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DIVISION

TECHNICAL MEMORANDUM

PREPARED BY: E. F. Perez

ABSTRACT

CYLINDRICAL RADIATOR WITH INTERNAL HEAT REJECTION

An analysis of a cylindrical radiator with heat rejection out of the ends is presented along with the resultant axial temperature profile and radiator weight. The analysis is an approximation, based on several assumptions, which permits a rapid evaluation by an iterative process. The results are based on a design point consistent with SNAP-8 prime radiator requirements during operation with maximum heat load from solar and planetary sources. Results were also obtained for operating temperature levels with no external heat load. A size and weight comparison is made with a cylindrical radiator having the interior surface completely insulated.

APPROVED:

DEPARTMENT HEAD P. I. Wood

DIVISION MANAGER C. C. Ross

NOTE: This document is considered preliminary and is subject to revision as analysis progresses and additional data are acquired. The general reader may encounter internal reference not available to him.

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CYLINDRICAL RADIATOR WITH INTERNAL HEAT REJECTION

I. INTRODUCTION

An analysis was made to evaluate a cylindrical radiator with heat rejection out of the ends. This analysis was made to permit a comparison to a cylindrical radiator in which the inside of the cylinder is insulated and with no heat rejection out of the ends.

This was a simplified analysis to obtain an approximate evaluation of the radiator performance, prior to the date when more accurate information can be supplied by NASA.

II. DISCUSSION

A. RADIATOR DESIGN CRITERIA

The radiator was designed to provide the required heat rejection, 337 kw thermal, at temperature levels corresponding to the system design point, and with the maximum external heat load from solar and planetary sources. This corresponds to a high-noon position in an earth orbit with the longitudinal axis of the cylinder normal to the solar flux. After sizing the radiator for maximum external heat load, an analysis was made to determine the radiator operating temperature levels if no external heat load were considered, i.e., operating in interplanetary space with the longitudinal axis parallel to the solar flux or in the shade.

B. APPROXIMATIONS AND ASSUMPTIONS

The analysis was performed by making certain simplifying assumptions and approximations to permit a hand evaluation of the problem. The basic assumptions are noted in the following:

1. The main radiator and the lube and coolant radiator were assumed to be in a cylindrical configuration with a 20 foot diameter. Only the main radiator was considered to have radiation to the inside of the cylinder. The inside of the L/C radiator was assumed to be insulated and the inside surface was assumed to have an emissivity and reflectivity of 0.5.

2. The main radiator surface emissivity was assumed to be 0.85. The surface absorptivity was assumed to be .60 due to solar radiation and .85 due to diffused radiation from within the cylinder and to planetary radiation.

3. The external heat sources, solar and planetary, were evaluated on the basis of a 500 mile circular earth orbit. The average absorbed flux was assumed to be uniform around the periphery of the cylinder.

4. Temperature of space was assumed to be zero.

5. The heat rejection from the fins was evaluated using the data of Mackay and Bacha*. The effects of irradiation between tubes and fins were not considered.

6. The mutual irradiation between different sections within the cylinder was assumed to be all diffused radiation. The effects of multiple reflections within the cylinder were neglected because of the low reflectivity of the surface material.

7. Longitudinal heat conduction along the tubes and fins was not considered.

8. Fin dimensions were assumed to be: δ_h = .080 in., δ_c = .020 in. and L_h = 3.5 in.

9. The length of the L/C radiator section was assumed to be 6 feet long.

10. The obstruction of the view of space due to the NS, PCS and payload was not considered. A preliminary calculation indicated that this was a reasonable approximation.

III. METHOD OF ANALYSIS

The method of analysis is described in the appendix. The procedure was as follows:

A. The temperature distribution and radiator size were evaluated by an iterative process for the maximum external flux condition.

B. Using the size determined from the maximum external heat flux case, the temperature distribution was evaluated for the case where there is no external heat flux.

* D. B. Mackay, C. P. Bacha, Space Radiator Design and Analysis Part I ASD Technical report 61-30 October 1961.

IV. RESULTS

The results of the analysis are shown in the Figure 1 which shows the fluid temperature distribution in the main radiator as well as the surface temperature on the thermal radiation shield for the L/C radiator.

The size required for the main radiator was found to be considerably smaller, 9.5 feet long by 20 feet diameter compared to the cylindrical radiator in which all of the interior surface is insulated, and has dimensions of 18 feet long by 20 feet in diameter. The weight of tube, armor and fins for the two radiators compare as follows: 1040 lb for the radiator with internal heat rejection and 1150 lb for the radiator with all the interior insulated.

