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Technical Report No. 415-1-F

# FEASIBILITY STUDY OF A HEAT-ACTUATED ENVIRONMENTAL CONTROL SYSTEM FOR THE NAVY F-4 FIGHTER AIRCRAFT

Final Report
15 August 1966

by

Dieter D. Huber
and
Alfred J. Gorman

Prepared under Contract NOw 66-0226-c for the Naval Air Systems Command (formerly Bureau of Naval Weapons), Department of the Navy, by Conductron Corporation, Western Development Center, 9060 Winnetka Avenue, Northridge, California

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## FOREWORD

This is the final report describing the work performed on a feasibility study of the Conductron Heat-Actuated Environmental Control System for the Navy F-4 Fighter Aircraft. The program was conducted by Conductron Corporation, Western Development Center, under Contract No. NOw66-0226-c, sponsored by the Department of the Navy, Naval Air Systems Command (formerly Bureau of Naval Weapons).

The objective of this study was to obtain a comprehensive evaluation of the feasibility of adopting the Conductron heat-actuated refrigeration system to high-performance Navy fighter aircraft environmental control. This program was undertaken as a first-phase of a step-by-step evaluation and development of this new and unique environmental control concept.

The study reported herein was performed during the period 20 December 1965 through 15 July 1966.

ABSTRACT

This report presents the results of an analytical study to investigate the feasibility of a new and unique environmental control concept for the cabin air conditioning requirements of Navy F-4 fighter aircraft. The heat-actuated Conductron environmental control concept is basically a vapor compression system which utilizes two thermodynamic loops and a unique device which replaces the conventional compressor.

It is shown that an environmental control system based on the Conductron concept can be adopted to F-4 fighter aircraft within the existing constraints on weight, size, and power. Such a system can provide both cooling and heating under all operating conditions, including ground-static at idle. Furthermore, significant reductions in the compressor bleed air requirements, as compared to the present F-4 air-cycle air conditioning systems, can be realized with the Conductron system.

The Conductron environmental control system can be used for cabin and equipment temperature control of all Navy aircraft, both high-performance and subsonic. The specific results of this study, however, are applicable only to the Navy F-4 fighter aircraft.

## LIST OF FIGURES

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## 1.0 INTRODUCTION

Environmental control of aircraft crew compartments involves the maintenance of temperature, absolute pressure, humidity, air movement, and air purity at safe and comfortable levels. In many applications, particularly in high-performance aircraft, it may be impractical to provide control of all these factors so as to assure ideal conditions because of the weight penalties associated with these control functions. The selection of the environmental control functions for any given aircraft application are governed by the requirement of a cabin environment which will not impair crew performance below the minimum levels required for successful mission accomplishments.

Each aircraft environmental control application requires a system designed especially for the cooling loads and temperature levels peculiar to that aircraft and its mission. The selection of the type of environmental control system to be used is based upon trade-offs between weight, volume, and power penalties and the performance, reliability, and maintainability characteristics of a particular system.

Most high-performance military aircraft use air cycle systems because of their compactness and light weight. However, these systems have poor performance characteristics at high speeds and on the ground, in addition to high power demands and poor reliability due to use of high-speed machinery. Vapor cycle systems are characterized by higher performance, higher weight penalties, and poor reliability due to use of mechanical compressors. Many of the inherent disadvantages of air cycle and vapor cycle systems can be minimized or eliminated, at least in theory, by the Conductron system. The most attractive features are:

- FULL PERFORMANCE at all operating conditions.
- HIGH RELIABILITY due to absence of high-speed machinery.
- LONG LIFETIME due to minimum amount of wearing parts.
- SYSTEM FLEXIBILITY allowing temperature control at multiple locations.
- VERSATILITY of installation due to reduced constraints on component placement.
- WASTE HEAT utilization to actuate the system.
- LOW NET WEIGHT by offering cooling, heating, and dehumidification from a single unit.

These attractive features of the Conductron refrigeration system make the evaluation and development of this concept for aircraft applications highly desirable.

## 1.1 Program Objective and Scope

The practical application of any environmental control system to existing high-performance aircraft must take into account many existing constraints, the most severe being no increase in weight and size and no modifications to the airframe. Under such severe constraints and limitations, an analytical design study was deemed to be absolutely essential in order to establish the compatibility of the Conductron system with a specific aircraft.

For this feasibility study of the Conductron system, the environmental control requirements as well as weight, size, and installation constraints of the McDonnell F-4 fighter aircraft were chosen as those typifying Navy high-performance aircraft. Therefore, the specific results and conclusions of this study are not directly applicable to other aircraft; since the differences between individual aircraft, even those with similar mission requirements and performance characteristics, can be quite substantial.

The scope of the study is reflected in the organization of this report. The remainder of this report consists of six sections, each covering a specific aspect of the feasibility study. Section 2 describes the basic concept, features, and possible mechanizations of the Conductron environmental control system. The functional requirements of the environmental control system for the F-4 fighter aircraft are presented in terms of operational constraints of the aircraft in Section 3. The system design parameters for the Conductron system are established in Section 4, together with size and weight constraints of the F-4 fighter aircraft. The basis for the selection of the refrigerant for the Conductron system is also presented. Section 5 discusses the performance characteristics of the major components of the Conductron system. In addition, the development status of each component is assessed. Section 6 describes the system mechanizations considered and the basis for selecting four systems for detailed analysis. The performance characteristics of each system are compared with the aircraft requirements, and the advantages and disadvantages of the candidate systems are pointed out. The system operation and control are presented, and a comparison of system size and weight is given for each system. Also included are schematic diagrams showing the thermal flow in the aircraft for each of the candidate systems. Section 7 contains the conclusions and recommendations resulting from this study.

## 1.2 Summary of Results

The results of this study show that an environmental control system based on the heat-actuated Conductron concept can be designed for the F-4 fighter aircraft meeting all performance requirements and within the established weight, size, and power limitations of the aircraft.

The Conductron system offers substantial advantages over the conventional air cycle system because of

- Full performance at all aircraft operating conditions.
- Substantial reduction in compressor bleed air requirements.
- Higher reliability and longer lifetime.

Four different system mechanizations of the Conductron system were analyzed in detail and performance characteristics determined. The differences between these systems consist in the type of cabin cooling loop selected (closed or open), in the utilization (or elimination) of ram air and bleed air expanders, and the variation in the efficiency of these components. State-of-the-art component characteristics were used throughout the analysis for all system components, except converters and vane motors. However, the projected performance improvements for these critical components of the Conductron system are realistic and attainable. A comparison of the four systems is given below:

TABLE 1-1

SUMMARY OF RESULTS

| | SYSTEM | | | |
|---|---|---|---|---|
| | A | B-1 | B-2 | C |
| (1) Type of cabin cooling loop | closed | open | open | open |
| (2) Ram air expander efficiency | 0.7 | 0.5 | 0.5 | NONE |
| (3) Bleed air expander efficiency | 0.8 | 0.5 | 0.8 | 0.8 |
| (4) Max. performance capability (100$^{o}$F cabin temperature) ICAO Standard Day - Altitude (feet) | 40,000 | 37,000 | 40,000 | 26,000 |
| Mach Number | 2.25 | 2.13 | 2.25 | 1.83 |
| (5) Bleed air requirements (% of bleed air used by air cycle) | 24 | 61 | 37 | 41 |
| (6) Estimated system weight (lbs) | 124.8 | 171.1 | 134.4 | 105.6 |
| Assumed converter efficiency: 0.35 | | | | |

Systems "A" and "B-2" are seen to meet the maximum cabin cooling requirements. Although Systems "B-1" and "C" do not meet the cooling requirements at the maximum flight conditions, they are clearly capable of providing the necessary cooling capacities under all "normal" flight speeds. The performance of these two systems can be considerably improved with a more optimum ram air scoop design.

The results of this design study lead to the conclusion that the environmental control requirements of other aircraft with less stringent design limitations and constraints can be met by a heat-actuated Conductron environmental control system. The adaptation of the Conductron system to medium range supersonic ($M \leq 1.8$) and certainly subsonic aircraft seems to be particularly attractive.

## 2.0 THE CONDUCTRON SYSTEM

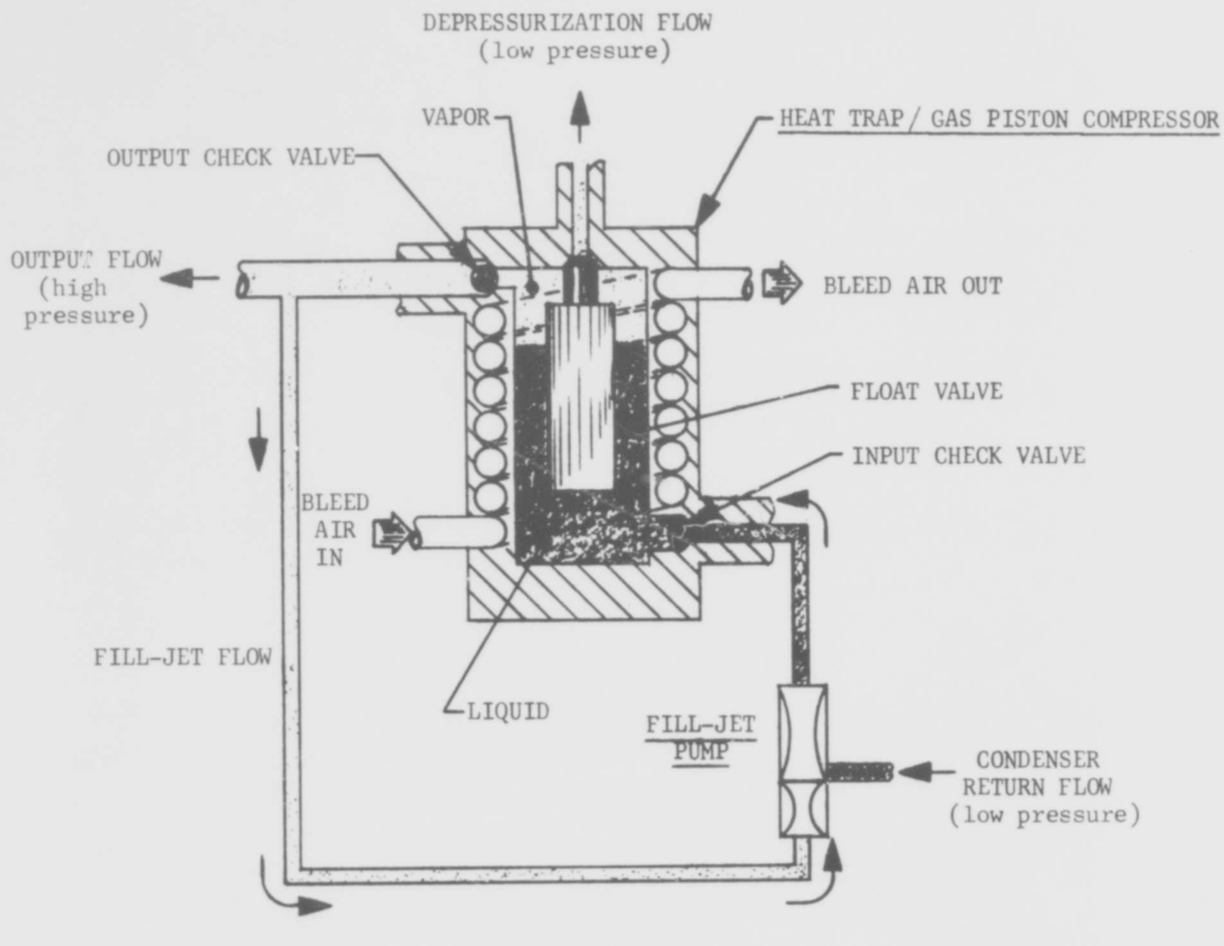
This section describes the fundamentals of the Conductron system. The thermodynamic processes involved are outlined, and a functional description of system components is given. Features of the Conductron concept are presented in terms of utilization of various heat sources and heat sinks; and possible system mechanizations for aircraft applications are briefly discussed.

### 2.1 Basic Concept

The Conductron environmental control system concept is basically a modified vapor compression refrigeration system. Thermodynamically, the difference between the Conductron concept and a conventional vapor compression cycle lies in the use of two basic thermodynamic cycles which are interconnected but serve two distinct functions: (1) thermal energy conversion (power cycle) and (2) refrigeration cooling (refrigeration cycle). The power cycle converts a steady input of heat energy into a flow of pressurized gas (refrigerant) which, in turn, powers the refrigeration cycle.

Physically, the Conductron system differs from a conventional vapor compression system in the following manner: (1) the conventional, mechanical compressor is replaced by a heat-actuated gas piston unit, incorporating a gas piston compressor, a heat trap and a fill-jet pump, (2) a converter is added, and (3) air motors can be added to drive auxiliary air blowers and/or to reduce temperature levels of the medium being processed. Other components are identical to those of the conventional system: condenser, evaporator, and expansion valve.

The gas piston unit (Figure 2.1) represents the prime-mover for the system. The unique part of this unit is the gas piston compressor, whose operation is based on the large volume expansion of liquid as it vaporizes. As the liquid boils off in a constant volume container, the pressure increases until a desired pressure level is reached. This high-pressure vapor flows out through an output check valve, thus supplying the system requirements. When the liquid is almost completely vaporized, the residual vapor in the gas piston compressor is depressurized to the condenser by means of a buoyant float valve. After depressurization, a new supply of liquid enters the container and the process is repeated. The second part of the gas piston unit is the heat trap, which is any device that supplies the required amount of heat to the liquid to be vaporized. The actual configuration of the heat trap depends largely upon the heat source being used. In the case of compressor bleed air or engine exhaust gas as the heat source, the heat trap would be an air-to-liquid boiling heat exchanger. The third part of the gas piston unit is a fill pump, which assures the supply of liquid to




TYPICAL GAS PISTON UNIT



FIGURE 2.1

the gas piston compressor after depressurization. This pumping action is accomplished by an ejector-like device, which uses a small quantity of high-pressure gas from the gas piston compressor as the motive flow to pump the desired amount of liquid from the condenser. Use of such a fill-jet pump makes the filling operation gravity insensitive.

The converter is the second component of special importance to the Conductron system. It is a device which raises the pressure of the refrigerant vapor leaving the evaporator to a point where condensation can occur at non-refrigerating or elevated temperatures. This function can be accomplished by many devices; however, serious interest is centered only in those devices having no moving parts, such as ejectors (Figure 2.2).

