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final report

TECHNICAL APPLICATION AND FEASIBILITY ASSESSMENT OF SCROLL COMPRESSOR/ EXPANDER FOR AIR CYCLE COOLING OF COMBAT VEHICLES

by

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ARTHUR D. LITTLE, INC.
CAMBRIDGE, MASS. 02140

NOVEMBER 1981

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An air cycle cooling system utilizing a Scroll compressor and expander was sized and its performance estimated. The design produced appears to have advantages and merits further investigation relative to practicality and verification of performance projections.

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FOREWORD

This work was performed under the sponsorship of the U.S. Army Mobility Equipment Research and Development Command (MERADCOM). The work was performed by Arthur D. Little, Inc. of Cambridge, MA 02140, under T.O. 0022 of contract number DAAK70-79-D-0036. Mr. John E. McCullough and John T. Dieckmann were the authors. This report was prepared under the guidance of Mr. Robert Rhodes of MERADCOM as the Technical Point of Contact and Mr. Jerry Dean of MERADCOM as the Contract Officer's Representative.

PREFACE

Arthur D. Little, Inc., has been working to develop Scroll equipment technology for the past 9 years. During this period, 16 separate Scroll prototype equipment development programs have been completed and four major companies have licensed this technology for commercial products. The list of clients sponsoring Scroll technology includes the U. S. Government, U. S. commercial clients, and a Japanese company; the total sponsorship of development work exceeds 3.0 millon dollars, to date.

It is our policy to actively pursue as many applications for this technology as are feasible and to seek licensees who can commercialize these hardware developments.

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Summary

Purpose

The purpose and components of this work task are given below, as they appear in the Description of Task paragraph in the contract "Statement of Work and Services":

The objective of this task is to conduct an analysis of Scroll air compressor and expander technology to establish the feasibility and potential advantages of applying this technology to air cycle (simple reverse Brayton cycle) cooling requirements in combat vehicles. In particular, an air cycle cooling system for this application must have the following relative merits:

- improved C.O.P. (minimum power input)
- reduced size
- reduced weight

This work would be undertaken in the following tasks:

Task 1. Perform preliminary analysis of air cycle to establish component requirements such as compressor air flow rate, pressure ratio, and air temperatures throughout the system.

Task 2. Perform preliminary sizing of the Scroll compressor and expander components to meet cycle requirements from Task 1 above.

Establish volumetric, mechanical, and overall isentropic efficiency of Scroll compressor and expander components to estimate motor power requirements.

Estimate Scroll compressor and expander size and weight.

Task 3. Optimize overall system to achieve maximum compliance with improvement criteria cited above. This task will consist of reworking analyses in tasks 1 and 2 to arrive at the best possible system.

Task 4. Document optimum air cycle system by showing size and general configuration of the system components as well as their probable arrangement, including internal duct and heat rejection air cooling fan.

Task 5. Write and submit a draft final report with recommendations.

Task 6. Meet at MERADCOM to discuss Government comments and finalize a report.

Conclusions

The feasibility study described within has resulted in an air cycle water chilling system with the following characteristics:

- COP = .60 (delivered cooling ÷ total electric power consumption)
- size = 26" x 14" x 11" (2.32 ft^3)
- weight = 120 lbm

The configuration is shown on Figures 13 and 14. The major advantages of this unit are estimated to be:

- Compact size due to compact size of Scroll components and close packaging of the individual components
- High overall COP, due to high efficiency of Scroll components
- Moderate weight

- Reliable, rugged mechanical design - similar to Scroll automotive refrigerant compressor

- Quiet, vibration free operation - all rotating components dynamically balanced.

Recommendations

ADL recommends that a breadboard prototype Scroll compressor/expander unit be designed, fabricated and tested to evaluate the practicality of this type of apparatus and measure actual performance efficiencies.

1.0 Air Cycle System Tradeoff Analysis

The recommended air cycle system configuration was the result of the following analytical and design steps:

- Preliminary air cycle analysis
- Preliminary component sizing
- Component and system optimization

A flow diagram of the air cycle cooling system is shown in Figure 1. Air at atmospheric pressure is compressed by the Scroll compressor. The hot compressed air is cooled in the air-to-air heat exchanger and expanded to near atmospheric pressure in the Scroll expander. The mechanical power recovered in the expander provides a portion of the power required to drive the compressor. The cold expander discharge air passes through the air-to-water heat exchanger, absorbing heat from the cooling water, and returns to the compressor inlet. Cooling water passing through the air to water heat exchanger is chilled to 50°F.

The design requirements of the system are summarized in Table 1. The performance characteristics of components, shown in Figure 1 are unspecified. For the purposes of system trade-off and design studies, the performance of each component of the air cycle system was modeled or characterized in terms of parameters shown in detail in Table 2. For the system shown in Figure 1, performance parameters, such as temperatures, pressures, air flow rate, and COP can be predicted and system weight and volume estimated. The objective of the trade-off and design studies was to determine the combination of component performance characteristics that provide the lowest system volume, and weight, and highest coefficient of performance (low electric power consumption).

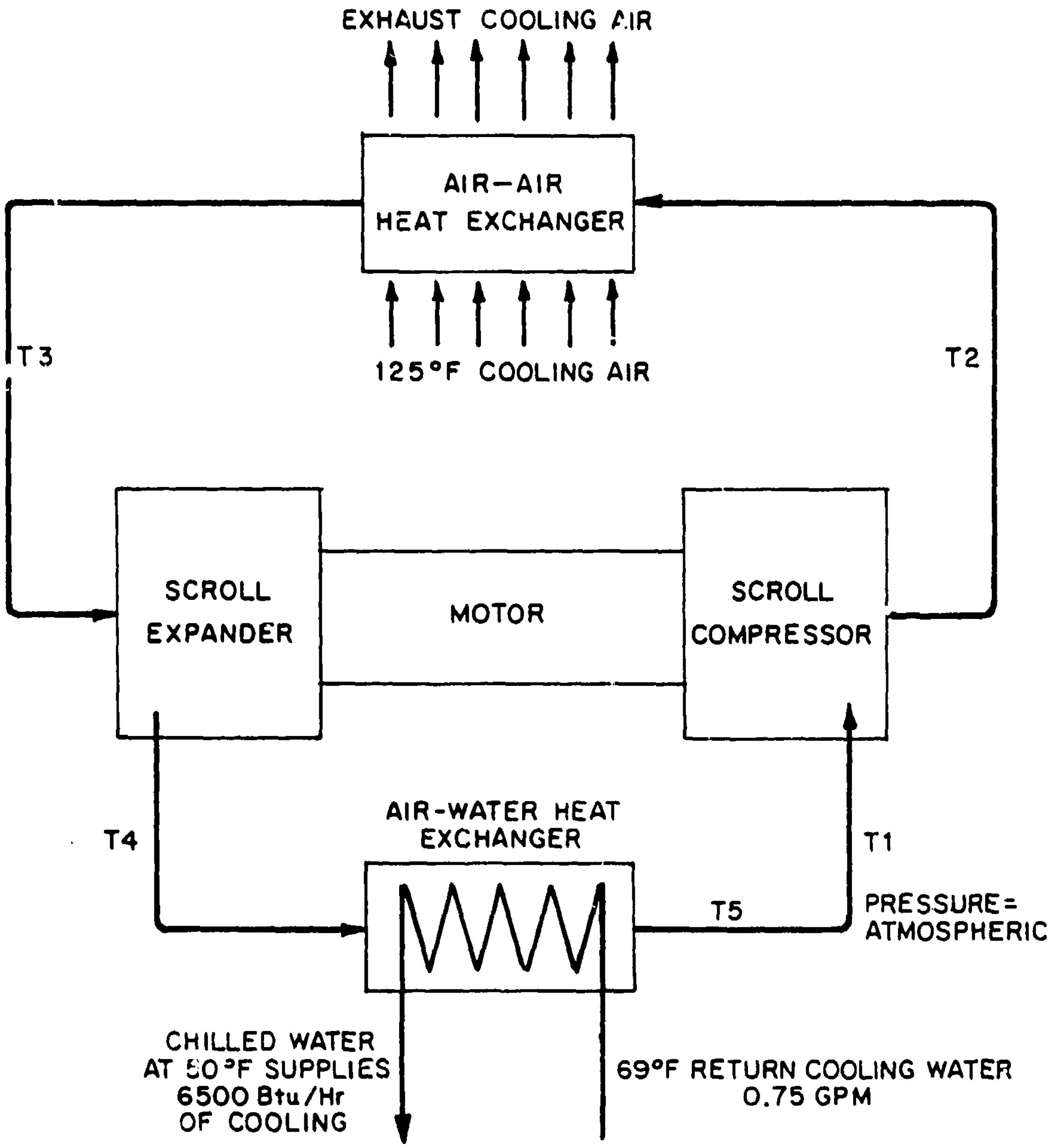


FIGURE 1 FLOW DIAGRAM OF SCROLL COMPRESSOR AND EXPANDER BASED AIR CYCLE COOLING SYSTEM.

Table 1 - Design Goals and Specifications

Design Goals:

- improved C.O.P. (minimum power input)
- reduced size
- reduced weight

The preliminary specification for the desired cooling system is:

- cooling capacity: 6,500 BTU/hr
- a closed cycle system is preferred
- the system to be analyzed will consist of the following components:
    - Scroll compressor and drive motor
    - air to air heat exchanger for heat rejection
    - fan, rejection heat exchanger w/motor
    - Scroll expander (connected to a Scroll compressor drive motor)
    - air to liquid heat exchanger to provide chilled water for cooling purposes
    - pump, chilled water w/motor
- motor speed -- that which will contribute to maximum overall system efficiency
- electric power - 28 VDC
- pressure ratio: approximately 2.7:1
- heat rejection sink temperature: 125°F
- chilled water temperature 50°F
- chilled water flow -- 0.75 gpm
- pressure drop, chilled water flow external to the air cycle chilling unit -- 15 psi

Table 2 - Component Characterization

Component	Performance parameters used
Compressor	Overall isentropic efficiency, pressure ratio
Expander	Overall isentropic efficiency, pressure ratio
Air to air heat exchanger	Heat transfer effectiveness in terms of cycle air side, $\Delta P/P_{absolute}$ of cycle air side
Water to air heat exchanger	Heat transfer effectiveness in terms of cycle air side, $\Delta P/P_{absolute}$ of cycle air side

The trade-off and design studies were conducted in the sequence listed above. In the preliminary air cycle analysis, an estimate was obtained of the range of the required air flow rates, component performances, and cycle pressure ratio. Based on this, preliminary component sizing was performed, providing a basis for developing weight-volume-performance relationships for the individual components. Component and system optimization used these component weight-volume-performance relationships. Each of the three steps is presented in detail in the following sections.

1.1 Preliminary Air Cycle Analysis

In this task, cycle calculations were performed to determine the effect on COP and the air flow rate required to obtain 6,500 Btu/hr cooling capacity for different combinations of heat exchanger effectiveness, pressure drop, compressor and expander efficiencies, and cycle pressure ratio. Cycle calculations were performed on the basis of an adiabatic closed cycle system.

Performance was calculated from the following relationships (T_1, T_2, T_3, T_4, T_5 as shown in Figure 1):

$$T2 = T1 + (T1_{absolute})\,(P_{ratio}{}^{.286} - 1)/\eta_c$$

$$T3 = T2 - \varepsilon_{air-air}\,(T2 - TSINK)$$

$$T4 = T3 - \eta_e(T3_{absolute}) \Big/ \left\{ \left[P_{ratio}\,(1 - \Delta P/P_{air-air})\,(1 - \Delta P/P_{air-water}) \right]^{.286} - 1 \right\}$$

$$T5 = T4 + \varepsilon_{air-water}\,(T_{water} - T4)$$

$$T1 = T5$$

$$\text{Cooling Capacity} = \dot{m}\,C_p\,(T5 - T4)$$

$$\text{Motor Shaft Power} = \dot{m}\,C_p\,\;(T2 - T1) - (T3 - T4)$$

The variables used in these equations that are not defined in Figure 1 are:

- P_{ratio} = pressure ratio of compressor
- ε = heat exchanger effectiveness

- η_c = compressor overall efficiency

- η_e = expander overall efficiency

- $\Delta P/P_{inlet}$ = heat exchanger pressure loss on the air cycle side inlet

- $\dot{m}$ = air mass flow rate through the cycle

- C_p = the specific heat of air at constant pressure, .24 Btu/lbm - ^oF

- T_{water} = 69°

- TSINK = 125°F

A computer program was written to perform these cycle computations. A typical output of the program is shown in Figure 2. The program was run for systems made up of a variety of individual component characteristics. The range of component performance characteristics considered was:

- Pressure ratios between 2.5 and 4.5

- Compressor and expander overall isentropic efficiencies between 80% and 90%

- Heat exchanger heat transfer effectivenesses between 80% and 90%

- Heat exchanger air cycle side pressure losses between .1% and 3% of inlet pressure

The results are shown in Figures 3 through 6. Figure 3 shows the COP to be quite sensitive to compressor and expander efficiency and mildly sensitive to the pressure ratio for a given level of compressor and

P1	TSINK	TWATRT	BTUH	RHO	CP	POLYN	PRATIO
14.700	125.000	69.000	6500.0	0.0762	0.2400	1.4000	3.2000
COMPEF	EXPEFF	ARHXEF	WTHXEF	ARHXDP	WTHXDP		
0.8750	0.8750	0.9000	0.9000	0.0050	0.0050		
CMPWRK	EXPWRK	NETWRK	COOLNG	MSSFLW	CFM	MTRPWR	COP
56.391	35.494	20.897	16.144	402.62	88.04	3.303	0.773
T1	T2	T3	T4	T5	HTREJ		
61.53	296.49	142.15	-5.74	61.53	14913.8		

