Numerical Model of Solar Dynamic Radiator for Parametric Analysis

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ABSTRACT

Extensive thermal and power cycle modeling capabilities have been developed which are powerful tools in Space Station Freedom electric power system design and analysis, but which prove cumbersome and costly for simple component preliminary design studies. In order to aid in refining the Solar Dynamic radiator to the mature design stage, a simple and flexible numerical model was developed. The model simulates heat transfer and fluid flow performance of the radiator and calculates area, mass, and impact survivability for many combinations of flow tube and panel configurations, fluid and material properties, and environmental and cycle variations. This paper presents a brief description and discussion of the numerical model, it's capabilities and limitations, and results of the parametric studies performed to date.

INTRODUCTION

Powerful numerical modeling tools have been developed and are being employed for design and analysis of Space Station Freedom's solar dynamic (SD) power modules. These models provide detailed analyses of the SD module's power cycle, thermal environment, structural compatibility, control system, and component performance. While they are invaluable to the progress of the SD module design, these codes prove cumbersome and costly for simple component preliminary design studies and parametric analyses of design options.

In order to aid in refining one component of the SD module - the radiator - to the mature design stage, a simple, flexible numerical model was developed. The model was built to simulate the fundamental functions of the SD radiator in order to facilitate parametric evaluation in terms of heat rejection, thermal integration, area, mass, pumping power, and orbital debris impact survivability. The model simulates functional performance of a simplified SD radiator for many combinations of flow-tube and panel configuration, fluid and material properties, and environmental and cycle variations.

This paper is intended to provide an illustrative example of the tailoring of current engineering models of heat transfer, fluid flow, and orbital debris impact survivability to a simplified space radiator for the purposes of preliminary design and parametric study. As design needs

evolve, so will the numerical model; this paper covers the code's development through the Spring of 1989.

BACKGROUND

The initial phase of Space Station Freedom will be powered by photovoltaic arrays [1]. Growth power requirements will be met through addition of solar dynamic power modules [2], which produce electric energy by means of a thermodynamic power cycle. Solar energy is captured by a solar concentrator and focused into a receiver, in which a helium-xenon gas mixture is heated. A closed Brayton cycle engine utilizes the gas mixture as a working fluid to produce electrical power for transmission to Station users. The SD module rejects waste heat from the closed Brayton cycle power conversion unit to space through the pumped-loop, multi-panel radiator. This process is illustrated schematically in Fig. 1.

The SD module radiator is one of many types of radiators to be deployed on Freedom, but has performance requirements which are unique. Since the radiator acts as the thermal sink portion of the SD power cycle, the heat rejected must be sufficient to maintain power cycle state points throughout the wide operating range of the Brayton cycle system. The radiator also functions as the active cooling system for electrical equipment on the SD utility plate.

The SD radiator design is driven by SD module integration requirements as well. Two SD modules are required to be transported to Freedom in one launch of the STS Orbiter, thus radiator mass, deployability, and launch packaging are critical. The radiator must also have adequate structural integrity and must be oriented such that shadowing of the SD concentrator is minimized. In addition, radiator fluid pumping power requirements must not exceed parasitic power allotments.

Station-level design drivers include reliability and maintainability; survival in orbital environment; and minimization of thermal and structural interactions, drag, and reaction control necessary for off-axis orientation of centers of mass, and, of course, costs.

The development of the SD radiator is based in part on Apollo, Skylab, and Space Shuttle Orbiter technologies [3]. Figure 2 illustrates the SD radiator components and baseline

configuration. The multi-panel radiator is auto-matically deployed using a motorized, scissor-arm and cable mechanism. A single-phase heat transfer fluid is pumped through the electrical equipment cold plate and the cycle gas cooler and on to the radiator panels, which are plumbed in parallel by flexible hoses. The flow loop is complete when the fluid returns to the pump inlet. System redundancy is maintained by two, wholly separate fluid loops - a primary loop and a secondary (back-up) loop.

Each radiator panel is configured with inlet and outlet flow manifolds. Flow tubes are connected to manifolds by perpendicular take-offs and run the length of the panel. The current baseline design calls for 18 active tubes per panel, alternating with 18 secondary tubes. Flow tubes are protected from orbital debris penetration by a bumpered, stand-off configuration. Primary and secondary flow tubes, which alternate through the width of the panel, are separated from each other by a honey-comb structure, and are bound together by a foam adhesive. The flow-tubes and honeycomb structure are both manufactured of aluminum and are sandwiched between two aluminum face-sheets, attached by adhesive.

