.140	2089.3	2150.8	2212.2	2273.7	2335.1
0.150	2157.2	2220.6	2284.1	2347.5	2411.0

2-YEAR RELIABILITY

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.070	0.45642	0.71762	0.87894	0.95579	0.98590
0.080	0.66264	0.87564	0.96310	0.99080	0.99801
0.090	0.79258	0.94399	0.98799	0.99785	0.99967
0.100	0.86851	0.97295	0.99562	0.99941	0.99993
0.110	0.91262	0.98577	0.99818	0.99981	0.99998
0.120	0.93884	0.99184	0.99915	0.99993	0.99999
0.130	0.95498	0.99491	0.99955	0.99997	1.00000
0.140	0.96528	0.99659	0.99974	0.99998	1.00000
0.150	0.97211	0.99756	0.99983	0.99999	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.070	27.336	13.645	9.057	6.772	5.398
0.080	18.111	8.938	5.878	4.353	3.433
0.090	12.783	6.241	4.047	2.961	2.311
0.100	9.536	4.596	2.934	2.112	1.625
0.110	7.448	3.530	2.220	1.564	1.179
0.120	6.064	2.815	1.742	1.204	0.886
0.130	5.088	2.329	1.411	0.951	0.674
0.140	4.396	1.983	1.177	0.771	0.524
0.150	3.895	1.732	1.007	0.641	0.420

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.070	34.728	35.187	35.802	36.475	37.159
0.080	32.076	32.576	33.193	33.848	34.507
0.090	30.685	31.228	31.844	32.496	33.158
0.100	29.997	30.572	31.196	31.849	32.518
0.110	29.714	30.311	30.954	31.611	32.285
0.120	29.699	30.312	30.977	31.651	32.335
0.130	29.827	30.469	31.149	31.836	32.527
0.140	30.069	30.737	31.436	32.138	32.839
0.150	30.392	31.085	31.803	32.525	33.247

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Table A-2 ORC Radiator System Trade Summary – Alternate A (Monogroove Heat Pipe and Wiffletree)

Required Number of Panels	33
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	2306.8	2374.6	2442.4	2510.3	2578.1
0.080	2343.8	2412.7	2481.6	2550.6	2619.5
0.090	2380.8	2450.8	2520.8	2590.8	2660.9
0.100	2417.8	2488.9	2560.0	2631.1	2702.2
0.110	2454.8	2527.0	2599.2	2671.4	2743.6
0.120	2491.8	2565.1	2638.4	2711.7	2784.9
0.130	2528.8	2603.2	2677.6	2751.9	2826.3
0.140	2565.8	2641.3	2716.7	2792.2	2867.7
0.150	2602.8	2679.4	2755.9	2832.5	2909.0

2-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	0.45642	0.71762	0.87894	0.95579	0.98590
0.080	0.66264	0.87564	0.96310	0.99080	0.99801
0.090	0.79258	0.94399	0.98799	0.99785	0.99967
0.100	0.86851	0.97295	0.99562	0.99941	0.99993
0.110	0.91262	0.98577	0.99818	0.99981	0.99998
0.120	0.93884	0.99184	0.99915	0.99993	0.99999
0.130	0.95498	0.99491	0.99955	0.99997	1.00000
0.140	0.96528	0.99659	0.99974	0.99998	1.00000
0.150	0.97211	0.99756	0.99983	0.99999	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	27.336	13.645	9.057	6.772	5.398
0.080	18.111	8.938	5.878	4.353	3.433
0.090	12.783	6.241	4.047	2.961	2.311
0.100	9.536	4.596	2.934	2.112	1.625
0.110	7.448	3.530	2.220	1.564	1.179
0.120	6.064	2.815	1.742	1.204	0.886
0.130	5.088	2.329	1.411	0.951	0.674
0.140	4.396	1.983	1.177	0.771	0.524
0.150	3.895	1.732	1.007	0.641	0.420

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	43.511	44.112	44.859	45.668	46.486
0.080	39.247	39.859	40.589	41.359	42.129
0.090	36.873	37.518	38.233	38.989	39.756
0.100	35.517	36.189	36.905	37.654	38.422
0.110	34.735	35.424	36.157	36.904	37.672
0.120	34.304	35.003	35.756	36.518	37.293
0.130	34.081	34.808	35.574	36.346	37.121
0.140	34.002	34.753	35.534	36.319	37.101
0.150	34.020	34.793	35.591	36.392	37.192

Table A-3 ORC Radiator System Trade Summary – Alternate B (Dual-Slot Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	31
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	1310.5	1351.4	1392.4	1433.3	1474.3
0.080	1365.6	1408.3	1451.0	1493.7	1536.3
0.090	1420.8	1465.2	1509.6	1554.0	1598.4
0.100	1477.4	1523.5	1569.7	1615.9	1662.0
0.110	1535.4	1583.4	1631.4	1679.4	1727.3
0.120	1596.4	1646.3	1696.1	1746.0	1795.9
0.130	1657.3	1709.1	1760.9	1812.7	1864.5
0.140	1719.7	1773.5	1827.2	1881.0	1934.7
0.150	1783.6	1839.3	1895.1	1950.8	2006.5

2-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	0.43323	0.69521	0.86409	0.94821	0.98272
0.080	0.62072	0.84831	0.95086	0.98657	0.99681
0.090	0.74187	0.91993	0.98014	0.99587	0.99926
0.100	0.81587	0.95366	0.99074	0.99846	0.99978
0.110	0.86127	0.97049	0.99504	0.99931	0.99992
0.120	0.88990	0.97951	0.99700	0.99963	0.99996
0.130	0.90859	0.98470	0.99799	0.99978	0.99998
0.140	0.92122	0.98787	0.99853	0.99985	0.99999
0.150	0.93003	0.98991	0.99886	0.99989	0.99999

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	28.530	14.271	9.492	7.107	5.677
0.080	19.875	9.845	6.498	4.827	3.820
0.090	14.873	7.306	4.774	3.514	2.761
0.100	11.823	5.761	3.724	2.713	2.116
0.110	9.872	4.772	3.052	2.201	1.704
0.120	8.559	4.105	2.604	1.858	1.423
0.130	7.651	3.644	2.296	1.620	1.228
0.140	7.014	3.313	2.075	1.455	1.092
0.150	6.550	3.073	1.914	1.335	0.993