TWATRT = Water return temperature, i.e. water inlet temperature to air-water heat exchanger

POLYN = Polytropic coefficient

CMPWRK, EXPWRK. NETWRK, COOLNG in Btu/lbm

MSSFLW in lbm/hr

MTRPWR in horsepower

COP = delivery cooling ÷ compressor/expander shaft power

HTREJ = heat rejected in air-air heat exchanger, in Btu/hr

Figure 2 - Sample output of results of AIRCYCLE Computer Program used for preliminary analysis of Air Cycle System

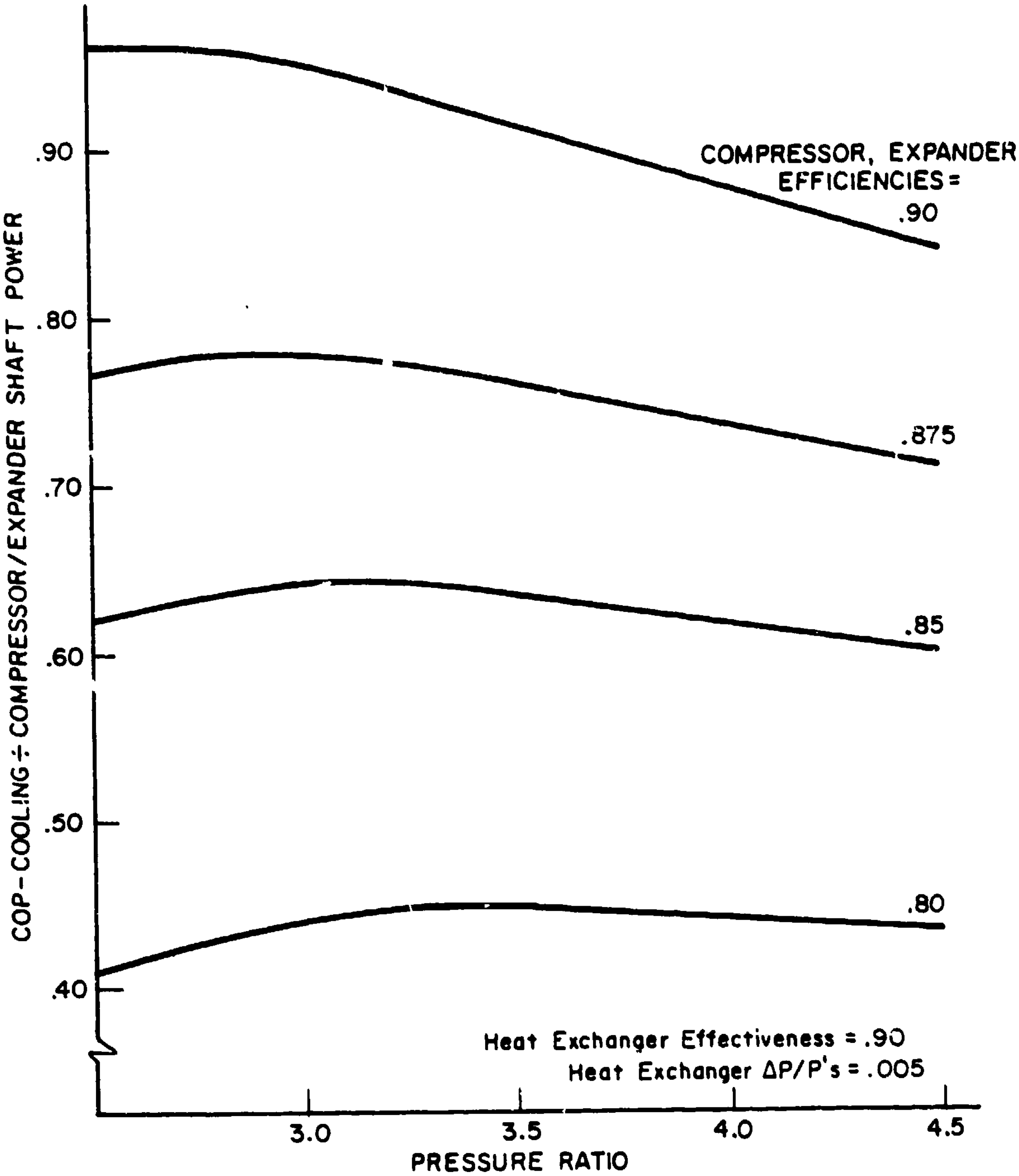


FIGURE 3 EFFECT OF PRESSURE RATIO AND COMPRESSOR AND EXPANDER EFFICIENCIES ON COP.

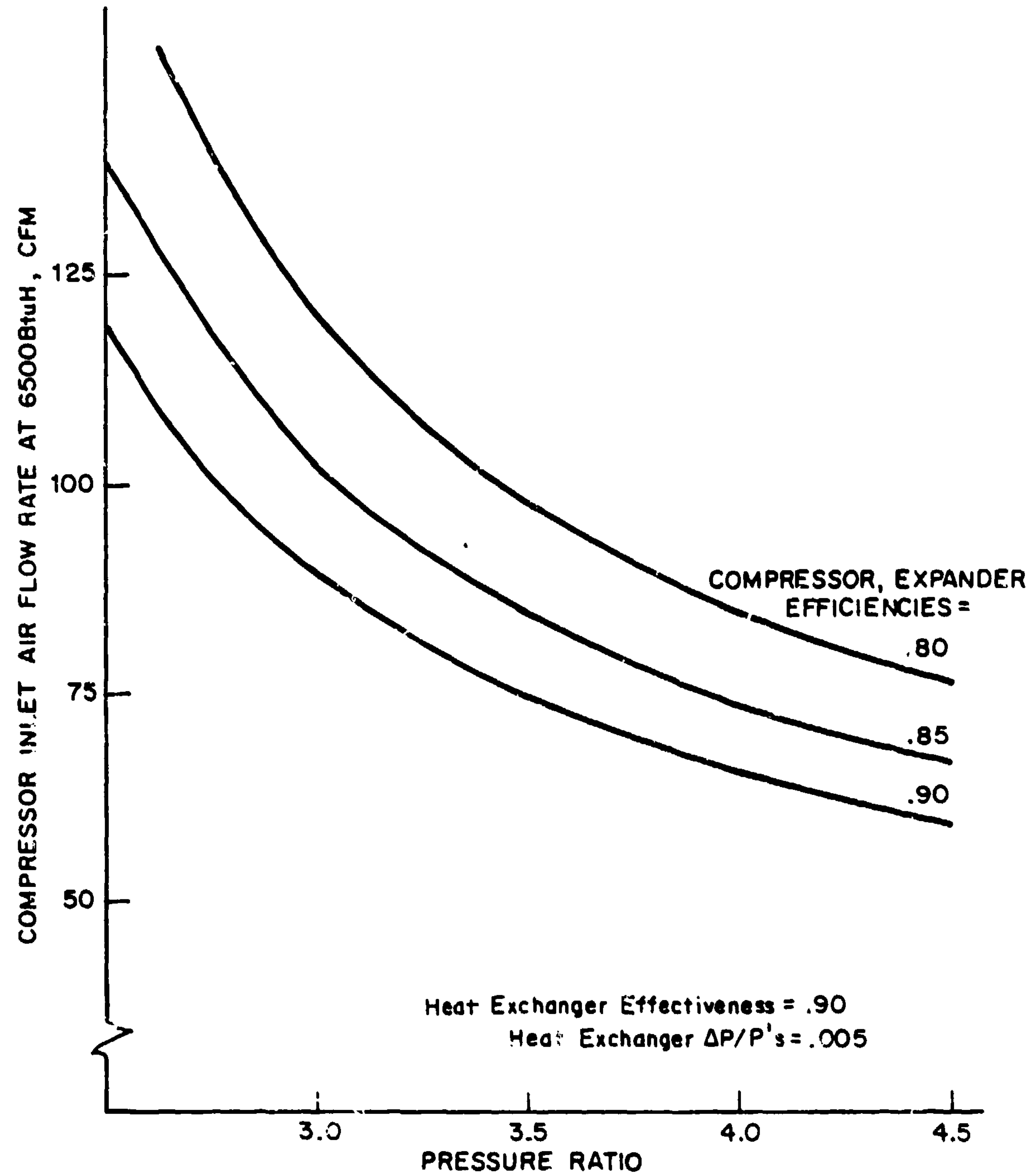


FIGURE 4 EFFECT OF PRESSURE RATIO AND COMPRESSOR/EXPANDER EFFICIENCIES ON CYCLE AIR FLOW RATE FOR 6500 Btu/Hr COOLING CAPACITY.

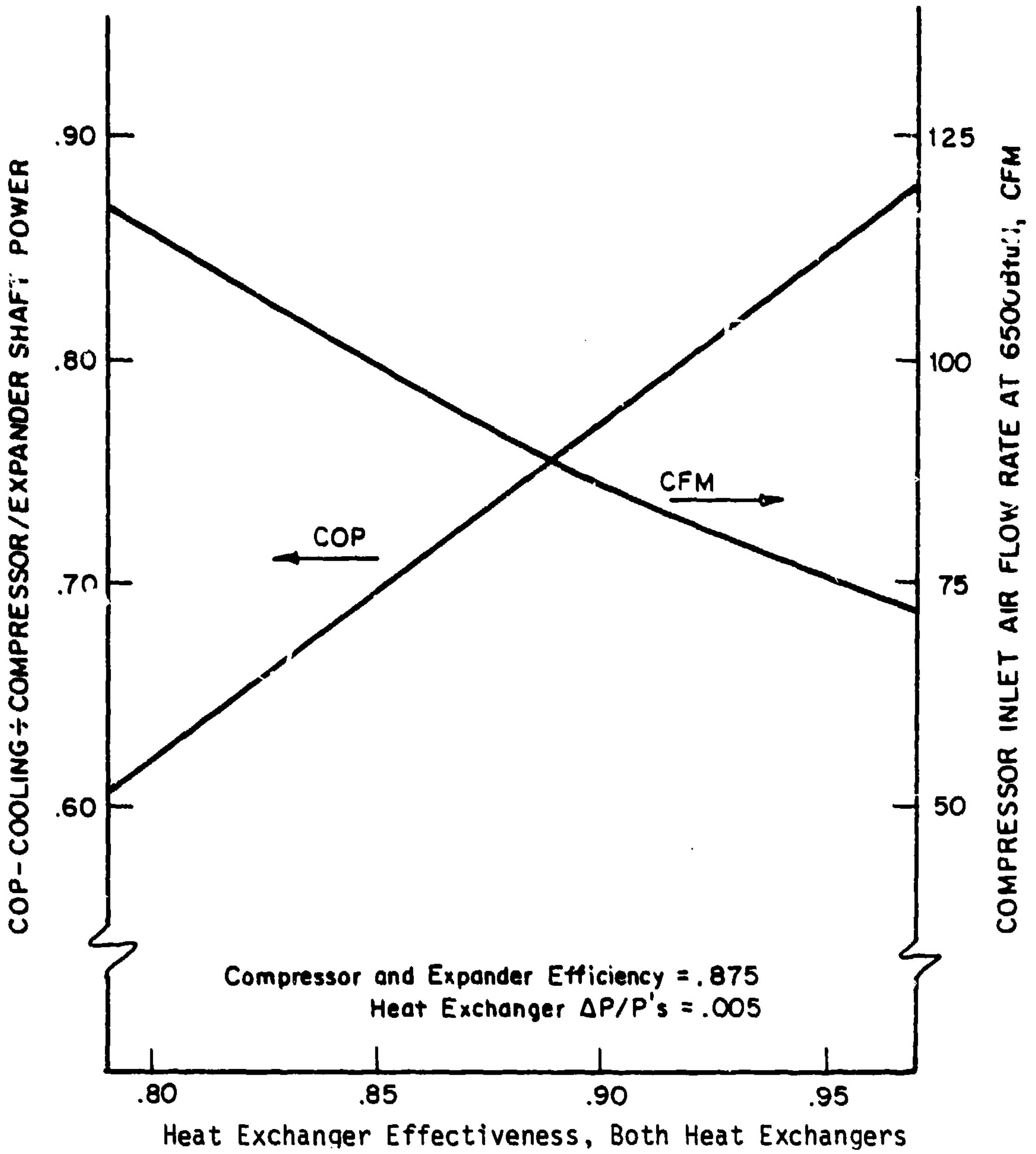


FIGURE 5 EFFECT OF HEAT EXCHANGER HEAT TRANSFER EFFECTIVENESS ON COP AND CYCLE AIR FLOW RATE.