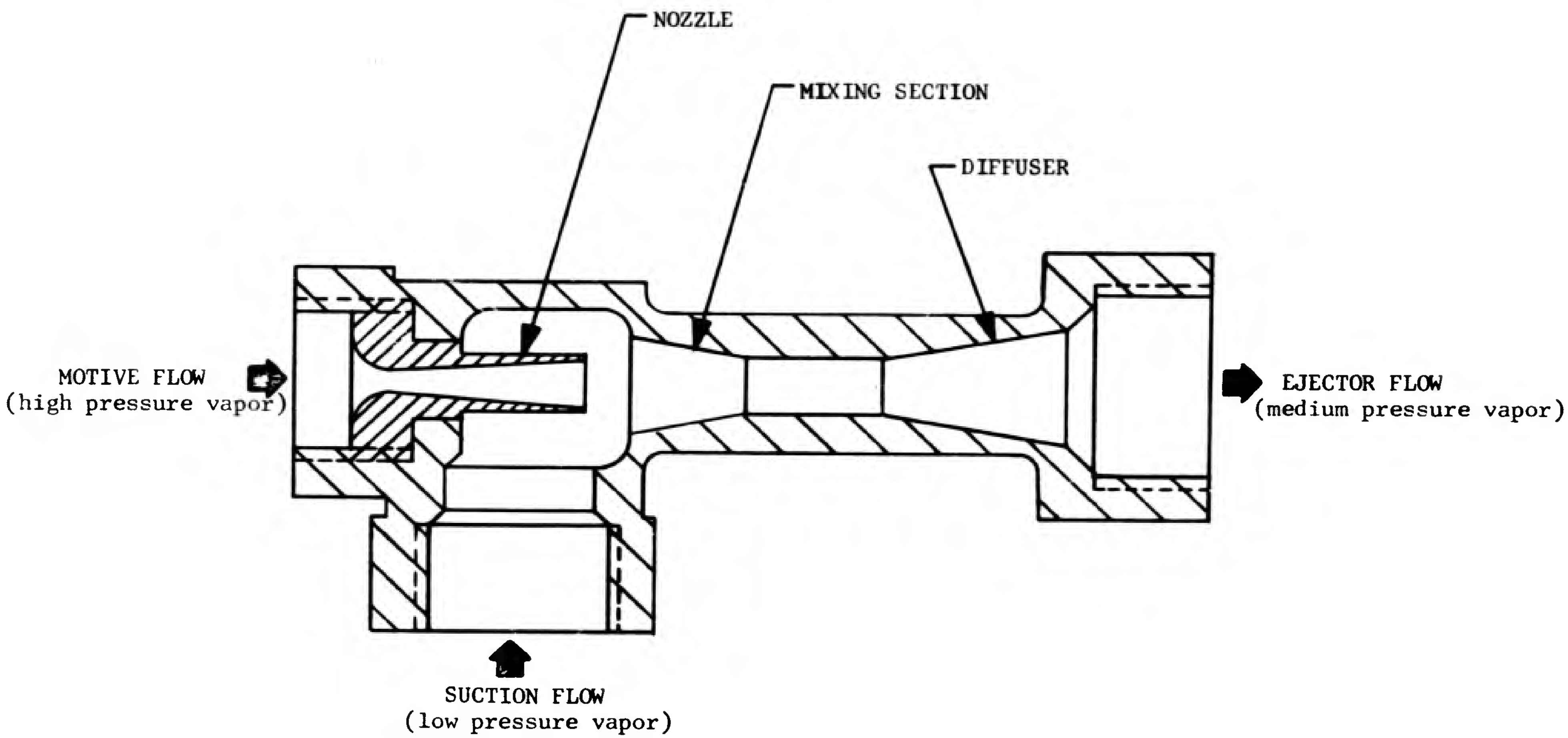
The third unusual, but not essential, component of the Conductron system is the air motor (Figure 2.3), which transforms the energy of flowing air into rotary energy. Four or six vane designs have efficient operating speeds in the ranges suitable for heat exchanger blowers. The expansion of the high-pressure air through the air motor is accompanied by an air temperature drop, allowing the utilization of the air motor as a means of obtaining a lower temperature heat sink for the condenser.

A schematic diagram of the basic Conductron environmental control system concept is shown in Figure 2.4. All major system components are included: gas piston unit, ejector, condenser, and evaporator on the refrigerant side; air motor on the air side. Statepoints corresponding to the thermodynamic cycle are indicated.

The idealized thermodynamic cycle of the Conductron environmental control system is shown in Figure 2.5. The constant volume and constant pressure processes (1-2 and 2-3, respectively) describe the functions of the gas piston compressor. The ejector operation is shown by the two isentropic processes 3-4 and 6-4. The gas piston compressor depressurization is a constant enthalpy process 3-7. The condenser is shown by the constant pressure processes 4-1 and 7-1. The expansion of the liquid through an expansion valve is represented by the constant enthalpy process 1-5. The evaporator function is the constant pressure process 5-6. It must be noted that the thermodynamic process in the ejector as shown is not a true representation of the actual process. A more correct description of the ejector thermodynamic process will be presented later.

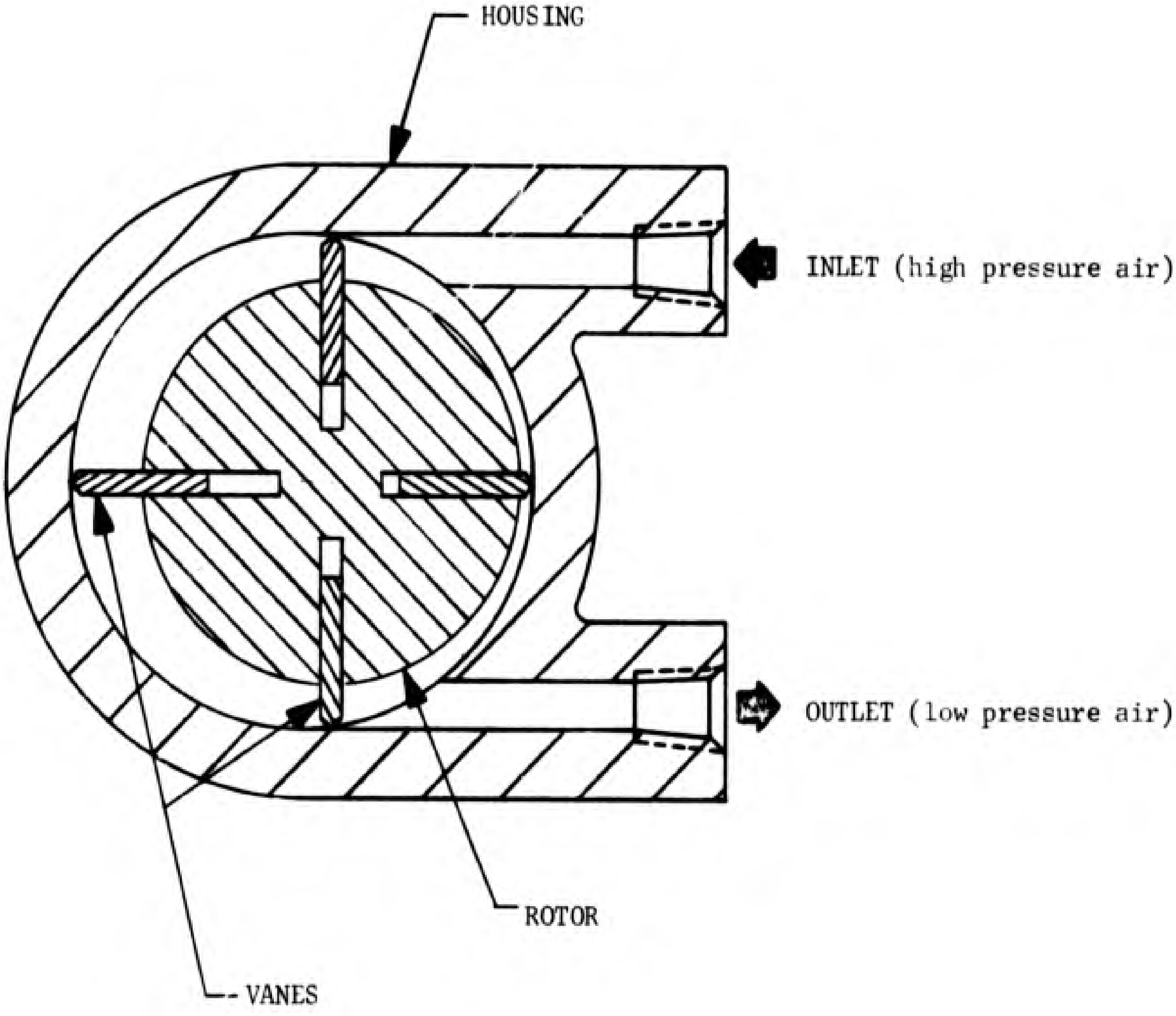
## 2.2 System Features

The mechanization of the Conductron environmental control concept is inherently related to the practical utilization of heat sources and heat sinks. With respect to aircraft applications, the availability of waste heat sources for system actuation represents a particular performance advantage. Since the available heat energy from most of these




TYPICAL EJECTOR CONVERTER

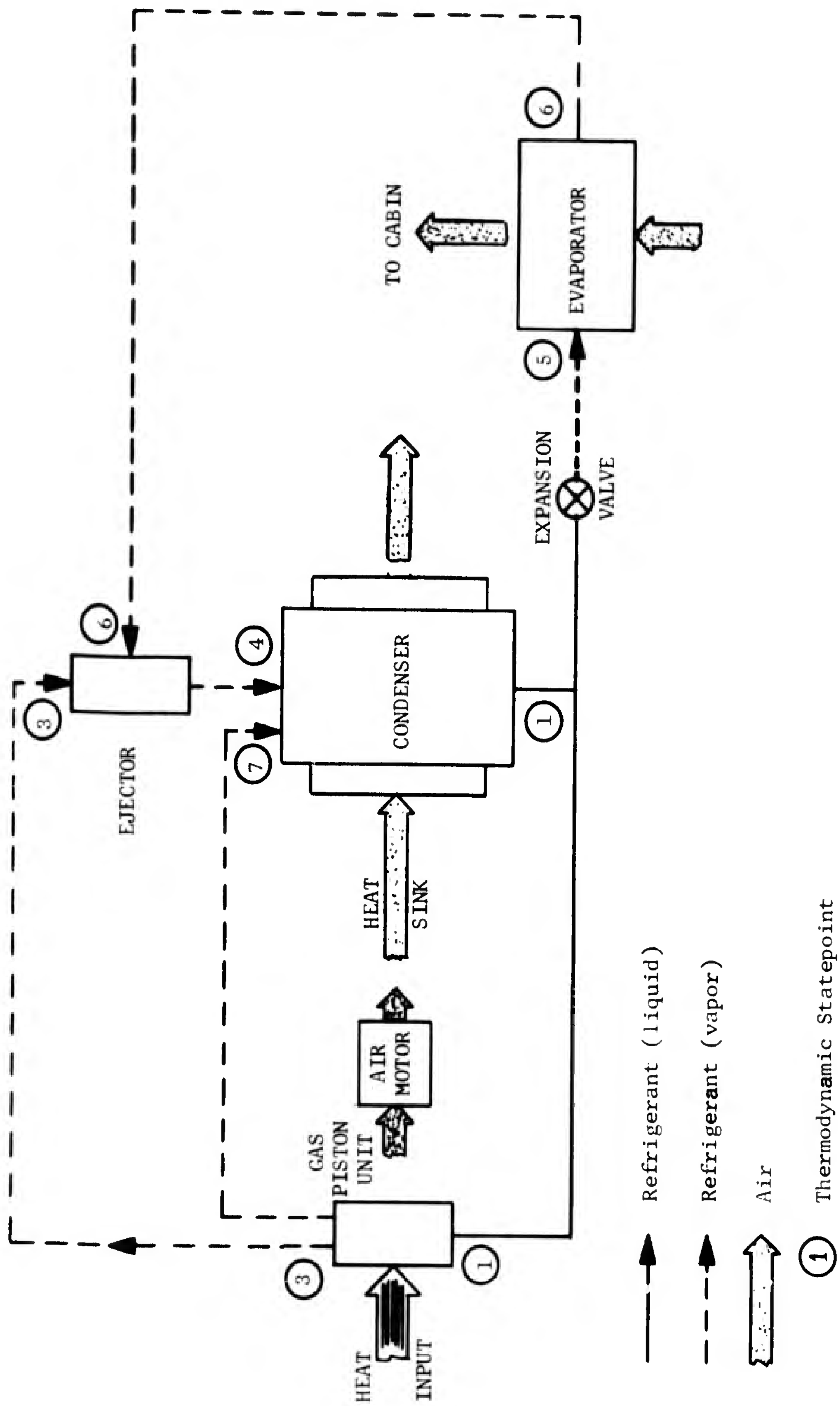
FIGURE 2.2




TYPICAL VANE MOTOR



FIGURE 2.3




BASIC CONDUCTRON SYSTEM
SCHEMATIC DIAGRAM



FIGURE 2.4

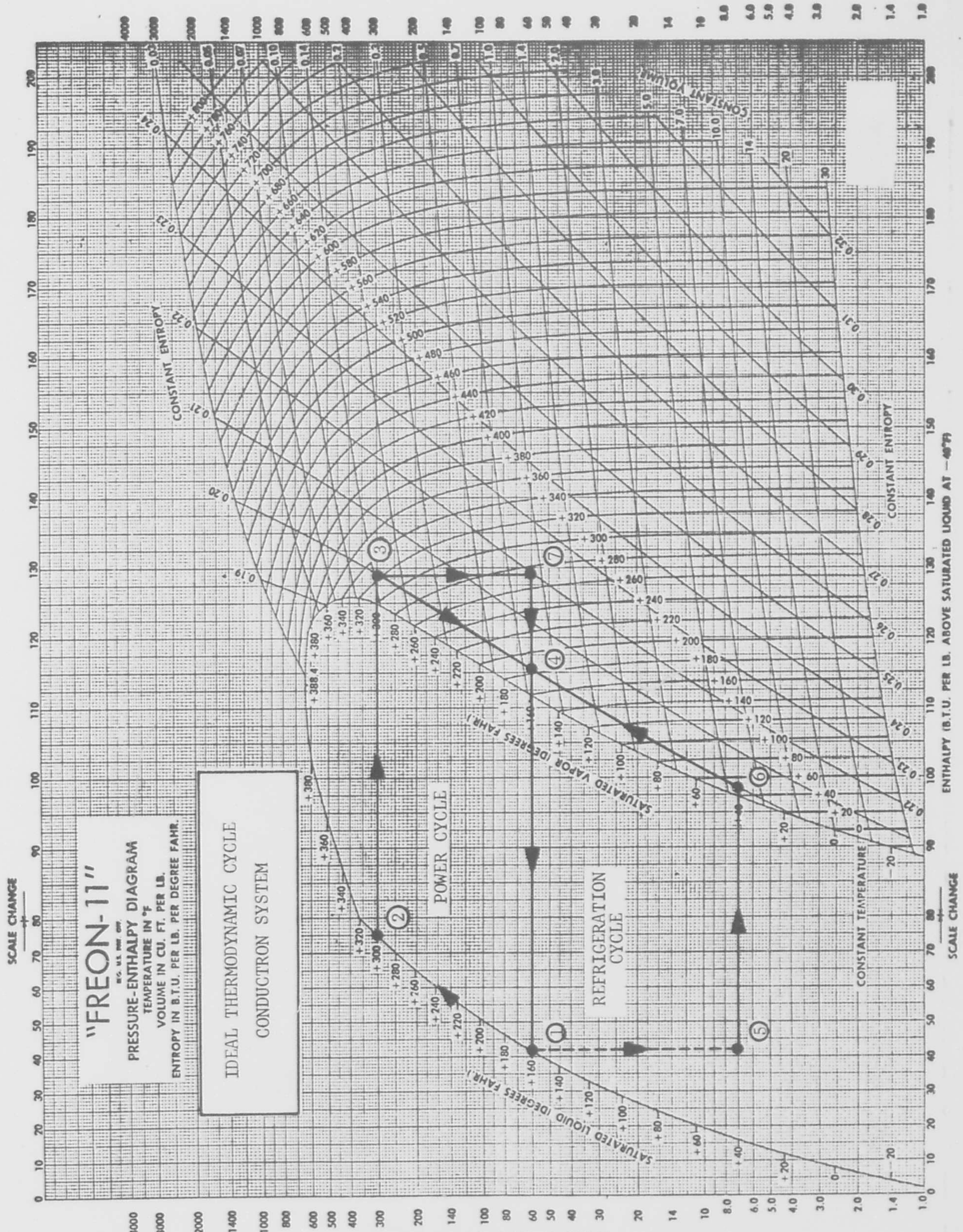

"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN °F
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
IDEAL THERMODYNAMIC CYCLE
CONDUCTRON SYSTEM
POWER CYCLE
REFRIGERATION CYCLE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT TEMPERATURE
SCALE CHANGE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT −40°F)

FIGURE 2.5

heat sources is much greater than required for system actuation, minimum system size and weight can be attained by trading weight and design convenience with system efficiency. The variety of heat source and heat sink combinations available in aircraft for use with the Conductron system allows considerable flexibility in component packaging and system installations in a size and weight smaller than that of conventional vapor cycle equipment, and comparable to lower capacity air cycle equipment. The selection of an optimum system design is therefore related to aircraft installation and operation constraints.