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	34.056	34.464	35.027	35.645	36.286
0.080	31.069	31.491	32.042	32.629	33.221
0.090	29.501	29.950	30.486	31.058	31.646
0.100	28.722	29.193	29.722	30.286	30.878
0.110	28.401	28.889	29.415	29.977	30.575
0.120	28.368	28.868	29.404	29.962	30.558
0.130	28.502	29.014	29.560	30.116	30.712
0.140	28.766	29.281	29.837	30.405	31.001
0.150	29.116	29.637	30.203	30.782	31.380

Table A-4 ORC Radiator System Trade Summary – Alternate C (Monogroove Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	31
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	1924.4	1984.5	2044.6	2104.8	2164.9
0.080	1959.2	2020.4	2081.6	2142.9	2204.1
0.090	1994.0	2056.3	2118.6	2181.0	2243.3
0.100	2028.8	2092.2	2155.6	2219.0	2282.4
0.110	2063.7	2128.2	2192.7	2257.1	2321.6
0.120	2098.5	2164.1	2229.7	2295.2	2360.8
0.130	2133.3	2200.0	2266.7	2333.3	2400.0
0.140	2168.2	2235.9	2303.7	2371.4	2439.2
0.150	2203.0	2271.8	2340.7	2409.5	2478.4

2-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	0.43323	0.69521	0.86409	0.94821	0.98272
0.080	0.62072	0.84831	0.95086	0.98657	0.99681
0.090	0.74187	0.91993	0.98014	0.99587	0.99926
0.100	0.81587	0.95366	0.99074	0.99846	0.99978
0.110	0.86127	0.97049	0.99504	0.99931	0.99992
0.120	0.88990	0.97951	0.99700	0.99963	0.99996
0.130	0.90859	0.98470	0.99799	0.99978	0.99998
0.140	0.92122	0.98787	0.99853	0.99985	0.99999
0.150	0.93003	0.98991	0.99886	0.99989	0.99999

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	28.530	14.271	9.492	7.107	5.677
0.080	19.875	9.845	6.498	4.827	3.820
0.090	14.873	7.306	4.774	3.514	2.761
0.100	11.823	5.761	3.724	2.713	2.116
0.110	9.872	4.772	3.052	2.201	1.704
0.120	8.559	4.105	2.604	1.858	1.423
0.130	7.651	3.644	2.296	1.620	1.228
0.140	7.014	3.313	2.075	1.455	1.092
0.150	6.550	3.073	1.914	1.335	0.993

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.070	43.018	43.576	44.277	45.036	45.819
0.080	38.495	39.034	39.699	40.403	41.108
0.090	35.981	36.533	37.167	37.839	38.531
0.100	34.550	35.114	35.730	36.384	37.073
0.110	33.735	34.309	34.913	35.558	36.247
0.120	33.280	33.859	34.468	35.100	35.780
0.130	33.054	33.639	34.254	34.878	35.549
0.140	32.985	33.566	34.187	34.819	35.485
0.150	33.016	33.596	34.223	34.862	35.524

Table A-5 ORC Radiator System Trade Summary — Alternate D (Dual-Slot Heat Pipe/Deployable System)

Nominal Required Heat Pipes	62
Pipe Material	Aluminum
Pipe Diameter (in.)	0.75
Nominal Condenser Length (ft.)	45.0
Area per Heat Pipe (sq.ft.)	24.0
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.070	1351.6	1380.6	1415.5	1452.7	1492.2
0.080	1406.9	1437.3	1473.9	1512.9	1554.4
0.090	1462.2	1494.0	1532.3	1573.1	1616.5
0.100	1518.9	1552.1	1592.3	1635.0	1680.4
0.110	1577.1	1611.8	1653.8	1698.4	1745.8
0.120	1638.2	1674.5	1718.3	1764.9	1814.5
0.130	1699.3	1737.2	1782.9	1831.5	1883.3
0.140	1761.8	1801.3	1849.0	1899.7	1953.6
0.150	1825.8	1867.0	1916.6	1969.4	2025.6

30-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.070	0.00000	0.00000	0.00001	0.00008	0.00051
0.080	0.00004	0.00072	0.00572	0.02594	0.07908
0.090	0.00271	0.02648	0.11191	0.28242	0.50320
0.100	0.03011	0.16681	0.42377	0.69055	0.86980
0.110	0.12213	0.42398	0.73650	0.91398	0.97897
0.120	0.28020	0.67187	0.90560	0.98152	0.99735
0.130	0.46111	0.83633	0.97056	0.99649	0.99970
0.140	0.62295	0.92449	0.99140	0.99936	0.99997
0.150	0.74761	0.96650	0.99754	0.99988	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.070	20.31	15.25	10.87	8.10	6.41
0.080	12.78	8.82	6.06	3.97	2.96
0.090	8.10	5.39	2.96	1.87	1.03
0.100	5.57	3.01	1.57	0.66	0.27
0.110	3.79	1.76	0.62	0.23	0.08
0.120	2.52	0.89	0.25	0.06	0.02
0.130	1.74	0.47	0.12	0.01	0.00
0.140	1.19	0.28	0.03	0.00	0.00
0.150	0.90	0.16	0.02	0.00	0.00

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.070	56.745	46.319	40.592	37.174	35.401
0.080	44.955	37.129	33.601	30.997	30.136
0.090	37.544	32.296	29.051	27.952	27.247
0.100	33.694	28.944	27.215	26.341	26.378
0.110	31.019	27.344	26.037	26.046	26.531
0.120	29.171	26.341	25.887	26.248	26.937
0.130	28.209	26.089	26.149	26.672	27.435
0.140	27.633	26.247	26.486	27.177	27.981
0.150	27.570	26.523	26.972	27.710	28.531

Table A-6 ORC Radiator System Trade Summary — Alternate E (Monogroove Heat Pipe/Deployable System)