DESCRIPTION OF NUMERICAL MODEL

The creation of a manageable engineering model from preliminary design information and proposed parametric study concepts required numerous simplifications and approximations to the radiator physical configuration. The model was developed with the objective of simulating operation of a variety of configurations, while maintaining simplicity, efficiency, and flexibility of the code. Results of the numerical model are, of course, limited by the assumptions and simplifications made in its development.

Thermal Model

First among these simplifications is the stipulation that all active radiator tubes perform equally. That is, the fluid inlet temperature, effective sink temperature, and mass flow rate are assumed equal for each of the radiator tubes, and for each radiator panel, regardless of location. This implies that heat transfer from panel manifolds and flex hoses is negligible. With reference to Fig. 2, this assumption is based in reality, given the tube-in-tube manifold configuration and the relatively short span of the flex hoses between panels. On orbit, flow distribution between panels is expected to vary because effective sink temperature, and thus heat transfer, will vary between panels. Nevertheless, the variation of effective sink temperature from panel to panel is small and was neglected here. Frictional pressure drop through manifolds and flex hoses affects the fluid flow distribution in the radiator, and thus the amount of heat transfer from each panel. This effect is neglected in the analysis, since the diameter of panel manifolds and flex hoses is large in relation to that of the flow-tubes, and the span between panels is short in relation to flow-tube length. Pressure drop through the radiator assembly is calculated for pump analysis (see pressure loss model).

The thermal model was constructed from a one-dimensional, steady-state heat balance performed on an element of radiator tube. The basis of this heat balance was developed previously [4] for an armored radiator tube. The focus of this paper is twofold: (1) discussion of the major extensions and modifications made to adapt the model to a particular application and, (2) presentation of some parametric study results. The interested reader is referred to the citation for development of the fundamental heat balance. The heat balance is slightly modified from Ref. 4 to account for a bumpered (rather than armored) tube and for radiation from both sides of the radiator panel node.

The model is formulated so that temperature distributions along a tube are found, marching in the direction of fluid flow. Starting with a radiator tube inlet temperature, which is known from the power cycle state point analysis, nodal temperatures and heat transfer are found by implicit solution of the discretized heat equation for the fin root temperature. An iterative (implicit) solution is necessary because the fin effectiveness is also a function of the fin root temperature (see radiation model discussion). Temperature dependent fluid properties are updated at each node, using a nodal average fluid temperature. The nodal heat rejection is calculated from the radiation heat balance and nodal outlet temperature is calculated from a heat balance on the bulk fluid. Total heat rejection from the radiator is estimated by summing the heat rejected by each node in the single flow-tube and multiplying by the number of tubes in the radiator. Details of the heat rejection cycle model - from bulk fluid heat loss, to tube wall convection, to conduction through the tube bumpering, and finally, to radiation to space - are described below.

Convection Model

At a constant mass flow rate, the flow through a radiator tube decelerates from entrance to exit as it gives off heat and encounters increased viscous drag, so that its Reynolds number can fall into the transition and even laminar flow regimes. Although low Reynolds number flow is undesirable from a standpoint of increased convection heat transfer, its occurrence is a result of design compromises between necessary heat rejection, maintenance of electrical equipment temperatures in the fluid loop (which determines the maximum radiator outlet temperature), pumping power capabilities and candidate fluid properties.

The convection model appears in the nodal heat balance equation as a thermal resistance term for heat transfer between the bulk fluid and the flow-tube wall. The model was developed using conservative estimates of the convective heat transfer coefficients. Entrance region effects were found to be negligible, so the flow is taken to be thermally and hydrodynamically fully developed.

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Nusselt Number equations for the turbulent, transition, and laminar regimes are used to approximate the heat transfer coefficient. It must be noted here that literature on flow retransition is notoriously ambiguous and lacking in useful models or experimental correlations. In addition, agreement does not exist as to where these regimes begin and end as functions of Reynolds number. The emphasis here, then, is on conservative (i.e., worst case) modeling until testing indicates the appropriateness of other models. The standard relation is used in the turbulent (Re > 6400) regime [5]. In the laminar flow regime (Re < 2000), a constant heat rate is assumed, Nu = 3.66, yielding a conservative estimate of heat transfer. An explicit viscosity correction is not made in the equations because the fluid properties (including viscosity) are updated at each computational node. In the flow transition regime (6400 > Re > 2000), an estimate, attributed to H. Hausen, was adapted from a widely used numerical model [6].

