1123 8-1
Du Lh. Dtd. 11/15/77

Topical Report

DOE/SF/01123--T40

MASTER

THERMAL-TRANSIENT ANALYSIS OF THE BRAYTON ISOTOPE POWER SYSTEM (BIPS) RECUPERATOR

76-13483

October, 1977

Prepared for

**AiResearch Manufacturing Company of Arizona
Phoenix, Arizona**

Contract No. EY-76-C-03-1123

AC03-76 SF 01123

**AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

Topical Report

THERMAL-TRANSIENT ANALYSIS OF THE BRAYTON ISOTOPE POWER SYSTEM (BIPS) RECUPERATOR

76-13483

October, 1977

Prepared for

AiResearch Manufacturing Company of Arizona
Phoenix, Arizona
Contract No. EY-76-C-03-1123

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

DISCLAIMER

This book was prepared as an internal work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, expressed or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of the information contained herein, or represents that it is to be used for any specific purpose, or that it may be copied, reproduced, or otherwise used for any specific purpose without the written permission of the United States Government. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

CONTENTS

<u>Section</u>	<u>Page</u>
INTRODUCTION	1
SUMMARY	1
THERMAL ANALYSIS	2
Preliminary Thermal Analysis for New Transients	2
1. Numerical Results	2
2. Preliminary Thermal Analysis Conclusions	3
Comparison Between Computed Results of HXT4 and Experimental Data	3
1. Effect and Corrections Applied to Test Data	4
a. Thermocouple Lag Correction	4
b. Recuperator Mass	6
c. Flow Maldistribution and Heat Leak	6
d. Thermal Radiation Losses	6
2. Conclusions	8
Detailed Thermal Study for New Transients	8
1. Theoretical Considerations	8
2. Discussion of Numerical Results	9
3. Conclusions	9
4. Recommendations	10
STRESS ANALYSIS	
Analytical Approach	10
Results and Discussions	11
Conclusions	14
REFERENCES	15

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

TORRENCE

TOPICAL REPORT
THERMAL TRANSIENT ANALYSIS OF THE
BRAYTON ISOTOPE POWER SYSTEM (BIPS)

INTRODUCTION

During the system startup transient, the recuperator temperature response varies in the fins, plates, header bars, and manifolds. Large temperature differences can be produced in adjacent structures due to the difference in their mass and in the local heat transfer coefficient. This results in high thermal stresses that can produce plastic deformation in local areas. Damage produced by plasticity and creep during each operating cycle accumulates, and after a sufficient number of cycles a low-cycle fatigue crack develops; this crack can propagate through the parent metal, weld, or braze joint. Cracking can result in internal bypassing of the working fluid (loss in system power output) or in loss of working fluid to space (system shutdown). Low-cycle fatigue was identified as the failure mechanism on the NASA Engine B recuperator. In this test, cracks were observed to develop in the header bar-to-tube plate joints after 15 to 30 cycles of operation.

Extensive transient analyses of the recuperator performance were previously conducted as part of the NASA Brayton-cycle heat exchanger technology program as reported in Reference 1. In this study, about 300 seconds were required for the low pressure gas inlet (turbine discharge) temperature to increase from room temperature to the 1330° F design point condition. Transient analysis of the Brayton Isotope Power System indicated that in this application the inlet temperature transient could be considerably more rapid. A faster startup transient could be expected to produce more severe thermal stresses with a corresponding reduction in cycle life.

The purpose of this study was, therefore, to: (1) examine the proposed system startup modes; (2) select the most favorable condition with regard to recuperator cyclic life; (3) perform a detailed transient heat transfer analysis; and (4) compute the resultant thermal stresses and low cycle fatigue life. In addition, a study was performed to compare the predicted metal temperature transients with experimental data.

SUMMARY

Proposed system startup modes were examined and the most favorable method was selected. Detailed transient temperature analyses were performed for the selected startup sequence and temperature maps were developed for the response of the heat exchanger elements, bars, plates, fins, and manifolds. This data was the basis for subsequent structural analysis.