The following sources of heat are suitable for actuating the Conductron environmental control system:

1) Direct engine exhaust heat
2) Indirect engine exhaust heat
3) Compressor bleed air
4) Fuel burning

Engine exhaust gases represent the largest heat source available. It can be readily shown that only a small portion of the available exhaust gas flow is required for system actuation. Effective tapping of this supply is readily possible in non-reaction type engines, such as reciprocating and turboprop engines. Extraction of heat from the exhaust gases of reaction turbojets is somewhat more difficult because disturbance of the controlled nozzle-like exhaust flow must be avoided. Thus, the utilization of engine exhaust gases for system actuation must consider certain constraints being placed on design freedom by particular flow characteristics of the engine exhaust.

Indirect engine exhaust heat refers to the fact that the surface temperatures of the engine exhaust ducting approach the high temperature levels of the exhaust gases, thereby allowing indirect extraction of heat from the exhaust gases by using the hot duct surfaces as heating elements. Some limitations exist due to space restrictions around engines; however, when space is available, heat extraction can be achieved without extensive engine or aircraft modifications. In general, only a portion of the total exhaust gas heat energy is recoverable in this manner.

Compressor bleed air is a particularly convenient source of heat. Under most engine operating conditions, the compressor bleed air temperature is sufficiently high for system actuation. The relatively high compressor bleed air pressure allows extremely compact heat trap designs. The limitations of this heat source relate to the relatively low bleed air temperature during engine idle conditions and the fact that bleed-off of compressor air represents a reduction in engine power. However, in most cases the compressor bleed air flow rates available are sufficient for actuating the Conductron system.

Direct fuel burning is an alternate source of heat which has the advantage of versatility. It is independent of engine operation and has no restrictions as to placement. In this case the heat energy is not "free" but then neither is the supply energy for any competitive system. Direct fuel burning systems become very desirable when the heat from waste sources becomes either inadequate or inconvenient.

From this discussion, it becomes apparent that heat actuation represents something of a natural for aircraft. This is especially true when it is recognized that delivered cooling requirements necessitate heat actuation energy levels which are only a relatively small portion of the heat available. As a result, considerable room exists for size-weight versus efficiency trade-offs.

Related to any refrigeration system is the need for adequate heat sinks for rejection of heat. In aircraft applications, the following heat sinks are possible:

1) Ram or ambient air
2) Compressor bleed air
3) Fuel
4) Airframe

Ram air is the most common source for heat rejection used in aircraft. It is readily available and usable at most flight conditions. During static conditions, the flow of ram air is non-existent, and flow of ambient air must be induced by positive force techniques. Electric or air motor-driven blowers can be used to provide the desired cooling air flow. In propeller-driven aircraft, propeller wash may be utilized to move air across heat exchangers. At supersonic flight conditions, the ram air temperature can reach high enough levels to make it completely ineffective as a heat sink. In order to utilize ram air as a heat sink, auxiliary means for lowering the high air temperatures must be provided.

Compressor bleed air offers the possibility of a heat sink if it is used as the actuating heat source for the Conductron system. The compressor bleed air leaving the heat trap is usually cooled to around $300^{o}F$, and expansion of this gas through an air motor or some other expansion device reduces this temperature to a level where it can serve as a suitable condenser heat sink. In addition, because of the relatively high pressure level of the compressor bleed air, a higher than usual pressure drop condenser can be used, resulting in a substantial decrease in condenser size.

Fuel is a heat sink which is worthy of consideration either by itself or in combination with other heat sinks. Although it is an inherently large heat sink, its usefulness is somewhat restricted by the rate of fuel consumption, fuel tank capacity, the accessibility to the fuel tanks, and placement of heat exchangers. A further limitation is due to the fact that usually this heat sink is already being utilized by other on-board systems, such as the hydraulic system.

Use of the aircraft frame and structure as a heat sink is possible, but mechanization may be difficult because of the large contact areas required for heat transfer. However, this method may be effective as a supplementary heat sink in combination with any of the other heat sinks.

## 2.3 System Mechanization

The versatility of the Conductron system lends itself to many and varied aircraft applications due to its relative independence on choice of heat source or heat sink and its extreme flexibility in component arrangements. This unique feature makes the system exceedingly attractive for applications where available space is at a minimum, allowing utilization of small unused spaces, as well as placement of components in an optimum manner: the gas piston unit near the heat source, the evaporator close to the cabin, and the condenser in such a position as to minimize pressure drops due to ducting.

The mechanization of the Conductron environmental control system involves the varying use of the available sources for system actuation, heat rejection, and cabin cooling. Any combination of heat sources and heat sinks previously mentioned can be mechanized. However, once specific aircraft constraints are introduced, most of the mechanizations must be eliminated as impractical, leaving only a small number for actual consideration. Two approaches to cabin cooling are possible with the Conductron system:

a) Open cycles, where ambient and/or compressor bleed air is processed continuously through the system components and rejected from the aircraft after absorption of heat in the cabin.

b) Closed cycles, where the cabin air is recirculated continuously through the evaporator. Since cabin leakage cannot be avoided, compressor bleed air must be used for make-up of leakage, resulting in a semi-closed system. Contaminant concentrations are normally kept within safe limits due to the relatively high leak rates of aircraft cabins.

The choice of system mechanization must be based on the particular requirements and constraints of the aircraft under consideration. Installation constraints, weight and size restrictions, desired environmental control system performance characteristics, and the aircraft operating conditions are all important factors which determine every system selection and mechanization.

## 3.0 PERFORMANCE REQUIREMENTS

The McDonnell F-4 fighter aircraft was selected as the aircraft best typifying the environmental control requirements of Navy high-performance aircraft. The F-4 environmental control system consists of two separate and independent units--a cabin package and an equipment package. Since this feasibility study of the Conductron system deals specifically with cabin environmental control, only the performance requirements of the cabin package are considered.

The following functions are performed by the cabin package: (1) Cabin cooling; (2) Cabin pressurization; (3) Pressure-suit cooling; and (4) Windshield rain removal. This section describes the requirements for these functions, thus establishing the performance specifications for the Conductron system.

Figure 3.1 is a schematic representation of the F-4 aircraft cabin environmental control system functions.

### 3.1 Cabin Cooling

Aircraft compartment cooling loads are composed of several elements with heat transfer by convection, radiation, and conduction occurring in the following manner:

a) Convection between the boundary layer and the outer skin of the aircraft,

b) Convection between the interior cabin surfaces and the cabin air,

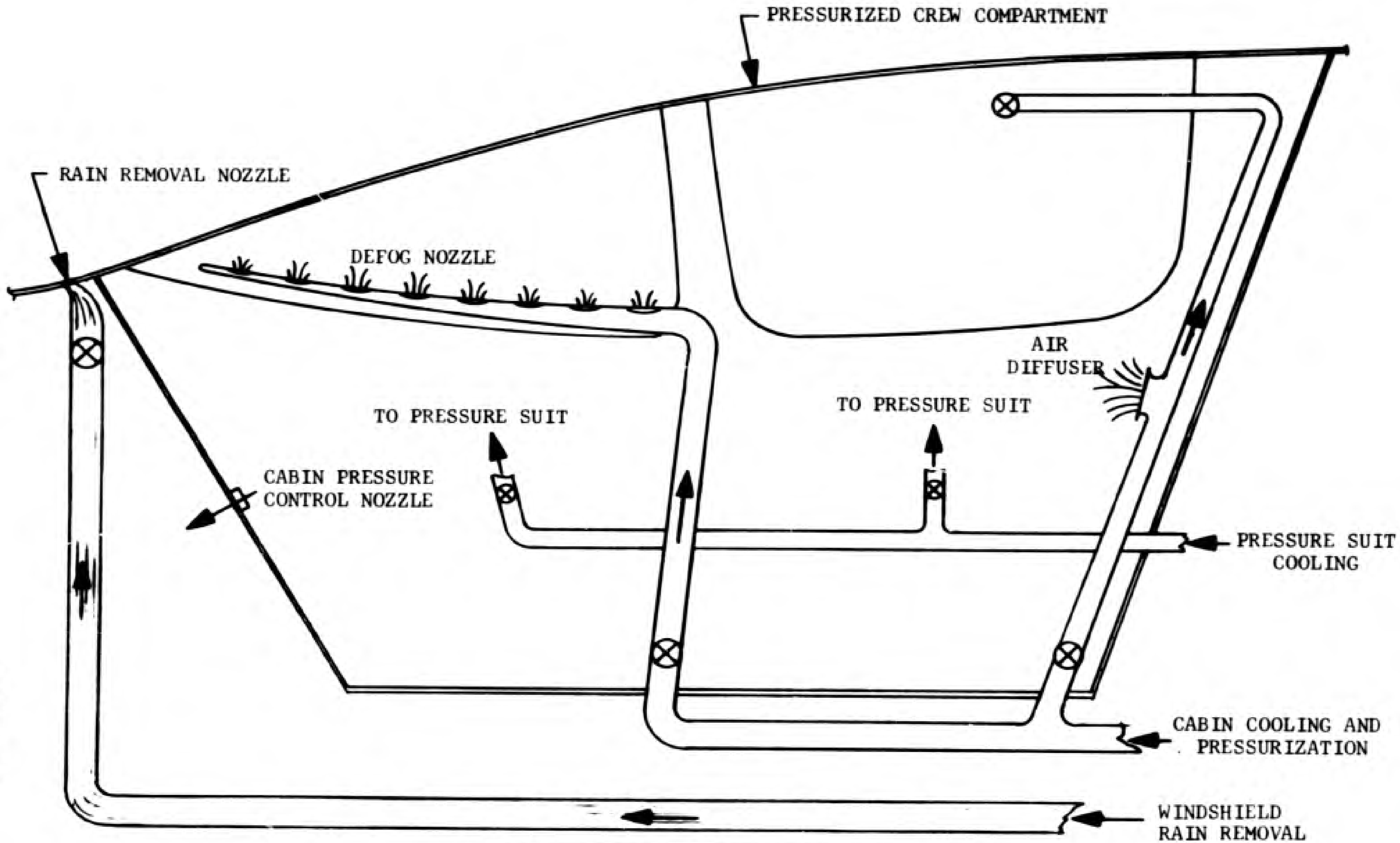
c) Convection between the cabin air and personnel or equipment,

d) Convection and radiation from internal heat sources, such as electronic equipment,

e) Radiation between the external skin of the aircraft and the external environment,

f) Solar radiation through transparent areas directly upon the crew, equipment, and the interior surfaces of the cabin,

g) Conduction through the cabin walls and structural members.




F-4 ENVIRONMENTAL CONTROL SYSTEM
CABIN PACKAGE
FUNCTIONAL SCHEMATIC

FIGURE 3.1

The cabin cooling load is generally considered to consist of heat transfer through external and internal compartment surfaces, solar radiation load, personnel load, and cabin electronics load. Since the peak electronics load, personnel load, and internal heat transfer rates are relatively constant, variations in cabin cooling load are primarily due to changes in the heat transfer rates through external compartment surfaces and solar radiation, and thus are a function of:

- Flight speed (or Mach number)
- Altitude
- Cabin temperature
- Temperature-Day

The cabin cooling loads for the F-4 fighter aircraft are expressed in terms of these variables in Figures 3.2 through 3.7. The data shown in these figures is taken from Reference 1. The effect of aerodynamic heating on cabin cooling load is so pronounced that the design cabin temperature is raised to 100°F at high supersonic speeds. For subsonic and low supersonic operating conditions, the design cabin temperature is 70°F.

The maximum cabin cooling load of 38,500 BTU/hr is experienced on a Standard Day at an altitude of 40,000 feet and maximum flight speed (Mach 2.25). The maximum cabin heating load of 18,500 BTU/hr occurs on a Cold Day at sea level and Mach 0.25.

Figure 3.8 shows the maximum cabin cooling loads for the F-4 fighter aircraft as a function of altitude and temperature-day. This figure is based on the design conditions of the F-4 refrigeration system, (Reference 2).

F-4 CABIN COOLING LOADS
FOR 70°F CABIN TEMPERATURE
ICAO STANDARD DAY

The following loads are included:

1. Heat transfer through external compartment surfaces.
2. Heat transfer through internal compartment surfaces.
3. Solar radiation load.
4. Personnel load.

NOTE: Negative values indicate heating loads.