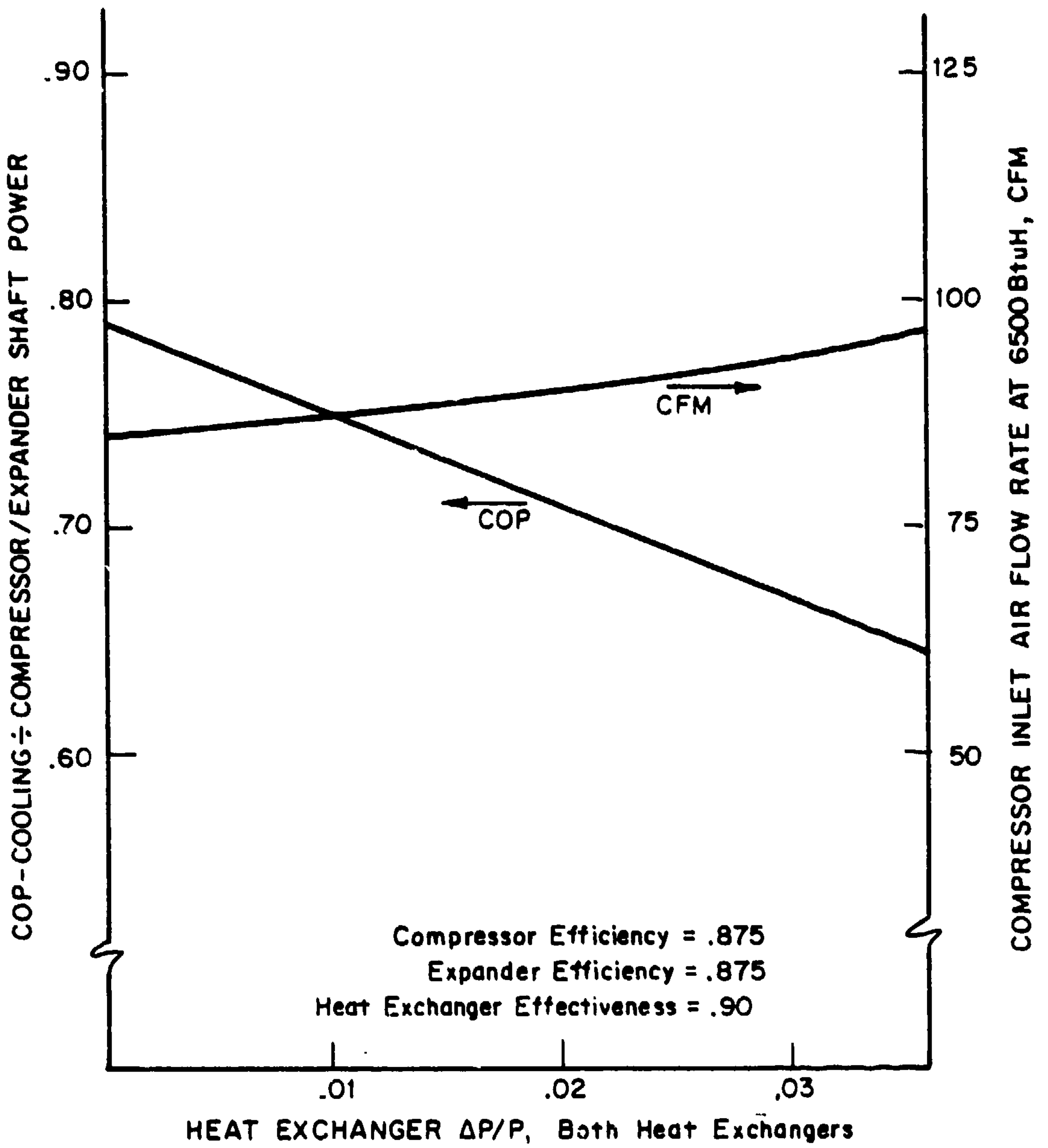


FIGURE 6 EFFECT OF HEAT EXCHANGER PRESSURE LOSS ON COP AND CYCLE AIR FLOW RATE.

expander efficiency. Maximum COP is attained at pressure ratios near 3.0, with relatively small reductions as the pressure ratio is increased. Figure 4 shows a decrease in the cycle air flow rate required to obtain a cooling capacity of 6500 Btu/hr with increasing compressor and expander efficiency. The air flow rate also decreases substantially as the pressure ratio is increased, with constant compressor and expander efficiency. A pressure ratio of 3.25 provides near maximum COP. Preliminary estimates of Scroll compressor and expander efficiencies indicated that efficiencies of .875 are attainable and that the best single stage, clearance sealed Scroll component efficiencies will be obtained at pressure ratios near 3.

Figure 5 shows a significant increase in COP and decrease in cycle air flow rate with increasing heat transfer effectiveness of the heat exchanger. Figure 6 shows a significant decrease in COP with increasing heat exchanger pressure loss. High performance heat exchangers, therefore, tend to benefit the system in terms of both reduced system air flow (and component volume) and increased COP.

1.2 Preliminary Component Sizing

The objective of this task was to make a preliminary specification of system component operating characterisitics providing a reasonable compromise between minimum volume and maximum COP. The volume of system components tends to increase with increasing cycle air flow rate, with increasing heat exchanger effectiveness, and with decreasing heat exchanger pressure drop. The following combination of components and operating conditions was selected with the expectation of reaching a reasonable compromise between maximum COP and minimum component volume:

- Pressure ratio = 3.2
- Heat exchanger effectiveness = 92%
- Heat exchanger $\Delta P/P$'s = .005
- Compressor/expander efficiencies = 87.5

An air cycle system based on this set of component performances was sized under the preliminary component sizing task, described below.

The Scroll compressor and expander and the heat exchangers, were sized on the basis of system and component performance characteristics defined by the preliminary analysis of the air cycle. These component performance requirements are listed in Table 3. As described in the following sections, each of these components was sized to meet these requirements. In addition, the relationship between volume and performance was developed for each component for use in the system optimization task.

Table 3 - System and Component Performance Characteristics Used as the Basis for Preliminary Component Design

Component	Performance Characteristic
Complete System	Pressure ratio = 3.2 Cooling capacity = 6,500 BTU/hr Air mass flow rate = 6.19 lbm/min
Compressor	Overall isentropic efficiency 87.5% Inlet volume flow rate 82.5 ACFM Built in volume ratio 2.29
Expander	Overall isentropic efficiency 87.5% Discharge vc' ne flow rate 70 ACFM Built in volume ratio 2.25
Air-Air Heat Exchanger	Heat transfer effectiveness 92% Pressure loss/inlet pressure .005
Air-Water Heat Exchanger	Heat transfer effectiveness 92% Pressure loss/inlet pressure .005

1.3 Scroll Compressor and Expander Design

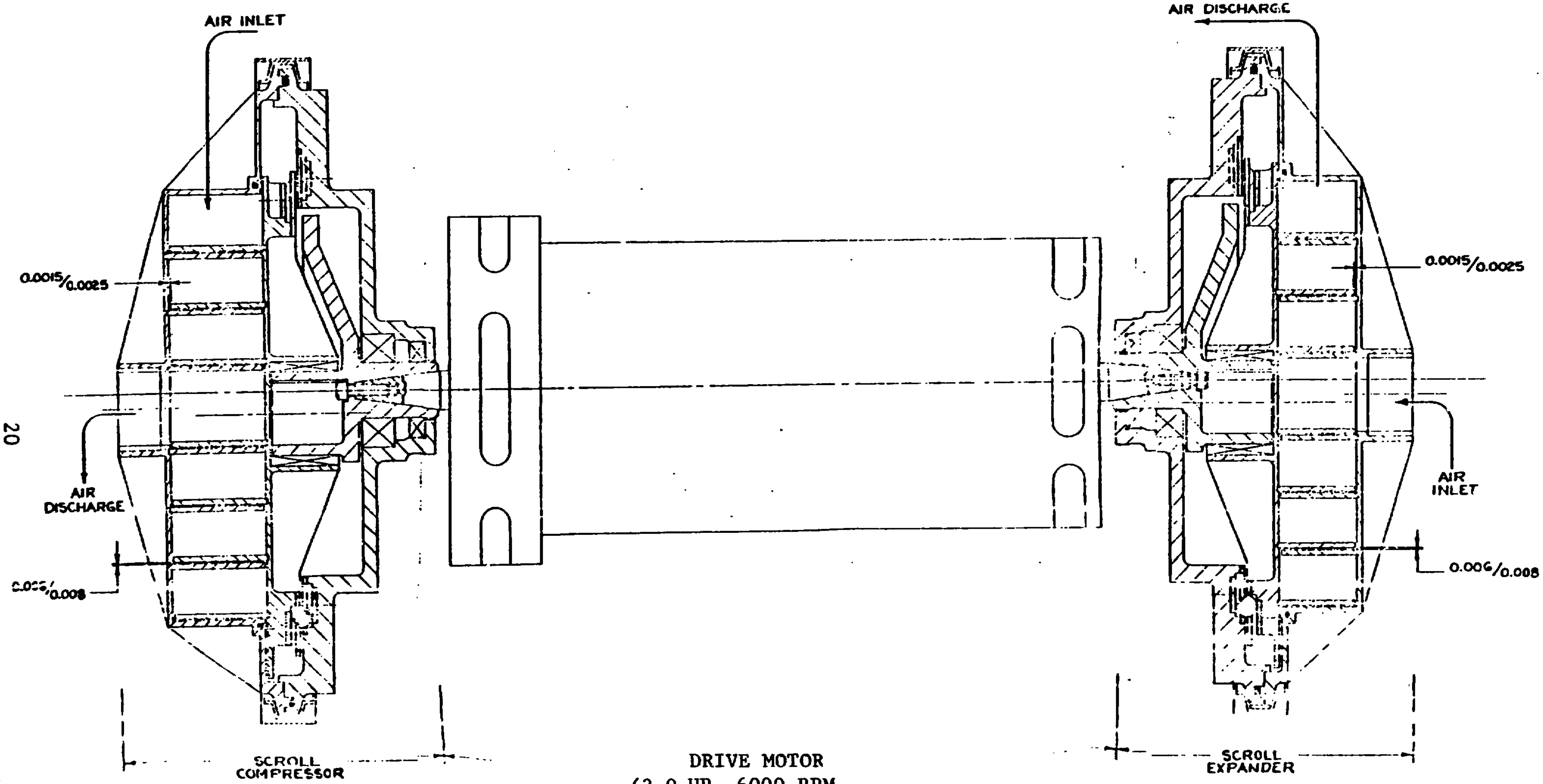
The preliminary design requirements of the compressor and expander are:

- mass flow rate = 6.2 lbm/min
- compressor inlet volume flow rate = 82.5 ACFM
- expander discharge volume flow rate = 70.0 ACFM
- compressor pressure ratio = 3.20
- expander pressure ratio = 3.17
- compressor and expander efficiencies: $\geq$ 87.5

To meet the Table 1 air cycle system performance requirements, the following compressor and expander performance characteristics must be achieved:

- High volumetric efficiency
- High mechanical efficiency
- High overall efficiency
- Low volume
- Oil free operation of the compressor and expander Scrolls (elements wetted by the cycle air)

The compressor-expander design approach shown in Figure 7 incorporates features which should provide these characteristics.



DRIVE MOTOR
(3.0 HP, 6000 RPM
THRU-VENT, EXPLOSION PROOF MOTOR,
WESTINGHOUSE ELECTRIC CORP.
MODEL 915F-614 SHOWN)

FIGURE 7

Scroll Compressor and Expander

The Scroll compressor and expander are integrated with the drive motor to minimize package volume, weight and mechanical losses. Each orbital drive mechanism is oil lubricated and uses rolling element bearings throughout, minimizing friction. The 6000 RPM operating speed is slow enough to moderate centrifugal force loads on bearings, and minimize breathing losses, but sufficiently fast to minimize leakage losses through Scroll internal clearances. The counter weight equipped drive cranks provide compact, vibration-free orbital motion. The design is arranged so that there is a minimal build-up of tolerances affecting the clearances between the fixed and orbiting Scrolls. A face seal prevents oil migration from the orbital drive mechanism to the cycle air handling portion of the Scroll elements.

The air cycle cooling system COP is highly sensitive to the overall efficiency of the compressor and expander. The efficiency of a Scroll compressor or expander is predicted by considering the magnitude of losses in the compression or expansion process and in the operation of the orbital drive machinery. The most significant loss in the compression or expansion process is leakage through clearances between the fixed and orbiting Scrolls. In a Scroll compressor, leakage causes a reduction in volumetric efficiency and an increase in the compression work. In a Scroll expander leakage causes an increase in the air mass flow rate and an increase (but a much smaller increase) in the expansion work recovered. The effect of leakage on Scroll component performance is estimated using the existing Scroll Leakage Model Computer Program. The Leakage Model computes the ratio of actual air mass flow rate to ideal air mass flow rate and the isentropic compression or expansion efficiency, given specified Scroll geometry, speed, operating pressures, and clearances between the Scroll elements. Clearances are estimated based on allowances for tolerances and differential thermal expansion of parts. For the preliminary air cycle compressor and expander design the expected tip clearance at operating

conditions would be .002 inch and the flank clearance would be .007 inch. The resulting isentropic compression or expansion efficiency would be 91.0 percent.

Mechanical losses are small. The orbiting Scroll drive bearings, which support the large centrifugal force loading of each orbiting Scroll, account for a significant portion of these losses. Overall mechanical efficiency is estimated at 98%.

A breathing loss of approximately 1.5% results from accelerating the compressor inlet gas to the flow velocity of the outermost pocket. A similar loss occurs at the discharge of the expander.

Overall isentropic compression or expansion efficiencies of 87.5% are predicted, based on the combined effect of leakage, mechanical, and breathing losses.

The total volume of the Scroll compressor and expander (drive motor excluded) is 770 in^3. The total weight is 27lbm. The bulk density of the compressor and expander is .035lbm/in^3. The volume and weight of geometricaly similar Scroll compressor and expander packages are directly proportional to the cycle air volume flow rate and to the square root of the pressure ratio.

