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THE GARRETT CORPORATION
AiResearch Manufacturing Division
LOS ANGELES 45, CALIFORNIA

REPORT NO. L-9375

RECUPERATOR DEVELOPMENT PROGRAM
SOLAR BRAYTON CYCLE SYSTEM
PROGRESS REPORT

July 19 to August 19, 1964

L-9375 August 31, 1964

NO. OF PAGES 9

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DATE August 31, 1964

EDITED BY E. F. Busch

Attachments:

1. Appendix I
   (AiResearch Report AM-288)
2. Drawing L198005

APPROVED BY K. O. Parker

Approved By R. D. Mueller

REVISIONS				ADDITIONS			
PAGE	DATE	PAGE	DATE	PAGE	DATE	PAGE	DATE

RECUPERATOR DEVELOPMENT PROGRAM
SOLAR BRAYTON CYCLE SYSTEM
PROGRESS REPORT
JULY 19 TO AUGUST 19, 1964

INTRODUCTION

This report describes the work accomplished by the AiResearch Manufacturing Division of The Garrett Corporation, Los Angeles, California, during the above reporting period under National Aeronautics and Space Administration Contract NAS3-2793. This contract is for the development of a recuperator to be utilized in a closed Brayton Cycle space power system which will use solar energy as the heat source and argon as the working fluid.

FINAL DESIGN SELECTION

On July 15, 1964, official confirmation of the NASA selection of a pure counterflow plate and fin heat exchanger as the unit to meet their specification was received. The final operating conditions for this unit were as follows:

	Temperature, °R	Pressure, psia
Cold Inlet	801	13.8
Hot Inlet	1560	6.73

Gas Flow Rate (each side) = 36.69 lb/min (argon)

Effectiveness = 0.9

Total Pressure Drop ($\Delta P/P$ for both sides), percent = 2.0

With the decision to use this type of matrix at these problem conditions, a very careful survey of the results obtained during the Parametric Survey (AiResearch Report L-9372) was made to ensure that the optimum core was selected. In examining the results of the survey, attention had to be paid to the pressure drop as well as the volume and weight of the matrix. As pointed out in the presentation of the results of the Parametric Survey, in pure counterflow plate fin heat exchangers which have the constraint that flow length for both fluids must be identical, the amount of available pressure drop on the high pressure side varies from matrix to matrix. In selecting a core it is, therefore, possible that the lightest matrix may use more pressure drop than a slightly heavier one. In this case, the selection of optimum may be strongly influenced by the size and pressure drop of the triangular ends required to introduce and remove the argon from both sides of the heat exchanger. With these ends becoming increasingly important in reaching the final selection, AiResearch wrote a computer program to analyze the pressure losses in these ends. The program was written to handle the use of rectangular ends. Rectangular ends are not suited to this application as they may only be used where pressure drop on one side of the heat exchanger is very high. A complete description of this program, together with the theory used to determine the pressure drop in the ends, is shown in Appendix I attached to this report.

This program has been used to predict the pressure losses in the ends of another very similar heat exchanger. Preliminary testing of this heat exchanger has substantiated the results of the computer program predictions

The careful study of the heat exchanger designs formulated under the Parametric Survey led to the selection of a heat exchanger matrix with a flow length of 7.28 in., a stackup height of 25.46 in., and a flow width of 25.19 in. This core consisted of 74 sandwiches of 12 rectangular offset fins per inch, 0.178 in. high on the low pressure side and 74 sandwiches of 16 rectangular offset fins per inch, 0.153 in. high on the high pressure side. This unit has the required effectiveness of 0.90 with a total pressure drop ($\Delta P/P$) of 0.71 percent. Using 6 in. and 8 in. diameter ducts, the total fixed pressure losses from ducts to manifolds is estimated to be 0.63 percent. This leaves a total of 0.66 percent available for the triangular ends. With this pressure drop available for the ends, the computer program was used to examine a series of triangle heights and splits between high and low pressure sides.