The average temperature variation due to the variation in external heat load, assuming that the fluid heat rejection is the same for both cases is approximately 20°F.

The surface temperature of the thermal radiation shield on the L/C radiator is approximately 265°F. This means that the insulation requirements are minimal since the temperature levels in the L/C radiator are 210°F to 250°F.

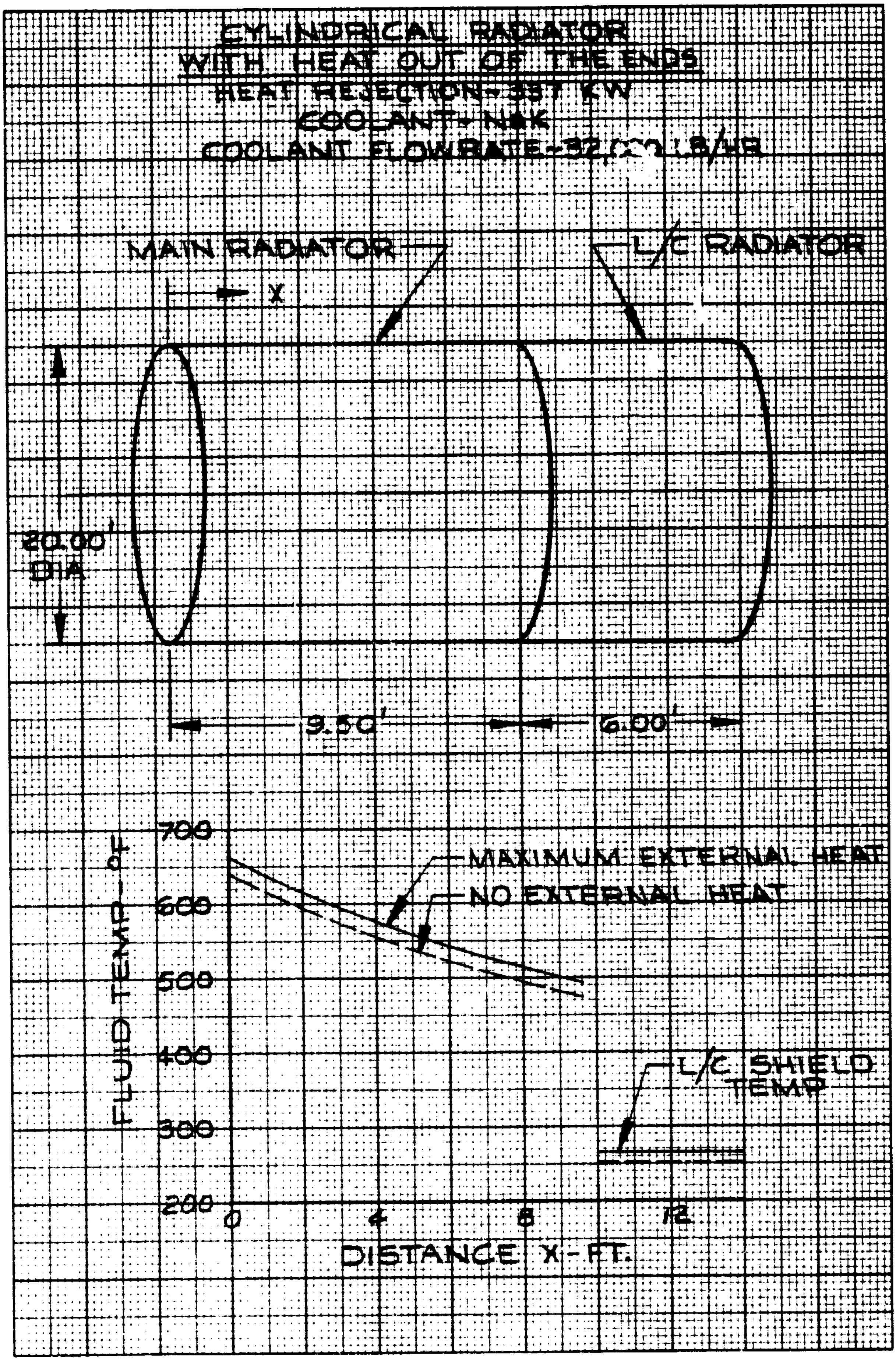
V. CONCLUSIONS

The conclusions which can be drawn from this preliminary analysis are:

A. The size and weight of the main radiator can be reduced by considering heat rejection from the interior of the cylinder. Although additional analysis must be made to evaluate the weight requirements for thermal radiation shielding on the PCS and the payload to fully assess the overall system weights using either radiator concept.