Recommendations for heat exchanger modifications to obtain a more uniform thermal response throughout the structure are proposed. These changes would result in reduced thermal stresses and should be considered for follow-on, advanced recuperator designs.

Good agreement between experimental data and transient temperature predictions is shown; the analytical techniques are verified.

The structural analysis identified certain critical areas within the core assembly where a high thermal stress is developed. Low cycle fatigue cracks could develop in some of the critical areas after accumulating less than 100 startup cycles (design life). These cracks, if they propagate through the structure, would result in an internal bypass leak that would degrade system performance; however, no external leakage with loss of working fluid would occur. It is concluded that the recuperator low cycle fatigue life is adequate for the BIPS application.

Additional analyses are proposed to more rigorously define the three-dimensional thermal stresses and the resultant cyclic life. These results will increase the confidence level in the present recuperator structural design.

THERMAL ANALYSIS

Preliminary Thermal Analysis for New Transients

Based upon a transient analysis of the Brayton Isotope Power System by the AiResearch-Arizona systems group (Reference 2), AiResearch-California conducted a preliminary thermal analysis. The AiResearch-Arizona analysis furnished new start cycles for the recuperator; the AiResearch-California preliminary thermal analysis assessed the impact of the new startup transients on the recuperator thermal response.

1. Numerical Results

A complete thermal investigation of the recuperator module required the use of two (2) separate computer programs. The first, a special purpose heat exchanger design program (HXT4), calculates gas temperatures and the tube-plate response. The program does not incorporate manifold effects or header bar response characteristics, but does provide the necessary boundary conditions for input to a second computer program. The second computer program, a general purpose thermal analyzer (H0910), investigates in greater detail the temperature distribution at critical regions where large thermal gradients could exist. In this analysis the header bar-tube plate joint, the manifold pan-header bar joint, and the hot side triangular end section-splitter joint were considered the critical regions where large thermal gradients could occur.

In view of the high value of specific heat for the xenon-helium (Xe-He) working fluid (0.593 Btu/lb·°F), observe that in Figure 1* a total of 86 x 10 thermal nodes are required for a stable solution (Reference 3).

Transient tube plate thermal responses for start cycles 101 and 107 from the AiResearch-Arizona analysis (Reference 2) are depicted in Figures 2 and 3.

*All figures are presented at the end of this report.

The temperature gradients are large compared to the results obtained using the NASA-specified startup transient which is the present basis for design. The following comparison is made at the time where the maximum gradient occurs:

	<u>Run 101</u>	<u>Run 107</u>
Normal to flow, Figure 2, $\Delta T/\Delta X \div (\Delta T/\Delta X)_{\text{NASA}}$	1.7	2.1
Flow direction, Figure 3, $\Delta T/\Delta X \div (\Delta T/\Delta X)_{\text{NASA}}$	1.9	2.3

Run 107, designed to produce a less severe operating condition for the recuperator, actually results in a greater thermal stress.

Analysis indicates that the recuperator would have a cycle life of less than 10 cycles for run 107 conditions, too severe to be considered for design purposes. Run 101 startup conditions are considered reasonable with a possible 100 cycle life, which is believed to be consistent with the BIPS design objectives. Detail calculations must be performed to determine the thermal stress in the critical areas before a firm estimate of cycle life can be established.

In Figures 4 and 5 a direct comparison is made between the AiResearch-Arizona program Subroutine Recup and AiResearch-California program HXT4. There is a noticeable difference for the high pressure gas outlet temperatures between results from the two programs.

2. Preliminary Thermal Analysis Conclusions

The preliminary thermal analysis indicates that start cycle 107 is so severe that little or no life can be expected of the unit under these conditions. Start cycle 101 imposes relatively milder thermal stresses, but higher than any of the previous start cycles analyzed.