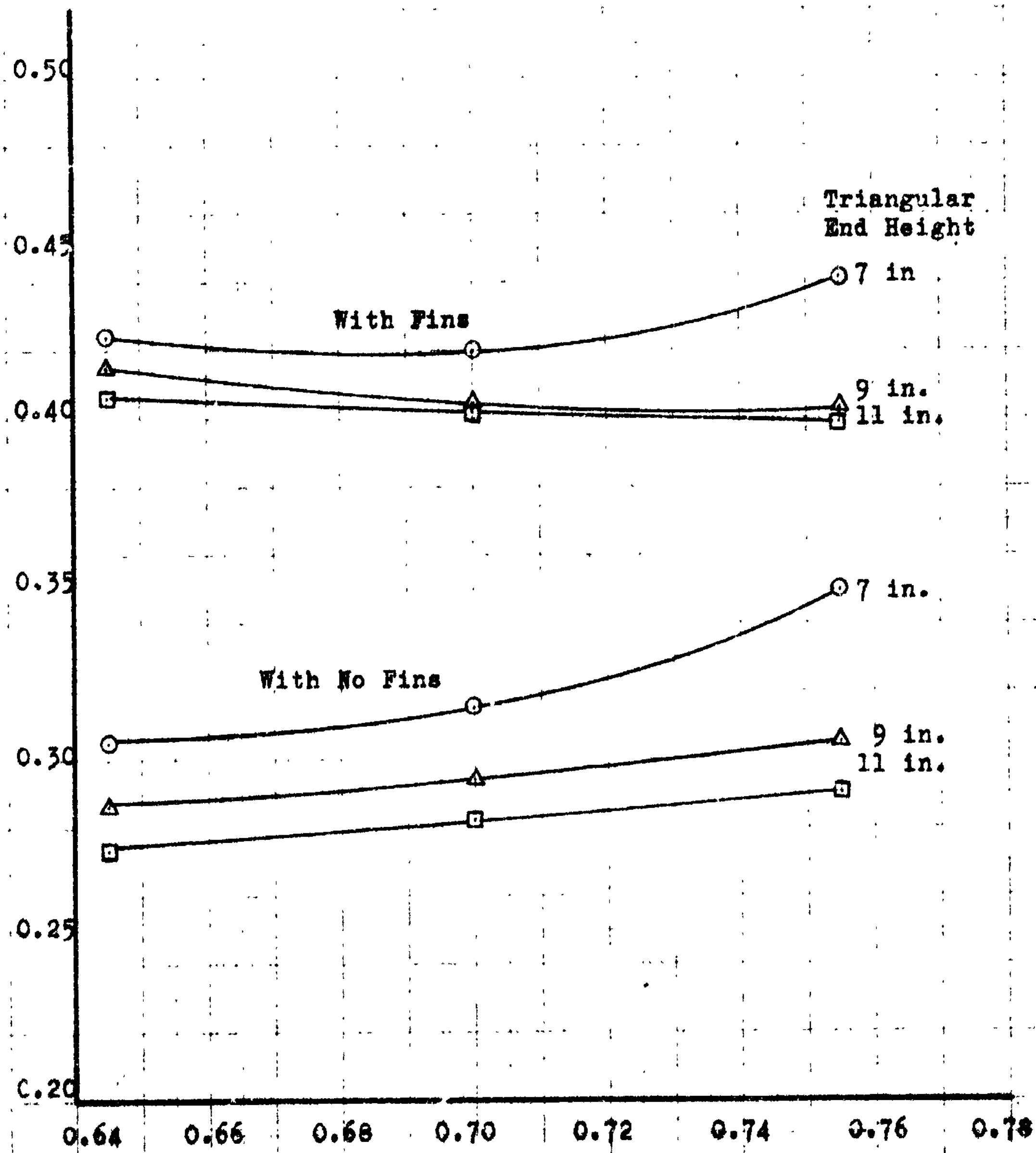
The fins used in the triangular ends require only to match the height of the appropriate passage. The fewer the fins used, the higher the hydraulic radius and the lower the pressure drop. The investigations of the ends included the examination of both 5 fins per inch on both sides and 5 fins per inch on the high pressure side and no fins at all in the low pressure side. The results of this investigations are shown in Figure 1. This figure shows that the ratio of the split of the core face between low and high pressure should be about 70 percent to 30 percent, respectively, if fins are used in both sides of the ends and to be about 65 to 35 percent if no fins are used in the low pressure side. The effect of varying height of the triangular ends is also clearly shown. The effect of eliminating the fins in the low pressure side is very pronounced, but until stress analysis and manufacturing details are completed, it cannot be fully determined whether or not these fins may be omitted.

The design selected from this investigation was to use a height of 7 in. and a split of 65 percent on the low pressure side and 35 percent on the high pressure side. With this selection, the estimated overall pressure drop for the complete heat exchanger is 1.77 percent (with fins) or 1.65 percent (without fins). Both these numbers are below the aimed-for 2 percent, but it is believed that this tolerance is desirable to allow for any possible developmental or manufacturing difficulties. The estimated total weight of this unit is 330 lb (with fins) and 303 lb (without fins). Both these weights are based on the use of 0.005 in. thick Hastelloy plates.

LAYOUT DRAWINGS

Layout drawings have been started for the recuperator design selected by NASA. The layout is based on the result of the final

NOTE: 1. Ratio defined in Appendix I
2. Core Geometry
Height=25.46 in., width=25.19 in., length=7.28 in.
3. Fins in ends, 5 per inch
4. No Fins only in low pressure side
5. Height shown for one end only



CALCULATED BY	IRM	8-64	PRESSURE DROP IN TRIANGULAR END SECTIONS	Figure 1
TRACED BY	RJ	8-64		
CHECKED BY	AFA	8-64		
APPROVED BY			AiResearch Manufacturing Division	L-9375
UNIT NO.				Page 4

design study discussed in the previous section. The final layout and detail drawings (Task II) will incorporate all modifications that the small scale test program indicates are necessary for optimum design.

It is during the layout phase of the program that many of the mechanical design and fabrication development problems are first considered. [Some of the main problems with the NASA recuperator are due to its size and weight.] The overall dimensions are approximately 25 in. wide by 25 in. high by 31 in. long with a calculated weight of 303 lbs. Because of the size and weight the core will probably have to be brazed in 2 or 3 sections which will then be welded together. In accordance with the NASA request, provision is also being made so that the manifold pans may be cut off and rewelded in place.

[Several possibilities are being considered in an effort to reduce the weight.] One is the use of hollow header bars for fluid containment. Another is the possibility of using fins only on the high pressure side of the triangular inlet sections. Finally, the use of 0.005 in. thick tube sheets results in minimum tube sheet weight. To show the progress which has been made to date in the preparation of the layout drawing, the drawing is shown in AiResearch Drawing L198005, included with this report. This drawing is very preliminary and only partially completed.

SMALL SCALE TESTING

At this time the only test approved by NASA is the axial conduction test. This test consists of measuring the product kA (k=thermal conductivity, A=effective cross sectional area) on a small section of the recuperator core using the same fins, tube plates, and braze alloy. As was discussed in the small scale test program (AiResearch Report L-9371) analytical procedures are available for estimating the effect on performance of axial conduction, however, the axial conduction parameter is a strong function of kA which is difficult to estimate for the following reasons:

1. The proper cross-sectional area to use with offset fins.

2. The effect on the thermal conductivity of the tube plates and fins due to diffusion of the braze alloy.

The test specimen for this test has been fabricated, using .005 in. thick Hastelloy C plates, and testing will be completed early in September. The test specimen is shown in Figure 2, this figure also shows a section of rectangular offset fins.

The product, kA, will be determined from measurements of the electrical resistance of the test specimens as shown in the test schematic on Figure 2.