B. The temperature variation from tube-to-tube, due to the varying external heat flux, should be minimized by having mutual irradiation on the inside of the cylinder. This will minimize the problems of flow distribution in the radiator tubes.

C. The structural weight should be reduced by virtue of the reduced radiator size.


CYLINDRICAL RADIATOR
WITH HEAT OUT OF THE ENDS
HEAT REJECTION-337 KW
COOLANT-NaK
COOLANT FLOWRATE-32, LB/HR
MAIN RADIATOR
L/C RADIATOR
X
20.00' DIA
9.50'
6.00'
MAXIMUM EXTERNAL HEAT
NO EXTERNAL HEAT
L/C SHIELD TEMP
FLUID TEMP-°F
700
600
500
400
300
200
0
4
8
12
DISTANCE X-FT.

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SUBJECT CYLINDRICAL RADIATOR BY E7P

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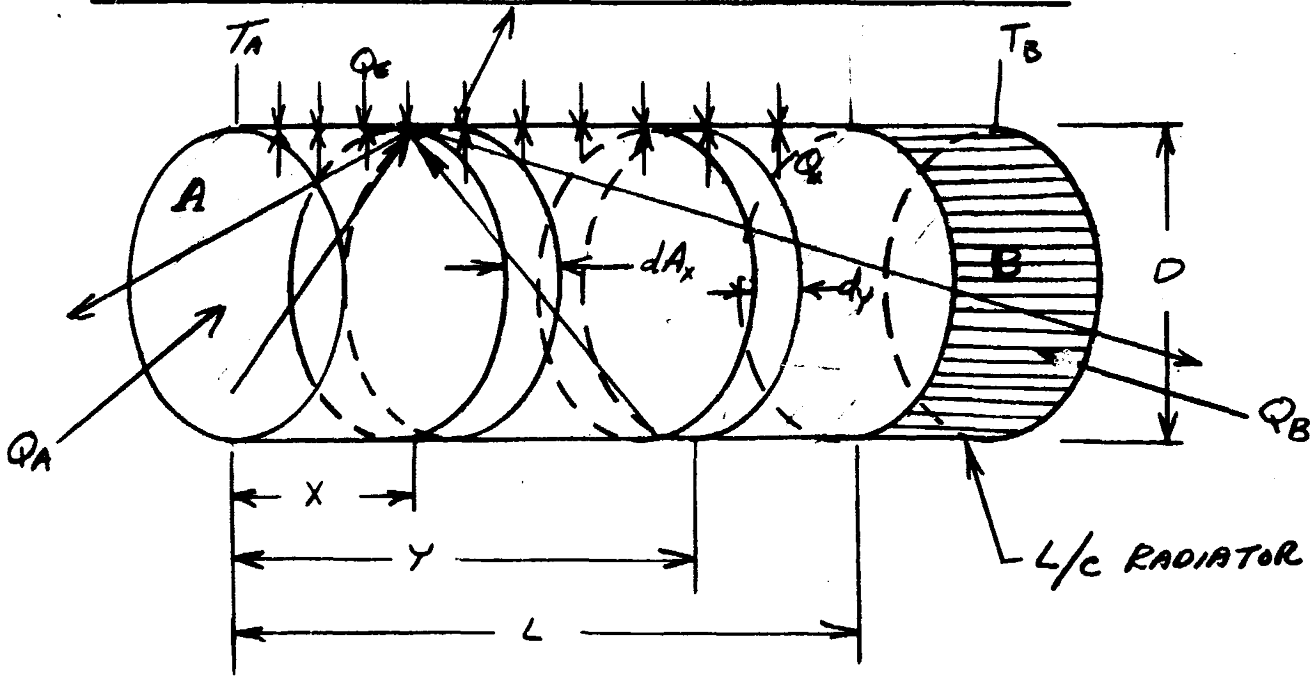
APPENDIX