1.4 Air To Air Heat Exchanger Design

The preliminary design requirements for the air to air heat exchanger are:

- Heat transfer effectiveness = 92%
- Maximum cycle side pressure loss =1/2% of inlet pressure
- Cycle air side flow rate = 6.2 lbm/hr
- Cycle air side pressure = 47 psia
- Maximum cooling fan shaft power = 1/4 HP, (assume fan is 50% efficient

A cross-flow plate fin heat exchanger, shown in Figure 8, was conceptually designed to meet these requirements. The major characteristics of the heat exchanger are:

- Overall dimensions: 12" long (including compressed air side headers), 4" high, 3.5" deep (parallel to cooling air flow)
- Total volume 171 in^3
- Total weight 2.25 lbm
- Average density .0130 lbm/in^3
- Cycle air side: 20 fins/inch in a staggered strip fin configuration
- Cooling air side: 12 fins/inch in a staggered strip fin configuration
- Cycle air side flow velocity is 21.5 ft/sec, pressure loss is 6.5 in. W.C.
- Cooling air side flow rate: 330 SCFM (365 ACFM) (four times cycle air side flow rate). Flow velocity: 45 ft/sec, pressure loss is 2.2 in. W.C.

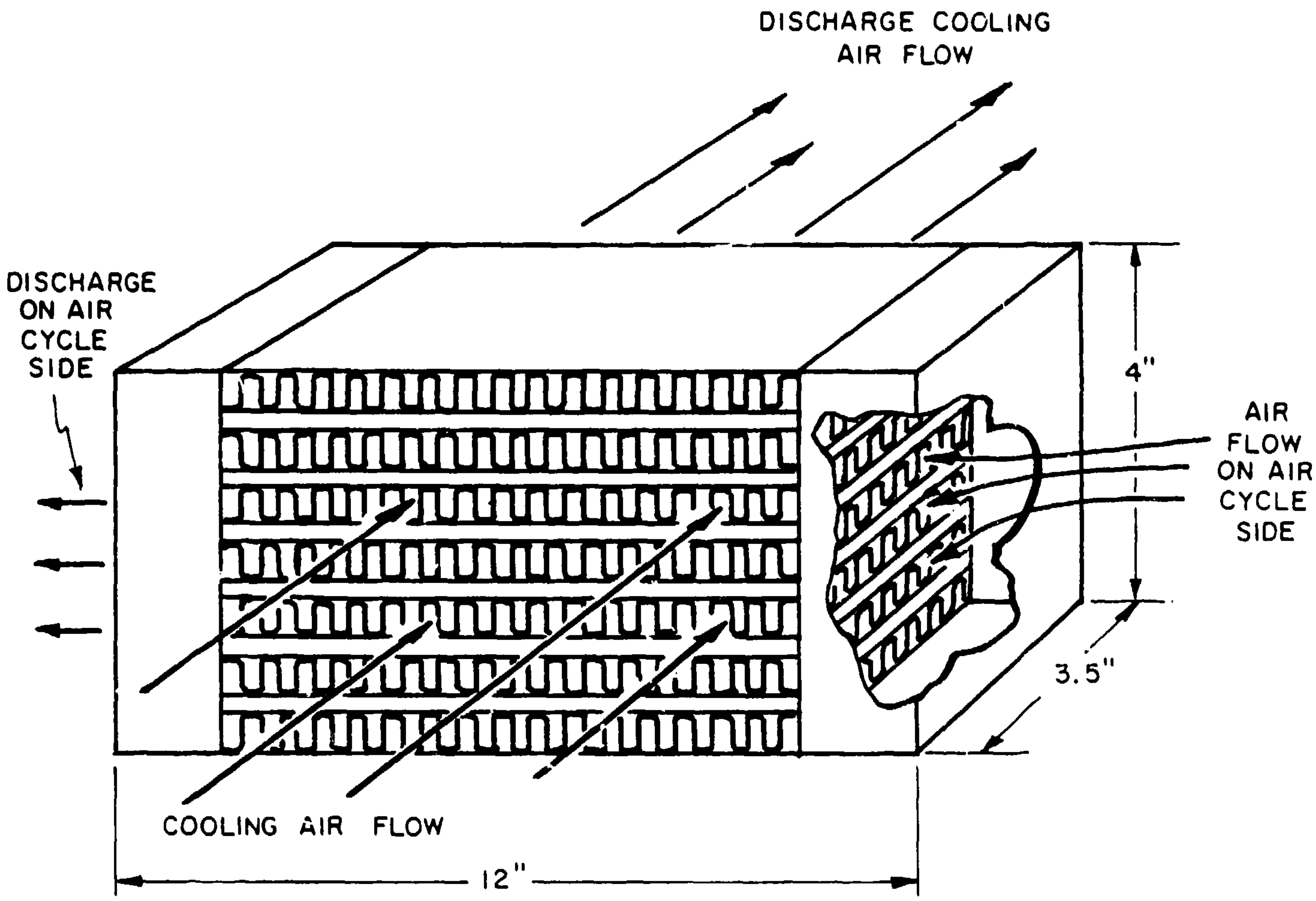


FIGURE 8 AIR-TO-AIR HEAT EXCHANGER CORE CONFIGURATION.

The basis for this heat exchanger design is presented in detail in Appendix A. It is a near minimum volume design for the performance requirements above.

This heat exchanger configuration may be used for other combinations of cycle air mass flow rate, heat transfer effectiveness and pressure drop that are reasonably close to those for the above design. The volume of the resulting heat exchanger would be:

$$V = \left(\frac{\dot{m}}{6.2}\right) \left[68.8 + 102.2 \left(\frac{.005}{\Delta P/P}\right)^{.146}\right] \left(\frac{.08}{1 - \varepsilon}\right)^{.684}$$

with,

V = heat exchanger volume in in^3

$\dot{m}$ = cycle side air mass flow rate in lbm/min

$\Delta P/P$ = cycle side air pressure loss/inlet pressure

ε = heat transfer effectiveness

T. development of this scaling relationship is presented in Appendix A.

1.5 Air to Water Heat Exchanger Design

The preliminary design requirements for the air to water heat exchanger are:

- Heat transfer effectiveness = 92%
- Maximum air side pressure drop =1/2% of inlet pressure
- Air side flow rate = 6.2 lbm/hr
- Air side discharge pressure = ambient air pressure
- Water flow rate = 0.75 GPM

A multipass crossflow-counterflow fin-tube heat exchanger, shown in Figure 9, was conceptually designed to meet these requirements. The main features of the heat exchanger are:

- Overall dimensions: 5-1/2 x 4-1/4 x 11 long (later changed to 11 x 2-1/8 x 10 for compact system package)
- Overall dimensions: 11 x 2-1/8 x 10
- Total volume is 234 in^3
- Total weight is 5.1 lbm
- Average density is .0219 lbm/in^3
- Air side: 20 fins/inch
- Fins are arranged in 9 rows of flat fins as shown in Figure 5
- Water side is a 1/4" ID tube making a total of 36 passes across the heat exchanger core
- Air side flow velocity is 23.7 ft/sec., pressure loss is 2" W.C.
- Water side flow velocity is 4.9 ft/sec, pressure loss is 5 psi

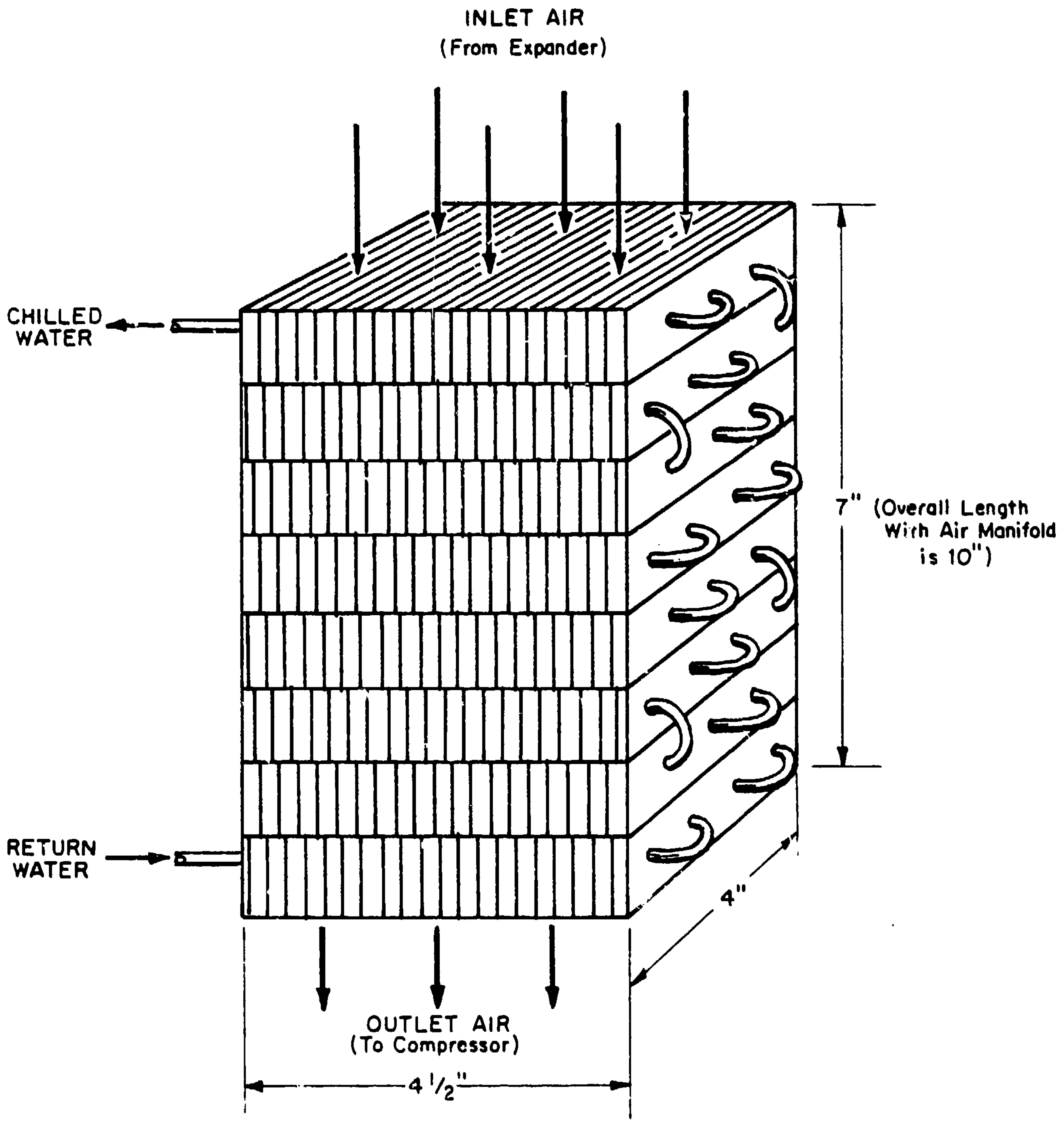


FIGURE 9 CONCEPTUAL DESIGN OF AIR TO WATER HEAT EXCHANGER CORE.

The basis for this heat exchanger design is presented in detail in Appendix B. It is a near minimum volume design for the performance requirements noted above.

This heat exchanger configuration may be used for other combinations of air side flow rate, heat transfer effectiveness, and air side pressure drop. The volume of the resulting heat exchanger would be

$$V = 233\left(\frac{\dot{m}}{6.2}\right) \left(\frac{.005}{\Delta P/P}\right)^{.23} \left(\frac{.08}{1 - \varepsilon}\right)^{.68}$$

with

V	=	heat exchanger volume in in^3
$\dot{m}$	=	air side mass flow rate in lbm/min
$\Delta P/P$	=	air side pressure loss/inlet pressure
ε	=	heat transfer effectiveness

The development of this scaling relationship is presented in Appendix B.

1.6 Component and System Optimization

The objective of applying a Scroll compressor and expander to air cycle refrigeration systems is to increase system COP while reducing system weight and volume. In a typical combat vehicle application, high COP and low system volume are more important than low system weight. As a general rule, the air cycle system COP increases as compressor and expander efficiency increase, pressure ratio decreases, heat exchanger effectivenesses increase, and heat exchanger pressure losses decrease. Air cycle system volume decreases as compressor and expander efficiency increase, pressure ratio increases, heat exchanger effectiveness (and volume) is decreased and heat exchanger pressure losses increase (further reducing heat exchanger volume). Except for maximizing compressor and expander efficiency, the requirements for improving the COP are opposite to the requirements for reducing volume. Therefore, the system selected will represent a compromise between minimum volume and maximum COP. The purpose of the system optimization task was to provide a quantitative assessment of the degree to which COP must be sacrificed to attain a given degree of volume reduction.

Trade-offs were conducted with heat exchanger performance (heat transfer effectiveness and pressure loss) and pressure ratio as the primary variables. The relationships presented in the previous sections for the weight and volume of the Scroll compressor and expander, Scroll drive motor, and heat exchangers, were used. The Scroll drive motor efficiency was assumed to be 80%. Other parameters, fixed on the basis of earlier program work, are as follows:

- Compressor and expander overall isentropic efficiency = 87.5%
- Meet system specifications of Table 1
- 0.75 GPM of chilled water supplied at 50°F
- Cooling load = 6,500 Btu/hr
- Cooling air temperature = 125°F

The cooling fan volume, weight, and power consumption were accounted for as follows:

- Volume = 31 in^3 x cycle air mass flow rate in lbm/min
- Weight = Volume x .03 lbm/in^3
- Power = 30 watts (electric) x cycle air mass flow rate in lbm/min

The water circulation pump was not considered in these trade-offs because the pump requirements are fixed, independent of the rest of the system. The preliminary air cycle analysis program was extended to incorporate the component weight and volume scaling relationships and run for enough combinations of component performance characteristics to establish the volume-COP trade-off.