The electrical resistance is related to the thermal conductivity by the following relation: $k = \mathcal{L} \sigma T + B$, where,

k = thermal conductivity at temperature T

σ = electrical conductivity at temperature T

T = absolute temperature

$\mathcal{L}$ = Lorenz number

B = lattice conduction constant.

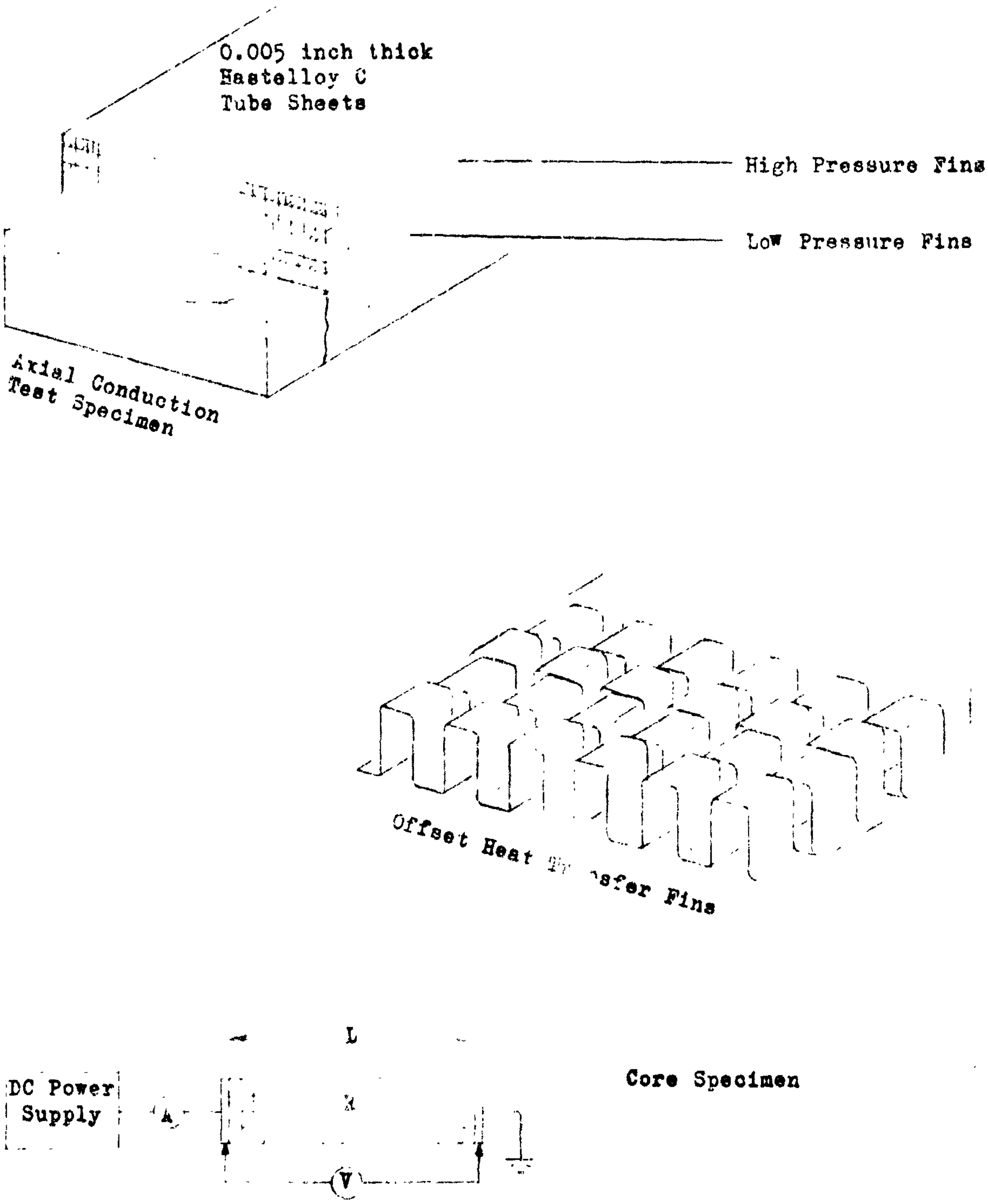
Since the electrical resistance and conductivity are related as follows:

$$R = \frac{L}{\sigma A}$$

It follows that:

$$kA = \mathcal{L} \frac{L}{R} T + B$$

The data on kA is determined from the procedure outlined above will be used to verify the estimates of kA used in the design analysis of the recuperator. Provided kA exp. and kA calc. are equal, it can be concluded that the effect of axial heat leak on the recuperator performance has been correctly estimated. The theory used for this test was take from C. S. Smith & E. W. Palmer, "Thermal & Electrical Conductivities of Copper Alloys", Trans. AIME, 117, (1935) and W. Hume-Rothery, The Metallic State, Oxford (1931).