Fluid Property Models

The SD radiator will operate over a wide temperature range; thus, an accurate model accounts for the variation of properties important to heat transfer - viscosity, specific heat, thermal conductivity, and density - over the temperature range. The fluid loop is a low pressure system (200 psi or less), therefore, the effect of pressure change on fluid properties has been neglected. A variety of methods have been developed to estimate fluid properties, due to availability of data and the number of fluids under study. Property tables from several sources [7 to 10] are used for the hydrocarbons under study (e.g., toluene, n-heptane). The code uses a linear interpolation routine between data points and a linear extrapolation beyond data points. The code contains an option to use curve-fit equations estimated by the methods described in Ref. 11. These equations are generally more precise than linear interpolation for a property, such as viscosity, which is a highly nonlinear function of temperature. Curve-fit equations are also used for property data of the more esoteric fluids under study, such as FC-75.

Flow-Tube Internal Geometry Models

The design constraints discussed previously will necessitate evaluation of some compact heat exchanger technology, so the numerical model was developed to allow for parametric studies of this type. Increase of the convective heat transfer coefficient may be accomplished by effectively increasing flow turbulence (by adding turbulators such as twisted-tape inserts), by decreasing the hydraulic radius (by altering the tube shape), or by adding internal fins. Several interesting tube modifications discussed in the literature [12 to 14] may be of use in refining the SD radiator flow-tube design. Of course the benefits of these modifications must be weighed against the resulting increased pumping power. The convection model was generically formulated to account for parameters (such as hydraulic radius, friction coefficient, and heat transfer coefficient) which can be different from that of a circular tube and for inclusion of turbulators or fins. Models specific to particular geometries are formulated in subroutines.

Conduction Model

Heat is conducted from the tube walls through the bumpered extrusion and into the panel face-sheet (Figs. 2 and 3). The bumpered extrusion design is a result of the need to protect the flow tubes from penetration by impact of micrometeoroids and space debris. The conduction model appears in the nodal heat balance equation as a thermal resistance term between the tube wall and the radiator face-sheet. A preliminary thermal resistance calculation showed that the major resistances in the conduction flow path were through the four flow-tube extrusion standoffs and across the adhesive barrier between the extrusion and the face-sheets. Indeed, the adhesive resistance is an order-of-magnitude larger than that of the standoffs, which is in turn, much greater than the resistances of the remainder of the conduction path. Contact resistance and surface coating resistance are neglected. Conduction from the extrusion through the aluminum honeycomb is not expected to be significant because of the relatively thick foam adhesive which attaches the honeycomb to the extrusion (thus the one-dimensional conduction formulation).

The radiator face-sheet and honeycomb are approximated as an equivalent mass, equivalent length, rectangular fin, in the limiting case where the effective sink temperature approaches the fin surface temperature [4].

Radiation Model

The radiation heat transfer is formulated to represent heat rejection from the 'prime' and 'extended' radiator surfaces in terms of the effective sink temperature and the fin root temperature. The prime surface is taken to be the face-sheet area which is directly attached to the flow-tube extrusion. The extended surface is defined as the face-sheet area which is directly attached to the honeycomb structure. The effective sink temperature is a function of the radiation environment of the radiator surfaces with respect to all other radiating surfaces (i.e., other station components, solar and earth radiation, and earth-reflected solar radiation) and is not a physical temperature, but an effective environmental temperature. The effective sink temperature for the SD radiator is predicted to range between approximately -70 and -125 °F, prior to addition of more SD modules as required for power growth.

As described in the conduction discussion, the extended surface is modeled as a rectangular fin having a mass equivalent to the combined honeycomb and extended face-sheet and a fin length equal to half the distance between active panel tubes. The fin thickness is modeled as the thickness equivalent to a solid fin of the same mass as the honeycomb and face-sheet. The extended

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surface is doubled to account for a fin on either side of the flow-tube.

This fin model approximation overestimates heat transfer because it neglects thermal resistances due to the adhesive and the honeycomb configuration. Therefore, an option was written into the code which effectively neglects the conduction effect of the honeycomb, and thus underestimates the fin thermal performance. The fin performance is bracketed in this way.