CYLINDRICAL RADIATOR ANALYSIS WITH HEAT REJECTION OUT OF THE CYLINDER ENDS



ASSUMMING ALL PRIME AREA THE HEAT BALANCE ON ELEMENT ON ELEMENT dA_x

$$(1)\quad dA_x \sigma \epsilon T_{w_x}^4 = dA_x Q_x + dA_x \epsilon^2 \int_0^{x/D} \sigma T_{w_x}^4 \, G\left(\frac{x-y}{D}\right) \frac{dy}{D} + dA_x \epsilon^2 \int_{x/D}^{L/D} \sigma T_{w_x}^4 \, G\left(\frac{y-x}{D}\right) \frac{dy}{D} + dA_x \sigma \epsilon T_A^4 F(x) + dA_x \sigma T_B^4 F(L-x) + dA_x Q_E + \epsilon_L \sigma T_L^4 F_{x-n}$$

FIRST TERM ON THE RIGHT HAND SIDE OF EQN (1) IS THE HEAT REJECTED BY THE FLUID

THE SECOND & THIRD TERMS ACCOUNT FOR HEAT INCIDENT INCIDENT ON dA_x FROM OTHER ELEMENTS RADIATING INSIDE THE RADIATOR. $G(x)$ IS A

COMPLEX FUNCTION DESCRIBING THE FORM FACTOR FROM RING ELEMENT dA_y TO ELEMENT dA_x

THE FOURTH & FIFTH TERMS DESCRIBE THE HEAT INCIDENT ON ELEMENT dA_x FROM EXTERNAL SOURCES. i.e. SOLAR FLUX, PLANETARY ALBEDO, PLANETARY BLACK BODY EQUIVALENT RADIATION. FUNCTION $F(X)$ IS THE FORM FACTOR FROM THE OPEN ENDS OF THE CYLINDER TO ELEMENT dA_x

THIS EQUATION IS DIFFICULT TO SOLVE FOR DIRECTLY SINCE WE DO NOT KNOW THE MANER IN WHICH Q_x AND T_x VARY AS A FUNCTION OF X. THEREFORE: WE MUST MAKE A SERIES OF SIMPLIFYING ASSUMPTIONS AND ACCOUNT FOR THE FACT THAT THE RADIATION IS FROM TUBES AND FINS INSTEAD OF AN ALL PRIME SURFACE. THE HEAT BALANCE IS MADE FROM A NUMERICAL INTEGRATION

SIMPLIFYING ASSUMPTIONS

1. ASSUME RADIATOR MUST BE DESIGNED FOR HIGHEST ABSORBED EXTERNAL FLUX. THIS OCCURS WHEN SOLAR FLUX IS NORMAL TO THE LONGITUDINAL AXIS OF THE RADIATOR AND THE PLANETARY LOAD IS MAXIMUM (EQUIVALENT TO A HIGH NOON POSITION)

2. ASSUME EXTERNAL FLUX IS UNIFORM AROUND THE PERIFERY OF THE CYLINDER. AND NO HEAT COMES IN THROUGH THE ENDS

3. ASSUME DATA OF McKAY APPLIES FOR DESCRIBING HEAT REJECTION FROM THE FINS. EXTERNAL HEAT ON FINS INCLUDES SOLAR & PLANETARY AS WELL HEAT FROM OTHER PORTIONS OF THE RADIATOR. NEGLECT EFFECT OF ADJACENT TUBES

4. DO NOT CONSIDER AXIAL CONDUCTION ALONG TUBES, ARMOR OR FINS

COMPUTATIONAL PROCEDURE

1. ESTIMATE SIZE OF RADIATOR AND SPECIFY FIN AND TUBE DIMENSIONS

2. BREAK RADIATOR INTO SEVERAL EQUAL LENGTH NODES AND EVALUATE FORM FACTORS FOR EACH NODES

3. ESTIMATE TEMPERATURE DISTRIBUTION AND FIN EFFECTIVENESS FOR EACH NODE AND EVALUATE ABSORBED FLUX ON ONE NODE COMMING FOR OTHER NODES INCLUDING FLUX FROM THE L/C RADIATOR SHIELD AND EXTERNAL SOURCES IF THEY ARE CONSIDERED

4. CALCULATE NEW TEMPERATURE DISTRIBUTION AND FIN EFFECTIVENESS FROM DATA OF <u>MACKAY</u>. ITERATE UNTIL UNTIL SUCCESSIVE TEMPERATURE PROFILES AGREE WITHIN A REASONABE TOLARENCE. CALCULATE TOTAL FLUID HEAT REJECTED IN RADIATOR