Nominal Required Heat Pipes	62
Pipe Material	Aluminum
Pipe Diameter (in.)	0.75
Nominal Condenser Length (ft.)	45.0
Area per Heat Pipe (sq.ft.)	24.0
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Heat Pipes 2	4	6	8	10
0.070	1966.9	2011.7	2065.8	2123.2	2184.4
0.080	2001.8	2047.5	2102.6	2161.2	2223.6
0.090	2036.7	2083.3	2139.5	2199.3	2262.9
0.100	2071.6	2119.1	2176.4	2237.3	2302.2
0.110	2106.5	2154.9	2213.3	2275.4	2341.5
0.120	2141.5	2190.7	2250.2	2313.4	2380.7
0.130	2176.4	2226.5	2287.1	2351.5	2420.0
0.140	2211.3	2262.3	2324.0	2389.5	2459.3
0.150	2246.2	2298.1	2360.9	2427.6	2498.5

30-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Heat Pipes 2	4	6	8	10
0.070	0.00000	0.00000	0.00001	0.00008	0.00051
0.080	0.00004	0.00072	0.00572	0.02594	0.07908
0.090	0.00271	0.02648	0.11191	0.28242	0.50320
0.100	0.03011	0.16681	0.42377	0.69055	0.86980
0.110	0.12213	0.42398	0.73650	0.91398	0.97897
0.120	0.28020	0.67187	0.90560	0.98152	0.99735
0.130	0.46111	0.83633	0.97056	0.99649	0.99970
0.140	0.62295	0.92449	0.99140	0.99936	0.99997
0.150	0.74761	0.96650	0.99754	0.99988	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Heat Pipes 2	4	6	8	10
0.070	20.31	15.25	10.87	8.10	6.41
0.080	12.78	8.82	6.06	3.97	2.96
0.090	8.10	5.39	2.96	1.87	1.03
0.100	5.57	3.01	1.57	0.66	0.27
0.110	3.79	1.76	0.62	0.23	0.08
0.120	2.52	0.89	0.25	0.06	0.02
0.130	1.74	0.47	0.12	0.01	0.00
0.140	1.19	0.28	0.03	0.00	0.00
0.150	0.90	0.16	0.02	0.00	0.00

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Heat Pipes 2	4	6	8	10
0.070	73.510	59.412	51.647	46.983	44.527
0.080	56.853	46.433	41.704	38.194	36.987
0.090	46.446	39.551	35.266	33.775	32.792
0.100	40.915	34.770	32.500	31.321	31.310
0.110	37.029	32.341	30.641	30.601	31.158
0.120	34.265	30.707	30.098	30.498	31.305
0.130	32.684	30.054	30.084	30.681	31.568
0.140	31.609	29.908	30.157	30.950	31.875
0.150	31.172	29.901	30.399	31.238	32.172

Table A-7 ORC System Life Cycle Cost Breakdown — Baseline (Dual-Slot Heat Pipe and Wiffletree)

Required Number of Panels	33
Number of Redundant Panels	2
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Heat Pipe Failure Rate (per year)	0.0039
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30
2-year System Reliability	0.9918
Avg. No. of Maintenance Sessions	2.81

Life Cycle Cost Breakdown					
Initial Manufacturing					
Radiator Panels	35 at	$100000.0 each	3.500		
Wiffletrees	35 at	$ 20000.0 each	0.700		
Heat Exchangers	35 at	$ 50000.0 each	1.750		
					5.950
Initial Launch					
Radiator Panels	35 at	87.0 lbs	3045.0 lbs		
Wiffletrees	35 at	30.0 lbs	1050.0 lbs		
Heat Exchangers	35 at	10.0 lbs	350.0 lbs		
					15.557
Initial Assembly (IVA)					
IVA Usage of 10.6 hours					0.162
Replacement Manufacturing					
Radiator Panels	3.7 at	$100000.0 each	0.374		
Wiffletrees	1.9 at	$ 20000.0 each	0.038		
Heat Exchangers	0.0 at	$ 50000.0 each	0.000		
					0.412
Replacement Launch					
Radiator Panels	3.7 at	87.0 lbs	325.1 lbs		
Wiffletrees	1.9 at	30.0 lbs	56.8 lbs		
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs		
					1.337
Maintenance (IVA)					
IVA Usage of 1.6 hours					0.078
Reboost (Drag)					
Projected Area of 1680.0 square feet					6.816
Total Cost per Module					30.312

Table A-8 ORC System Life Cycle Cost Breakdown — Alternate A (Monogroove Heat Pipe andAuto)

Required Number of Panels	33
Number of Redundant Panels	2
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Heat Pipe Failure Rate (per year)	0.0039
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30
2-year System Reliability	0.9918
Avg. No. of Maintenance Sessions	2.81

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	35 at	$100000.0 each	3.500	
Wiffletrees	35 at	$ 20000.0 each	0.700	
Heat Exchangers	35 at	$ 50000.0 each	1.750	
				5.950
Initial Launch				
Radiator Panels	35 at	121.6 lbs	4256.0 lbs	
Wiffletrees	35 at	30.0 lbs	1050.0 lbs	
Heat Exchangers	35 at	10.0 lbs	350.0 lbs	
				19.796
Initial Assembly (IVA)				
IVA Usage of 10.6 hours				0.162
Replacement Manufacturing				
Radiator Panels	3.7 at	$100000.0 each	0.374	
Wiffletrees	1.9 at	$ 20000.0 each	0.038	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.412
Replacement Launch				
Radiator Panels	3.7 at	121.6 lbs	454.4 lbs	
Wiffletrees	1.9 at	30.0 lbs	56.8 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
				1.789
Maintenance (IVA)				
IVA Usage of 1.6 hours				0.078
Reboost (Drag)				
Projected Area of 1680.0 square feet				6.816
Total Cost per Module				35.003

Table A-9 ORC System Life Cycle Cost Breakdown — Alternate B (Dual-Slot Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	31
Number of Redundant Panels	2
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Heat Pipe Failure Rate (per year)	0.0039
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30
2-year System Reliability	0.9795
Avg. No. of Maintenance Sessions	4.10