A-7422

FIGURE 2

TEST SPECIMEN FOR AXIAL CONDUCTION TEST

FUTURE WORK

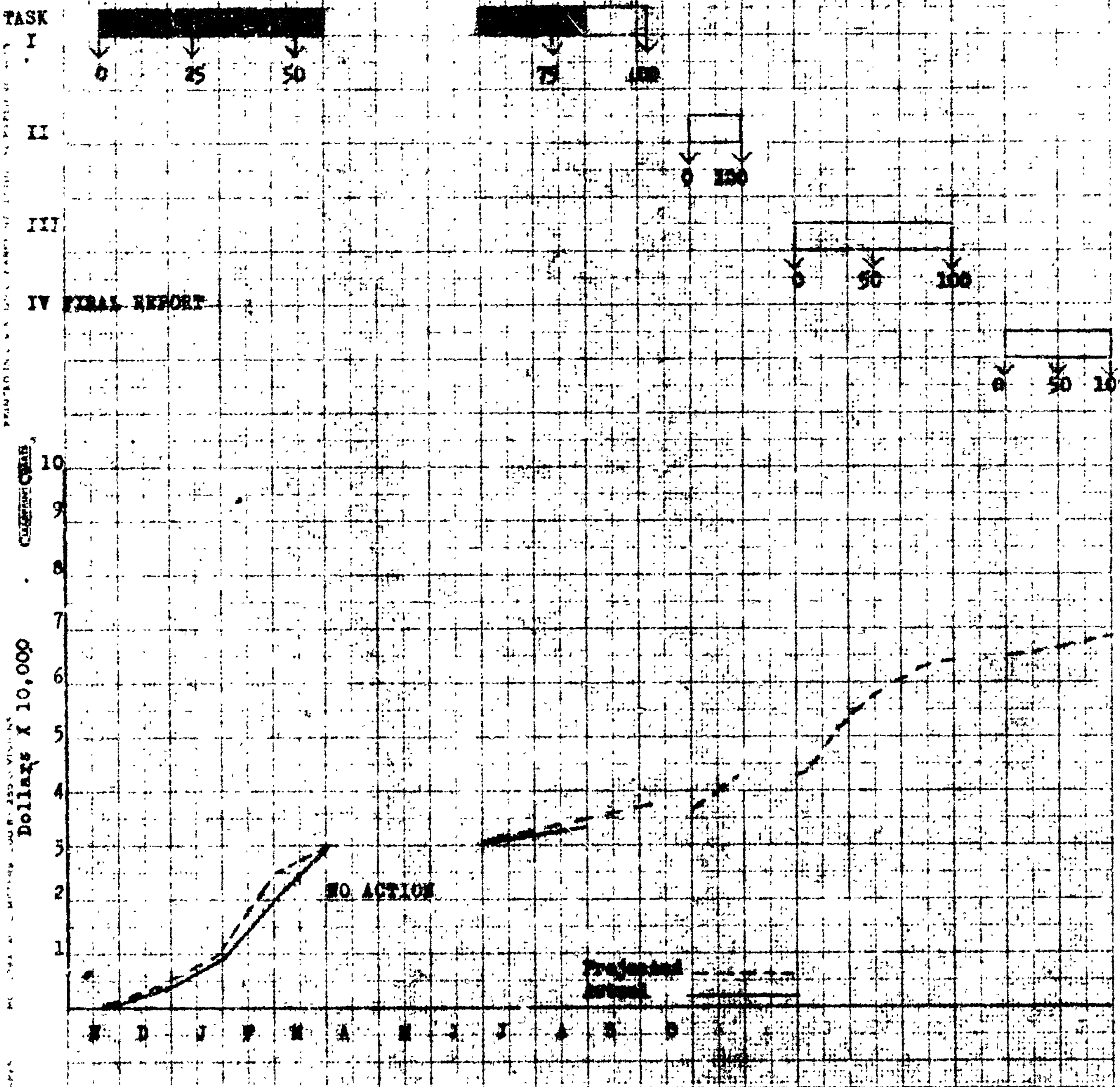
All analytical work in this program has now been completed. During the next reporting period, the majority of the work will concentrate on the preparation of the manufacturing details. This work will progress with the layout drawings and small scale tests. It is hoped that during the next reporting period NASA will make a decision with regard to the flow distribution test.

COST MANAGEMENT

The percentage of the tasks completed are shown in Figure 3. Also shown in this figure is a comparison between the estimated and actual cost of the program to date.

COST MANAGEMENT REPORT

Percentage of Task Completion



CALCULATED BY	NASA-SPACE ENGINES	
TRACED BY	BRAYTON CYCLE HEAT EXCHANGER	
CHECKED BY		
APPROVED BY	AiResearch Manufacturing Division	L-9375
UNIT NO.		Page 9

APPENDIX I

IBM 7074 PROGRAM H-1400

END SECTION PRESSURE DROPS - COUNTERFLOW PLATE-FIN HEAT EXCHANGERS

(AiResearch Report AM-288)

IBM 7074 PROGRAM H1400

END SECTION PRESSURE DROPS - COUNTERFLOW
PLATE-FIN HEAT EXCHANGERS

INTRODUCTION

With the increasing demand for very high effectiveness heat exchangers, the use of pure counterflow plate-fin designs has become fairly widespread. [A computer program, Plate-fin 7B (H1010) has been written to design the heat transfer matrix required to meet a specified problem.] With the sizing of this matrix, the overall design is, however, not completed, as the fluids on both sides of the heat exchanger have to be introduced and removed from the core. As in counterflow heat exchangers, the flow face area is common to both fluids, simple manifolds are not sufficient to accomplish the fluid distribution. Two prime design concepts are available to accomplish the required flow distribution, and these are illustrated in Figure 1. Where the pressure drop available is low, the triangular-shaped ends of Figure 1a are generally preferred, but if pressure drop is not limited, then the rectangular design of Figure 1b may be preferred. For both design concepts, the ends are fabricated as an integral part of the heat transfer matrix. The plates used throughout cover the entire flow passage areas, but the fins used in the end sections need not necessarily have the same configuration as the fins in the counterflow core. Only the fin height must be maintained throughout. In most cases, as the temperature differences in the ends are small and as the flow is almost entirely cross-flow, the heat transfer in these sections is negligible (or is assumed to give extra "safety margin" to the design). The dimensions of the ends must be minimized to reduce heat exchanger weight; however, as the size of the ends are reduced, the pressure drop increases. As the geometry of the ends is not fully fixed by the design of the core matrix, in order to determine the optimum selection for minimum weight and pressure drop, a large number of geometries must be investigated. In order to facilitate this investigation, a computer program has been written to determine the pressure drops in the end sections of a given configuration.

DESCRIPTION OF PROGRAM

As previously mentioned, there are two types of end design, triangular and rectangular, which may be considered. The program will calculate the pressure losses in the four individual ends of either type of design. The approach taken to determining the pressure losses in the two designs is slightly different, and they are described separately below. It must be stressed that at this time, the approach being used is theoretically only. In the triangular designs, there is some indication from a brief series of tests that the approach being used does fairly accurately determine the true losses. However, until a more comprehensive test program is conducted, or at least until some units have been built from this program and tested, the pressure losses obtained must be treated as approximations.