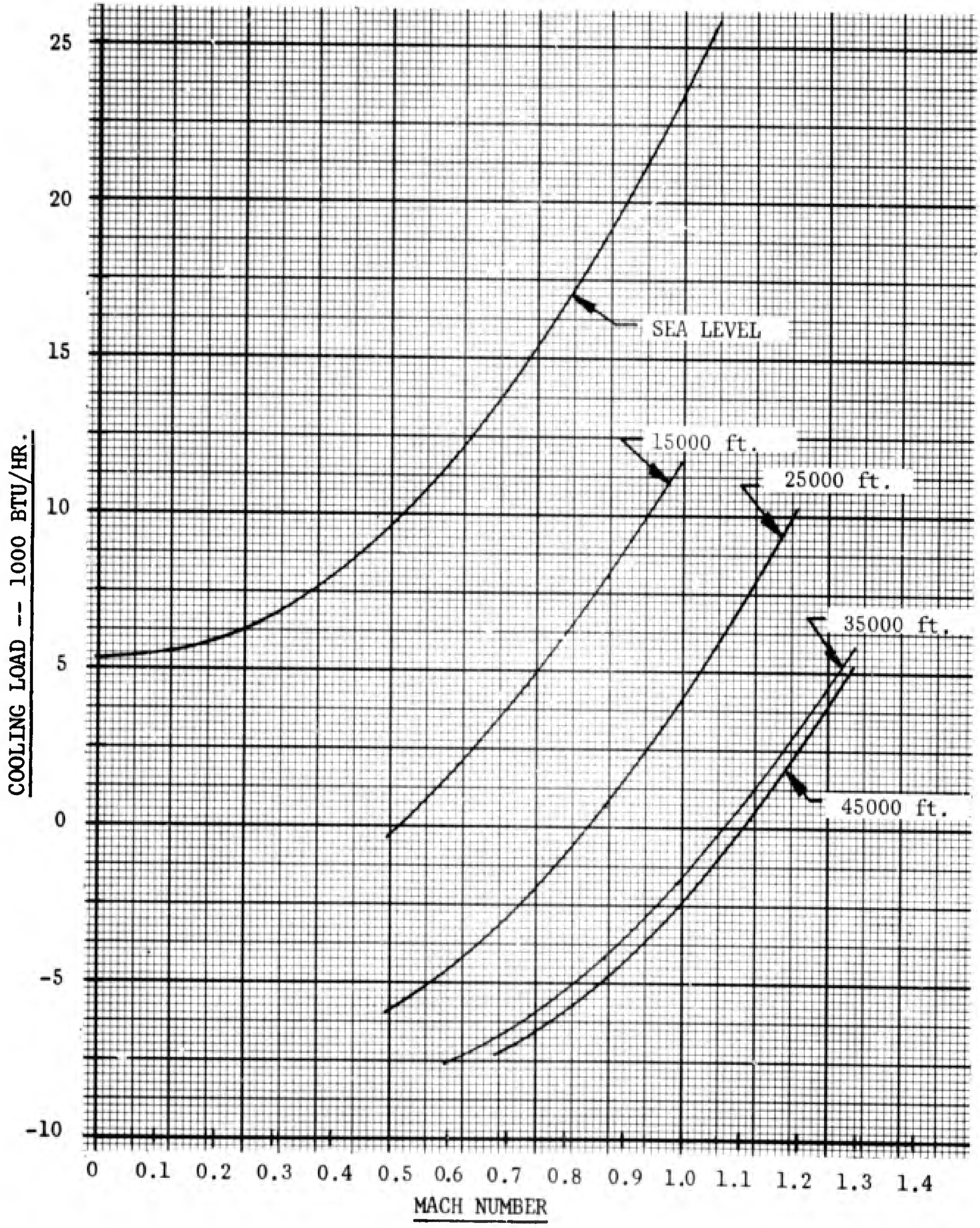





FIGURE 3.2

F-4 CABIN COOLING LOADS
FOR 70°F CABIN TEMPERATURE
ANA HOT DAY

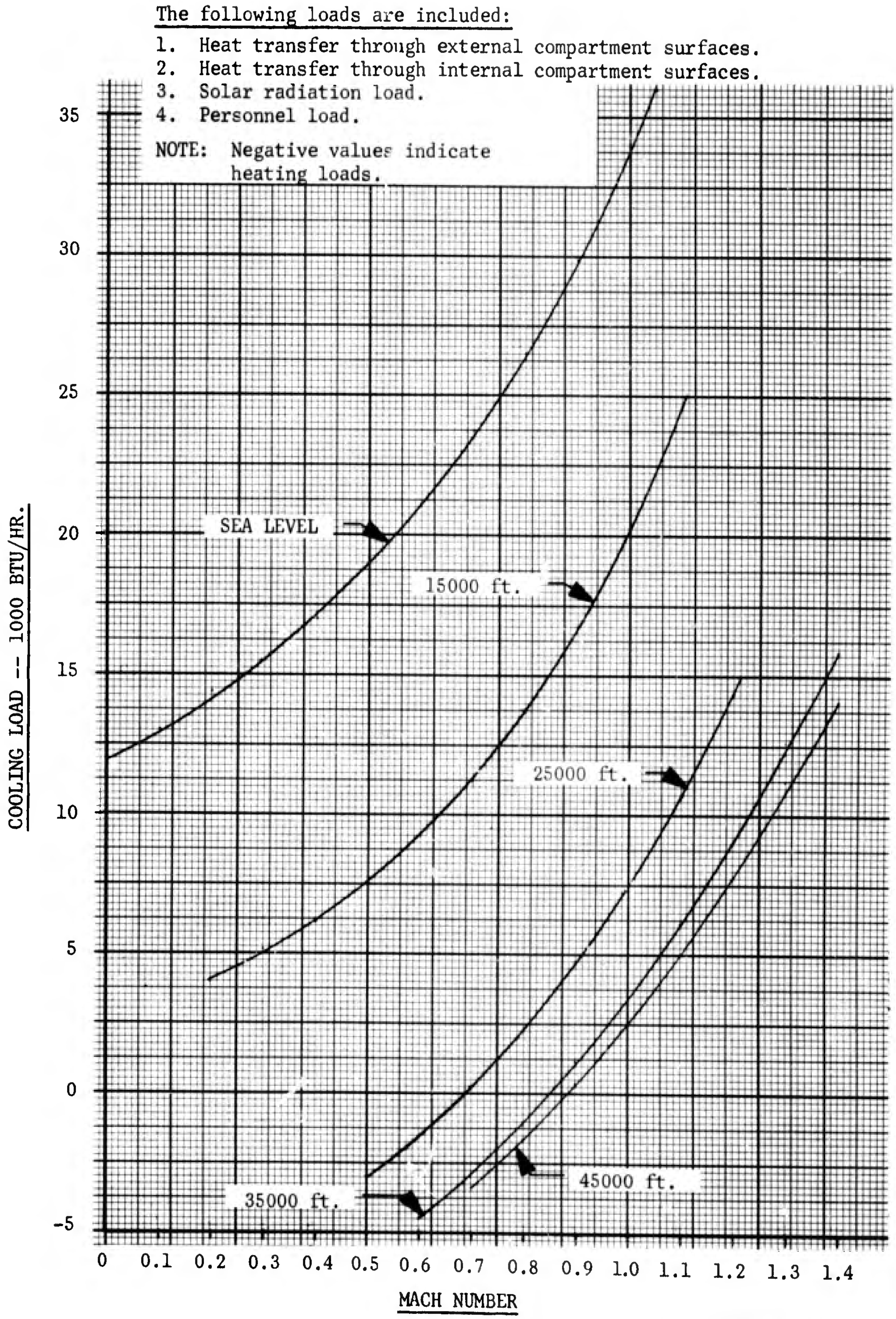





FIGURE 3.3

F-4 CABIN COOLING LOADS
FOR 70°F CABIN TEMPERATURE
ANA COLD DAY

The following loads are included:

1. Heat transfer through external compartment surfaces.
2. Heat transfer through internal compartment surfaces.
3. Solar radiation load.
4. Personnel load.

NOTE: Negative values indicate heating loads.