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	33 at	$100000.0 each	3.300	
Disconnects	33 at	$ 20000.0 each	0.660	
Heat Exchangers	33 at	$ 50000.0 each	1.650	
				5.610
Initial Launch				
Radiator Panels	33 at	87.0 lbs	2871.0 lbs	
Disconnects	33 at	13.0 lbs	429.0 lbs	
Heat Exchangers	33 at	10.0 lbs	330.0 lbs	
			3630.0 lbs	12.705
Initial Assembly (IVA)				
IVA Usage of 10.1 hours				0.154
Replacement Manufacturing				
Radiator Panels	3.6 at	$100000.0 each	0.821	
Disconnects	4.6 at	$ 20000.0 each	0.164	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.985
Replacement Launch				
Radiator Panels	3.6 at	87.0 lbs	714.3 lbs	
Disconnects	4.6 at	13.0 lbs	106.7 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
			821.1 lbs	2.874
Maintenance (IVA)				
IVA Usage of 2.3 hours				0.114
Reboost (Drag)				
Projected Area of 1584.0 square feet				6.427
Total Cost per Module				28.868

Table A-10 ORC System Life Cycle Cost Breakdown — Alternate C (Monogroove Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	31
Number of Redundant Panels	2
Heat Pipe Material	Aluminum
Heat Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	45.0
Area per Panel (sq.ft.)	48.0
Heat Pipe Failure Rate (per year)	0.0039
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30
2-year System Reliability	0.9795
Avg. No. of Maintenance Sessions	4.10

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	33 at	$100000.0 each	3.300	
Disconnects	33 at	$ 20000.0 each	0.660	
Heat Exchangers	33 at	$ 50000.0 each	1.650	
				5.610
Initial Launch				
Radiator Panels	33 at	121.6 lbs	4012.8 lbs	
Disconnects	33 at	13.0 lbs	429.0 lbs	
Heat Exchangers	33 at	10.0 lbs	330.0 lbs	
			4771.8 lbs	16.701
Initial Assembly (IVA)				
IVA Usage of 10.1 hours				0.154
Replacement Manufacturing				
Radiator Panels	3.6 at	$100000.0 each	0.821	
Disconnects	4.6 at	$ 20000.0 each	0.164	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.985
Replacement Launch				
Radiator Panels	3.6 at	121.6 lbs	998.4 lbs	
Disconnects	4.6 at	13.0 lbs	106.7 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
			1105.1 lbs	3.868
Maintenance (IVA)				
IVA Usage of 2.3 hours				0.114
Reboost (Drag)				
Projected Area of 1584.0 square feet				6.427
Total Cost per Module				33.859

Table A-11 ORC System Life Cycle Cost Breakdown — Alternate D (Dual-Slot Heat Pipe/Deployable System)

Number of Radiator Panels	9
Number of Heat Pipes/Panel	8
Number of Redundant Heat Pipes	6
Pipe Material	Aluminum
Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	42.3
Area per Heat Pipe (sq.ft.)	24.0
Heat Pipe Failure Rate (per year)	0.0019
Design Life (years)	30.0
30-Year System Reliability	0.9056
Avg. No. of Maintenance Sessions	0.25

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	9 at	$375757.6 each	3.382	
Heat Exchangers	9 at	$187878.8 each	1.691	
Deployment Mechanism	1 at	$500000.0 each	0.500	
				5.573
Initial Launch				
Radiator Panels	9 at	326.9 lbs	2942.2 lbs	
Heat Exchangers	9 at	37.6 lbs	338.2 lbs	
Deployment Mechanism	1 at	500.0 lbs	500.0 lbs	
			3780.4 lbs	13.231
Initial Assembly (IVA)				
IVA Usage of 1.5 hours				0.023
Replacement Manufacturing				
Radiator Panels	0.3 at	$375757.6 each	0.095	
Heat Exchangers	0.3 at	$187878.8 each	0.048	
				0.143
Replacement Launch				
Radiator Panels	0.3 at	326.9 lbs	82.9 lbs	
Heat Exchangers	0.3 at	37.6 lbs	9.5 lbs	
			92.4 lbs	0.324
Maintenance (IVA)				
IVA Usage of 0.5 hours				0.008
Reboost (Drag)				
Projected Area of 1623.3 square feet				6.586
Total Cost per Module				25.887

Table A-12 ORC System Life Cycle Cost Breakdown — Alternate E (Monogroove Heat Pipe/Deployable System)

Number of Radiator Panels	9
Number of Heat Pipes/Panel	8
Number of Redundant Heat Pipes	6
Pipe Material	Aluminum
Pipe Diameter (in.)	0.75
Wall Thickness (in.)	0.12
Condenser Length (ft.)	42.3
Area per Heat Pipe (sq.ft.)	24.0
Heat Pipe Failure Rate (per year)	0.0019
Design Life (years)	30.0
30-Year System Reliability	0.9056
Avg. No. of Maintenance Sessions	0.25

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	9 at	$375757.6 each	3.382	
Heat Exchangers	9 at	$187878.8 each	1.691	
Deployment Mechanism	1 at	$500000.0 each	0.500	
				5.573
Initial Launch				
Radiator Panels	9 at	456.9 lbs	4112.3 lbs	
Heat Exchangers	9 at	37.6 lbs	338.2 lbs	
Deployment Mechanism	1 at	500.0 lbs	500.0 lbs	
			4950.5 lbs	17.327
Initial Assembly (IVA)				
IVA Usage of 1.5 hours				0.023
Replacement Manufacturing				
Radiator Panels	0.3 at	$375757.6 each	0.095	
Heat Exchangers	0.3 at	$187878.8 each	0.048	
				0.143
Replacement Launch				
Radiator Panels	0.3 at	456.9 lbs	115.9 lbs	
Heat Exchangers	0.3 at	37.6 lbs	9.5 lbs	
			125.4 lbs	0.439
Maintenance (IVA)				
IVA Usage of 0.5 hours				0.008
Reboost (Drag)				
Projected Area of 1623.3 square feet				6.586
Total Cost per Module				30.098

Table A-13 CBC Radiator System Trade Summary – Baseline (Titanium Heat Pipe and Wiffletree)