The results of the Volume-COP trade-offs are plotted in Figure 10. For each curve, corresponding to a fixed pressure ratio, heat exchanger effectiveness increases and pressure losses decrease from left to right. Minimum and near minimum component volume occurs over a broad range of increasing COP.

The results of the trade-off between component weight and COP are plotted in Figure 11 and are similar in character to the Figure 10 plots.

Minimum component volume, component weight, and COP are plotted versus pressure ratio in Figure 12.

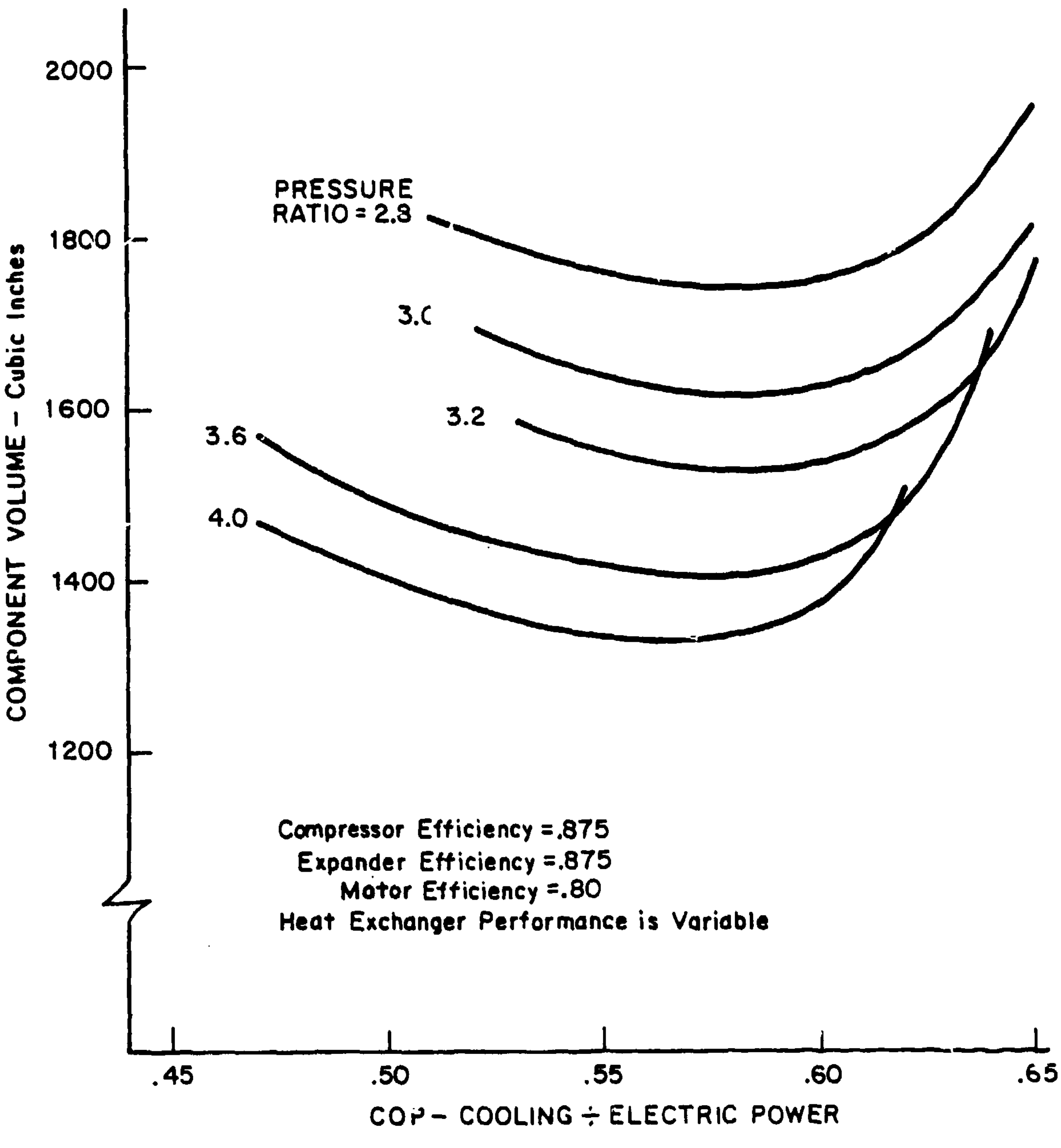


FIGURE 10 TRADE-OFF BETWEEN COMPONENT VOLUME AND COP FOR DIFFERENT PRESSURE RATIOS.

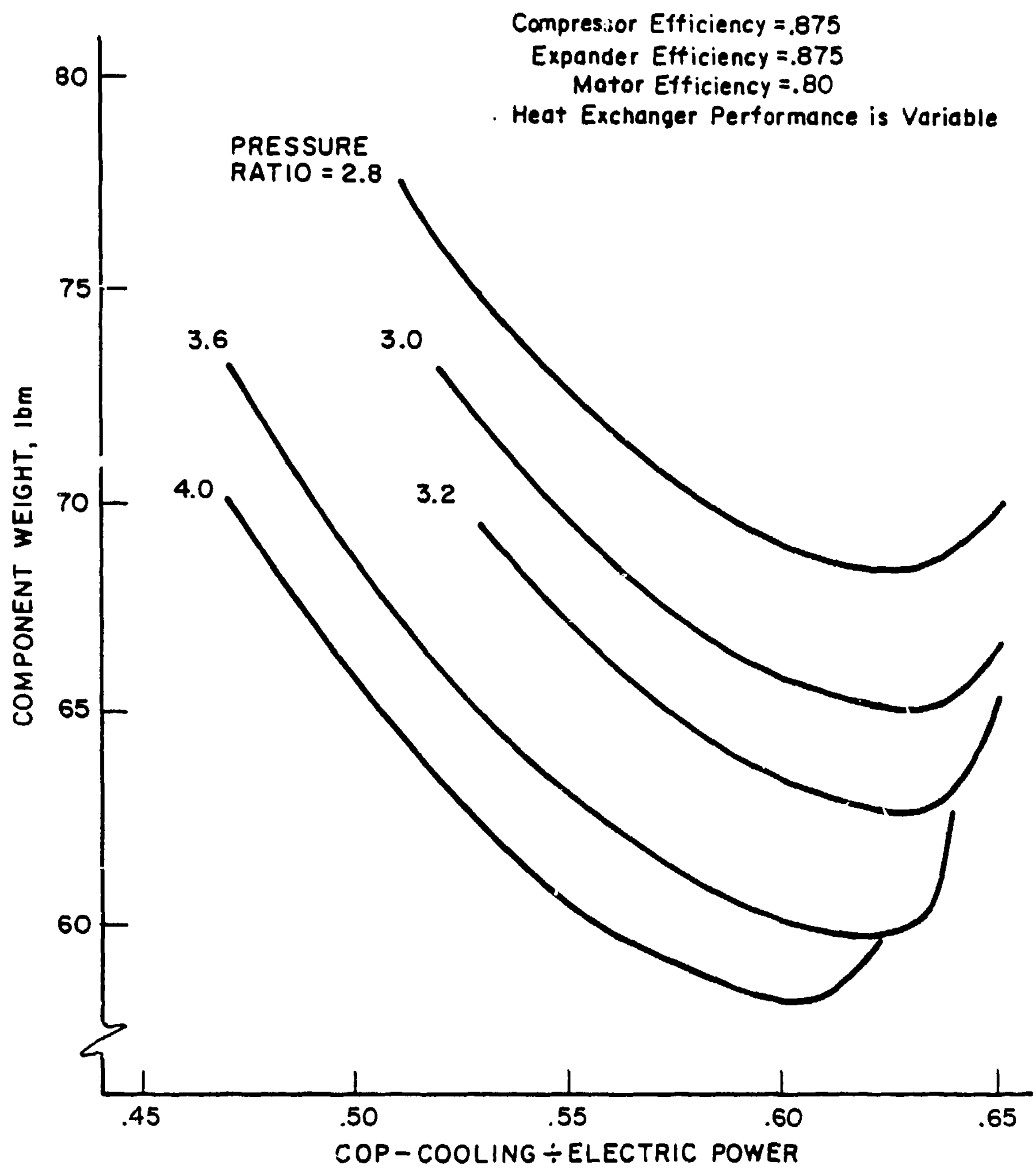


FIGURE 11 TRADE-OFF BETWEEN COMPONENT WEIGHT AND COP FOR DIFFERENT PRESSURE RATIOS.

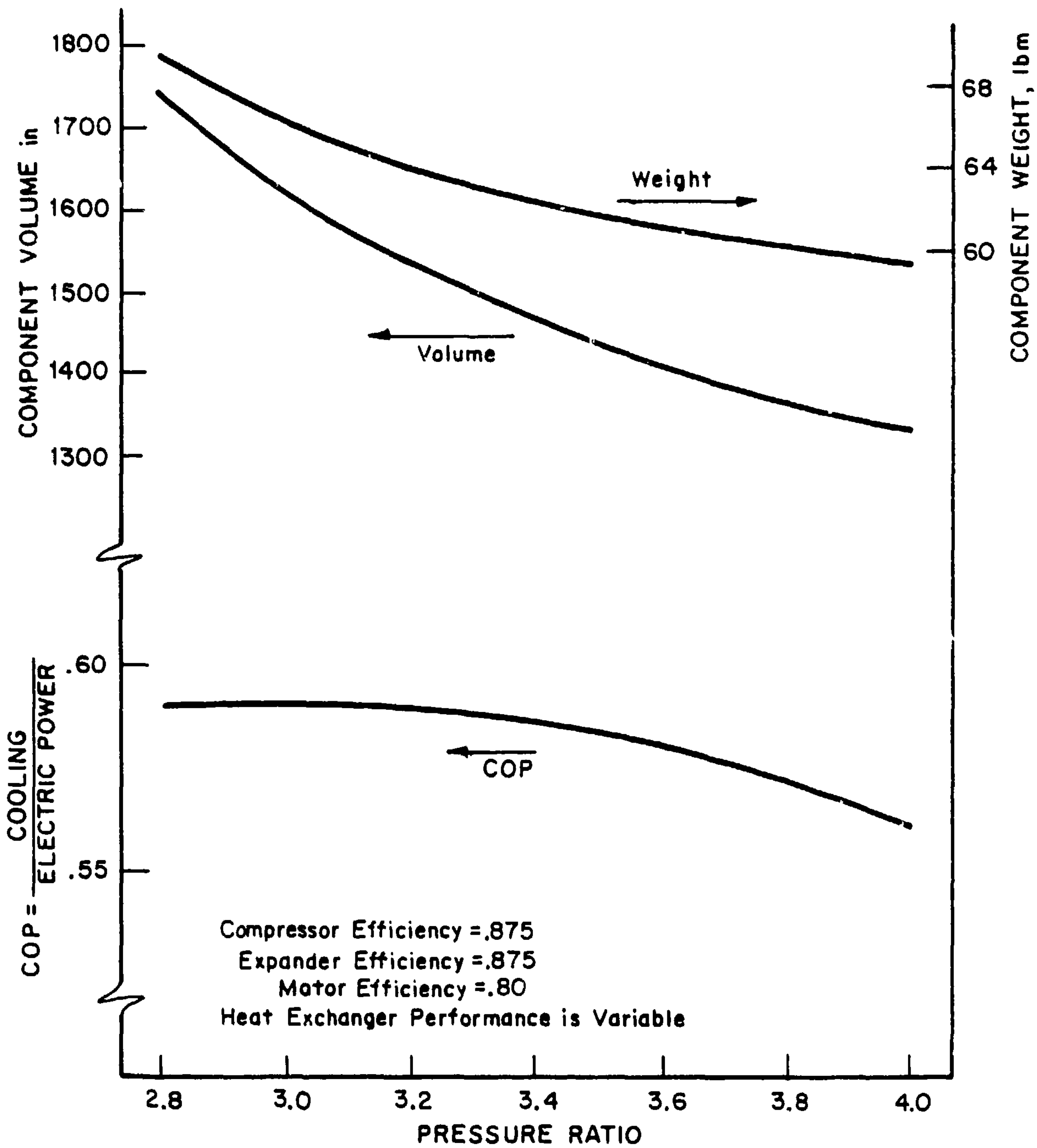


FIGURE 12 MINIMUM COMPONENT VOLUME, COMPONENT WEIGHT, AND COP VS. PRESSURE RATIO.

1.7 Recommended System

The recommended system for further development has the following component performance characteristics:

- Pressure ratio = 3.20
- Compressor and expander efficiencies = 87.5%
- Heat exchanger heat transfer effectivenesses = 93%
- Heat exchanger pressure losses are .5% of the inlet pressure.

The pressure ratio of 3.2 was selected to limit the compressor discharge temperature to 300°F, consistent with long term temperature limits of aluminum. The resulting system is close to the highest attainable COP and lowest attainable volume for compressor and expander efficiencies of 87.5%.

The system and component performance characteristics and dimensions are summarized in Table 4.