The analysis indicates that system startup studies should be continued with a goal to reduce the temperature transients imposed for the recuperator. The program should be modified to more accurately reflect the recuperator response characteristics.

Comparison Between Computed Results of HXT4 and Experimental Data

A brief thermal study was made to assess the accuracy of computer program HXT4, comparing the numerical results (high pressure air outlet temperatures) of the quarter-size Mini-Brayton recuperator with available experimental data.

An earlier report (Reference 4) stated that the test laboratory could not simulate the low pressure air inlet transient temperatures shown in Figures 6 and 7. Runs with computer program HXT4 currently use inlet conditions identical to those of the experiment. Numerical results for the long and short ramps are shown in Figures 3 and 4, respectively. For interest, the numerical

results of a second computer program (H0912) are also shown. Good agreement is observed between the results of the two programs.

Discrepancies between predicted values and test results are attributed to phenomena associated with experimental data as follows:

- (1) Thermocouple lag
- (2) Recuperator mass
- (3) Flow maldistribution in stack-wise direction, and heat leak
- (4) Thermocouple radiation losses to the surrounding environment (conduction losses are neglected for practical purposes)

The first two are inherent transient startup characteristics. The third may be applicable to both transient and steady-state conditions. The fourth is usually dominant at elevated temperatures (steady states) when small variations between wall and gas temperatures result in large losses in radiative heat transfer from the thermocouple wires to the surrounding wall. The effect and corrections applied to test data, to account for these phenomena, are discussed below.

1. Effect and Corrections Applied to Test Data

a. Thermocouple Lag Correction

The thermocouple time constant τ is found from Equation 1 below.

$$\tau = \frac{\rho_w C_w V}{A_w H_c} \quad (1)$$

where ρ_w = thermocouple material density

C_w = thermocouple specific heat at constant pressure

V = exposed volume

A_w = exposed surface area

H_c = average heat transfer coefficient (convective)

H_c is determined as in Reference 5,

$$H_c = .44 \frac{K}{D} \sqrt{\frac{DG}{\mu}}$$

where K = gas thermal conductivity, Btu/hr·ft·°F

D = thermocouple wire diameter, ft

G = mass velocity, lb/ft²-sec

μ = gas viscosity, lb/ft·sec

For a 0.03-in.-dia Chromel-Constantan thermocouple wire, 7 lb/min gas flow through a 2.375-in.-dia pipe, H_c is found to be 115 Btu/hr·ft²·°F. Substituting the various parameters in Equation 1, a value of about 1.08 seconds is found for the thermocouple time constant τ .

For a linear increase in gas temperature ($T_g = T_o + A\theta$) the thermocouple temperature is given below as a function of time θ .

$$T = A\theta - A\tau + T_{g_o} + \left(T_o - T_{g_o} + A\tau\right) e^{-\frac{\theta}{\tau}} \quad (2)$$

where T = thermocouple temperature at time = θ

T_o = thermocouple temperature at time = 0

T_{g_o} = initial gas temperature

A = slope of gas temperature response (7.33°/sec and 9.9°/sec for the long and short ramps, respectively)

Substituting the appropriate values in Equation 2 above gives

$$T = 7.33\theta + 72.08 + 7.92e^{-\frac{\theta}{1.08}} \text{ (long ramp)}$$

$$T = 9.9\theta + 69.31 + 10.69e^{-\frac{\theta}{1.08}} \text{ (short ramp)}$$

$$\therefore \Delta T_E = 7.92 - 7.92e^{-\frac{\theta}{1.08}} \text{ (long ramp)}$$

$$\Delta T_E = 10.69 - 10.69e^{-\frac{\theta}{1.08}} \text{ (short ramp)}$$

where ΔT_E is the difference between T/C reading and true gas temperature at time = θ .