Pressure Drop Model

The primary reason for calculating pressure drop through the radiator assembly is to determine pumping power requirements. The power required to pump the radiator fluid through the cooling loop is a parasitic power loss, and therefore must be minimized. Increased thermal performance from the radiator can be achieved through increased mass flow rate and or alteration of flow tube cross-sectional area, at a cost of increased pumping power. Since convective performance is a primary consideration in radiator mass reduction, careful consideration is being given to the balance of pumping power and the combined effects of mass flow rate and flow-tube cross-sectional area.

Pressure loss due to friction through the flow-tube is calculated for each node as a function of nodal fluid properties and a friction factor based on the Reynolds number. The nodal pressure drops are summed for the tube to find the total tube pressure drop. The standard equation [4] for pressure loss due to pipe friction is used. The friction factor for turbulent flow [15] is assumed in the transition as well as turbulent flow regimes (Re > 2000), and the laminar friction factor [15] is used below Re = 2000. As with heat rejection, pressure drop through each of the radiator tubes is assumed to be equivalent.

To estimate frictional pressure drop through the radiator assembly, the tube pressure drop is added to pressure drops for the inlet, or 'hot' radiator header and the outlet, or 'cold' radiator header. The headers, composed of the panel manifolds and flex hoses, run from the radiator base to the end of the top panel. Pressure drop through the headers is calculated in the same manner as for the tubes, except that losses through fittings (elbows, tees, etc.) are included. Fluid conditions for the 'hot' side are calculated at the radiator inlet temperature and for the 'cold' side at the radiator outlet temperature.

Pumping power is the product of the total pressure loss and the volumetric flow rate, divided by the pump efficiency. Fluid conditions for power calculations are evaluated at the pump inlet temperature, which is assumed to be the same as the radiator outlet temperature.

Mass and Area Calculations

The radiator surface area, neglecting manifold and flex-hose areas, is a product of the panel width, panel length, and number of panels. The radiator mass calculation is based on an algorithm developed by LTV Missiles and Electronics for the baseline radiator configuration. This algorithm uses radiator dimensions and material densities to estimate mass.

Orbital Debris Impact Survivability Model

The prediction of radiator flow-tube survivability from orbital debris impacts is important to preliminary design studies because there is a minimum survivability requirement to which the radiator is expected to comply. Penetration of a flow-tube would cause loss of one fluid loop, necessitating use of the redundant fluid loop. Any alteration to tube bumpering or flow tube geometry for the purpose of heat transfer enhancement may affect the degree to which the radiator is protected from orbital debris penetrations. Thus, penetration survivability is a parameter which must be met to optimize radiator design.

The prediction of survivability in low-earth-orbit from micrometeoroid and space debris impacts is difficult because of uncertainties in: (1) the determination of the size, mass and velocity of a particle which will penetrate a component, and (2) the prediction of the actual debris environment that the component will encounter in terms of type of particle (size, mass, direction, and velocity), population of particles in orbit (currently and over the life of a component), and flux of particles (by altitude, direction, and size).

A great deal of experimentation and analysis has been performed to quantify penetration thresholds (lethal projectile sizes) of components which will experience orbital debris impacts. Penetration has been found to depend upon such parameters as particle diameter, mass, velocity and orientation to the component (normal or oblique); and upon component material, density and the geometry particular to its shielding. In addition, current experimental capabilities are limited to velocities which only approach the lower limit of predicted orbital debris velocities. Thus, prediction of the lethal particle size for a particular component in low-earth-orbit is highly uncertain.

Determination of the debris environment and mitigation of its effects is currently a subject of international concern and much debate. The models to which the Space Station Freedom is being designed are expected to be revised to reflect a more severe environment. The current baseline debris flux models are used in the numerical model and will be updated when the program requirements are modified. It should be noted that the environment is composed of two distinct types of particles which differ in flux, orientation, average velocity, size, and mass. Micrometeoroids occur naturally, while space debris has been deposited in orbit as a result of human activity and is expected to increase.

The general method of predicting survivability is to determine the minimum particle diameter which would penetrate a component; use orbital

flux models to predict the population of particles at least as large as the lethal particle size; and determine the probability of no penetration of the vulnerable area over the component lifetime.