Table 4 Summary of Characteristics of Recommended Air Cycle System

Compressor/Expander Efficiencies	.875	
Heat Exchanger Effectiveness	.93	
Heat Exchanger $\Delta P/P$	.005	
Pressure Ratio	3.2	
COP = Cooling ÷ Compressor/Expander Shaft Power	.818	
COP = Cooling ÷ Total Electric Power	.625	
Total System Weight	120	lbm
Cycle Air Mass Flow Rate	6.19	lbm/min
Compressor Inlet Volume Flow Rate	81.47	ACFM
Expander Discharge Volume Flow Rate	70.11	ACFM
Compressor Inlet Temperature	63.5	°F
Compressor Discharge Temperature	299.5	°F
Expander Inlet Temperature	137.2	°F
Expander Discharge Temperature	-9.5	°F
Compressor-Expander-Motor-Fan Package		
Weight	45	lbm
Net Shaft Power (incl. fan)	3.37	HP
Fan Shaft Power	.25	HP
Motor Electric Power	3,000	Watt Elec.
Overall Length	24	inches
Overall Diameter	10½	inches

Table 4 (Cont.)

Air-Water Heat Exchanger

Weight	5.10	lbm
Water Side ΔP	5	psi
Overall Length (parallel to air flow)	10	inches
Overall Width	11	inches
Overall Thickness	2-1/8	inches

Air-Air Heat Exchanger

Weight	2¼	lbm
Cooling Air Flow Rate	330	SCFM
Cooling Air Pressure Drop	2.2	in. W.C.
Overall Length (parallel to cycle air flow)	12	inches
Overall Width	4	inches
Overall Height	3½	inches

Water Pump

Flow Rate	.75	GPM
ΔP	20	psi
Shaft Power (33% eff)	.027	HP
Electric Power (50% eff)	40	watts
Weight	3	lbm
Overall Length	5	inches
Overall Diameter	3	inches

2.0 Air Cycle Cooling System Operation

The air cycle cooling system shown in Figures 13 and 14 is a compact arrangement of components grouped around the main element; the motor driven Scroll compressor and expander. The process air flows from the compressor to the air to air heat exchanger, where it is cooled, to the expander inlet. After expansion the air flows to the air to water heat exchanger then back to the compressor. Outside air is used for cooling the drive motor and process air after compression. Outside air is drawn into the drive motor housing, through the motor windings and then through the air to air heat exchanger before it discharges back outside. Since the motor will have an overall efficiency of 80% or more, only about 10% of the total heat rejected comes from the motor. Therefore, using the motor fan (enlarged as necessary) to cool the motor and the heat rejection heat exchanger, simplifies the overall design without degrading heat exchanger performance significantly.

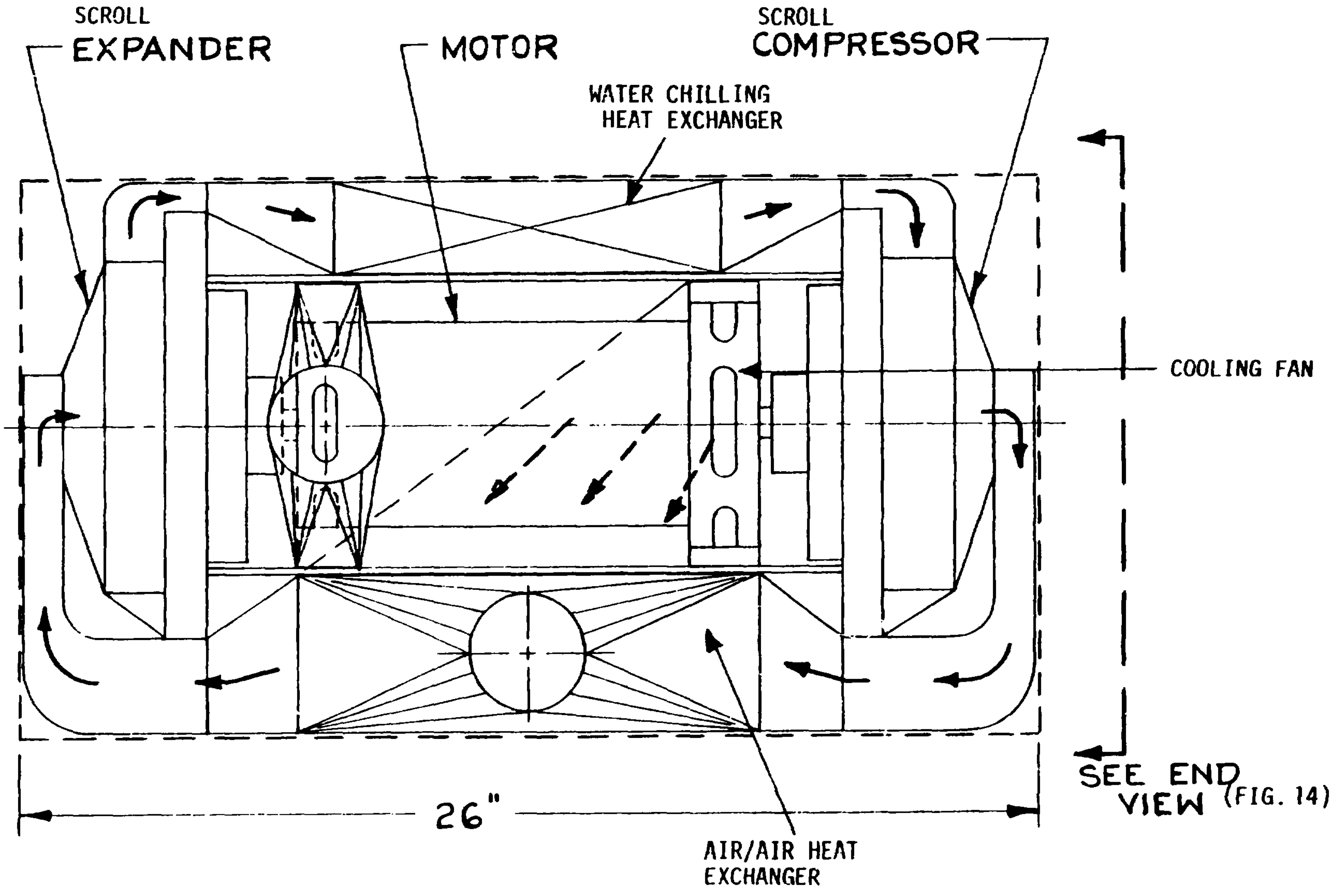


FIGURE 13. 6500 BTU/HR Scroll Air Cycle Chilled Water Cooling System

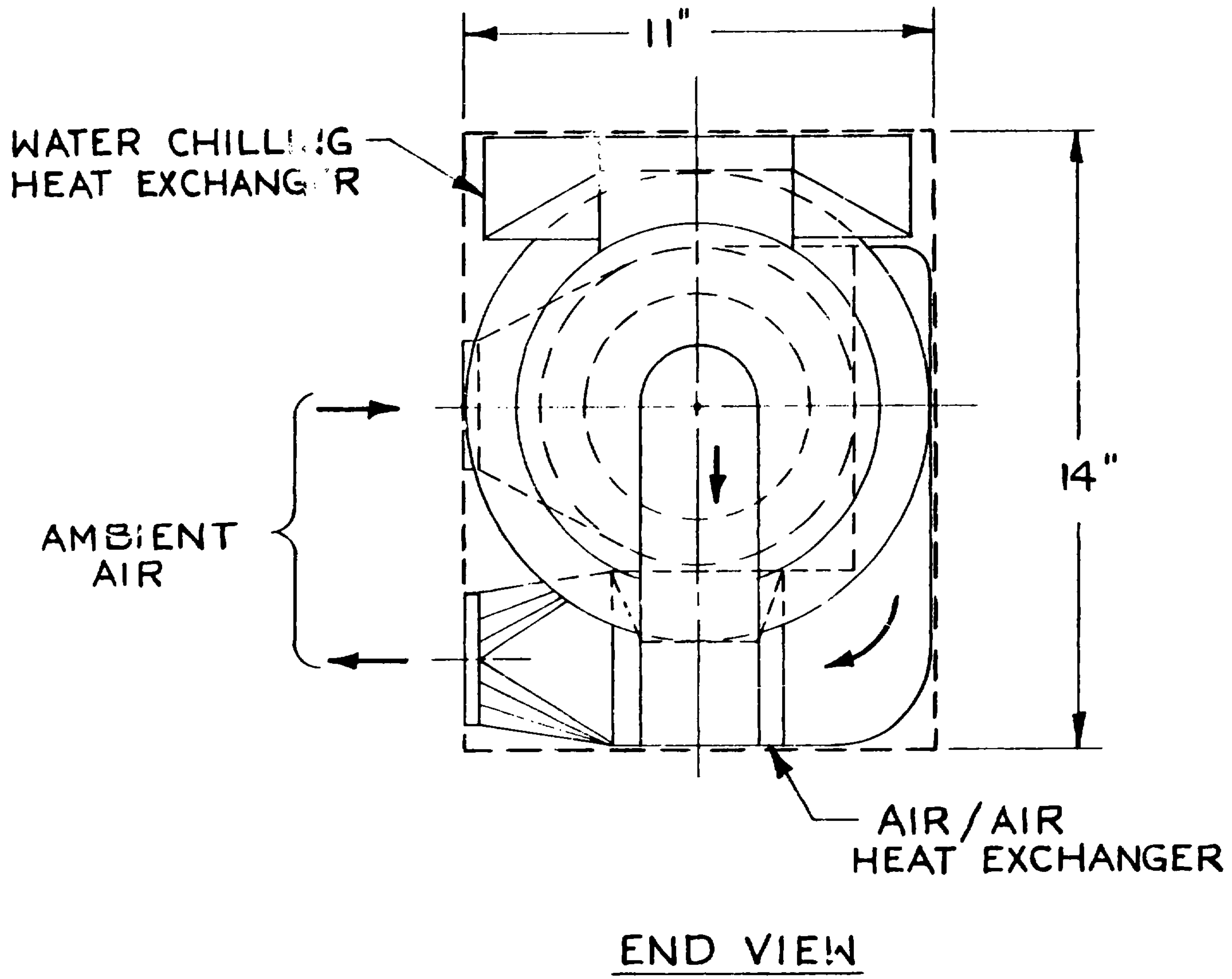


FIGURE 14. End View of Figure 13

3.0 Air Cycle Cooling System Weight Estimate

A preliminary estimate of the weight for the air cycle cooling system shwon is as follows:

Component	Estimated Weight
• Drive motor	28.0
• Scroll compressor	15.0
• Scroll expander	12.0
• Air to air heat exchanger	2.3
• Air to water heat exchanger	5.1
• Chilled water pump	3.0
• Component mounting structure	15.0
• Air ducting	18.0
• External, sheet metal cover	22.0
	120.4 lbm.

APPENDIX A

Air to Air Heat Exchanger Design and Optimization

To meet the two main air cycle system design requirements of low volume and high COP, the air to air heat exchanger must be as compact as possible while performing with high heat transfer effectiveness and low pressure drop. Cross flow plate-fin construction generally offers the most compact gas to gas heat exchanger configuration and the conceptual design and sizing were based on this arrangement, as shown in Figure 8. A great deal of design flexibility is available to meet a given heat exchanger design requirement; options include:

- Fin spacing and configuration on each side of heat exchanger
- Ratio of cooling air mass flow rate to air cycle air mass flow rate
- Air velocity on each side

A staggered strip fin configuration was selected for each side, as shown in Figure A-1. The staggered fin arrangement continously interrupts the growth of a thermal boundary layer in the air flowing over the fins, providing significantly higher heat transfer coefficients than a straight fin arrangement. The fin configurations for the two sides of the air to air heat exchanger were selected from reference 1. Heat transfer and flow friction data for the cycle air and the cooling air sides are shown in Figures 10-60 and 10-58, respectively, of Reference 1.

A cooling air mass flow rate equal to four times the compressed cycle air mass flow rate results in nearly minimum heat exchanger core volume, as shown in Figure A-2, without requiring an excessively large volume cooling air fan.

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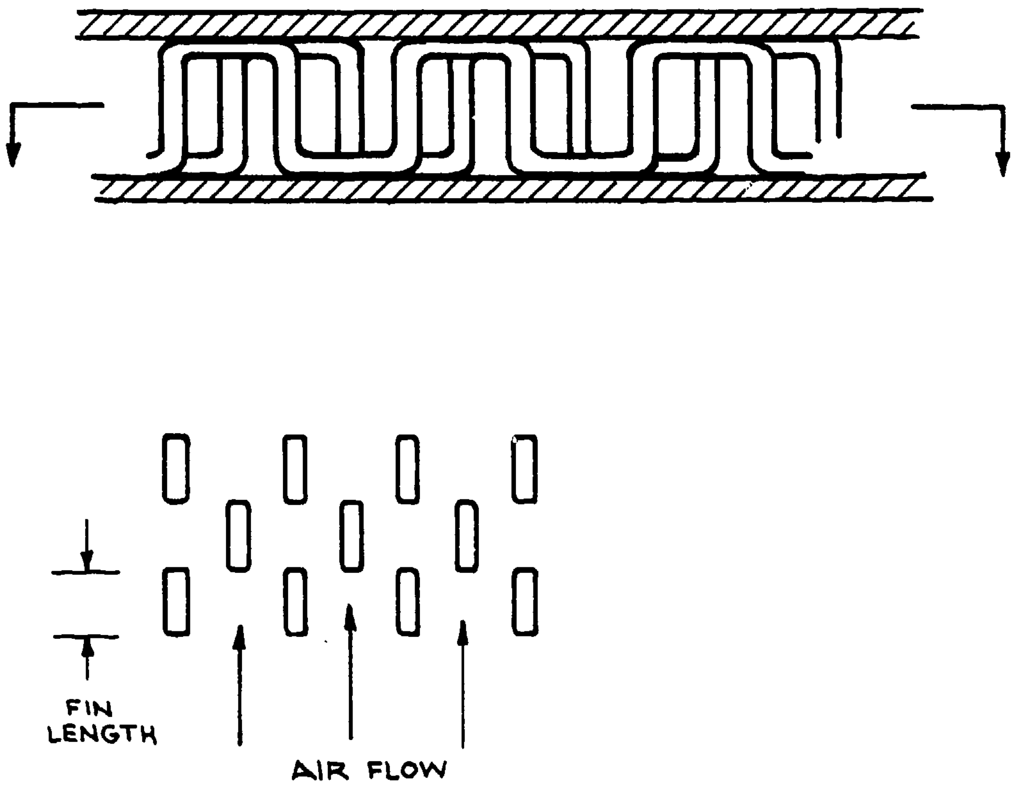
[1] Kays and London Compact Heat Exchangers, Mc-Graw-Hill, New York.