b. Recuperator Mass

The quarter-size recuperator measured weight is 42-3/8 lb, including manifolds, pans, and splitter plates. HXT⁴ numerical calculations do not include these parts, thus the results are based upon a lighter unit. The computer output indicates a total weight of 31.965 lb. To have a comparable unit a second case was run with HXT⁴ by increasing the total weight through an increase in header bar mass. This is an approximation, but the method is considered to be adequate for computing the gas outlet temperatures.

c. Flow Maldistribution and Heat Leak

Computed results are based upon the assumption of equal amount of gas flow through each of the fluid passages (25 for low pressure side and 24 for high pressure side). This is an ideal situation which, in reality, is very difficult to achieve. There is usually some degree of flow maldistribution as a result of pressure nonuniformity between gas inlet and outlet points at different stack-wise locations. There is also some flow maldistribution in the plane of a fluid passage between various fluid streams (Reference 6). It has also been noted in Reference 6 that heat losses to the surrounding environment, as a result of imperfect insulation, may account for noticeable drops in recuperator effectiveness and hence high pressure air outlet temperatures.

d. Thermal Radiation Losses

Temperatures measured by thermocouples need to be corrected for radiative and conductive heat losses to the surrounding wall. The true gas temperature T_g is given by:

$$T_g = T_w + (T_E - T_w) \left(\frac{H_c + H_r}{H_c} \right) B \quad (3)$$

where T_E = thermocouple tip temperature

T_w = wall temperature

H_c = convective heat transfer coefficient

H_r = radiative heat transfer coefficient

$$= \sigma \frac{F_{1-w} (T_E^4 - T_w^4)}{T_E - T_w}$$

$$F_{1-w} = \text{gray body shape factor} \approx \frac{1}{\frac{\rho}{\epsilon} + \frac{1}{F_{1-w}}}$$

ρ = reflectivity

ϵ = emissivity (.8)

F_{1-w} = geometric view factor

B = conduction losses (may be neglected for practical purposes, B = 1)

Thermocouple reading corrections are given in Table 1, assuming a 400°F difference between the gas and wall temperatures during transient startup and a 80°F difference for steady state.

TABLE 1
THERMOCOUPLE CORRECTIONS

Time (θ), sec	T/C Reading, °F		True Gas Temperature (Equation 3)	
	Long Ramp	Short Ramp	Long Ramp	Short Ramp
0	80	80	80	80
40	430	440	438	448
90	630	750	646	773
120	720	-	741	-
Steady State	1220	1220	1239	1239

The effects of radiation and thermal lag corrections (Figures 8 and 9) are favorable when compared with the computed results of HXT4 and H0912. The steady-state corrections due to radiation losses produce a discrepancy of only 34°F between predicted and experimental values which is less than 3%. The overestimation of predicted temperatures during transient startup is attributed mostly to the recuperator mass difference (42-3/8 lb experimental, 31.965 lb predicted), stated above. In Figures 10 and 11 the results of the second run with HXT4 are shown with the recuperator mass increased to make it comparable to that of the test unit. Improved agreement is observed. The maximum variation between predicted and experimental temperature is about 20°F for the long ramp and about 50°F for the short ramp case. This variation, representing about 6% difference between experimental and predicted, is attributed to flow maldistribution. The predicted temperatures are underestimated at the beginning of the start cycle as a result of uncertainty of metal temperatures at the start of the cycle.

2. Conclusions

A comparison was made between the numerical results of the computer program HXT4 and experimental data for the high pressure air outlet temperature of the quarter-size Mini-Brayton recuperator. The thermocouple measured temperatures were corrected for thermal lag and radiation losses to the surrounding wall. The computer program HXT4 was run a second time for larger recuperator total mass to account for the manifolds, pans, and splitter plates. The final corrected values indicated that there is less than 3 percent difference between measured and predicted temperatures at steady states and 6 percent difference during transient startups. In both instances the measured values were underestimated as compared to predicted values. This trend is probably associated with flow maldistribution and possible heat leak due to imperfect insulation.

The good agreement between predicted (program HXT4) and the measured high pressure air outlet temperature substantiated the use of computer program HXT4 in assessing the recuperator metal temperatures.