Lethal Particle Size Model

The minimum particle diameter which would penetrate the radiator fluid tube wall is calculated for micrometeoroids and space debris using a model which is considered to yield a conservative prediction [16] for double-walled (bumpered) structures. The calculation is based on empirical results of Nysmith [17] for a normal impact to bumpered aluminum, where the particle diameter is predicted as a function of target wall thickness, wall spacing, and particle velocity, with modification for material density. Micrometeoroids and space debris have different average densities and impact velocities, thus the model predicts different threshold diameters for each type of particle [18].

Hypervelocity impact testing of the preliminary design configuration of the SD radiator is currently underway. Results of these tests are expected to further define the radiator's survivability from orbital debris impacts.

Particle Flux Model

Once the threshold penetration diameter is known, the flux (impacts per year per unit area) of both micrometeoroids and space debris particles large enough to penetrate the radiator is found from the flux models in Ref. 19. These models were developed in particular for the Space Station Freedom orbital altitude and inclination.

The flux of micrometeoroids is based on an 'exposed' area [18], which for a radiator with round tubes, is the product of the tube circumference, tube length, and number of radiator tubes. The flux of space debris is based on a 'projected' area [18], which is the total tube area projected on a plane perpendicular to the space debris plane. For simplicity, the projected area is taken as the product of the tube diameter, tube length and number of tubes.

Survival Probability Model

The probability of no impact of a particle greater than or equal to the lethal particle size is calculated according to the method in Ref. 18 using a 10 year expected lifetime, where separate probabilities are found for micrometeoroids and space debris, and the total probability is the product of the two. Since the radiator has an entirely redundant fluid loop, the survival probability is based on loss of both loops.

EXAMPLES OF RESULTS

Initial runs of the model were made to examine convergence and accuracy of the model. It was found that a convergence of the nodal root temperatures to 0.5 °R was adequate for parametric study purposes. Division of the tube length into 100 nodes yields consistent values for heat rejection.

Next, the model was verified against results of other numerical models [20, 21]. These results are shown in Table 1 for the baseline radiator and show good agreement.

Several parametric studies have been performed to examine optimization of the baseline radiator configuration. One study examined the variation of face-sheet thickness versus spacing between active radiator tubes. Face-sheet thickness was incrementally increased from the baseline thickness of 0.01 in., and the number of active tubes in a radiator panel was incrementally decreased from the baseline value of 18. The width of the radiator panel was held at a constant 7.5 ft. This effectively varied the fin performance. It was found that a thicker face-sheet provided a slight performance improvement at a cost of much increased mass. Reduced mass and equivalent thermal performance could be achieved by reducing the number of tubes per panel and increasing frictional head loss (at a constant mass flow rate). The model indicated that a 16 percent mass reduction can be obtained using 16 tubes per panel and a frictional head loss increase of 40 percent.

This information prompted a study of a parametric variation of the flow-tube diameter and the number of active tubes per panel, holding all other parameters constant. This also effectively varies the fin length. It was found that mass can be reduced by decreasing the number of tubes per panel and increasing the tube diameter, at a cost of slightly increased pumping power (within the current 25 psi allotment).

One of the recent design efforts focused on improvement of radiator performance (especially convective heat transfer in the flow-tubes) is the selection of a fluid. Rocketdyne conducted a fluid trade study analysis [22] in which fluids were evaluated against performance, safety, and material criteria. The numerical model was used to verify the performance results of the contractor's evaluation for the final fluid candidates: FC-75, toluene, and N-heptane. These results are shown in Tables 1 and 2. Small performance differences are attributed to the small differences in fluid properties used in each model and the 'overestimating' fin model used (see radiation model discussion). The results are in agreement with the contractor's conclusion that toluene and N-heptane exceed FC-75 in performance.

SUMMARY

A simple, flexible numerical model has been developed to analyze a variety of SD radiator configurations by parametric study. The model has been verified against results of other available models, and has proven useful in verification of contractor trade-study analyses and preliminary design studies.

ACKNOWLEDGEMENTS

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TABLE 1. – RESULTS FOR BASELINE RADIATOR CONFIGURATION

Parameters	Baseline SD[a] radiator results		
	Ref. 20	Rhatigan	Ref. 21
Number of panels	8	8	8
Outlet temperature, °F	52.4	53.3	52.5
Heat rejected, kW	99.1	98.5	99.1
Panel length, ft	24.7	24.7	25.9
Panel area, ft^2	1482	1482	1554
Total mass, lb	2808	2804	[b]2354
Flow rate, lb/hr	4137	4137	4137
Pressure drop across panel, psi	14.5	15.4	[c]16.8

[a]Baseline: FC-75 heat transfer fluid, 18 active tubes
   per panel, 5 in. spacing between active tubes,
   0.07 in tube i.d., 7.5 ft panel width, T_{IN} = 348 °F,
   T_{SINK} = -70 °F.
[b]Mass does not include redundant loop, fittings,
   hoses, etc.
[c]Includes header losses.