Required Number of Panels	36
Heat Pipe Material	Titanium
Heat Pipe Diameter (in.)	0.80
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Panels: 1	2	3	4	5
0.030	1177.1	1208.9	1240.7	1272.6	1304.4
0.040	1236.7	1270.1	1303.5	1337.0	1370.4
0.050	1297.4	1332.5	1367.6	1402.6	1437.7
0.060	1360.7	1397.5	1434.2	1471.0	1507.8
0.070	1425.0	1463.5	1502.0	1540.5	1579.0
0.080	1490.4	1530.7	1571.0	1611.2	1651.5
0.090	1557.2	1599.3	1641.4	1683.4	1725.5
0.100	1625.1	1669.1	1713.0	1756.9	1800.8
0.110	1695.5	1741.3	1787.1	1832.9	1878.7

2-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Panels: 1	2	3	4	5
0.030	0.13899	0.31760	0.52287	0.70446	0.83626
0.040	0.59842	0.83359	0.94420	0.98425	0.99614
0.050	0.83754	0.96226	0.99308	0.99895	0.99986
0.060	0.92394	0.98858	0.99866	0.99987	0.99999
0.070	0.95705	0.99529	0.99960	0.99997	1.00000
0.080	0.97152	0.99749	0.99983	0.99999	1.00000
0.090	0.97869	0.99839	0.99991	1.00000	1.00000
0.100	0.98262	0.99882	0.99994	1.00000	1.00000
0.110	0.98496	0.99905	0.99995	1.00000	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Panels: 1	2	3	4	5
0.030	51.551	25.873	17.325	13.050	10.490
0.040	20.805	10.289	6.785	5.041	3.983
0.050	10.884	5.271	3.389	2.464	1.901
0.060	6.867	3.220	2.014	1.405	1.050
0.070	4.947	2.258	1.359	0.910	0.636
0.080	3.940	1.752	1.016	0.651	0.430
0.090	3.367	1.463	0.822	0.504	0.319
0.100	3.019	1.288	0.704	0.415	0.252
0.110	2.796	1.175	0.628	0.357	0.209

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Panels: 1	2	3	4	5
0.030	31.396	31.462	31.873	32.368	32.902
0.040	25.033	25.322	25.752	26.222	26.693
0.050	23.212	23.586	24.027	24.501	24.974
0.060	22.738	23.148	23.612	24.084	24.571
0.070	22.758	23.202	23.681	24.168	24.655
0.080	23.003	23.475	23.971	24.476	24.982
0.090	23.366	23.860	24.375	24.898	25.430
0.100	23.795	24.309	24.843	25.384	25.938
0.110	24.274	24.808	25.360	25.919	26.493

Table A-14 CBC Radiator System Trade Summary – Alternate A (Stainless Steel Heat Pipe and Wiffletree)

Required Number of Panels	36
Heat Pipe Material	Stainless Steel
Heat Pipe Diameter (in.)	0.80
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30

SYSTEM WEIGHT (KG)

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.030	1310.5	1345.9	1381.4	1416.8	1452.2
0.040	1417.9	1456.2	1494.6	1532.9	1571.2
0.050	1526.3	1567.6	1608.8	1650.1	1691.3
0.060	1638.2	1682.5	1726.8	1771.1	1815.3
0.070	1752.7	1800.0	1847.4	1894.8	1942.2
0.080	1869.3	1919.8	1970.3	2020.9	2071.4
0.090	1989.8	2043.6	2097.3	2151.1	2204.9
0.100	2111.3	2168.3	2225.4	2282.4	2339.5
0.110	2236.3	2296.7	2357.2	2417.6	2478.0

2-YEAR RELIABILITY

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.030	0.13899	0.31760	0.52287	0.70446	0.83626
0.040	0.59842	0.83359	0.94420	0.98425	0.99614
0.050	0.83754	0.96226	0.99308	0.99895	0.99986
0.060	0.92394	0.98858	0.99866	0.99987	0.99999
0.070	0.95705	0.99529	0.99960	0.99997	1.00000
0.080	0.97152	0.99749	0.99983	0.99999	1.00000
0.090	0.97869	0.99839	0.99991	1.00000	1.00000
0.100	0.98262	0.99882	0.99994	1.00000	1.00000
0.110	0.98496	0.99905	0.99995	1.00000	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.030	51.551	25.873	17.325	13.050	10.490
0.040	20.805	10.289	6.785	5.041	3.983
0.050	10.884	5.271	3.389	2.464	1.901
0.060	6.867	3.220	2.014	1.405	1.050
0.070	4.947	2.258	1.359	0.910	0.636
0.080	3.940	1.752	1.016	0.651	0.430
0.090	3.367	1.463	0.822	0.504	0.319
0.100	3.019	1.288	0.704	0.415	0.252
0.110	2.796	1.175	0.628	0.357	0.209

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Number of Redundant Panels

Pipe Wall Thickness	1	2	3	4	5
0.030	33.799	33.898	34.343	34.873	35.441
0.040	27.134	27.454	27.914	28.416	28.916
0.050	25.392	25.802	26.276	26.786	27.293
0.060	25.150	25.601	26.107	26.620	27.150
0.070	25.474	25.970	26.500	27.040	27.578
0.080	26.064	26.599	27.159	27.727	28.297
0.090	26.809	27.379	27.970	28.570	29.179
0.100	27.628	28.232	28.854	29.485	30.129
0.110	28.513	29.150	29.803	30.465	31.142

Table A-15 CBC Radiator System Trade Summary — Alternate B (Hybrid Heat Pipe System and Wiffletree)

	High Temp Panels	Low Temp Panels
Required Number of Panels	15	13
Heat Pipe Material	Titanium	Aluminum
Heat Pipe Diameter (in.)	0.80	0.75
Condenser Length (ft.)	25.0	45.0
Area per Panel (sq.ft.)	28.0	48.0
Wiffletree Failure Rate (per year)	0.0020	0.0020
Design Life (years)	30	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness (High/Low)	Number of Redundant Panels (High /Temp Panels/Low Temp Panels) 0/1	1/1	2/1	1/2	2/2
0.05/0.10	1255.5	1290.6	1325.6	1342.7	1377.8
0.07/0.10	1357.4	1395.9	1434.4	1451.6	1490.1
0.09/0.10	1464.3	1506.4	1548.5	1565.9	1608.0
0.05/0.12	1255.5	1290.6	1325.6	1342.7	1377.8
0.07/0.12	1357.4	1395.9	1434.4	1451.6	1490.1
0.09/0.12	1464.3	1506.4	1548.5	1565.9	1608.0
0.05/0.14	1255.5	1290.6	1325.6	1342.7	1377.8
0.07/0.14	1357.4	1395.9	1434.4	1451.6	1490.1
0.09/0.14	1464.3	1506.4	1548.5	1565.9	1608.0