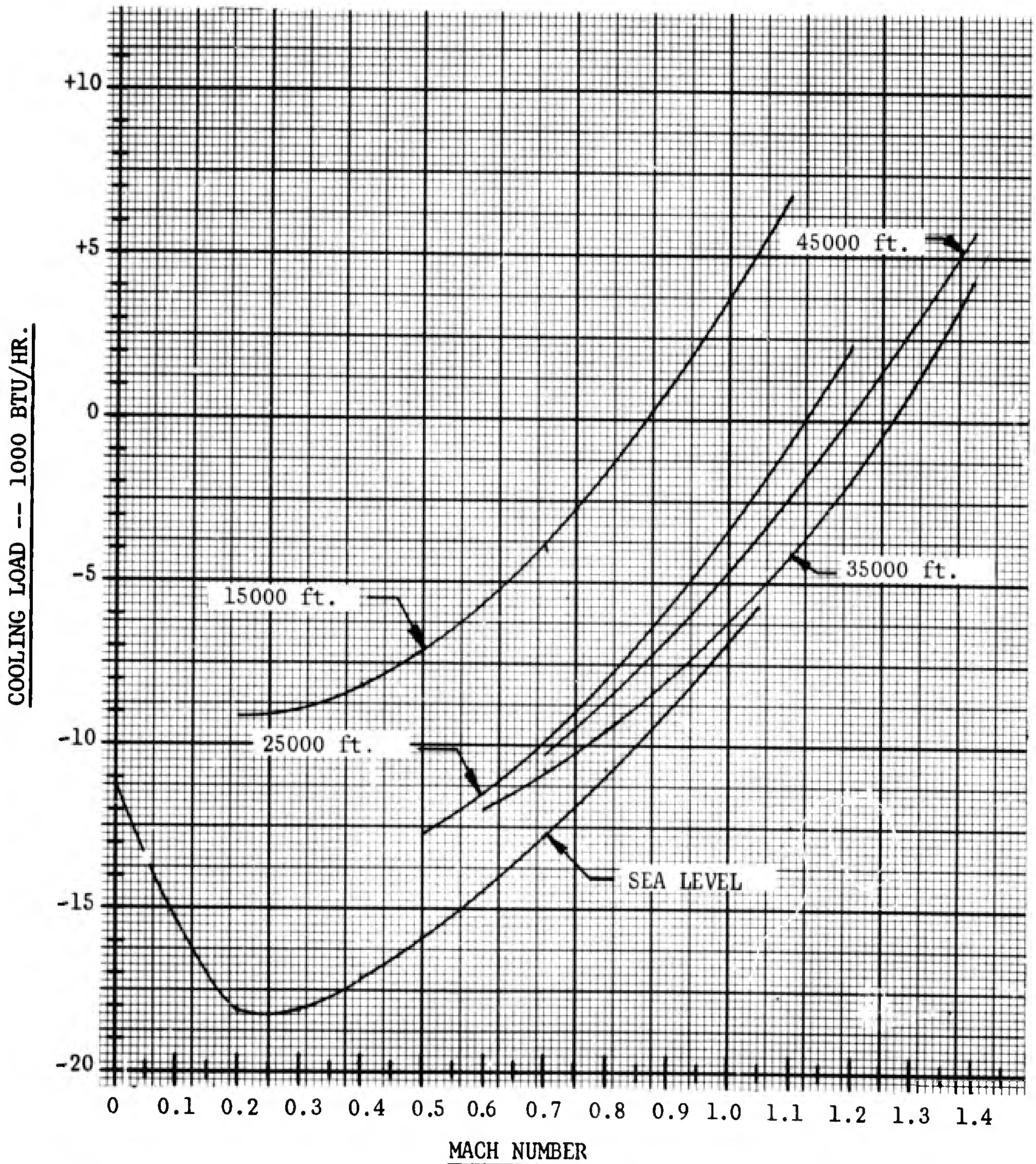





FIGURE 3.4

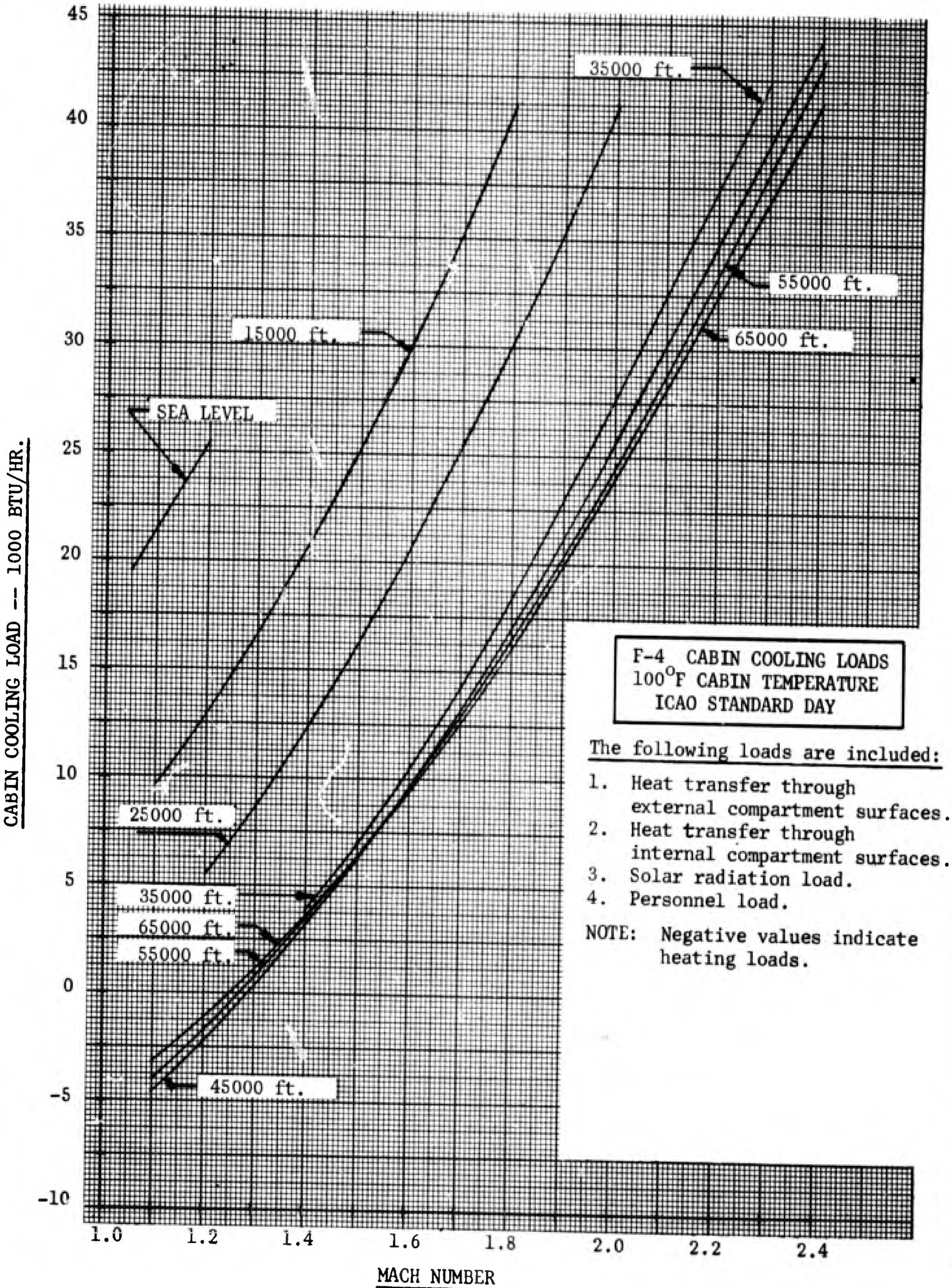
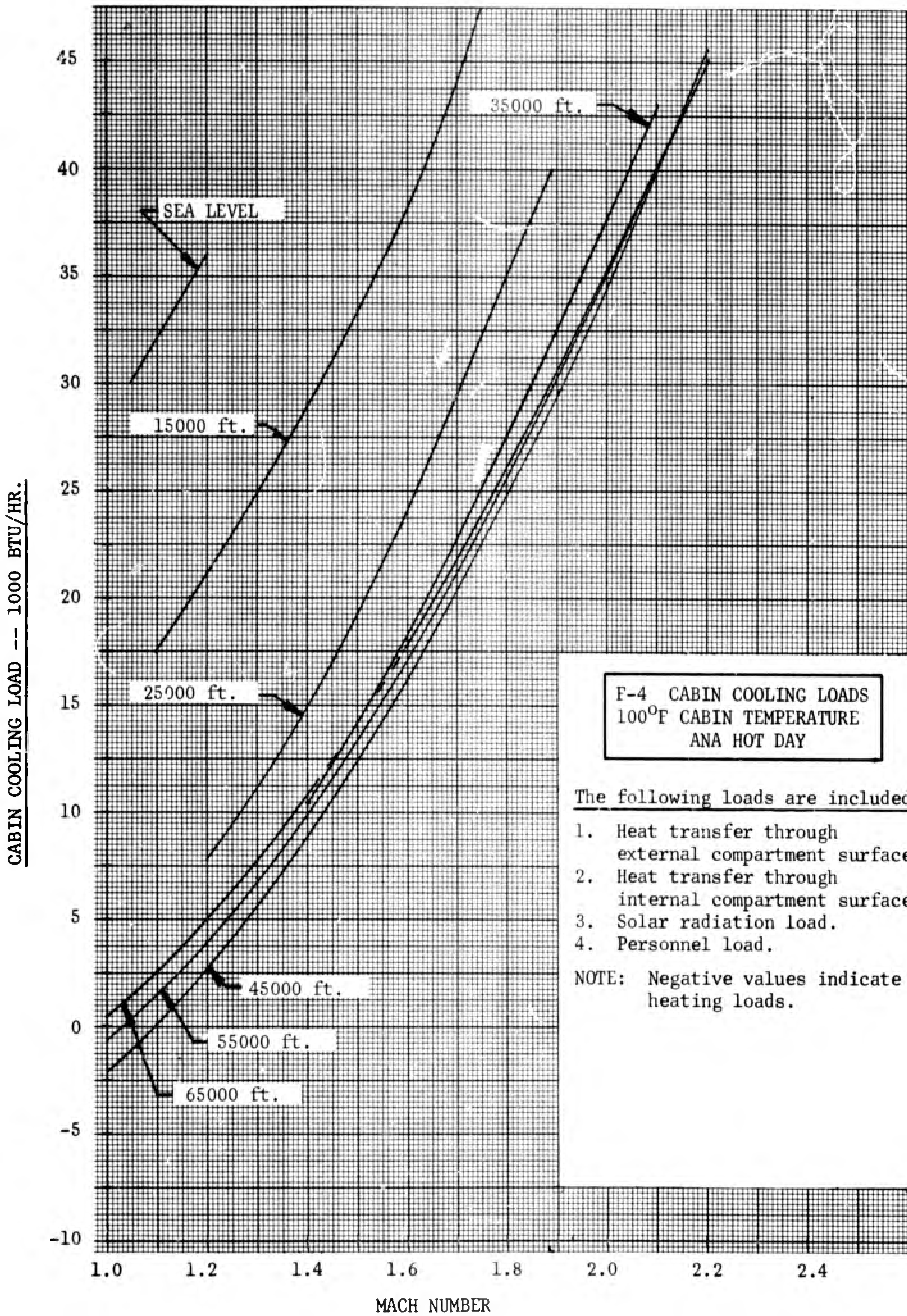

F-4 CABIN COOLING LOADS
$100^{o}$F CABIN TEMPERATURE
ICAO STANDARD DAY
The following loads are included:
1. Heat transfer through external compartment surfaces.
2. Heat transfer through internal compartment surfaces.
3. Solar radiation load.
4. Personnel load.
NOTE: Negative values indicate heating loads.
CABIN COOLING LOAD -- 1000 BTU/HR.
MACH NUMBER
SEA LEVEL
15000 ft.
25000 ft.
35000 ft.
45000 ft.
55000 ft.
65000 ft.
45
40
35
30
25
20
15
10
5
0
-5
-10
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4



FIGURE 3.5




F-4 CABIN COOLING LOADS
$100^oF$ CABIN TEMPERATURE
ANA HOT DAY

The following loads are included:

1. Heat transfer through external compartment surfaces.
2. Heat transfer through internal compartment surfaces.
3. Solar radiation load.
4. Personnel load.

NOTE: Negative values indicate heating loads.



FIGURE 3.6

F-4 CABIN COOLING LOADS
$100^{o}F$ CABIN TEMPERATURE
ANA COLD DAY

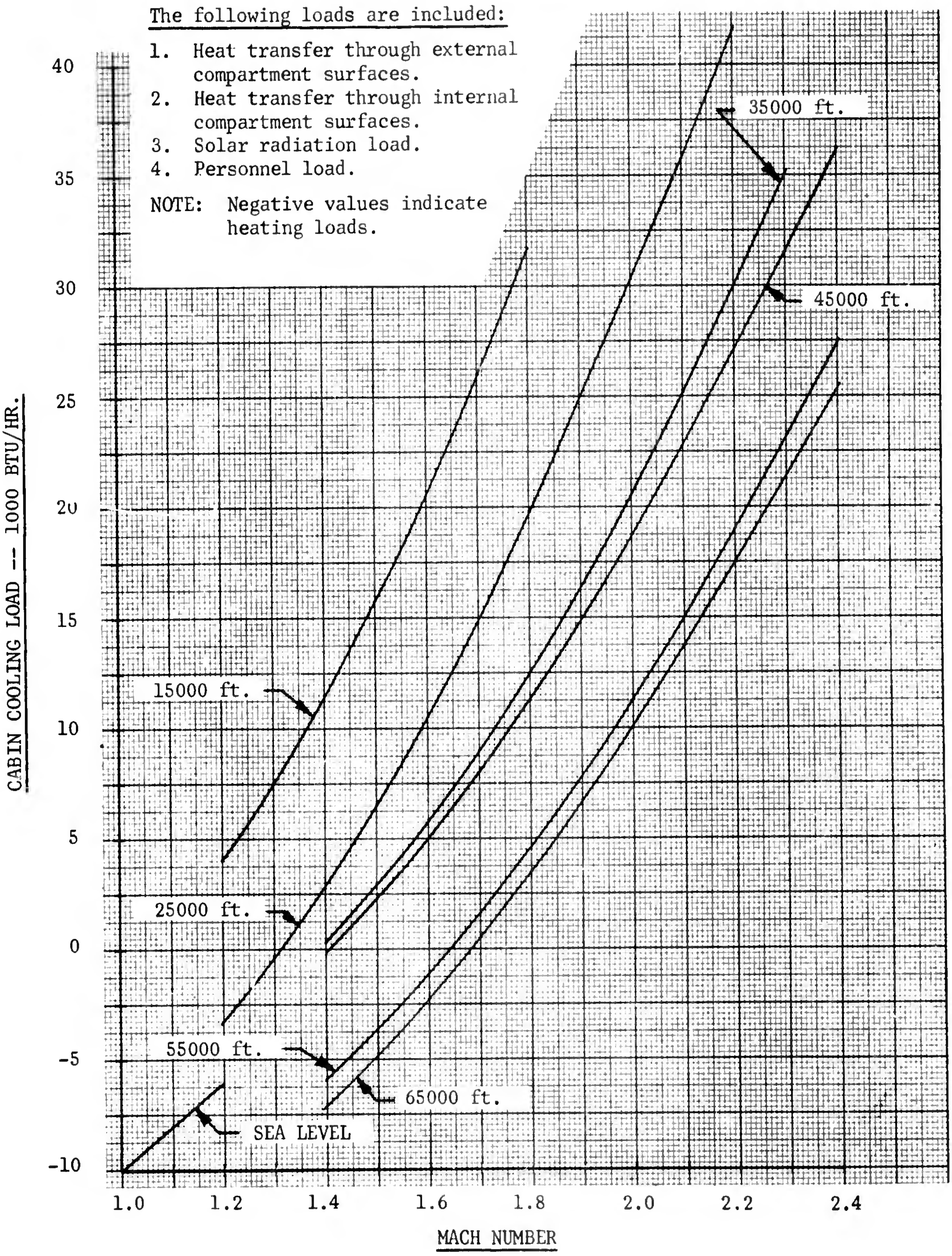





FIGURE 3.7

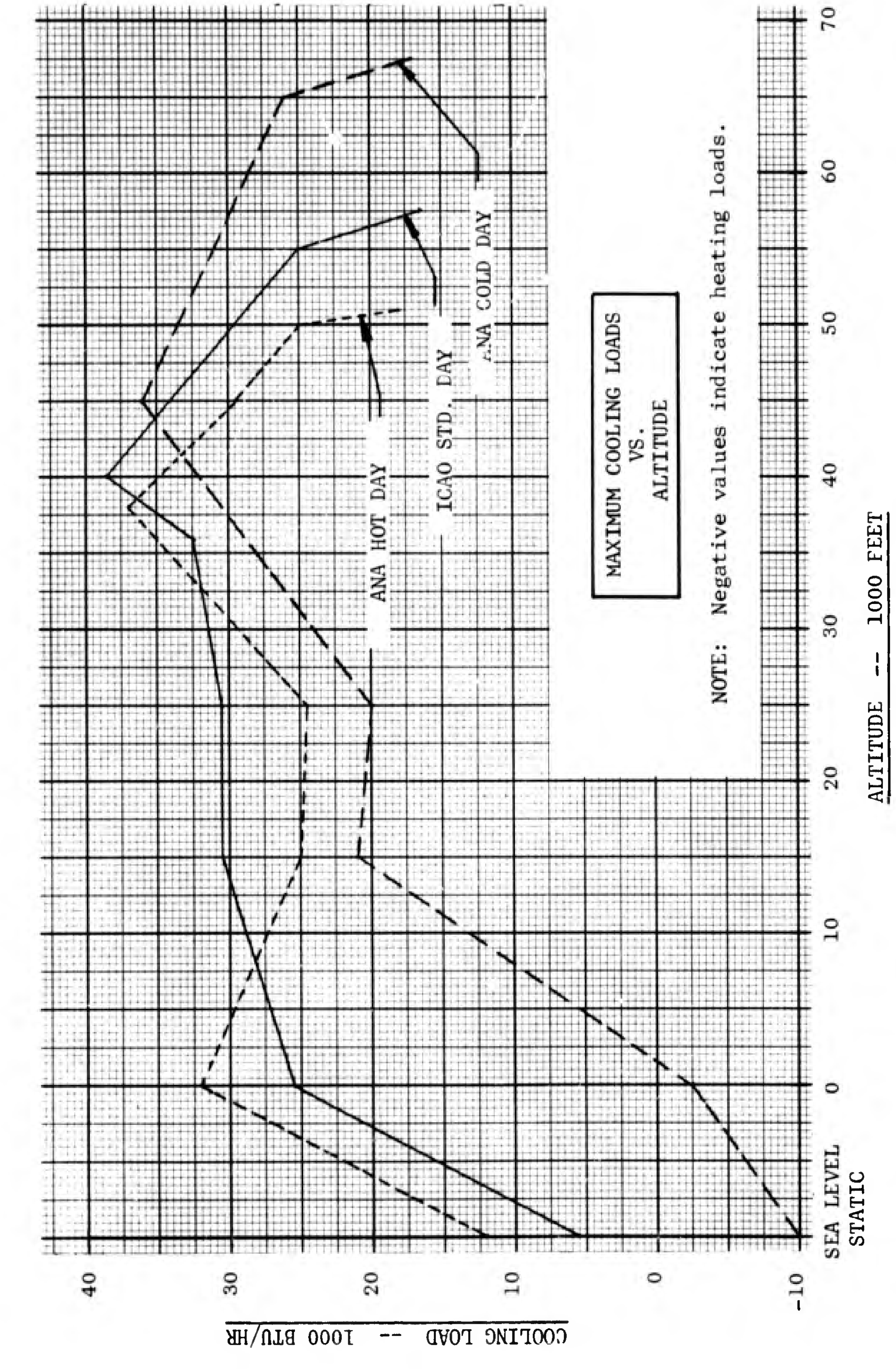

MAXIMUM COOLING LOADS
VS.
ALTITUDE
ANA HOT DAY
ICAO STD. DAY
ANA COLD DAY
NOTE: Negative values indicate heating loads.
COOLING LOAD -- 1000 BTU/HR
ALTITUDE -- 1000 FEET
SEA LEVEL STATIC
0
10
20
30
40
50
60
70
-10
0
10
20
30
40

FIGURE 3.8

The air flow rates required for cabin cooling are a function of cabin cooling load, desired cabin temperature, and cooling air inlet temperature.

The cooling load is related to the required air flow by the following expression:

$$(1)\ldots\ldots\ldots\ldots \quad Q_c = \dot{w}_c c_p (t_o - t_i)$$

where:

$Q_c$ = cabin cooling load (BTU/hr)

$\dot{W}_c$ = cabin air flow (lb/hr)

$C_p$ = specific heat of air at constant pressure (BTU/lb- $^{o}$F)

$t_o$ = air outlet temperature ($^{o}$F)

$t_i$ = air inlet temperature ($^{o}$F)

$t_c$ = cabin temperature ($^{o}$F)

Assuming the cabin temperature to be equal to the maximum or outlet air temperature:

$$t_c = t_o$$

$$(2)\ldots\ldots\ldots\ldots \quad Q_c = \dot{w}_c c_p (t_c - t_i)$$

or

$$(3)\ldots\ldots\ldots\ldots \quad \dot{w}_c = \frac{Q_c}{c_p(t_c - t_i)}$$

for the cabin temperatures of interest (70$^{o}$F to 100$^{o}$F),

$C_p$ = 0.240 BTU/lb$_m$ - $^{o}$F.