TABLE 2. – RESULTS FOR ALTERNATE RADIATOR FLUIDS

Parameter	N-Heptane			Toluene		
	Ref. 20	Rhatigan	Ref. 21	Ref. 20	Rhatigan	Ref. 21
Number of panels	6	6	6	6	6	6
Outlet temperature, °F	52.0	40.0	52.5	53.0	39.0	52.5
Heat rejected, kW	99.1	102.3	99.1	99.5	102.4	99.5
Panel length, ft	26.0	26.0	26.6	25.5	25.5	26.6
Panel area, ft^2	1170	1170	1197	1148	1148	1197
Total mass, lb	2186	2187	[a]1810	2173	2172	[a]1810
Flow rate, lb/hr	1878	1878	1878	2545	2545	2545
Pressure drop across panel, psi	14.3	14.9	[b]15.2	19.8	19.4	[b]21.2

[a]Does not include redundant loop, fitting hoses, etc.
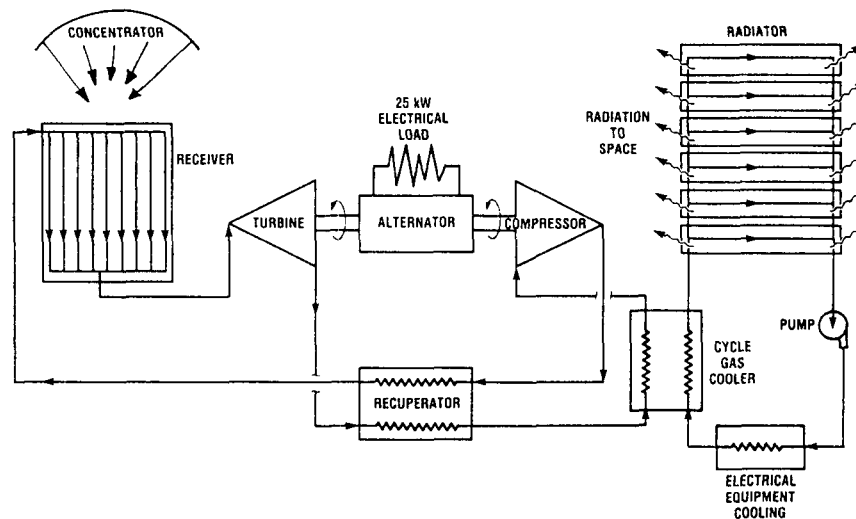[b]Includes header losses.