Triangular End Shape Designs

It is first necessary to define the geometry of the ends. Some of this is obtained directly from the design of the counterflow matrix. Information used from the core design includes core width, stackup height, number of passages on both sides, and plate spacing on both sides. In addition, the height (h) of the ends must be defined, together with the number of fins and the fin thickness to be used on both sides of the ends. One further parameter is required to define the end geometry, and the one chosen is the ratio a/w defined on Figure 2. With the end geometry defined, the effective flow width and length in the ends for both the high pressure and low pressure fluids is calculated. These effective dimensions are also defined in Figure 2. With both flow rates and all terminal pressure and temperature conditions of the heat exchanger known, the pressure drops are computed by the following steps. Mass velocity on one side of one end is computed using the appropriate flow and based on the effective flow width. Reynolds number is computed from viscosity (read from curve) and hydraulic diameter of the fin spacing selected (calculated within the program). Friction factor is read from appropriated stored curve (Reynolds number versus friction factor for surface to be considered is part of program input). An entrance shock loss coefficient is determined from the area ratio (end section free-flow area to frontal area) and a stored curve. The curves used by the program are the laminar expansion and contraction coefficients, and the turbulent expansion and contraction coefficients (Re = 3000) of Kays and London Figure 20. If the end section Reynolds number is less then 2000, the laminar curves are used. A second expansion or contraction loss is allowed for between end section and straight core based on the area ratio between end section free-flow area and core free-flow area. At this junction, there is also a turning loss based on the angle θ defined in Figure 2. The coefficient for this turning loss is taken from Figure 14 of SAE 23. The overall pressure loss is then computed from the sum of friction term, the face shock loss, and the velocity head change and turning loss at the junction of the ends with the core. Each of the four ends, low pressure inlet and outlet and high pressure inlet and outlet, are computed separately at the appropriate fluid properties and with the appropriate type of shock loss (expansion or contraction). The two ends of the heat exchanger are identical in the calculations, but if non-similar ends are required, the results obtained from different solutions may be combined.

Rectangular End Shape Design

With this type of design and with the core geometry specified, only the height (h) and the fin characteristics of the ends need be defined. In the straight-through (low pressure) side of the unit, the pressure loss in the ends is computed from a friction term and from a single shock loss based on the free flow to frontal area ratio. In the high pressure side where the flow enters at right angles to its flow path through the core, the pressure loss calculations are rather more complicated. Two velocity heads are computed, one based on entrance and exit areas (that is, based on h), and one based on the second set of fins in the end (uses core width, w). Using the velocity head based on h, a friction term is calculated for the first set of fins, a shock loss for the entrance or exit, and a shock loss (expansion or contraction) from these fins into the second set. A turning loss coefficient, also based on the "h" velocity head, is added to the pressure loss. This coefficient is an input quantity and should normally be based on a 90° turn (1.6 from SAE 23, Figure 14). The turning coefficient was left as an input quantity so that it may be varied at the user's discretion. This approach will be of particular advantage if test data is obtained, as the coefficient may be varied until agreement with the test data is achieved. The velocity head in the second set of fins is used only to compute a friction drop through that section.

INPUT INSTRUCTIONS

A typical input sheet for this program is shown in Figure 3. In order to clarify the sheet, the following instructions have been prepared.

CARD 1. Heading (80 Holerith characters available)

TABLES

Control Card, (with numbers of pairs of points in each of nine tables)

All data stored 8 words per card and uses LAGIN 2.30 pairs of points maximum.

NVIS NVISH FF FFH EXL CØNL EXT CØNT TURN (9I5)

Table 1. Temperature (°R) vs Viscosity (lb/sec ft) for low pressure side fluid.

Table 2. Temperature (°R) vs Viscosity (lb/sec ft) for high pressure side fluid.

Table 3. Reynolds No. vs Friction Factor for low pressure side fins.

Table 4. Reynolds No. vs Friction Factor for high pressure side fins.

Table 5. Area Ratio vs Laminar Expansion Coefficients.

Table 6. Area Ratio vs Laminar Contraction Coefficients.

Table 7. Area Ratio vs Turbulent Expansion Coefficients.

Table 8. Area Ratio vs Turbulent Contraction Coefficients.

(Tables 5–8: Kays and London Figure 20)

Table 9* Turning Angle (sin θ) vs Turning Coefficient (SAE 23 Fig. 14)

*In this table it is the sine of the angle θ that is stored in the program.

CARD 2. Control Card for Other Variables (315)

J1	No. of sets of cards 3 and 4 (5 max)
J2	No. of sets of cards 5 and 6 (12 max)
J3	No. of sets of cards 7 (50 max)

CARD 3.

WIDTH	Heat exchanger core width - in.
ANPL	Total No. of passages on low pressure side of heat exchanger
ANPH	Total No. of passages on high pressure side of heat exchanger
ALN	Stack-up height of heat exchanger, in.
HPL	Plate spacing, low pressure side, in.
HPH	Plate spacing, high pressure side, in.
TEST 1	= 0.0 if triangular ends = 1.0 if rectangular ends
BWL	Ac/bw for fins in counterflow core, low pressure side.

CARD 4.

BWH	AC/bw for fins in counterflow core, high pressure side
WTF	Weight factor, Weight of ends = WTF X Volume
AK	Turning loss coefficient for rectangular ends (Normally = 1.6 rectangular 0.0 triangular)

CARD 5.