Therefore,

$$(4)\ldots\ldots\ldots\ldots \quad \dot{W}_c = \frac{Q_c}{0.24\,(t_c - t_i)}$$

$$(5)\ldots\ldots\ldots\ldots \quad \dot{W}_c = 4.166\,\frac{Q_c}{(t_c - t_i)}\;, \text{ LBS/HR}$$

or

$$(6)\ldots\ldots\ldots\ldots \quad \dot{W}_c = 0.0694\,\frac{Q_c}{(t_c - t_i)}\;, \text{ LBS/MIN}$$

The cabin air flow requirements for maximum cooling and heating loads as a function of air inlet temperature are shown in Figures 3.9 and 3.10, respectively. The curves are based on the following flight conditions:

| | | |
|---|---|---|
| Cooling loads: | Curve 1 | ICAO Standard Day<br>40,000 ft; Mach 2.25<br>38,500 BTU/hr |
| | Curve 2 | ANA Hot Day<br>38,000 ft; Mach 2.0<br>37,000 BTU/hr |
| | Curve 3 | ANA Cold Day<br>45,000 ft; Mach 2.4<br>36,000 BTU/hr |
| Heating loads: | Curve 1 | ICAO Standard Day<br>35,000 ft; Mach 0.6<br>-7,000 BTU/hr |
| | Curve 2 | ANA Hot Day<br>35,000 ft; Mach 0.6<br>-5,000 BTU/hr |
| | Curve 3 | ANA Cold Day<br>Sea level; Mach 0.25<br>-18,500 BTU/hr |

CABIN AIR FLOW REQUIREMENTS
FOR
MAXIMUM COOLING LOADS

CABIN TEMPERATURE: 100°F

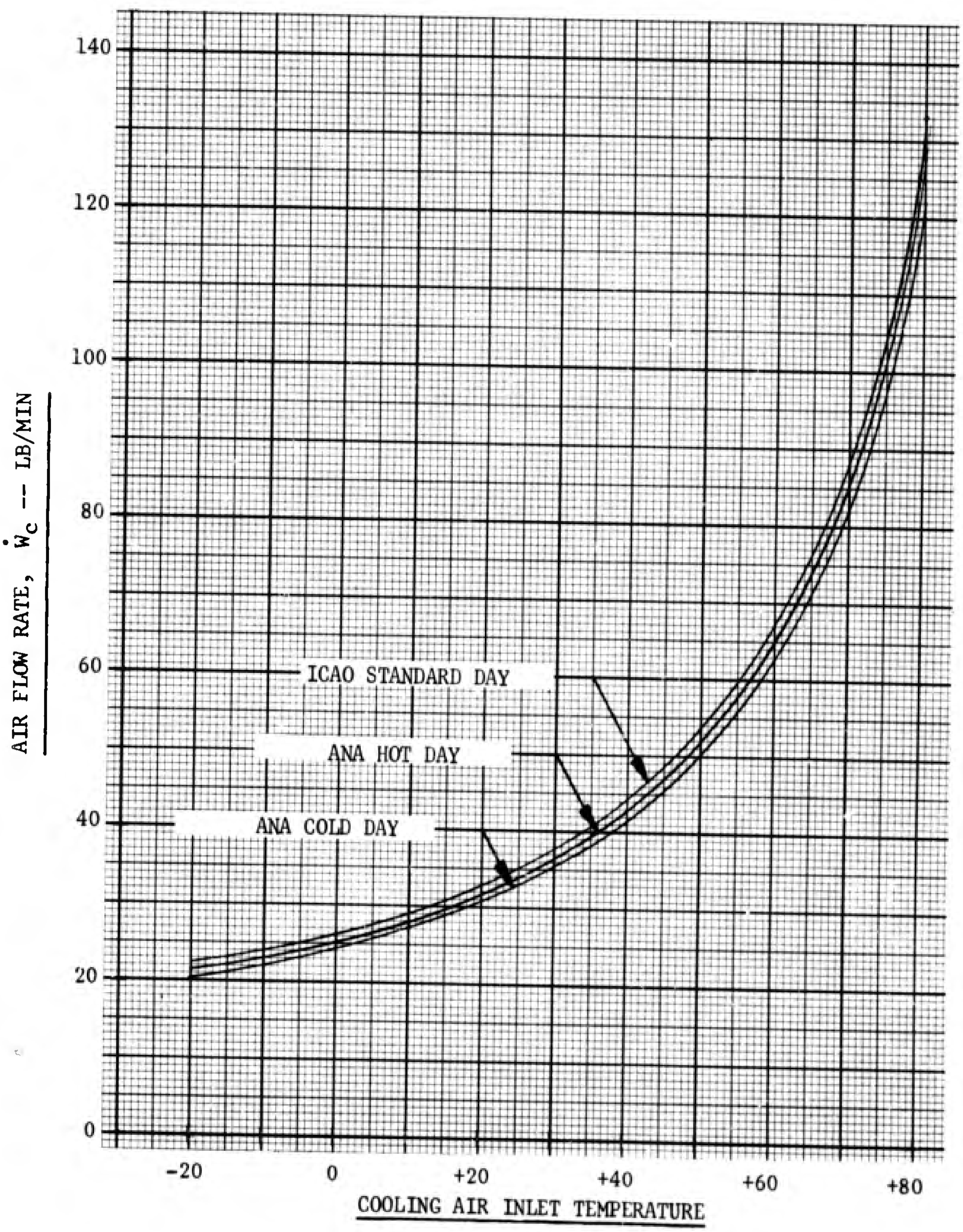



FIGURE 3.9

CABIN AIR FLOW REQUIREMENTS
FOR
MAXIMUM HEATING LOADS

CABIN TEMPERATURE: 70°F

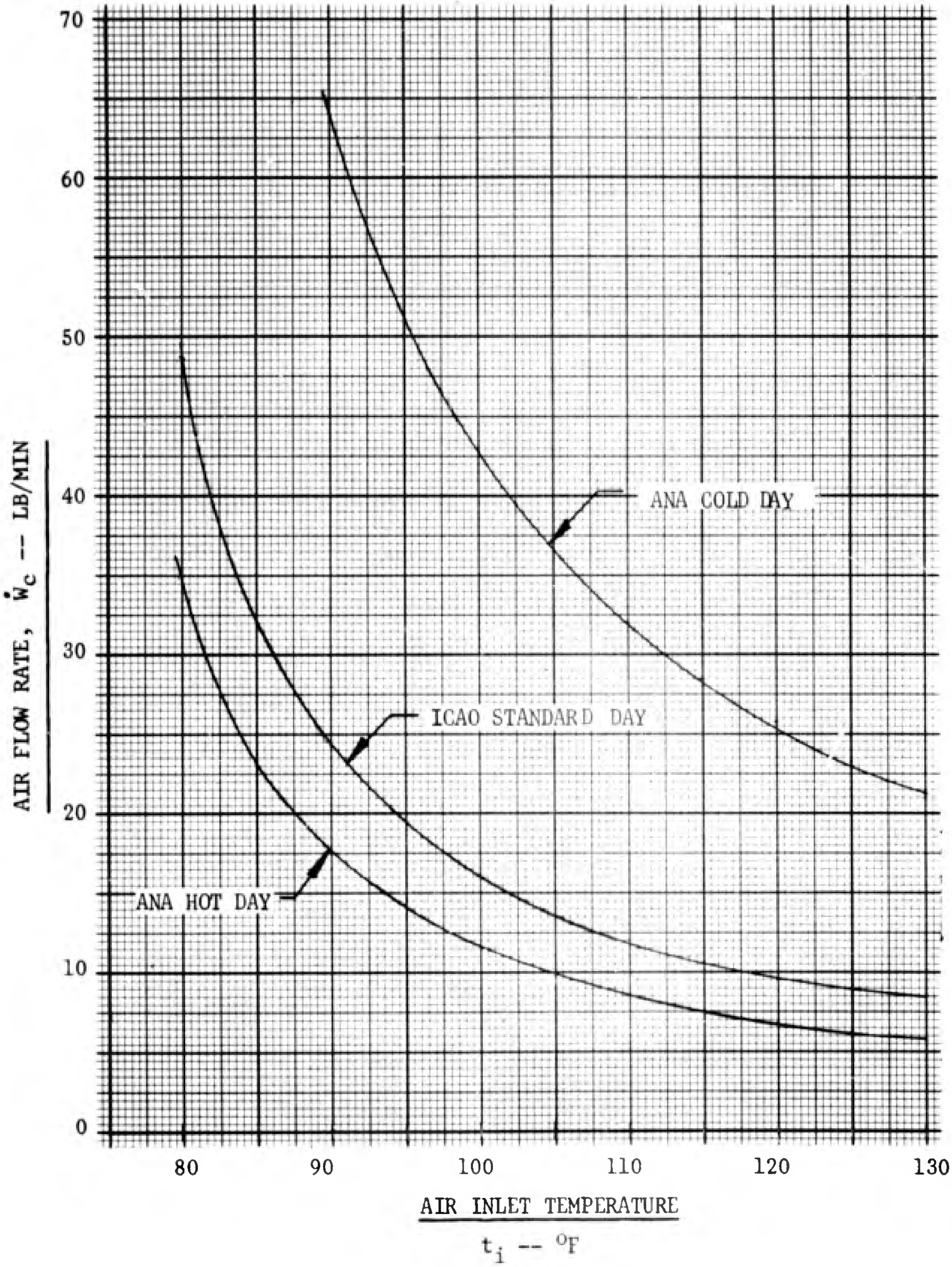





FIGURE 3.10

## 3.2 Cabin Pressurization

The cabin pressurization requirements of the F-4 fighter aircraft are based on those established by the armed services for all combat aircraft (Reference 3 ). Briefly stated, the cabin is unpressurized from sea level to 8,000 feet, is maintained at 8,000 feet pressure altitude until the differential to ambient reaches 5 psig, and is maintained at this differential pressure thereafter.

Figure 3.11 shows the required cabin pressure levels as a function of altitude.

The desired cabin pressure is obtained by controlling the exhaust of the cabin air. The cabin pressurization air flow is that required to make up the leakage from a hole with an effective area, CA, of 0.42 sq.in. and with the existing pressure differentials. (Reference 4).

The general form of the equation describing the flow of a compressible fluid through an orifice or a nozzle is given by:

(1)............ $$\dot{W} = \frac{C_D C_M A_o P_c}{\sqrt{T_c}} \quad , \text{ LBS/SEC}$$

where:

$C_D$ = orifice coefficient

$C_M$ = weight flow parameter $\sim f(P_a/P_c \; ; \sqrt{{}^\circ R}/\text{SEC})$

$A_o$ = throat area $(in^2)$

$P_c$ = upstream pressure $(lb/in^2)$

$P_a$ = downstream pressure $(lb/in^2)$

$T_c$ = upstream temperature $({}^oR)$

The weight flow parameter, $C_m$, may be expressed as a function of the pressure ratio across the orifice or nozzle, as shown in Figure 3.12 (Reference 5 ).

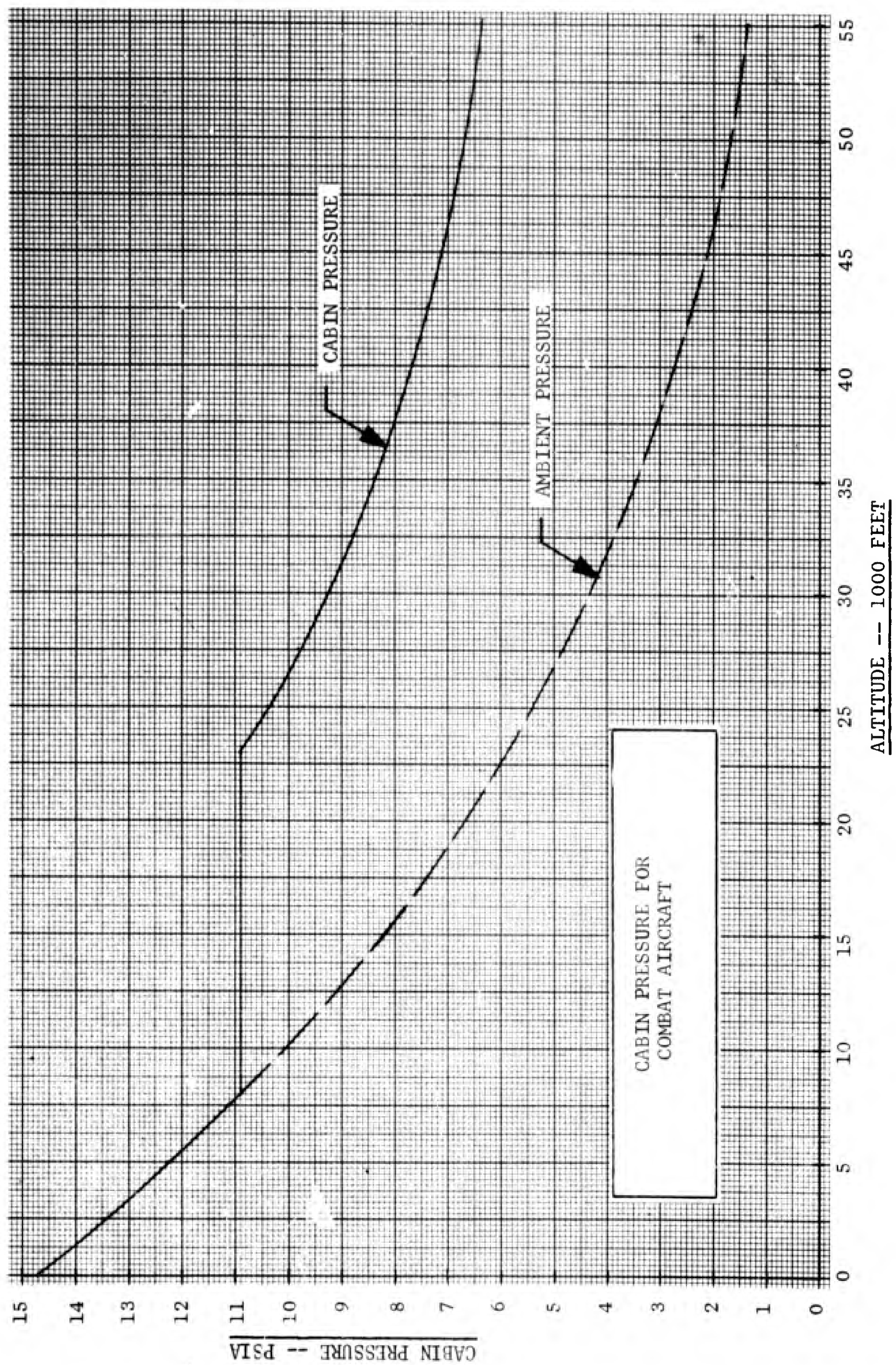
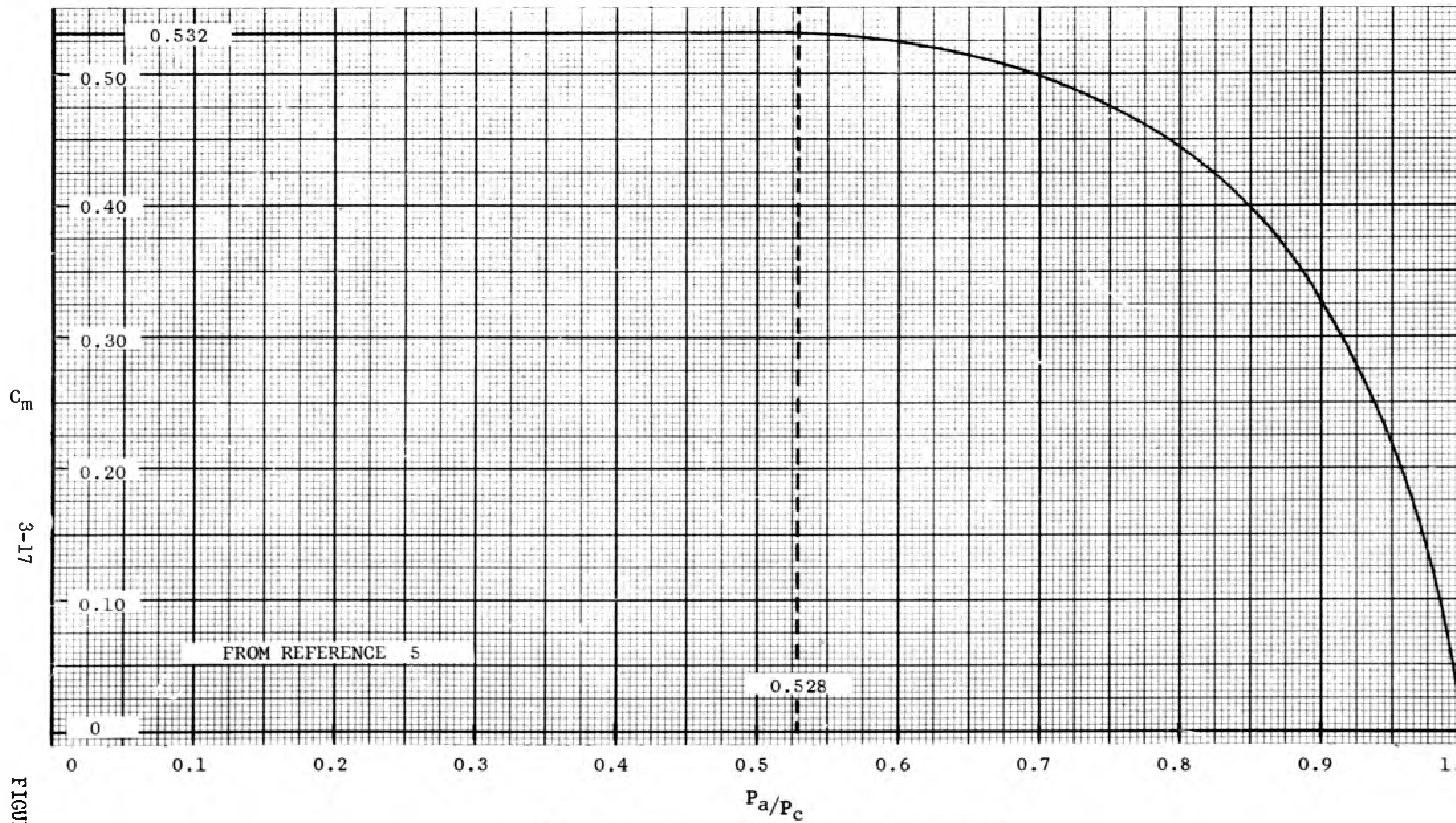