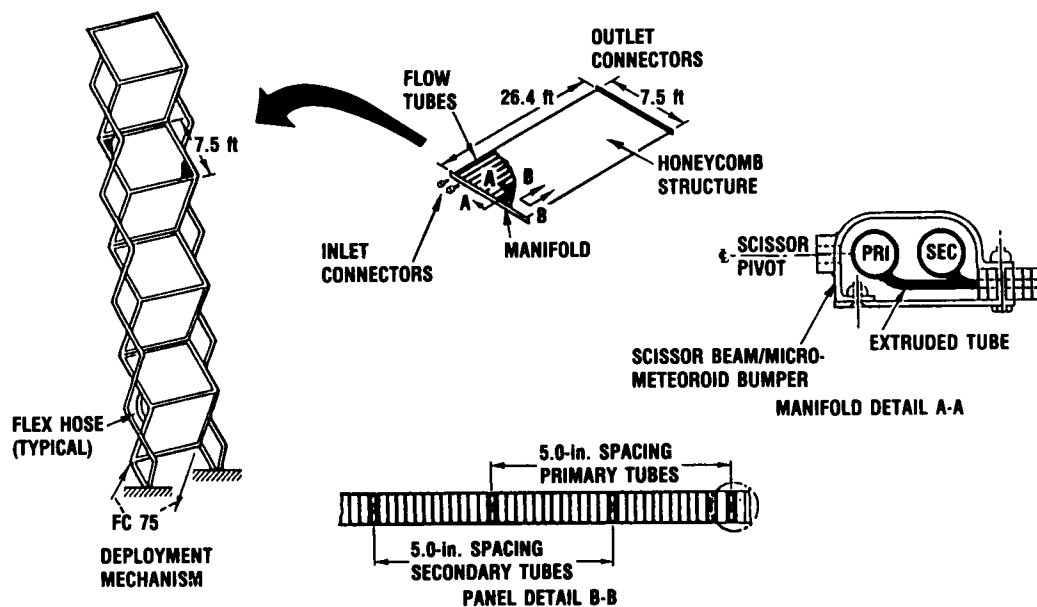FIGURE 1. – SIMPLIFIED SOLAR DYNAMIC POWER CYCLE.
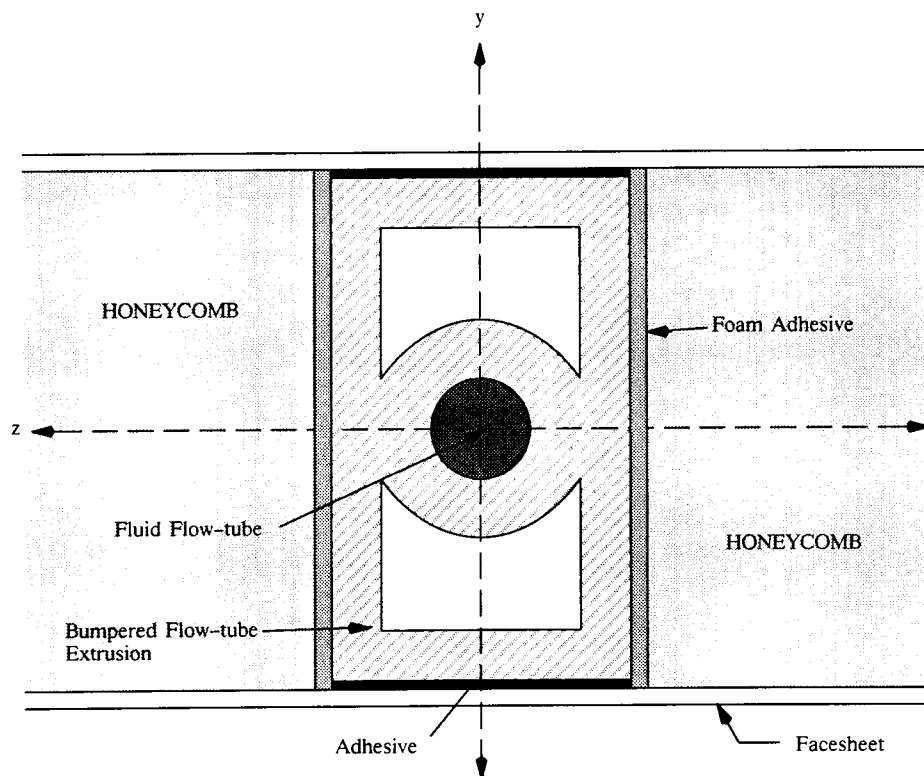
FIGURE 2. – SOLAR DYNAMIC MODULE RADIATOR.

FIGURE 3. – SECTION OF BASELINE SD RADIATOR FLOW-TUBE (FLOW IS IN X-DIRECTION).

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16. Abstract

Growth power requirements for Space Station Freedom will be met through addition of 25 kW solar dynamic (SD) power modules. The SD module rejects waste heat from the power conversion cycle to space through a pumped-loop, multi-panel, deployable radiator. The baseline radiator configuration was defined during the Space Station conceptual design phase and is a function of the state point and heat rejection requirements of the power conversion unit. Requirements determined by the overall station design such as mass, system redundancy, micro-meteoroid and space debris impact survivability, launch packaging, costs, and thermal and structural interaction with other station components have also been design drivers for the radiator configuration. Extensive thermal and power cycle modeling capabilities have been developed which are powerful tools in Station design and analysis, but which prove cumbersome and costly for simple component preliminary design studies. In order to aid in refining the SD radiator to the mature design stage, a simple and flexible numerical model was developed. The model simulates heat transfer and fluid flow performance of the radiator and calculates area mass and impact survivability for many combinations of flow tube and panel configurations, fluid and material properties, and environmental and cycle variations. This paper presents a brief description and discussion of the numerical model, it's capabilities and limitations, and results of the parametric studies performed.

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