2-YEAR RELIABILITY

Pipe Wall Thickness (High/Low)	Number of Redundant Panels (High /Temp Panels/Low Temp Panels) 0/1	1/1	2/1	1/2	2/2
0.05/0.10	0.516	0.734	0.926	0.963	0.991
0.07/0.10	0.644	0.761	0.968	0.969	0.993
0.09/0.10	0.689	0.770	0.976	0.966	0.996
0.05/0.12	0.580	0.813	0.952	0.982	0.995
0.07/0.12	0.727	0.828	0.979	0.984	0.999
0.09/0.12	0.761	0.843	0.985	0.987	0.998
0.05/0.14	0.621	0.847	0.961	0.986	0.997
0.07/0.14	0.758	0.871	0.981	0.991	0.999
0.09/0.14	0.798	0.881	0.988	0.992	0.999

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness (High/Low)	Number of Redundant Panels (High /Temp Panels/Low Temp Panels) 0/1	1/1	2/1	1/2	2/2
0.05/0.10	8.237	5.488	3.091	2.699	2.000
0.07/0.10	5.982	4.409	1.741	2.111	1.276
0.09/0.10	5.306	4.197	1.218	2.021	1.009
0.05/0.12	6.819	4.152	2.495	1.994	1.492
0.07/0.12	4.484	3.050	1.477	1.384	0.948
0.09/0.12	3.804	2.854	1.080	1.300	0.724
0.05/0.14	6.204	3.463	2.238	1.602	1.255
0.07/0.14	3.734	2.422	1.243	1.103	0.698
0.09/0.14	3.222	2.154	0.976	0.898	0.570

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness (High/Low)	Number of Redundant Panels (High /Temp Panels/Low Temp Panels) 0/1	1/1	2/1	1/2	2/2
0.05/0.10	21.594	21.894	22.230	22.657	23.103
0.07/0.10	21.490	21.857	21.892	22.582	22.854
0.09/0.10	21.740	22.204	22.059	22.939	23.067
0.05/0.12	21.501	21.843	22.306	22.601	23.085
0.07/0.12	21.382	21.792	22.143	22.516	22.988
0.09/0.12	21.633	22.138	22.379	22.874	23.243
0.05/0.14	21.695	22.017	22.575	22.765	23.313
0.07/0.14	21.540	21.982	22.420	22.757	23.204
0.09/0.14	21.828	22.301	22.727	23.037	23.546

Table A-16 CBC Radiator System Trade Summary — Alternate C (Titanium Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	35
Heat Pipe Material	Titanium
Heat Pipe Diameter (in.)	0.80
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.030	867.8	891.9	916.0	940.1	964.2
0.040	925.7	951.4	977.1	1002.9	1028.6
0.050	984.8	1012.2	1039.5	1066.9	1094.2
0.060	1046.4	1075.4	1104.5	1133.6	1162.6
0.070	1108.9	1139.7	1170.5	1201.3	1232.1
0.080	1172.6	1205.1	1237.7	1270.3	1302.9
0.090	1237.6	1271.9	1306.3	1340.7	1375.1
0.100	1303.7	1339.9	1376.1	1412.3	1448.5
0.110	1372.1	1410.2	1448.3	1486.4	1524.5

2-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.030	0.12679	0.29601	0.49672	0.68010	0.81766
0.040	0.53923	0.78911	0.92130	0.97517	0.99319
0.050	0.76797	0.93296	0.98460	0.99705	0.99951
0.060	0.86045	0.97034	0.99503	0.99931	0.99992
0.070	0.90047	0.98259	0.99761	0.99973	0.99997
0.080	0.91996	0.98762	0.99850	0.99985	0.99999
0.090	0.93049	0.99006	0.99889	0.99990	0.99999
0.100	0.93666	0.99140	0.99909	0.99992	0.99999
0.110	0.94050	0.99219	0.99920	0.99993	0.99999

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.030	53.243	26.746	17.916	13.509	10.868
0.040	23.408	11.621	7.686	5.726	4.540
0.050	13.786	6.742	4.385	3.220	2.514
0.060	9.887	4.769	3.050	2.202	1.692
0.070	8.040	3.829	2.419	1.715	1.298
0.080	7.074	3.330	2.086	1.462	1.095
0.090	6.521	3.044	1.896	1.318	0.979
0.100	6.179	2.873	1.780	1.229	0.905
0.110	5.959	2.763	1.705	1.172	0.858

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Panels 1	2	3	4	5
0.030	32.770	32.789	33.159	33.624	34.130
0.040	24.712	24.917	25.273	25.676	26.081
0.050	22.386	22.655	23.008	23.402	23.794
0.060	21.744	22.046	22.402	22.790	23.183
0.070	21.719	22.037	22.404	22.785	23.182
0.080	21.972	22.297	22.675	23.062	23.467
0.090	22.366	22.697	23.085	23.479	23.892
0.100	22.836	23.178	23.573	23.974	24.391
0.110	23.366	23.718	24.122	24.530	24.952

Table A-17 CBC Radiator System Trade Summary – Alternate D (Titanium Heat Pipe/Deployable System)

Nominal Required Heat Pipes	70
Pipe Material	Titanium
Pipe Diameter (in.)	0.80
Nominal Condenser Length (ft.)	25.0
Area per Heat Pipe (sq.ft.)	14.0
Design Life (years)	30

SYSTEM WEIGHT (KG)