	AIR CYCLE SIDE	COOLING AIR SIDE
FINS/INCH	20	12
FIN HEIGHT	.201	.353
FIN LENGTH	.125	.178

FIGURE A-1 FIN CONFIGURATION USED IN AIR TO AIR HEAT EXCHANGER.

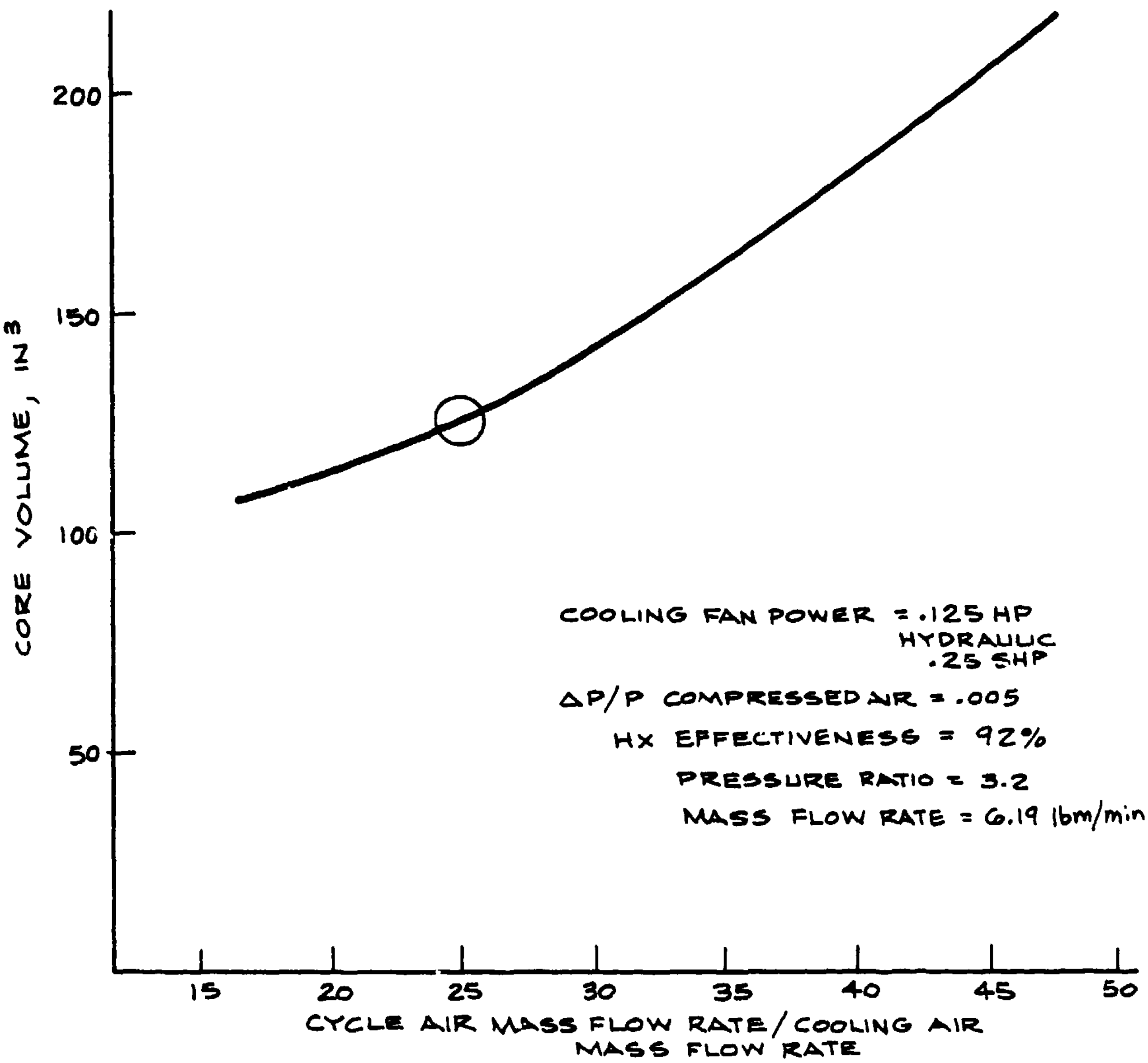


FIGURE A-2 CORE VOLUME vs. COOLING AIR FLOW RATE

A required level of heat transfer effectiveness consistent with a maximum pressure drop can be obtained by limiting the air flow velocity to an appropriate level. Figure A-3 is a plot of the maximum cycle air side flow velocity allowable for various combinations of required heat transfer effectiveness and allowable pressure drop. A similar relationship between allowable air velocity and heat transfer effectiveness exists for the cooling air side. The cooling air side pressure drop was assumed to be 2.2" W.C. corresponding to a fan shaft power requirement of .25 HP, and a fan efficiency of 50%.

Given allowable air flow velocities through the core on each side and the required heat transfer effectivness, the heat transfer surface on both sides is readily sized. For the baseline case of the preliminary component sizing task (cycle air flow rate of 6.2 lbm/min, effectiveness of 92%, $\Delta P/P$ of .5%) the resulting heat exchanger characteristics and volume are summarized in Table A-1.

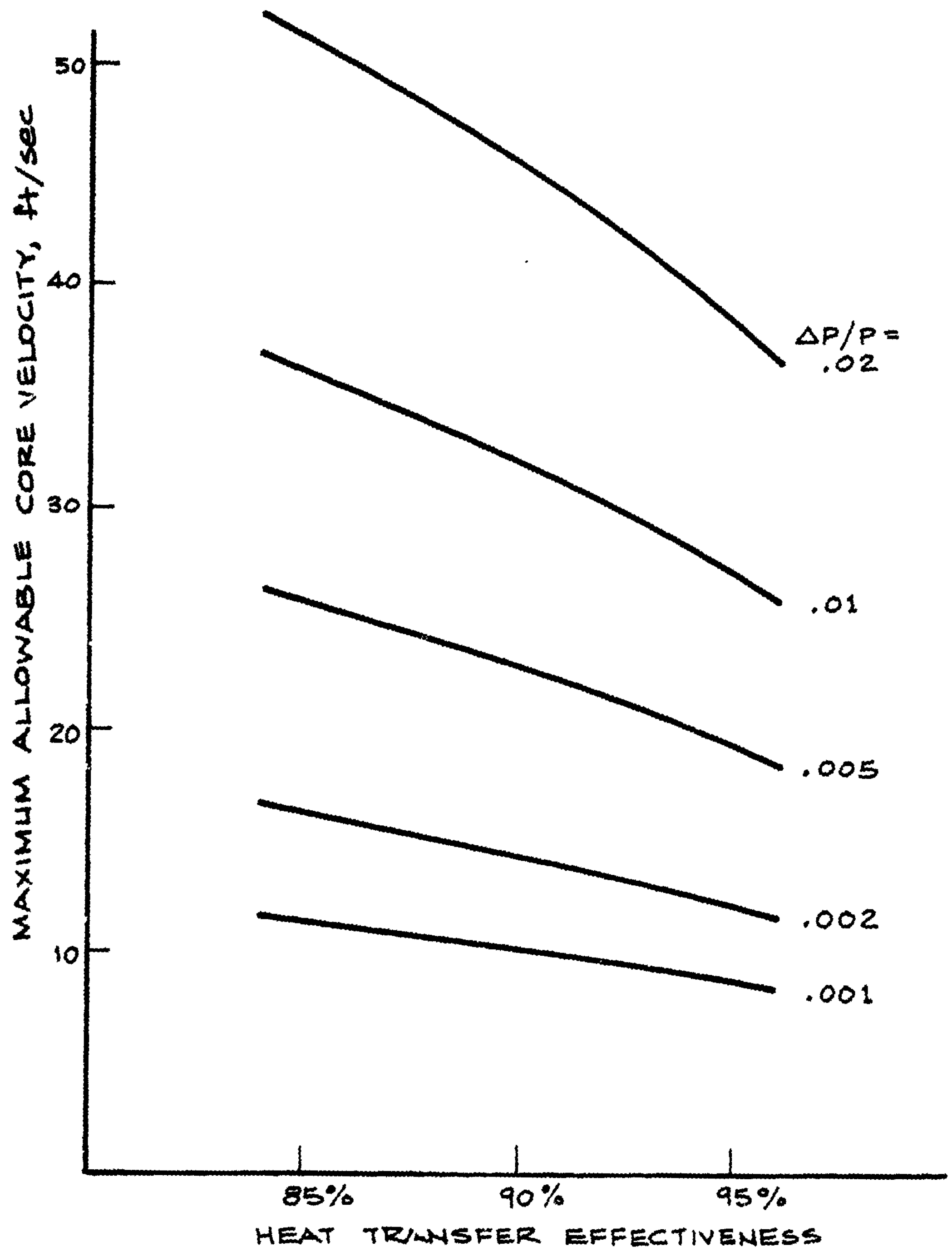


FIGURE A-3 ALLOWABLE CYCLE AIR SIDE CORE VELOCITY IN THE AIR TO AIR HEAT EXCHANGER VS REQUIRED HEAT TRANSFER EFFECTIVENESS AND ALLOWABLE CYCLE AIR SIDE PRESSURE LOSS.

TABLE A-1 CHARACTERISTICS OF AIR TO AIR HEAT EXCHANGER DESIGNED FOR THE PRELIMINARY COMPONENT SIZING TASK

	Cooling Air Side	Cycle Air Side
Air mass flow rate, lbm/min	24.8	6.2
Air velocity, ft/sec	45.5	21.5
Heat transfer coefficient, Btu/hr-ft^2-°F	30.4	46.5
Heat transfer area, ft^2	18.6	16.2
Core volume, in^3	80.3	44.0
Frontal area, in^2	19.3	4.2
Core density, lbm/in^3	.011	.018
Core volume, in^3	125	
Core dimensions	9 in x 4 in x 3.5 in	
Overall core density, lbm/in^3	.0138	
Dimensions including manifolds	12 in x 4 in x 3.5 in	
Average bulk density of heat exchanger including manifolds, lbm/in^3	.013	

A scaling relationship was developed to relate the air to air heat exchanger core volume to the heat transfer effectiveness, pressure drop, and cycle air mass flow rate. The heat exchanger core volume scales in direct proportion to the cycle air side mass flow rate, for constant pressure drop and heat transfer effectiveness. Following the procedure outlined above, heat exchanger core volumes were computed for heat transfer effectiveness between 84% and 96% and ΔP/P's between .001 and .02. The computed volumes, heat transfer effectivenesses, and ΔP/P's are listed in Table A-2 and plotted in Figure A-4. This data was fit to an equation of the form

$$V = \left(\frac{\dot{m}}{\dot{m}_{ref}}\right) \left[V_1 + V_2 \left(\frac{\Delta P/P_{ref}}{\Delta P/P}\right)^{K_1}\right] \left(\frac{1 - \varepsilon_{ref}}{1 - \varepsilon}\right)^{K_2}$$

using the nomenclature of Section 1.4. The subscript "ref" denotes a "reference" design, which was taken as the 92% effective, .005 ΔP/P design described above. Thus,

$\dot{m}_{ref}$ = 6.2 lbm/min

$\Delta P/P_{ref}$ = .005

$1 - \varepsilon_{ref}$ = .08

V_1 and V_2 were selected such that the sum of V_1 and V_2 is the heat exchanger core volume of the reference design.