WL	Flow rate, low pressure side, lb per sec
WH	Flow rate, high pressure side, lb per sec
TINL	Inlet temperature, L.P. side, ^{o}R
TINH	Inlet Temperature, H.P. side, ^{o}R
TOUTL	Outlet Temperature, L.P. side, ^{o}R
TOUTH	Outlet temperature, H.P. side, ^{o}R
PINL	Inlet pressure, L.P. side, psia
PINH	Inlet pressure, H.P. side, psia

CARD 6.

POUTL	Outlet pressure, L.P. side, psia
POUTH	Outlet pressure, H.P. side, psia
ROEL	Density factor, L.P. side ρ = ROEL $\frac{P,\ psia}{T,\ ^{\circ}R}$
ROEH	Density factor, H.P. side

CARD 7.

HEIGHT	Height of triangle or rectangle, defined in Figure 1.
RATIO	a/w as defined in Figure 1, if greater than 0.5 wider face will be low pressure side, if less than 0.5 wider face will be high pressure side.
ANFL*	No. of fins per inch, in low pressure ends.
ANFH	No. of fins per inch, in high pressure ends.
TFL	Fin thickness of fins in low pressure ends, in.
TFH	Fin thickness of fins in high pressure ends, in.

*If desired to look at zero fins on either side ANF = 0.0 and also TF = 0.0. Care should be taken in this case to be sure that friction factor is for flow between flat plates.

OUTPUT CLARIFICATION

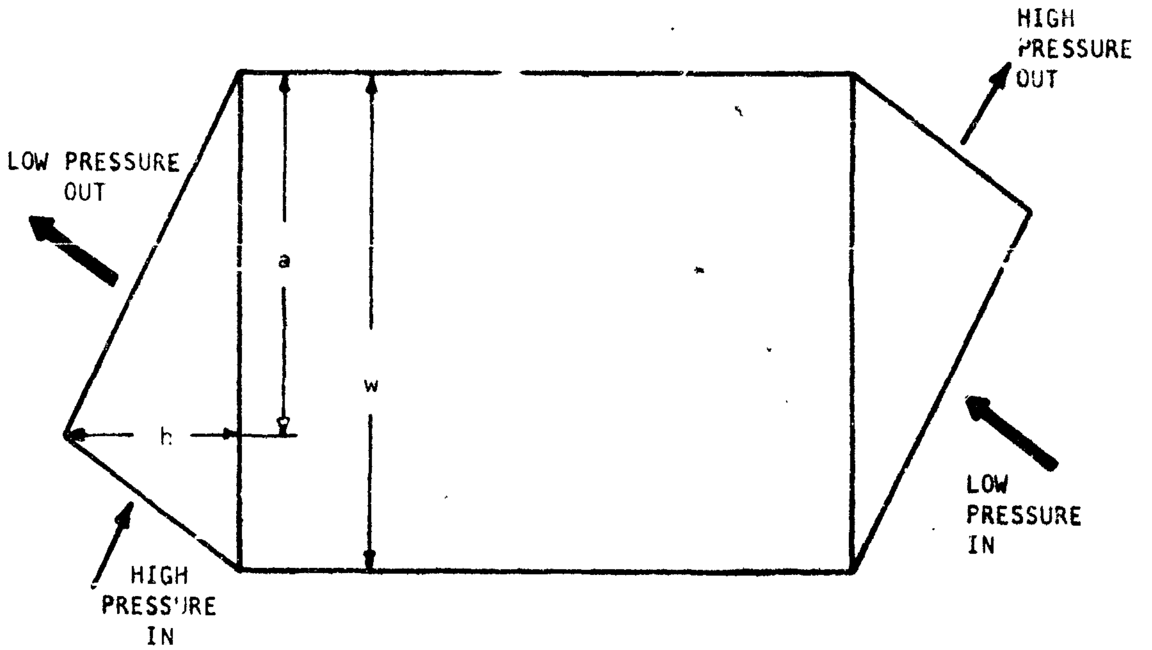
A typical output sheet is attached as Figure 4. The first line of data shows the flow, temperature, pressure data being examined. The second line defines the end geometry being examined while the third line identifies the counterflow core.

Line 4 presents the solutions where all four end pressure drops are shown both in psia and as a percentage of the INLET pressure on the appropriate side.