CABIN PRESSURE
AMBIENT PRESSURE
CABIN PRESSURE FOR
COMBAT AIRCRAFT
ALTITUDE -- 1000 FEET
CABIN PRESSURE -- PSIA
0
5
10
15
20
25
30
35
40
45
50
55
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15



FIGURE 3.11




ORIFICE WEIGHT FLOW PARAMETER, $C_m$
VS.
PRESSURE RATIO ACROSS ORIFICE, $P_a/P_c$

FIGURE 3.12

For an orifice with a CA of 0.42 sq.in., the cabin pressurization air flow is given by:

$$(2)\ldots\ldots\ldots\ldots \dot{W}_p = 25.2 \frac{C_M P_c}{\sqrt{T_c}} \quad , \text{ LBS/MIN}$$

For cabin temperatures of 70°F and 100°F, this equation reduces to:

$$(3)\ldots\ldots\ldots\ldots \dot{W}_p = 1.095\ C_M P_c \quad , \text{ LB/MIN} \quad (\text{for } t_c = 70\,°F)$$

$$(4)\ldots\ldots\ldots\ldots \dot{W}_p = 1.065\ C_M P_c \quad , \text{ LB/MIN} \quad (\text{for } t_c = 100\,°F)$$

The required cabin pressurization air flows are shown as a function of altitude in Table 3-1 and Figure 3.13. The effect of cabin temperature on pressurization air flow is shown to be very small for the temperature range of interest (70°F to 100°F). The variation of pressure and temperature with altitude for various atmosphere standards was taken from References 6 and 7.

Leakage rates exceeding those specified above can be handled by the Conductron system. The additional compressor bleed air flows are readily available, and the components in the cabin pressurization line can be sized to accommodate the required flow rates.

TABLE 3-1

CABIN PRESSURIZATION AIR FLOW REQUIREMENTS

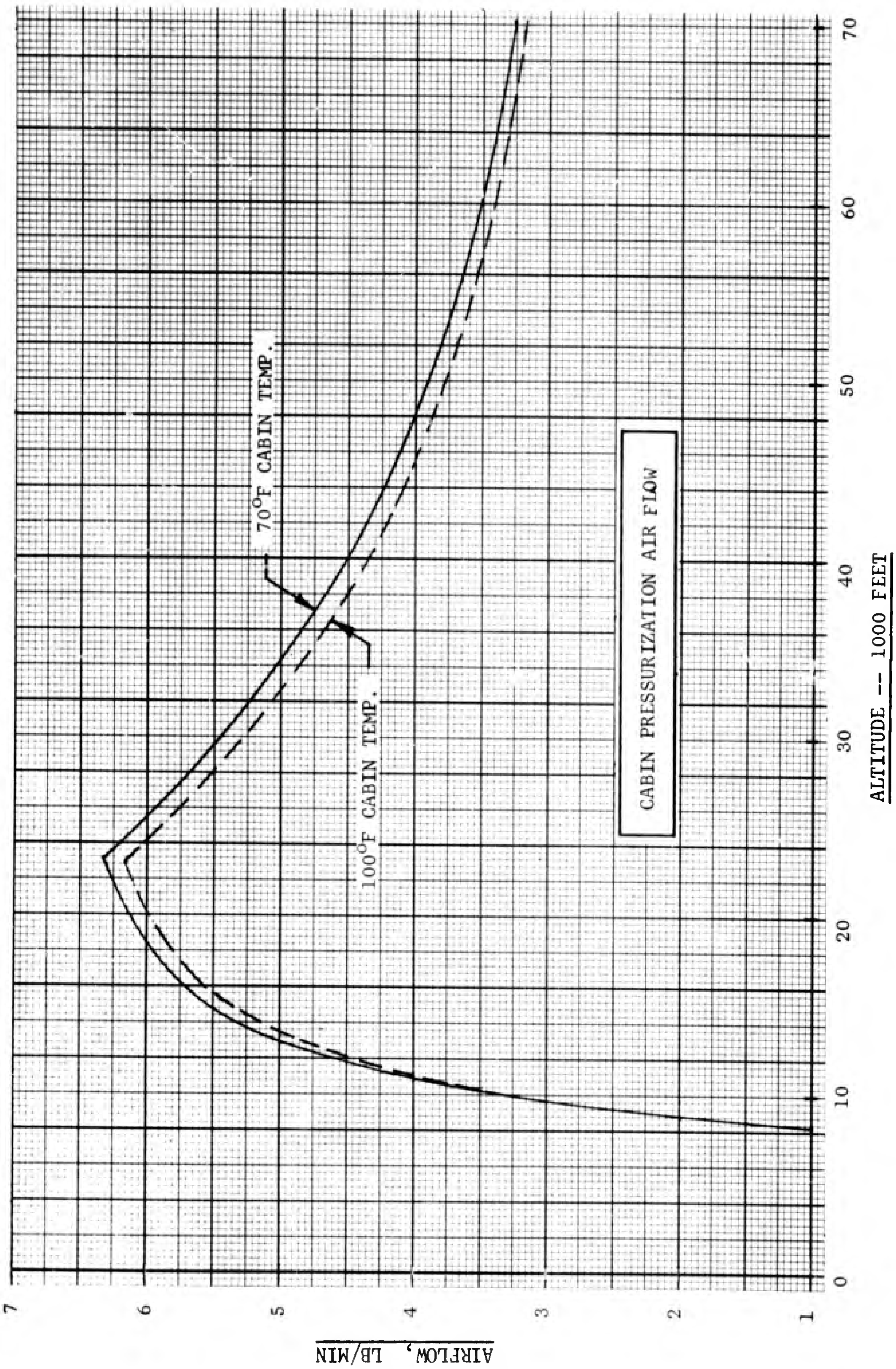
| ALTITUDE (feet) | $P_a$ (psia) | $P_c$ (psia) | $P_a/P_c$ | $C_m$ ($\sqrt{°R}$/sec) | $\dot{W}_p$ (lb/min) | |
|---|---|---|---|---|---|---|
| | | | | | $t_c$=70°F | $t_c$=100°F |
| 8,000 | 10.90 | 10.90 | 1.0 | 0 | 0 | 0 |
| 10,000 | 10.10 | 10.90 | .927 | .27 | 3.22 | 3.14 |
| 12,000 | 9.35 | 10.90 | .858 | .38 | 4.53 | 4.42 |
| 16,000 | 7.96 | 10.90 | .730 | .48 | 5.72 | 5.57 |
| 20,000 | 6.75 | 10.90 | .620 | .51 | 6.09 | 5.92 |
| 23,000 | 5.90 | 10.90 | .542 | .53 | 6.32 | 6.15 |
| 30,000 | 4.36 | 9.36 | .466 | .532 | 5.45 | 5.30 |
| 35,000 | 3.46 | 8.46 | .410 | .532 | 4.93 | 4.79 |
| 40,000 | 2.72 | 7.72 | .352 | .532 | 4.50 | 4.36 |
| 45,000 | 2.14 | 7.14 | .300 | .532 | 4.17 | 4.04 |
| 50,000 | 1.68 | 6.68 | .252 | .532 | 3.89 | 3.78 |
| 55,000 | 1.32 | 6.32 | .209 | .532 | 3.68 | 3.58 |
| 60,000 | 1.04 | 6.04 | .172 | .532 | 3.52 | 3.42 |

NOTE: $P_a$ = ambient pressure

$P_c$ = cabin pressure

$t_c$ = cabin temperature

$\dot{W}_p$ = pressurization air flow



70°F CABIN TEMP.
100°F CABIN TEMP.
CABIN PRESSURIZATION AIR FLOW
ALTITUDE -- 1000 FEET
0
10
20
30
40
50
60
70
AIRFLOW, LB/MIN
1
2
3
4
5
6
7




FIGURE 3.13

## 3.3 Pressure-Suit Cooling

The pressure-suit air flow and pressurization requirements for the F-4 fighter aircraft are essentially identical to those established by the armed services for all combat aircraft (References 3 and 4 ). The design air flow is 20.0 CFM with a temperature range selectible from +50°F to +120°F. During normal use of the pressure-suit, when the cabin is pressurized, and when the cabin is unpressurized up to a cabin altitude of 35,000 feet, the pressure-suit inlet pressure must be maintained at 3.0 psi above cabin pressure. During emergency use of the pressure-suit when the cabin altitude exceeds 35,000 feet, the inlet pressure to the pressure-suit must be regulated at 6.5 psia.

The desired pressure levels at the inlet to the pressure-suit are shown in Figure 3.14 as a function of altitude for normal and emergency conditions.

The pressure-suit cooling air mass flow rate, $\dot{W}_s$ , is a function of pressure, and is related to volume flow and density as follows:

(1).......... $\dot{W}_s = \dot{Q}_s \rho$ , LB/MIN

where: $\dot{Q}_s$ = volume flow rate, CFM

$\rho$ = air density, $lb/ft^3$

but, $\rho = 1.325\ P_s/T_s$ ($P_s$ in "Hg ; $T_s$ in °R)

or

$\rho = 2.697\ P_s/T_s$ ($P_s$ in psia ; $T_s$ in °R)

therefore:

(2)............. $\dot{W}_s = 2.697 \dfrac{P_s \dot{Q}_s}{T_s}$ , LB/MIN

since $\dot{Q}_s$ = 20 CFM and $510 \leq T_s \leq 580$ °R

the minimum and maximum pressure-suit flow rates are:

(3)........... $\dot{W}_s = 0.1057\ P_s$ , LB/MIN (for minimum inlet temperature of +50°F)

and

(4)........... $\dot{W}_s = 0.0930\ P_s$ , LB/MIN (for maximum inlet temperature of +120°F)

Figure 3.15 shows the pressure-suit ventilation requirements under normal and emergency operating conditions.