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.030	884.4	899.8	918.0	937.2	957.5
0.040	942.4	959.1	978.9	999.7	1021.8
0.050	1001.6	1019.7	1041.1	1063.7	1087.6
0.060	1063.5	1083.0	1106.1	1130.6	1156.4
0.070	1126.0	1147.0	1171.8	1198.1	1225.8
0.080	1189.8	1212.3	1238.9	1267.0	1296.7
0.090	1254.9	1278.9	1307.4	1337.4	1369.1
0.100	1321.0	1346.6	1376.9	1408.8	1442.5
0.110	1389.7	1416.9	1449.1	1483.0	1518.9

30-YEAR RELIABILITY

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.030	0.00000	0.00000	0.00000	0.00000	0.00000
0.040	0.00000	0.00007	0.00076	0.00459	0.01840
0.050	0.01037	0.07626	0.24856	0.49678	0.72719
0.060	0.17738	0.52853	0.82062	0.95264	0.99081
0.070	0.51083	0.86862	0.97952	0.99791	0.99985
0.080	0.76678	0.97144	0.99809	0.99992	1.00000
0.090	0.89812	0.99405	0.99982	1.00000	1.00000
0.100	0.95608	0.99871	0.99998	1.00000	1.00000
0.110	0.98066	0.99970	1.00000	1.00000	1.00000

AVERAGE NUMBER OF MAINTENANCE SESSIONS

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.030	42.66	34.13	25.78	20.43	17.11
0.040	15.40	11.45	7.94	5.64	4.34
0.050	6.82	4.06	2.32	1.15	0.58
0.060	3.25	1.35	0.53	0.13	0.03
0.070	1.51	0.46	0.07	0.01	0.00
0.080	0.88	0.12	0.01	0.00	0.00
0.090	0.46	0.02	0.00	0.00	0.00
0.100	0.21	0.02	0.00	0.00	0.00
0.110	0.13	0.00	0.00	0.00	0.00

LIFE CYCLE COST (MILLIONS OF DOLLARS)

Pipe Wall Thickness	Number of Redundant Heat Pipes: 2	4	6	8	10
0.030	67.703	54.741	46.697	41.696	38.850
0.040	36.394	30.752	27.375	25.288	24.321
0.050	26.540	22.933	21.352	20.414	20.194
0.060	22.572	20.253	19.685	19.652	20.009
0.070	20.798	19.663	19.616	20.010	20.506
0.080	20.453	19.725	20.048	20.530	21.052
0.090	20.373	20.116	20.567	21.072	21.609
0.100	20.510	20.626	21.099	21.622	22.174
0.110	20.918	21.148	21.655	22.193	22.762

Table A-18 CBC System Life Cycle Cost Breakdown — Baseline (Titanium Heat Pipe and Wiffletree)

Required Number of Panels	36
Number of Redundant Panels	2
Heat Pipe Material	Titanium
Heat Pipe Diameter (in.)	0.80
Wall Thickness (in.)	0.07
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Heat Pipe Failure Rate (per year)	0.0025
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30
2-year System Reliability	0.9953
Avg. No. of Maintenance Sessions	2.26

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	38 at	$100000.0 each	3.800	
Wiffletrees	38 at	$ 20000.0 each	0.760	
Heat Exchangers	38 at	$ 50000.0 each	1.900	
				6.460
Initial Launch				
Radiator Panels	38 at	44.9 lbs	1707.0 lbs	
Wiffletrees	38 at	30.0 lbs	1140.0 lbs	
Heat Exchangers	38 at	10.0 lbs	380.0 lbs	
				11.294
Initial Assembly (IVA)				
IVA Usage of 11.4 hours				0.174
Replacement Manufacturing				
Radiator Panels	2.5 at	$100000.0 each	0.250	
Wiffletrees	2.0 at	$ 20000.0 each	0.040	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.290
Replacement Launch				
Radiator Panels	2.5 at	44.9 lbs	112.1 lbs	
Wiffletrees	2.0 at	30.0 lbs	60.6 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
				0.604
Maintenance (IVA)				
IVA Usage of 1.3 hours				0.063
Reboost (Drag)				
Projected Area of 1064.0 square feet				4.317
Total Cost per Module				23.202

Table A-19 CBC System Life Cycle Cost Breakdown — Alternate A (Stainless Steel Heat Pipe and Daffletree)

Required Number of Panels	36
Number of Redundant Panels	2
Heat Pipe Material	Stainless Steel
Heat Pipe Diameter (in.)	0.80
Wall Thickness (in.)	0.07
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Heat Pipe Failure Rate (per year)	0.0025
Wiffletree Failure Rate (per year)	0.0020
Design Life (years)	30
2-year System Reliability	0.9953
Avg. No. of Maintenance Sessions	2.26

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	38 at	$100000.0 each	3.800	
Wiffletrees	38 at	$ 20000.0 each	0.760	
Heat Exchangers	38 at	$ 50000.0 each	1.900	
				6.460
Initial Launch				
Radiator Panels	38 at	64.4 lbs	2449.1 lbs	
Wiffletrees	38 at	30.0 lbs	1140.0 lbs	
Heat Exchangers	38 at	10.0 lbs	380.0 lbs	
				13.892
Initial Assembly (IVA)				
IVA Usage of 11.4 hours				0.174
Replacement Manufacturing				
Radiator Panels	2.5 at	$100000.0 each	0.250	
Wiffletrees	2.0 at	$ 20000.0 each	0.040	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.290
Replacement Launch				
Radiator Panels	2.5 at	64.4 lbs	160.8 lbs	
Wiffletrees	2.0 at	30.0 lbs	60.6 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
				0.775
Maintenance (IVA)				
IVA Usage of 1.3 hours				0.063
Reboost (Drag)				
Projected Area of 1064.0 square feet				4.317
Total Cost per Module				25.970

Table A-20 CBC System Life Cycle Cost Breakdown — Alternate B (Hybrid Heat Pipe System and Wiffletree)

	High Temp Panels		Low Temp Panels
Required Number of Panels	15		13
Number of Redundant Panels	2		1
Heat Pipe Material	Titanium		Aluminum
Heat Pipe Diameter (in.)	0.80		0.75
Wall Thickness (in.)	0.07		0.12
Condenser Length (ft.)	25.0		45.0
Area per Panel (sq.ft.)	28.0		48.0
Heat Pipe Failure Rate (per year)	0.0025		0.0039
Wiffletree Failure Rate (per year)	0.0020		0.0020
Design Life (years)	30		30
2-year System Reliability		0.9792	
Avg. No. of Maintenance Sessions		1.48	