It is recommended that future experiments be undertaken with a few wall measuring thermocouples imbedded inside surrounding walls whenever gas temperatures are measured. These thermocouples will provide a more accurate calculation of radiation losses to surrounding walls. Alternatively, an aspirating-type thermocouple probe should be utilized.

Detailed Thermal Study for New Transients

The detailed thermal analysis was performed for run 101 with Xe-He as the working fluid. Emphasis was placed upon proper construction of the thermal model, which must be compatible with the stress model and have the flexibility of incorporating design changes in areas with potentially high thermal stresses. The numerical results from the thermal analysis were used to identify critical areas with large thermal gradients and to identify possible design changes for attenuating these gradients.

1. Theoretical Considerations

The orientation of the hot end high-pressure outlet and low-pressure inlet ports with respect to the manifold pan is shown in Figure 12. Unlike the previous unit analyzed (Reference 1), the present configuration has both ports pointing in the same direction (down). The original design is shown in Figure 13 where the ports are aligned in opposite directions (one up, one down). Having the two ports pointing in the same direction is detrimental from a thermal standpoint. The manifold pan temperature lags noticeably behind the triangular section of the recuperator core. This effect will be further amplified with the design shown in Figure 12 (two ports down). The low pressure fluid stream provides good convection (forced) heat transfer at the entrance of the pan (bottom section of Figure 12); however, as the fluid stream flows upwards, the flow rate (and hence local velocity) is gradually reduced. When it reaches the "dead end", only a small amount of the fluid remains to heat the large mass of the pan. The situation is identical for the high-pressure gas outlet. The result is a "cold pan" at the upper portion of the pan and a "hot pan" at the

bottom portion near the port entrances. The average pan temperature is the coldest at section "A" (Figure 13) where there are small flow rates for the low-pressure gas. The numerical results shown below will provide a comparison between the thermal response at section A of Figure 13 to the configuration shown in Figure 12.

2. Discussion of Numerical Results

A record of nodal identification (for the thermal analyzer program) is shown in Figures 14 and 15. Boundary conditions for the thermal nodes are imposed from the output of program HXT4. The reason for considering a total length of 5 inches is determined from Figure 16 where it is noted that the recuperator core is hot for a distance of only 4 inches in the direction of low-pressure fluid flow. The remainder of the core (18.6 in.) is below 100°F during the entire transient process of 180 sec. The model shown in Figure 14 provides the capability of moving the manifold pan joints to the core back and forth (in the flow direction), if need be.

The response of the main recuperator heat transfer matrix is shown in Figures 17 through 19 at 3 different stations along the direction of the low pressure gas flow. It is seen that the location where the pan first joins the core (0.92 in.) of Figure 18, is contributive to the largest header bar lag. The splitter plate thermal response is indicated in Figures 20 and 21 which show the results of both configurations with respect to port orientation. There are distinctively large temperature gradients along the splitter plate. The improved response in the case where ports have opposite directions (Figure 2, Section A) is noted. The temperature distribution along the pan is depicted in Figure 21 for both configurations of Figures 12 and 13. The improved temperature response for the case of Figure 13 (one port up, one down) is noted for the low pressure fluid inlet side.

The results presented in Figures 17 through 21 indicate that for the design configuration of Figure 12, the largest thermal lag between recuperator core and manifold pan occurs between 120 and 180 seconds after start. For the case where the ports have opposite directions (Figure 13) the largest gradient along the splitter plate occurs at about 90 seconds after start. Temperature maps for 120, 150, and 180 sec were selected for structural analyses of the design configuration of Figure 12 (both ports down). In the case of Figure 13 (one port up, one down) three sets of temperature maps were selected for Sections A, B, and C at 90 seconds.