5. ITERATE ON RADIATOR SIZE UNTIL DESIRED HEAT REJECTION IS OBTAINED

NOMENCLATURE & SYMBOLS

A_F RADIATING AREA OF FIN

A_T RADIATING AREA OF TUBE

C_2 EXTERNAL RADIATION ABSORBED ON SURFACE OF FIN

D DIAMETER OF CYLINDRICAL RADIATOR

D_o OUTSIDE DIAMETER OF ARMCK ON TUBES

F_{L-L} FORM FACTOR FOR L/C SECTION UPON ITSELF

F_{L-x} FORM FACTOR FROM L/C SECTION TO ELEMENT

F_{n-x} FORM FACTOR FROM OTHER SECTIONS TO ELEMENT AT POINT X

F_{n-L} FORM FACTOR FROM OTHER SECTIONS TO THE L/C SECTION

F_{T-F} FORM FACTOR FROM TUBE TO SPACE WITH BLOCKAGE FROM ADJACENT TUBES AND FINS

F_{F-T} FORM FACTOR FROM FINS TO SPACE WITH BLOCKAGE FROM ADJACENT FINS AND TUBES

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N_T TOTAL NUMBER OF TUBES

q_E AVERAGE EXTERNAL ABSORBED FLUX

Q_T HEAT LOST BY THE FLUID THROUGH THE TUBES

Q_F HEAT LOST BY THE FLUID THROUGH THE FINS

q_x HEAT LOSS FROM FLUID AS A FUNCTION OF x

q_P EXTERNAL HEAT FLUX FROM PLANETARY SOURCES

q_S EXTERNAL HEAT FLUX FROM SUN

T_A EFFECTIVE BLACK BODY TEMPERATURE OF RADIATION COMMING IN THROUGH OPENING A

T_B EFFECTIVE BLACK BODY TEMPERATURE OF RADIATION COMMING IN THROUGH OPENING B

T_h TEMPERATURE AT SURFACE OF ARMOR AND AT ROOT OF FIN

$T_{L/C}$ EFFECTIVE TEMPERATURE AT SURFACE OF L/C SECTION

$T_{W(x)}$ WALL TEMPERATURE AS A FUNCTION OF X

δ_h FIN THICKNESS AT ROOT

δ_c FIN THICKNESS AT TIP

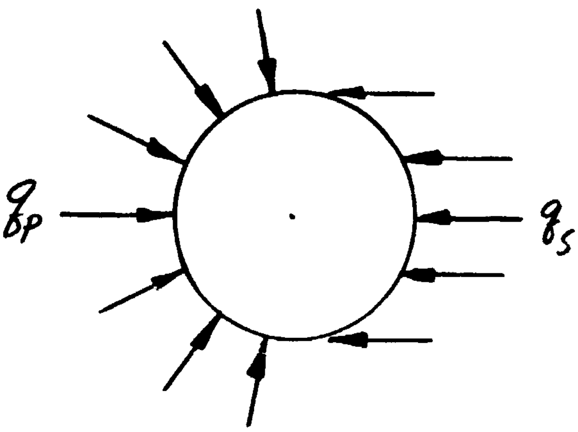
ϵ EMISSIVITY

σ STEFAN-BOLTZMANN CONSTANT

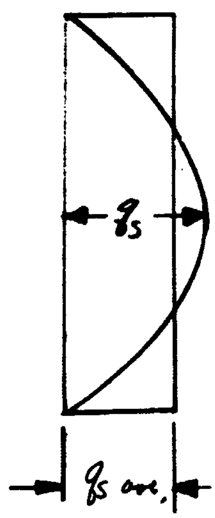
Ω FIN EFFECTIVENESS - RATIO OF HEAT REJECTED FROM FIN TO HEAT WHICH COULD BE REJECTED IF THE FIN WERE AT CONSTANT TEMPERATURE

EXTERNAL RADIATION INCIDENT ON RADIATOR

IN ANALYZING THE CYLINDRICAL RADIATOR AN APPROXIMATION WAS MADE TO ACCOUNT FOR THE EXTERNAL ABSORBED FLUX FROM SOLAR AND PLANETARY SOURCES. IT WAS APPROXIMATED THAT THE AVERAGE FLUX ACTED UNIFORMLY AROUND THE PERIPHERY OF THE CYLINDER, THE AVERAGE FLUX IS AS FOLLOWS



SOLAR FLUX DISTRIB.



$$q_{s\,ave} = \frac{2}{\pi} q_s$$

AVERAGING THE EXTERNAL HEAT LOAD

$$Q_E \cong \frac{\alpha\, q_s \frac{2}{\pi} + \epsilon\, q_p}{2}$$

$$= \frac{.60 \times .636 \times 460 \quad + 130 \times .85}{2} = 147.2 \ BTU/HR\text{-}FT^2$$