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	31 at	\$100000.0 each	3.100	
Wiffletrees	31 at	\$ 20000.0 each	0.620	
Heat Exchangers	31 at	\$ 50000.0 each	1.550	
				5.270
Initial Launch				
Radiator Panels (1)	17 at	44.9 lbs	763.6 lbs	
Radiator Panels (2)	14 at	87.0 lbs	1159.2 lbs	
Wiffletrees	31 at	30.0 lbs	930.0 lbs	
Heat Exchangers	31 at	10.0 lbs	310.0 lbs	
			3162.8 lbs	11.276
Initial Assembly (IVA)				
IVA Usage of 9.6 hours				0.145
Replacement Manufacturing				
Radiator Panels	1.9 at	\$100000.0 each	0.185	
Wiffletrees	1.2 at	\$ 20000.0 each	0.025	
Heat Exchangers	0.0 at	\$ 50000.0 each	0.000	
				0.210
Replacement Launch				
Radiator Panels (1)	1.0 at	44.9 lbs	42.9 lbs	
Radiator Panels (2)	0.9 at	87.0 lbs	78.0 lbs	
Wiffletrees	1.2 at	30.0 lbs	36.8 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
			157.8 lbs	0.552
Maintenance (IVA)				
IVA Usage of 2.7 hours				0.032
Reboost (Drag)				
Projected Area of 1148.0 square feet				4.658
Total Cost per Module				22.143

Table A-21 CBC System Life Cycle Cost Breakdown — Alternate C (Titanium Heat Pipe and Heat Pipe Disconnect)

Required Number of Panels	35
Number of Redundant Panels	2
Heat Pipe Material	Titanium
Heat Pipe Diameter (in.)	0.80
Wall Thickness (in.)	0.07
Condenser Length (ft.)	25.0
Area per Panel (sq.ft.)	28.0
Heat Pipe Failure Rate (per year)	0.0025
Disconnect Failure Rate (per year)	0.0050
Design Life (years)	30
2-year System Reliability	0.9826
Avg. No. of Maintenance Sessions	3.83

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	37 at	$100000.0 each	3.700	
Disconnects	37 at	$ 20000.0 each	0.740	
Heat Exchangers	37 at	$ 50000.0 each	1.850	
				6.290
Initial Launch				
Radiator Panels	37 at	44.9 lbs	1662.0 lbs	
Disconnects	37 at	13.0 lbs	481.0 lbs	
Heat Exchangers	37 at	10.0 lbs	370.0 lbs	
				8.796
Initial Assembly (IVA)				
IVA Usage of 11.2 hours				0.170
Replacement Manufacturing				
Radiator Panels	2.5 at	$100000.0 each	0.766	
Disconnects	5.1 at	$ 20000.0 each	0.153	
Heat Exchangers	0.0 at	$ 50000.0 each	0.000	
				0.919
Replacement Launch				
Radiator Panels	2.5 at	44.9 lbs	344.0 lbs	
Disconnects	5.1 at	13.0 lbs	99.6 lbs	
Heat Exchangers	0.0 at	10.0 lbs	0.0 lbs	
				1.553
Maintenance (IVA)				
IVA Usage of 2.1 hours				0.107
Reboost (Drag)				
Projected Area of 1036.0 square feet				4.203
Total Cost per Module				22.037

Table A-22 CBC System Life Cycle Cost Breakdown — Alternate D (Titanium Heat Pipe/Deployable System)

Number of Radiator Panels	10
Number of Heat Pipes/Panel	8
Number of Redundant Heat Pipes	6
Pipe Material	Titanium
Pipe Diameter (in.)	0.80
Wall Thickness (in.)	0.07
Condenser Length (ft.)	23.6
Area per Heat Pipe (sq.ft.)	13.2
Heat Pipe Failure Rate (per year)	0.0012
Design Life (years)	30
30-Year System Reliability	0.9795
Avg. No. of Maintenance Sessions	0.07

Life Cycle Cost Breakdown

Initial Manufacturing				
Radiator Panels	10 at	$378378.4 each	3.784	
Heat Exchangers	10 at	$189189.2 each	1.892	
Deployment Mechanism	1 at	$500000.0 each	0.500	
				6.176
Initial Launch				
Radiator Panels	10 at	170.0 lbs	1699.7 lbs	
Heat Exchangers	10 at	37.8 lbs	378.4 lbs	
Deployment Mechanism	1 at	500.0 lbs	500.0 lbs	
			2578.1 lbs	9.023
Initial Assembly (IVA)				
IVA Usage of 1.5 hours				0.023
Replacement Manufacturing				
Radiator Panels	0.1 at	$378378.4 each	0.027	
Heat Exchangers	0.1 at	$189189.2 each	0.014	
				0.041
Replacement Launch				
Radiator Panels	0.1 at	170.0 lbs	12.2 lbs	
Heat Exchangers	0.1 at	37.8 lbs	2.7 lbs	
			15.0 lbs	0.052
Maintenance (IVA)				
IVA Usage of 0.1 hours				0.002
Reboost (Drag)				
Projected Area of 1059.5 square feet				4.298
Total Cost per Module				19.616

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16. Abstract

This report presents the results of a concept development study of heat rejection systems for Space Station solar dynamic power systems. The heat rejection concepts are based on recent developments in high thermal transport capacity heat pipe radiators. The thermal performance and weights of each of the heat rejection subsystems has been addressed in detail, and critical technologies which require development tests and evaluation for successful demonstration were assessed and identified. Baseline and several alternate heat rejection system configurations and optimum designs were developed for both Brayton and Rankine cycles. The thermal performance, mass properties, assembly requirements, reliability, maintenance requirements and life cycle cost were determined for each configuration. A specific design was then selected for each configuration which represents an optimum design for that configuration. The final recommendations of heat rejection system configuration for either the Brayton or Rankine cycles depend on the priorities established for the evaluation criteria.

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