3. Conclusions

The numerical results indicate that for the present design, start cycle 101 creates large temperature differences between the header bar and the tube sheet at the junction between manifold pan and main core. Large temperature gradients also occur along the splitter plate between the manifold pan and the triangular end section of the recuperator. In the hot gas (low pressure) leading edge response shown in Figure 22, there appears to be a significant lead at the header bar at 10 sec after start. This is a potentially high stressed area and should be analyzed in a subsequent analysis.

4. Recommendations

The large temperature gradients cited above may be attenuated by considering the following changes:

- (a) Increase Hot Side Triangular End Section Fin Thickness from 0.008 in. to 0.016 in.--This mass addition will enable a slower response of the core and hence less temperature difference between core and manifold pan. A better splitter plate thermal gradient should also result.
- (b) Move the Pan Joint At the Recuperator Core Downward in the Low Pressure Gas Flow Direction--This change will result in less thermal lag in the header bar, indicated in Figure 18.
- (c) Remove the Seal Plate and Change the Header Bars to Channel Headers at the Manifold Attaching Flange--This mass reduction will reduce the large lag between the manifold pan and the recuperator tube sheet.
- (d) Introduce Channel Type Finned Surfaces Around the Pan Assembly and Along the Splitter Plate--This approach will provide some turbulence and hence better convective heat transfer between the gas and the pan.

STRESS ANALYSIS

Analyses of the Brayton Isotope Power System (BIPS) start-up transients have defined a new transient (Run No. 101), which is more severe than the original transient defined by NASA for the Mini-Brayton recuperator design. In Run No. 101, the recuperator, initially soaked at minus 10°F, is subjected to a low pressure inlet gas temperature of 1600°F instantaneously, decreasing to 1110°F in 180 seconds. These two transients are compared in Figure 23. This new start transient is considerably faster than that defined by NASA and hence, very high thermal stress are possible.

A two-dimensional finite element model was constructed and temperature distribution based on the results of the thermal analysis, discussed earlier in this report, were applied to determine thermal stress distributions. Maximum transient stresses were compared for two different manifold port configurations. Local plastic deformation and creep relaxation effects were considered and included in the analysis. Cyclic life estimates for several potentially critical areas in the recuperator were obtained.

Analytical Approach

In the finite element model shown in Figure 24, a plane stress state was assumed. Since the temperature difference between the core and manifold in the cold end is small, the manifold in the cold end is not included in the model.

In the portion of the recuperator with small thermal gradients, beam elements were employed to simulate local stiffness of the header bars and elements of larger size were used to save computation time. The location and the time where and when the largest thermal lag between core and manifold would most likely occur were selected for analysis because this condition would produce the largest thermal stress. Four cases were studied. In Case A (Figure 24) the recuperator has one manifold port up and the other down; analysis was performed for the time at 90 sec after start. In Cases B, C, and D, (Figure 25) the ports have the same direction; analysis was performed for the time at 90, 120, and 180 sec after start (for Cases B, C, and D, respectively). Effective values of stiffness and thickness were employed in the finite element model to represent the composite structure in the recuperator core. Temperature distributions were based on the thermal analysis discussed earlier in this report and an interpolation of these results to obtain the nodal point temperatures required for the finite element computer program. The Sandor-Topper and Goodman's mean stress correction method were employed to estimate the fatigue lives.

It should be noted that in the adjacent areas of the core-to-manifold joints, the stack height stresses were estimated and combined with the 2-D analysis results by means of the 3-D von Mises formula for equivalent stress to produce the fatigue life estimates in these areas.

Results and Discussion

The temperature distribution, thermal distortion, and equivalent stress distribution for Case A are shown in Figures 27 through 29. Figure 28 shows the manifold distortion due to its lagging transient thermal response. The splitter, flanges, and their vicinities are highly stressed. There are significant stress concentrations in the vicinity of the header bars and at the transition between the plain fins in the triangular turning section and the rectangular offset fins in the counterflow section due to the abrupt change in fin density.