TABLE A-2

Air To Air Heat Exchanger Core Volumes
Computed For Derivation Of Scaling Relationship

Effectiveness, ε	1 - ε	ΔP/P	Heat Exchanger Core Volume, in^3 From Design Calculations	Heat Exchanger Core Volume, in^3 From Scaling Relationship
.92*	.08	.005*	124.3	124.3
.84	.16	.005	75.2	77.4
.88	.12	.005	94.2	94.2
.96	.04	.005	187.5	199.7
.92	.08	.001	144.0	144.0
.92	.08	.003	129.8	130.1
.92	.08	.0075	120.4	120.0
.92	.08	.010	117.2	117.15
.92	.08	.015	112.7	113.3
.92	.08	.020	109.7	110.7

*"Reference" design

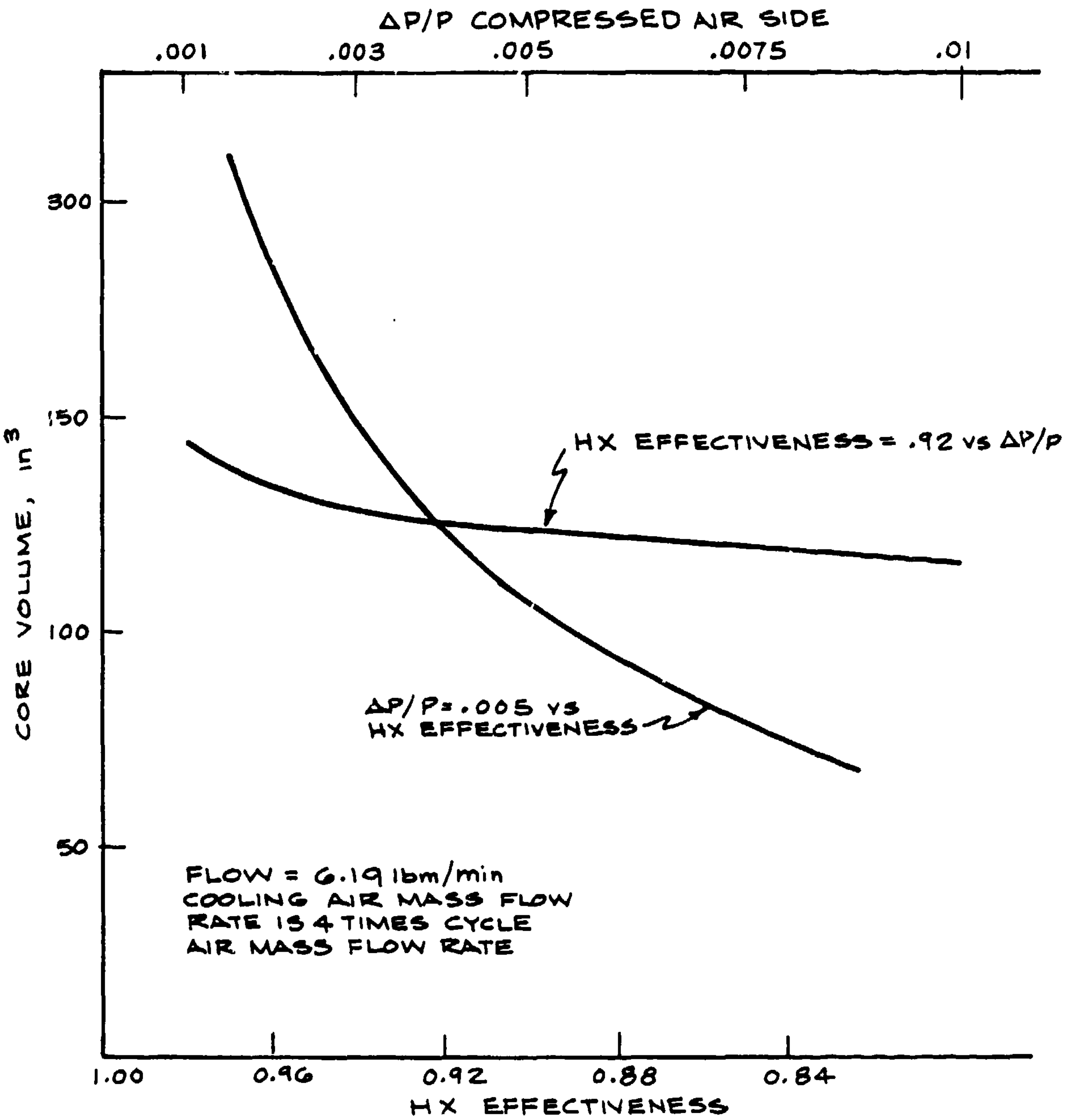


FIGURE A-4 TRADE-OFF BETWEEN AIR TO AIR HEAT EXCHANGER CORE VOLUME AND HEAT EXCHANGER PERFORMANCE.

The values of V_1, V_2, K_1, and K_2 were selected by trial and error to obtain the best correlation of the data in Table A-2. The best correlation was attained with values of

$$V_1 = 50.0 \text{ in}^3$$

$$V_2 = 74.3 \text{ in}^3$$

$$K_1 = .146$$

$$K_2 = .684$$

The resulting scaling relationship is

$$V = \left(\frac{\dot{m}}{6.2}\right) \left[50.0 + 74.3 \left(\frac{.005}{\Delta P/P}\right)^{.146}\right] \left(\frac{.08}{1 - \varepsilon}\right)^{.684}$$

The core volumes computed using the scaling relationship are listed in Table A-2 for comparison with the core volumes obtained directly from design calculations. To allow for the volume of the cycle air inlet and outlet headers, the overall heat exchanger volume is increased by 37.5 percent, so that

$$V_1 = 68.8$$

$$V_2 = 102.2$$

and the scaling relationship is

$$V = \left(\frac{\dot{m}}{6.2}\right) \left[68.8 + 102.2 \left(\frac{.005}{\Delta P/P}\right)^{.146}\right] \left(\frac{.08}{1 - \varepsilon}\right)^{.684}$$

as presented in Section 1.4.

APPENDIX B

Appendix B - Air to Water Heat Exchanger Design and Optimization

The design requirement of the air to water heat exchanger is similar to that for the air to air heat exchanger -- meet pressure drop and heat transfer effectiveness requirements with a low volume heat exchanger. A design approach similar to that outlined in Appendix A was followed.

- Define the basic fin and core geometry.
- Calculate air velocities consistent with required heat exchanger heat transfer effectiveness and pressure loss
- Calculate the core size for the preliminary component sizing task baseline case
- Develop a relationship to scale the core volume to other performance requirements.

The core geometry shown in Figure 9 in Section 1.5 of the report consists of several layers of plain fins with several passes of water tubing per layer. The water circuit passes through the heat exchanger core layers counter to the direction of the cycle air flow. The performance of this arrangement approaches counterflow.

Core design proceeded on the basis of using approximately 25% of the available heat transfer temperature difference for heat transfer from the tubing wall to the water and the remaining 75% for heat transfer from the cycle air to the fins. The cycle air side air velocity consistent with a range of required heat transfer effecctiveness and allowable pressure losses is plotted in Figure B-1.

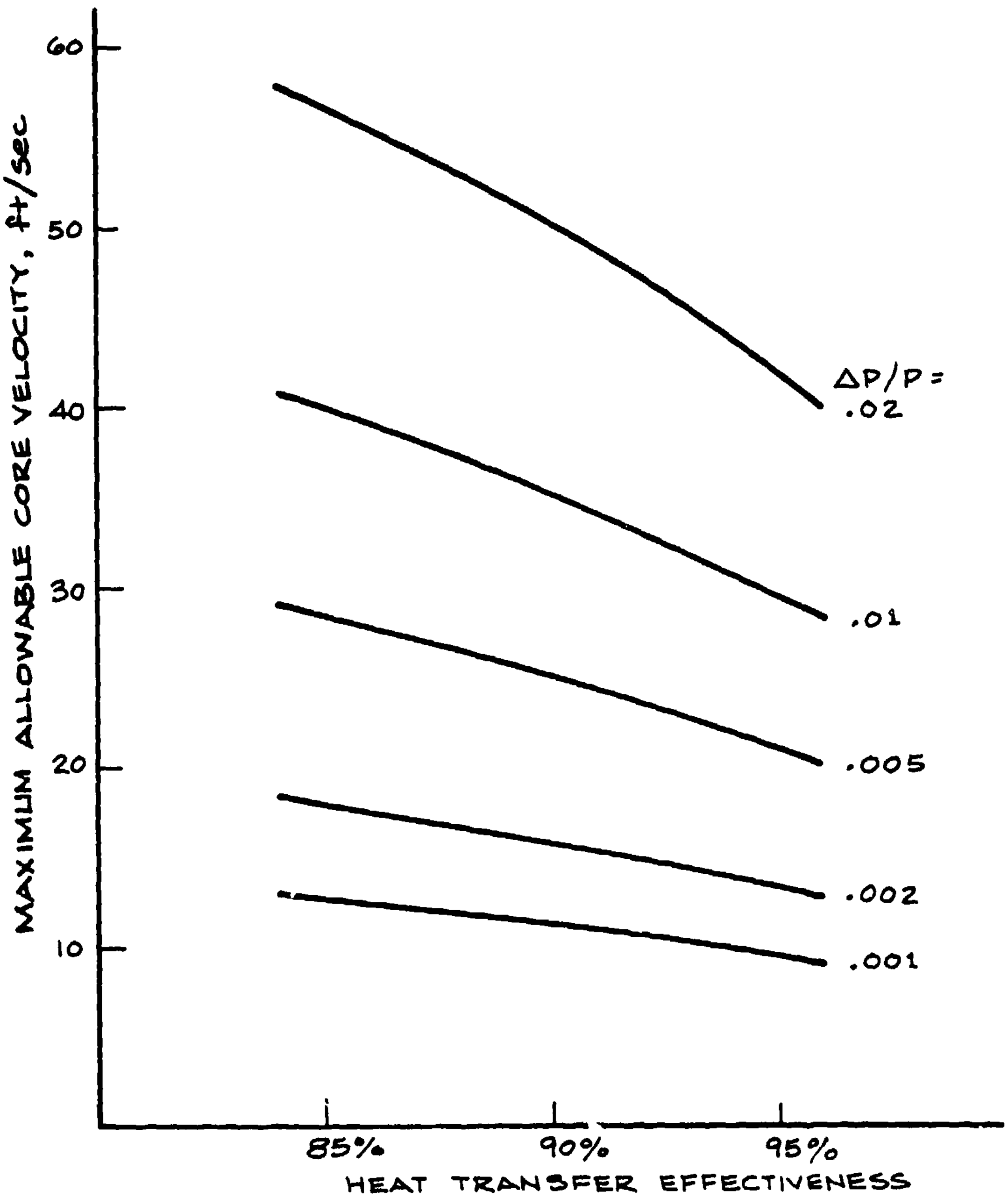


FIGURE B-1 ALLOWABLE CYCLE AIR SIDE CORE VELOCITY IN THE AIR TO WATER HEAT EXCHANGER VS REQUIRED HEAT TRANSFER EFFECTIVENESS AND ALLOWABLE CYCLE AIR SIDE PRESSURE LOSS.

The heat exchanger was sized to meet the preliminary component sizing task performance requirements (heat transfer effectiveness of 92%, air side pressure loss of .5% of inlet pressure). The characteristics of the resulting heat exchanger are summarized in Table B-1.

A scaling relationship was developed to relate the air to water heat exchanger volume to the heat transfer effectiveness, pressure drop, and cycle air mass flow rate. The heat exchanger volume scales in direct proportion to the cycle air side mass flow rate, for constant pressure drop and heat transfer effectiveness. Following the procedure outlined above, heat exchanger core volumes were computed for heat transfer effectiveness between 84% and 96% and $\Delta P/P$'s between .001 and .02. The computed volumes, heat transfer effectivenesses, and $\Delta P/P$'s are listed in Table B-2. This data was fit to an equation of the form

$$V = V_{ref} \left(\frac{m}{\dot{m}_{ref}}\right) \left(\frac{\Delta P/P_{ref}}{\Delta P/P}\right)^{K_1} \left(\frac{1 - \varepsilon_{ref}}{1 - \varepsilon}\right)^{K_2}$$

using the nomenclature of Section 1.5. The subscript "ref" denotes a "reference" design, which was taken as the 92% effective, .005 $\Delta P/P$ design described above, so that

$$V_{ref} = 233 \text{ in}^3$$

$$\dot{m}_{ref} = 6.2 \text{ lbm/min}$$

$$\Delta P/P_{ref} = .005$$

$$1 - \varepsilon_{ref} = .08$$

TABLE B-1 CHARACTERISTICS OF AIR TO WATER HEAT EXCHANGER DESIGNED FOR THE PRELIMINARY COMPONENT SIZING TASK

	Water Side	Cycle Air Side
Mass flow rate, lbm/min	6.2	6.2
Velocity, ft/sec	4.9	23.7
Heat transfer coefficient, Btu/hr-ft^2-°F	1,470	20.0
Heat transfer area, ft^2	.82	22.0
Core volume, in^3	12.0	100.0
Frontal area, in^2	.05	14.5
Core density, lbm/in^3	--	--
Core volume, in^3	112.0	
Core dimensions	7.25in x 4.0in x 4.0 in	
Overall core density, lbm/in^3	.0296	
Dimensions including manifolds	10in x 5.5in x 4.25 in	
Average bulk density of heat exchanger including manifolds, lbm/in^3	.0219	

TABLE B-2

Air To Water Heat Exchanger Volumes
Computed For Derivation Of Scaling Relationship

			Heat Exchanger Volume, in^3	
Effectiveness, ε	1 - ε	ΔP/P	From Design Calculations	From Scaling Relationship
.92*	.08	.005*	233.0	233.0
.84	.16		141.0	145.4
.88	.12		176.6	176.9
.96	.04		351.5	373.3
.92	.08	.001	337.3	337.4
		.002	287.5	287.7
		.003	262.1	262.0
		.0075	212.3	212.3
		.010	195.9	198.7
		.015	180.0	181.0
		.020	169.5	169.4

*"Reference" design

The values of K_1 and K_2 were selected by trial and error to obtain the best correlation of the data in Table A-2. The best correlation was attained with $K_1 = .23$ and $K_2 = .68$.

The resulting scaling relationship is

$$V = 233\left(\frac{\dot{m}}{6.2}\right) \left(\frac{.005}{\Delta P/P}\right)^{.23} \left(\frac{.08}{1 - \varepsilon}\right)^{.68}$$

The core volumes computed using the scaling relationship are listed in Table B-2 for comparison with the core volumes obtained directly from design calculations.