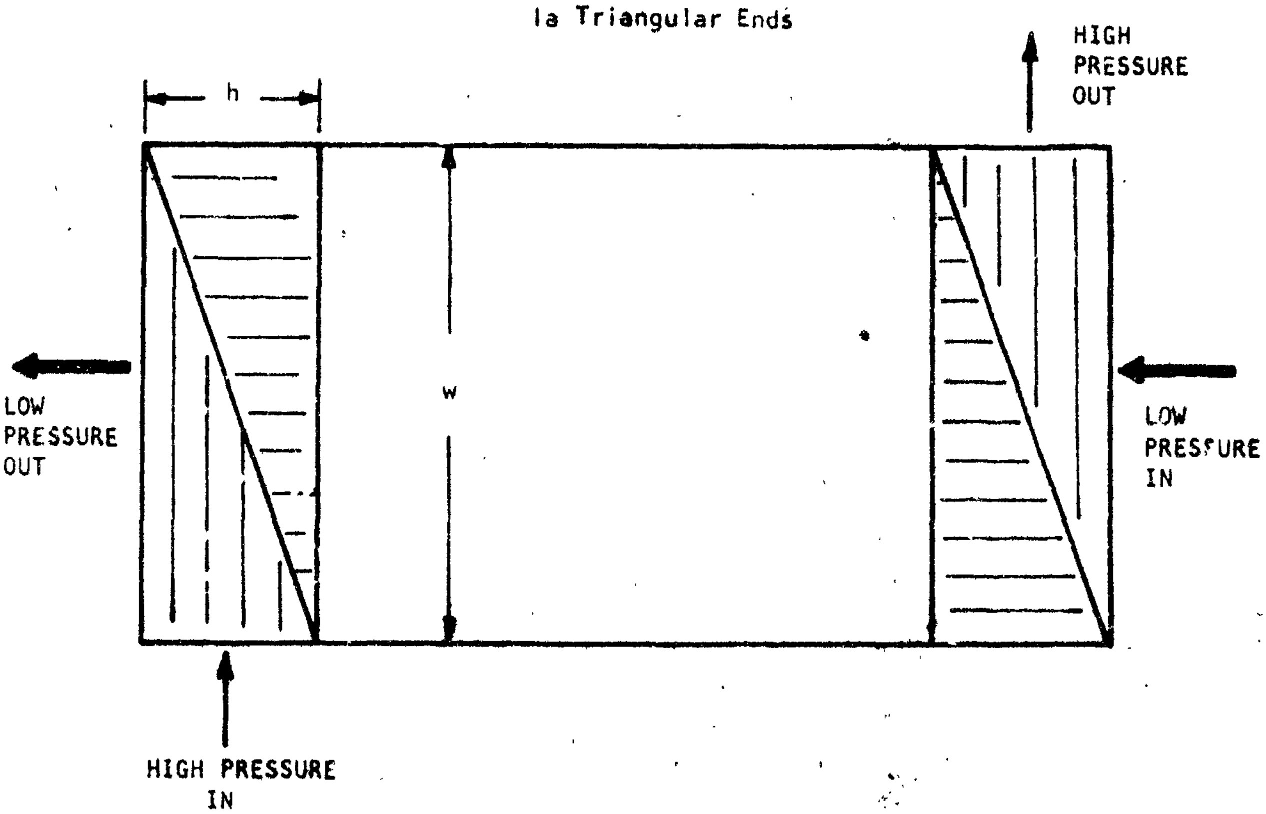
Line 5 presents additional information including volume of <u>one</u> end, weight of <u>one</u> end and the length of the two sides of the triangular ends. If rectangular ends the number under "dimensions" are the width of the core and the height of the ends. Also shown in Line 5 are the hydraulic radii in the ends and the Reynolds numbers in the ends.

NOTE:

At the time of writing a minor error in output format exists in Line 3 where the passage heights are shown as *153 and *178 this should read 0.153 and 0.178 and will be corrected if the program is recomplished for some other reason.

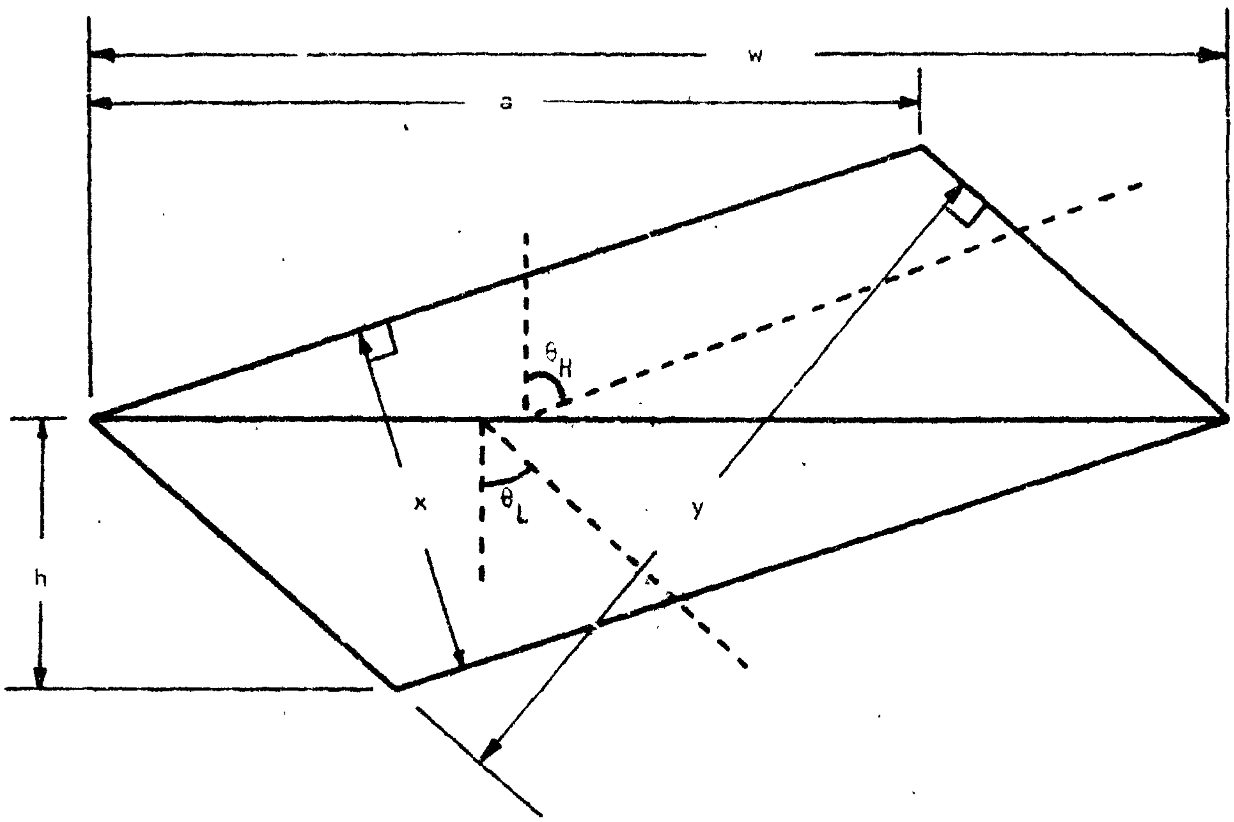


1a Triangular Ends



1b Rectangular Ends

Figure 1. Counterflow Design Concepts



EFFECTIVE FLOW WIDTH (LP) = y
EFFECTIVE FLOW WIDTH (HP) = x
EFFECTIVE FLOW LENGTH (LP) PER END = x/2
EFFECTIVE FLOW LENGTH (HP) PER END = y/2
TURNING ANGLE (LP) = θ_L } SIN θ IS USED IN PROGRAM
TURNING ANGLE (HP) = θ_H }

NOTE: SKETCH SHOWS TWO ENDS OF TRIANGULAR DESIGN TOGETHER, WITH CORE REMOVED

Figure 2. Triangular End Geometry

IBM 7074 PROGRAM H 1400

END SECTIØN PRESSURE DRØPS - CØUNTERFLØW PLATE-FIN HEAT EXCHANGERS

CHARGE No.		DATE		RUN TIME	
NAME		EMPLOYEE No.			