Temperature distribution, distortion, and equivalent stress are shown in Figures 30 through 38 for Cases B, C, and D. From these sets of plots (three each for Cases B, C, and D), involving the same port configuration, the change in stress level can be seen as a function of elapsed time. Maximum thermal stresses, based on von Mises equivalent values, occur between 90 and 180 seconds in the start-up transient and are reasonably constant during this time interval.

It is noted that only elastic analyses have been performed, and the stress levels may be lower in reality, since the calculated stress levels exceed the yield point in some portions of the recuperator.

The maximum stresses and temperatures for different conditions and different manifold configurations are shown in Table 2. The elements where these maximum stresses occur are shown in Figure 39.

The cyclic lives for elements in the critical areas are shown in Table 3. Element locations are shown on Figure 39. Note that these analyses are based on the stress analysis for the one port up, one port down model (Case A).

TABLE 2

MAXIMUM STRESSES IN CRITICAL ELEMENTS OF THE 2-D FINITE ELEMENT ANALYSIS

Model Configuration	One Port Up One Port Down	Both Ports Down		
Case	A	B	C	D
Time	90 Sec.	90 Sec.	120 Sec.	180 Sec.
σ_x , psi	42913.5* E427**	42931.9 E427	44313.5 E431	43684.5 E431
σ_y , psi	37416.7 E260	36877.8 E260	30257.7 E234	21772.2 E129
σ_{xy} , psi	18237.3 E399	17436.1 E399	15536.3 E413	12563.4 E481
Max. σ , psi	28628.0 E255	28553.6 E255	25426.0 E242	26087.0 E491
Min. σ , psi	46935.6 E432	47104.3 E432	48230.1 E432	45899.3 E431
Max. Shear σ , psi	25092.4 E432	25137.2 E432	25031.0 E432	22330.3 E432
Temp., °F	1072.70 E494	1071.8 E494	1067.3 E502	1057.66 E502
Equivalent σ , psi	48641.7 E432	48766.7 E432	49171.7 E432	45125.4 E431

*Maximum magnitude

**Element number

TABLE 3

CYCLIC LIFE ESTIMATES FOR CRITICAL AREAS IN THE RECUPERATOR

Area	Element Analyzed	Mean Stress Correction Method Used	Life Due to Plastic Strains Only, Cycles	Locked-in Stress Due to Plastic Strains, psi	Creep Damage Fraction per Cycle	Cyclic Life Estimates
Core	E427	S.T.* G.*	2574 2019	39615 47225	0.00901 0.001113	70~140 60~120
Splitter Joint	E539	S.T. G.	252 112	63589 67234	0.01222 0.013	40~80 30~60
Brazed Area of Splitter Joint	E531	S.T. G.	2938 2611	43373 47392	0.00808 0.00896	80~160 70~140
L.P. Pan Joint	E379	S.T. G.	428 296	62841 64477	0.0119 0.01228	20~40 15~30
H.P. Pan Joint	E391	S.T. G.	12672 26214	31526 39688	0.00496 0.00656	130~260 100~200

*S.T.: Based on the Sandor & Topper's correction.
G.: Based on the Goodman's correction.

Conclusions

Port configuration has little influence on stresses based on a two-dimensional core analysis. A three-dimensional manifold temperature distribution is required to produce a fatigue life comparison which more accurately represents the influence of port orientation.

The most critical area is at element 379, located at the junction between the core and the low pressure pan. The computed fatigue life is 15 to 40 cycles and dependent on the mean stress correction method used in the analysis.

The cyclic life represents the point at which the first crack would open. Unless the stresses were relieved, the crack would propagate and after a few additional cycles a leak path would develop. For any of the critical elements noted on Table 2 the leak path would be between the high and low pressure streams; there would be no external leakage. Any bypass leakage would result in a decrease in system efficiency, but no working fluid would be lost.

It is believed that the BIPS recuperator can withstand 100 startup cycles using Run 101 conditions without developing external leakage. Some internal bypass leakage could occur, however, due to low cycle fatigue cracking in critical areas.