HEAT BALANCE ON TUBE SECTION FOR ONE NODE

$$\varphi_T = N_T \Big[\pi F_{T-F} \epsilon \sigma T_n^4 - \sum F_{n-x} \sigma \epsilon^2 \left(\frac{A_T + A_F \Omega}{A_T + A_F} \right) T_{n_n}^4$$
$$- \varphi_E - \sigma \epsilon \epsilon_L F_{L-x} T_L^4 \Big] D_0 \Delta X \qquad (1)$$

φ_T = FLUID HEAT LOST FROM THE TUBE ONLY

F_{T-F} IS TUBE FORM FACTOR WITH BLOCKAGE FROM ADJACENT FINS & TUBES

F_{n-x} IS FORM FACTOR FROM NODE n TO NODE AT DISTANCE x

THE TERM $\sum F_{n-x} \sigma \epsilon^2 \left(\frac{A_T + A_F \Omega}{A_T + A_F} \right) T_{n_n}^4$ IS THE EFFECTIVE ABSORBED FLUX FROM OTHER NODES. THE BRACKETED TERM IS A RATIO WHICH IS USED TO DESCRIBE AN AVERAGE FLUX RADIATING FROM THE TUBE AND FINS.

THE LAST TERM INSIDE THE BRACKET IN EQN. (1) IS THE FLUX WHICH IS RE-EMITTED FROM THE LUB & COOLANT RADIATION SHEILD. WHICH IS OBTAINED FROM A HEAT BALANCE ON THAT SECTION

$$\sigma \epsilon_L T_{L/c}^4 = \sigma \epsilon \sum F_{n-L} \left(\frac{A_T + A_F \Omega}{A_T + A_F} \right) T_n^4 + \epsilon_L^2 F_{L-L} T_{L/c}^4$$
$$(2)$$

HEAT BALANCE ON FIN

$$Q_{FIN} = 4 F_{F-T} \epsilon \sigma \Omega L_h \Delta L T_h^4 N_T \qquad (3)$$

WHERE: F_{F-T} IS FORM FACTOR FOR FIN WITH BLOCKAGE FROM ADJACENT TUBES AND FINS

Ω IS FIN EFFECTIVNESS WHICH TAKES INTO ACCOUNT THE FLUX INCIDENT ON THE FINS FROM EXTERNAL SOURCES AND FROM ADJACENT NODES ON THE INSIDE OF THE CYLINDER

N_T IS TOTAL NUMBER OF TUBES

TOTAL EXTERNAL FLUX ON FINS

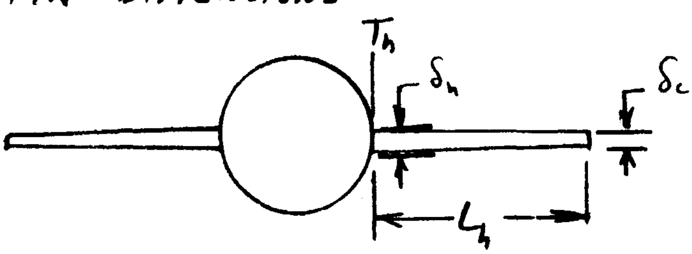
$$C_2 = Q_E + \sum F_{n-x} \epsilon^2 \left(\frac{A_T + \Omega A_F}{A_T + A_F}\right) T_{h(n)}^4 + F_{n-L} \epsilon \epsilon_L T_L^4 \qquad (4)$$

WHERE;

F_{n-x} IS FORM FACTOR FROM ELEMENT n TO ELEMENT AT x

F_{n-L} IS FORM FACTOR FOR L/C RADIATOR

FIN EFFECTIVENESS CAN BE APPROXIMATED USING THE DATA OF MacKay & BACHA * BY DEFINING THE FIN DIMENSIONS



*D.B. MACKAY AND C.P. BACHA "SPACE RADIATOR ANALYSIS AND DESIGN" PART I ASD 61-30 USAF

01-071-002

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PROFILE NUMBER IS DEFINED

$$\xi_p = \frac{C_1 T_h^3 L_h^2}{k \delta_h}$$

ENVIRONMENTAL PARAMETER IS DEFINED

$$E_p \quad \frac{C_2}{C_1 T_h^4}$$

WHERE $C_1 = \sigma(\epsilon_A + \epsilon_B)$ (EMISIVITITIES ON TWO SIDES OF FIN

FROM McKay $\Omega = f\left(\xi_p, E_p, \frac{\delta_c}{\delta_h}\right)$