Card								
1	HEADING CARD (80 Holerith columns)							
	INSERT TABLES							
2	J1 (5)	J2 (10)	J3 (15)	(3I5) Control Card				
3	WIDTH	ANPL	ANPH	ALN	HPL	HPH	TEST I	BWL
4	BWH	WTF	AK					
5	WL	WH	TINL	TINH	TØUTL	TØUTH	PINL	PINH
6	PØUTL	PØUTH	RØEL	RØEH				
7	HEIGHT	RATIØ	ANFL	ANFH	TFL	TFH		

Figure 3. Input Sheet

```
FIRST TRY  END DESIGN COUNTERFLOW HEAT EXCHANGERS   ANDERSEN 8/5/64
                                                                                  R  1 0.15600E 04
                                                                                  R 21 0.14860E 04
            LOW  PRESSURE  SIDE                                              HIGH  PRESSURE  SIDE
FLOW     INLET TEMP  OUTLET TEMP  PRESS IN   PRES OUT           FLOW    INLET TEMP  OUTLET TEMP  PRESS IN  PRES OUT
LB/SEC      R            R         PSIA        PSIA            LB/SEC      R            R         PSIA      PSIA
 0.6120   1560.000     875.000      6.730       6.960          0.6120    901.000     1486.000    13.800    13.730

HEIGHT,IN    RATIO    TEST1   RECT.TURN CON  NO.FIN(LP)  NO.FIN(HP)  FIN THICK(LP),IN  FIN THICK(HP),IN
  9.000     0.5000    0.0      [illegible]     0.00        5.00         0.0000           0.0040

WIDTH,IN=23.2600    NO.FLOW,IN= 23.290    PASSAGE HT.(LP)=.178    PASSAGE HT.(HP)=.153

        LOW PRESSURE SIDE,PRESSURE DROPS                               HIGH PRESSURE SIDE,PRESSURE DROPS
INLET,PSIA  OUTLET,PSIA  INLET,PERCENT  OUTLET,PERCENT    INLET,PSIA  OUTLET,PSIA  INLET,PERCENT  OUTLET,PERCENT
  0.0067      0.0028       0.0996          0.0414           0.0081      0.0225       0.0586          0.1631

VOLUME,CU.IN  WEIGHT,LB   DIMENSIONS,IN   L.P. RH,FT  H.P. RH,FT  IN RE (LP)  OUT RE (LP)  IN RE (HP)  OUT RE (HP)
 2437.7643    2437.7643   14.706 14.706   0.007417    0.003527     351.47      522.20       646.71      422.50
                           * * * * * * * * * * * * * * * * * * * *
                                                                                  R  1 0.15600E 04
                                                                                  R 21 0.14860E 04
            LOW  PRESSURE  SIDE                                              HIGH  PRESSURE  SIDE
FLOW     INLET TEMP  OUTLET TEMP  PRESS IN   PRES OUT           FLOW    INLET TEMP  OUTLET TEMP  PRESS IN  PRES OUT
LB/SEC      R            R         PSIA        PSIA            LB/SEC      R            R         PSIA      PSIA
 0.6120   1560.000     875.000      6.730       6.960          0.6120    801.000     1486.000    13.800    13.730

HEIGHT,IN    RATIO    TEST1   RECT.TURN CON  NO.FIN(LP)  NO.FIN(HP)  FIN THICK(LP),IN  FIN THICK(HP),IN
  9.000     0.6500    0.0        0.000         0.00        5.00         0.0000           0.0040

WIDTH,IN=23.2600    NO.FLOW,IN= 23.290    PASSAGE HT.(LP)=.178    PASSAGE HT.(HP)=.153

        LOW PRESSURE SIDE,PRESSURE DROPS                               HIGH PRESSURE SIDE,PRESSURE DROPS
INLET,PSIA  OUTLET,PSIA  INLET,PERCENT  OUTLET,PERCENT    INLET,PSIA  OUTLET,PSIA  INLET,PERCENT  OUTLET,PERCENT
  0.0047      0.0020       0.0702          0.0290           0.0118      0.0321       0.0855          0.2327

VOLUME,CU.IN  WEIGHT,LB   DIMENSIONS,IN   L.P. RH,FT  H.P. RH,FT  IN RE (LP)  OUT RE (LP)  IN RE (HP)  OUT RE (HP)
 2437.7643    2437.7643   17.595 12.136   0.007417    0.003527     290.05      430.94       773.78      505.51
                           * * * * * * * * * * * * * * * * * * * *
                                                                                  R  1 0.15600E 04
                                                                                  R 21 0.14860E 04
            LOW  PRESSURE  SIDE                                              HIGH  PRESSURE  SIDE
FLOW     INLET TEMP  OUTLET TEMP  PRESS IN   PRES OUT           FLOW    INLET TEMP  OUTLET TEMP  PRESS IN  PRES OUT
LB/SEC      R            R         PSIA        PSIA            LB/SEC      R            R         PSIA      PSIA
 0.6120   1560.000     875.000      6.730       6.960          0.6120    801.000     1486.000    13.800    13.730

HEIGHT,IN    RATIO    TEST1   RECT.TURN CON  NO.FIN(LP)  NO.FIN(HP)  FIN THICK(LP),IN  FIN THICK(HP),IN
  9.000     0.7500    0.0        0.000         0.00        5.00         0.0000           0.0040
```

Figure 4. Typical Output