REFERENCES

1. Killackey, J. J., Coombs, M. G., Graves, R. F., and Morse, C. J., Brayton-Cycle Heat Exchanger Technology Program, NASA CR 135158, August, 1976.
2. Transient Analysis Isotope Brayton Flight System, AiResearch-Arizona Report 75-311279, May 19, 1975.
3. A Study of Grid Size Effect on Accuracy of Counterflow Heat Exchangers, Transient Thermal Analysis Computer Programs HXT4, H0912, and H0910, AiResearch Office Memo 74-10832, Sept. 30, 1974.
4. Experimental Versus Computed Temperatures for Mini-Brayton Recuperator - HXDA Project, AiResearch Office Memo 74-10700, Aug. 26, 1974.
5. Moffat, R. J., Temperature - Its Measurement and Control, Science and Industry, Vol. 3, Part 2, 1962, pp. 553-571.
6. Design and Fabrication of the Mini-Brayton Recuperator (MBR), AiResearch Monthly Progress Report 74-10188(12), Feb. 28, 1975.

M. B. R.
THERMAL NODES (XE-HE)

Figure 1. Thermal Nodes (XE-HE)

Figure 2. Transient Thermal Response

Figure 3. Transient Thermal Response

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Figure 4. New Transient Startups
(Run 101)

Figure 5. New Transient Startups
(Run 107)

Figure 6. Long Ramp Low-Pressure Air Inlet Transient Temperature

AIRSEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Figure 7. Short Ramp Low-Pressure Air Inlet Transient Temperature

Figure 8. Long Ramp High-Pressure Air Outlet Temperature Transient

Figure 9. Short Ramp High-Pressure Air Outlet Temperature Transient

Figure 10. Long Ramp High Pressure Air Outlet Temperature Transient Response

Figure 11. Short Ramp High Pressure Air Outlet Temperature Transient Response

S-11553

Figure 12. Orientation of Ports in Same Direction

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

S-11552

Figure 13. Orientation of Ports in Opposite Direction

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

4.993"

Figure 15. Record of Nodal Identification

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Figure 16. Thermal Response New Start Cycle (Run 101)

Figure 17. Thermal Response New Start Cycle (Run 101) -- First Station

Figure 18. Transient Thermal Response New Start Cycle (Run 101) -- Second Station

Figure 19. Thermal Response New Start Cycle (Run 101) -- Third Station

Figure 20. Splitter Plate Thermal Response -- Hot End (Run 101)

Figure 21. Manifold Pan Thermal Response (Run 101)

Figure 22. Hot Gas Low Pressure Leading Edge

Figure 23. Start-up Temperature Transients from BIPS and MBR Spec.

Figure 24. Finite Element Model

Figure 25. Section Analyzed for the Configuration with One Manifold Port Up and the Other Down (Case A)

Figure 26. Section Analyzed for the Configuration with Both Manifold Ports Down (Cases B, C, and D)

Figure 27. Recuperator Hot End Isotherms, Case A

Figure 28. Recuperator Hot End Thermal Distortion, Case A

Figure 29. (Recuperator Hot End) Equivalent Stress Contours, Case A

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

Figure 30. Recuperator Hot End Isotherms, Case B

Figure 31. Recuperator Hot End Thermal Distortion, Case B

Figure 32. (Recuperator Hot End) Equivalent Stress Contours, Case B

Figure 33. Recuperator Hot End Isotherms, Case C

Figure 34. Recuperator Hot End Thermal Distortion, Case C

Figure 35. (Recuperator Hot End) Equivalent Stress Contours, Case C

Figure 36. Recuperator Hot End Isotherms, Case D

Figure 37. Recuperator Hot End Thermal Distortion, Case D

Figure 38. (Recuperator Hot End) Equivalent Stress Contours, Case D

Figure 39. Elements with Maximum Stresses and Elements Analyzed for Thermal Fatigue