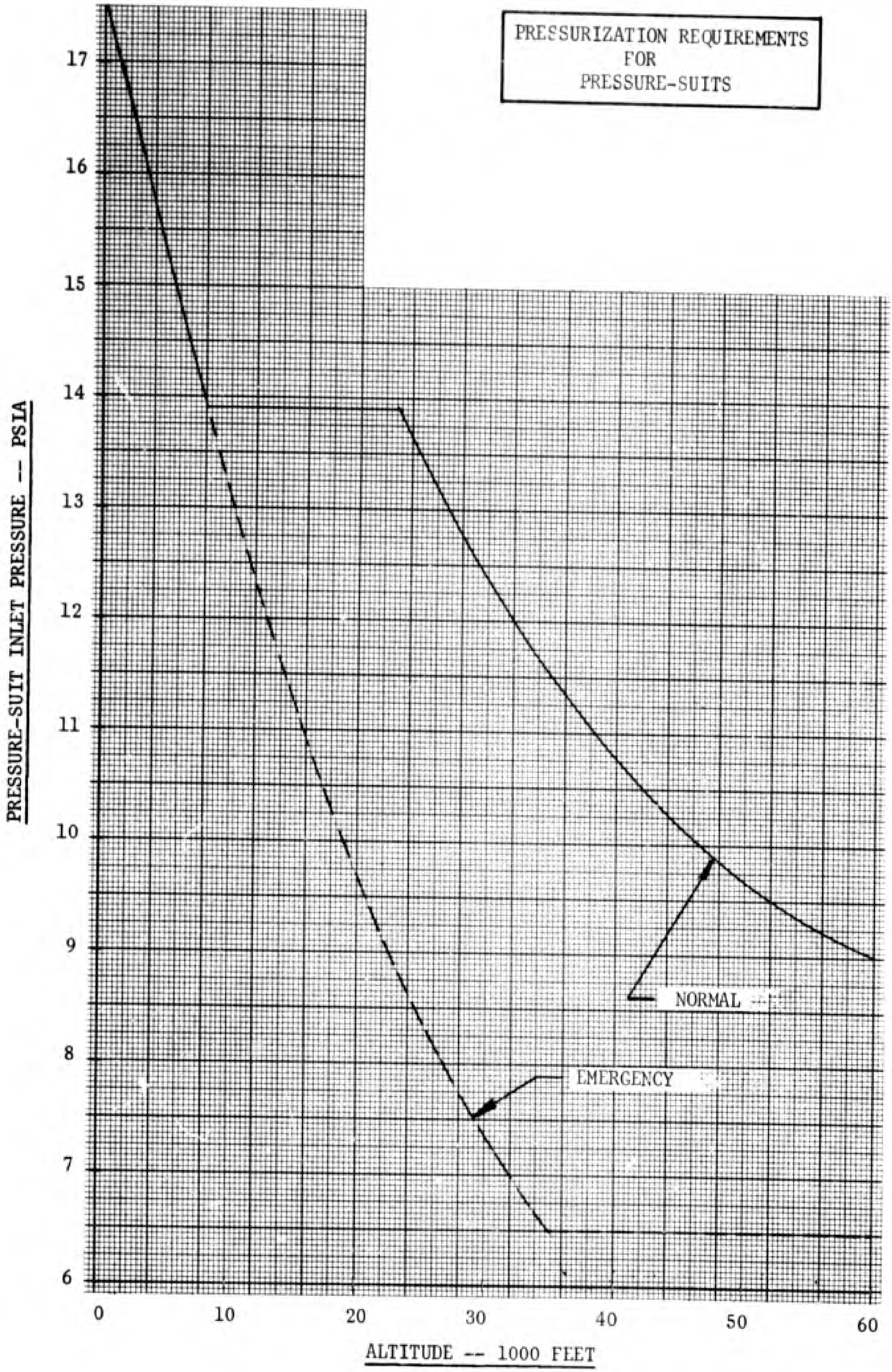

PRESSURIZATION REQUIREMENTS
FOR
PRESSURE-SUITS
PRESSURE-SUIT INLET PRESSURE -- PSIA
17
16
15
14
13
12
11
10
9
8
7
6
0
10
20
30
40
50
60
ALTITUDE -- 1000 FEET
NORMAL
EMERGENCY



FIGURE 3.14

PRESSURE-SUIT VENTILATION REQUIREMENTS

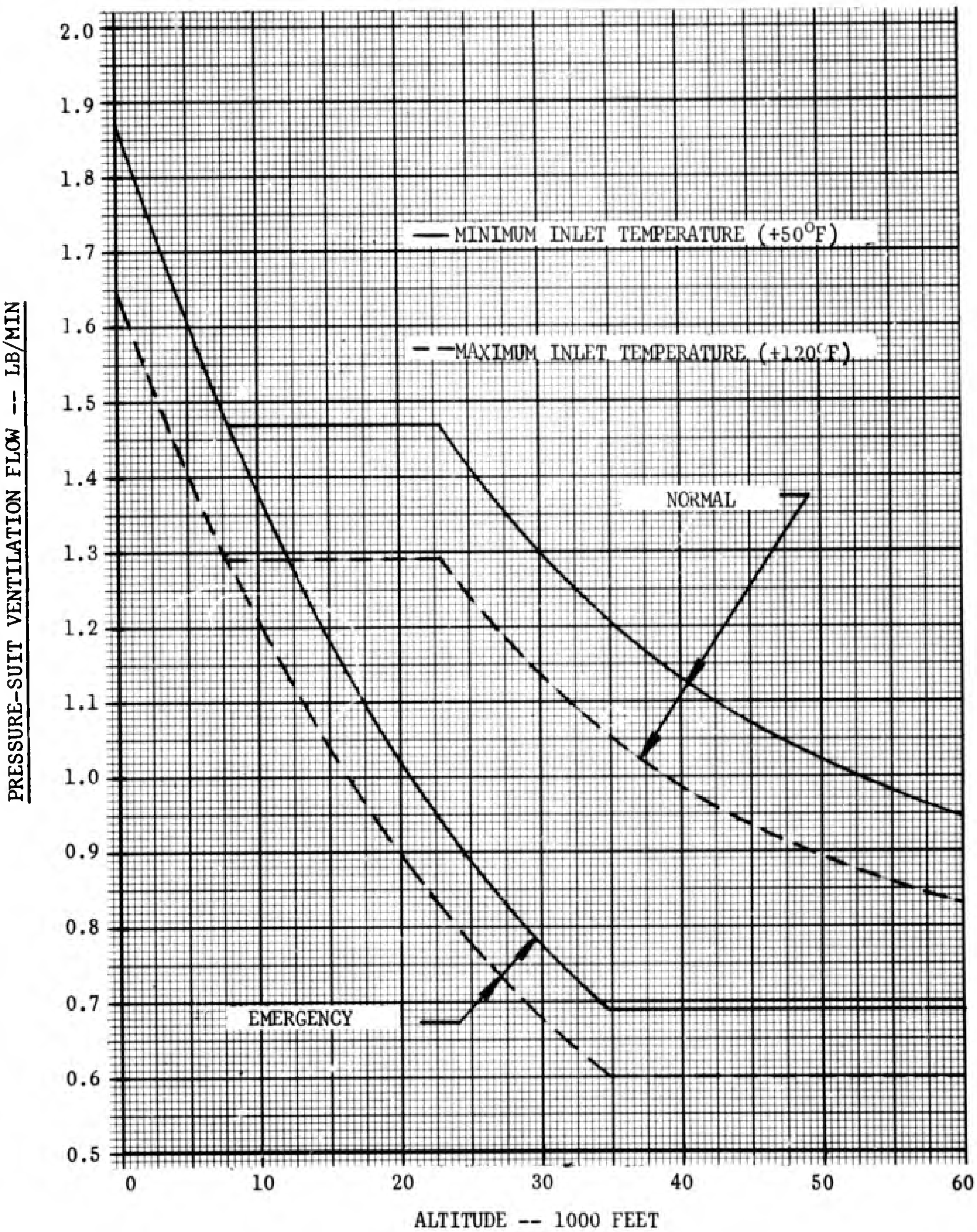





FIGURE 3.15

### 3.4 Windshield Rain Removal

During landing, take-off, and taxiing, the cabin package must be capable of delivering rain removal air across the windshield through a nozzle with a CA of approximately 1.2 sq. inches. The desired air temperature is 400 to 450°F; and normal air flow requirements are 51 to 90 lb/min, depending upon aircraft engine power setting, aircraft speed, and other operational characteristics (Reference 4).

Because of the temperature and flow requirements, compressor bleed air is the only available source that can meet the rain removal requirements. Since the total compressor bleed air flow is limited, the availability of compressor bleed air for other environmental control functions will be drastically reduced or non-existing whenever rain removal is required. Bleed air for boundary layer control, as required, is provided.

## 4.0 SYSTEM DESIGN PARAMETERS

This section discusses the criteria used in the selection of a basic mechanization of the Conductron system concept and the determination of the system design conditions by considering the effect of aircraft operating conditions on the design requirements of the Conductron environmental control system. In addition, the installation constraints of the F-4 aircraft, based on the air cycle system presently installed, are briefly discussed in terms of size and weight limitations. The selection of the refrigerant to be used with the Conductron system greatly affects system performance as well as other operational characteristics. The basis for refrigerant selection is discussed in detail.

### 4.1 Selected System Mechanization

The specific mechanization of the Conductron environmental control concept, involving the varying use of available sources for system actuation, heat rejection, and cabin cooling, is greatly dependent upon aircraft constraints. In the case of the F-4 fighter aircraft, the controlling factors are: (1) installation constraints, allowing no modifications to the existing airframe and no increase in weight or size over the presently installed air cycle system; and (2) cabin package performance requirements, which are detailed in Section 3. The following specific considerations apply in the selection of a mechanization of the Conductron system:

System actuation - Engine compressor bleed air is being used by the existing air cycle system, thus making it a readily available source which does not require modifications to the aircraft or its subsystems. The use of any of the other possible heat sources would result not only in undesirable modifications, but also in an increase in system weight.

Heat rejection - Ambient or ram air is the <u>only</u> available heat sink, because the fuel heat sink is already being completely utilized by the hydraulic system of the aircraft. Since ram air is used as the heat sink for the air cycle system, only minor modifications to the existing ducting would be required.

Cabin cooling - Open or semi-closed loop operation is suitable. The open system exhausts all of the cabin cooling air, which is supplied by compressor bleed air, whereas the semi-closed system recirculates the cabin air, with compressor bleed air being used for cabin pressurization.

Cabin cooling - (cont'd)

The simplicity of the open system is offset by the fact that under most operating conditions the semi-closed loop requires only a portion of the available compressor bleed air for system actuation and all cabin package functions, except cabin cooling. Since bleed air extraction is accompanied by a loss in thrust from the propulsion unit, any reduction in bleed air requirements will improve aircraft performance characteristics and reduce the aircraft performance penalty charged to the environmental control system.

Taking all these advantages and disadvantages into consideration, the basic mechanization of the Conductron system for the F-4 fighter aircraft should be one which utilizes compressor bleed air for system actuation, cabin pressurization, pressure-suit ventilation, and windshield rain removal, and which accomplishes system heat rejection by ambient or ram air. A semi-closed system is preferred for the cabin cooling function, whenever cabin leakage is sufficiently high to eliminate the need for air purification components.

## 4.2 Aircraft Operating Conditions

The large variations in operating conditions of high-performance aircraft result not only in associated changes in cooling requirements but also in temperature, pressure, and flow rate changes of the heat source (compressor bleed air) and heat sink (ambient or ram air). Tables 4-1, 4-2, and 4-3 show the aircraft design operating conditions, including parameters affecting the environmental control system. Shown are the compressor bleed air flow rates, temperatures, and pressures. The available potential for system actuation is determined from the following relationship:

$$q = \dot{W}_B c_p (t_{B1} - t_{B2})$$

where:

$q$ = available thermal energy (BTU/hr)

$\dot{W}_B$ = bleed air flow (lb/hr)

$c_p$ = specific heat of bleed air (BTU/lb-$^{o}$F)

$t_{B1}$ = bleed air temperature entering the gas piston unit ($^{o}$F)

$t_{B2}$ = bleed air temperature leaving the gas piston unit ($^{o}$F)

The bleed air temperature leaving the gas piston unit was arbitrarily taken as 300$^{o}$F.

TABLE 4-1

HEAT SOURCE AND HEAT SINK POTENTIALS
(ICAO STANDARD DAY)

| AIRCRAFT OPERATING CONDITION | | | | | COMPRESSOR BLEED AIR | | | | RAM AIR | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| NO. | ALTITUDE (ft) | MACH NO. - | AMBIENT TEMP. (°F) | AMBIENT PRESS. (psia) | FLOW (lb/min) | TOTAL TEMP. (°F) | TOTAL PRESS. (psia) | THERMAL ENERGY (Btu/hr) | TOTAL TEMP. (°F) | TOTAL PRESS. (psia) | CABIN COOLING LOAD (Btu/hr) | REMARKS |
| 1 | S.L. | 0 | 59 | 14.69 | 44.0 | 305 | 33.4 | 3,200 | 59 | 14.69 | 5,500 | Min. Thermal Energy |
| 2 | S.L. | 0 | 59 | 14.69 | 74.4 | 473 | 68.7 | 190,350 | 59 | 14.69 | 5,500 | |
| 3 | S.L. | 0 | 59 | 14.69 | 116.2 | 750 | 154.8 | 798,450 | 59 | 14.69 | 5,500 | |
| 4 | S.L. | .345 | 59 | 14.69 | 51.5 | 486 | 83.5 | 141,675 | 70 | 15.96 | 7,000 | |
| 5 | S.L. | .47 | 59 | 14.69 | 64.2 | 527 | 98.4 | 216,850 | 81 | 17.10 | 9,000 | |
| 6 | S.L. | 1.20 | 59 | 14.69 | 65.2 | 860 | 295.0 | 564,100 | 207 | 35.65 | 25,500 | |
| 7 | 15,000 | 1.60 | 5 | 8.29 | 59.8 | 908 | 270.0 | 565,000 | 243 | 35.20 | 30,500 | |
| 8 | 25,000 | .58 | -30 | 5.45 | 49.1 | 471 | 52.5 | 124,175 | -1 | 6.90 | -4,500 | |
| 9 | 25,000 | .748 | -30 | 5.45 | 56.6 | 493 | 60.4 | 161,900 | 18 | 7.95 | -2,000 | |
| 10 | 25,000 | 1.00 | -30 | 5.45 | 73.0 | 748 | 122.8 | 499,400 | 56 | 10.39 | 4,000 | |
| 11 | 25,000 | 1.80 | -30 | 5.45 | 58.6 | 911 | 231.0 | 556,400 | 249 | 31.50 | 30,500 | |
| 12 | 36,000 | 2.10 | -70 | 3.28 | 48.5 | 926 | 192.0 | 473,650 | 274 | 30.10 | 32,500 | |
| 13 | 38,000 | .765 | -70 | 2.99 | 38.1 | 485 | 43.1 | 104,250 | -24 | 4.41 | -5,500 | |
| 14 | 40,000 | 2.25 | -70 | 2.72 | 44.3 | 952 | 162.0 | 451,450 | 325 | 31.50 | 38,500 | Max. Cooling Load & Ram Air Temperature |
| 15 | 43,000 | .88 | -70 | 2.35 | 33.1 | 540 | 39.8 | 118,450 | -10 | 3.89 | -4,000 | |
| 16 | 50,000 | 1.00 | -70 | 1.68 | 26.1 | 680 | 38.0 | 149,950 | 8 | 3.20 | -5,000 | |
| 17 | 50,000 | 1.20 | -70 | 1.68 | 30.6 | 743 | 48.1 | 207,000 | 42 | 4.10 | -2,000 | |
| 18 | 55,000 | 1.50 | -70 | 1.32 | 30.0 | 800 | 50.9 | 230,400 | 105 | 4.85 | 6,500 | |
| 19 | 55,000 | 2.00 | -70 | 1.32 | 31.9 | 910 | 74.6 | 302,400 | 242 | 10.32 | 25,000 | |
| 20 | 57,500 | 1.80 | -70 | 1.17 | 30.0 | 877 | 59.0 | 267,950 | 183 | 6.73 | 16,500 | |

TABLE 4-2

HEAT SOURCE AND HEAT SINK POTENTIALS
(ANA HOT DAY)

| AIRCRAFT OPERATING CONDITION | | | | | COMPRESSOR BLEED AIR | | | | RAM AIR | | CABIN COOLING LOAD | REMARKS |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| NO. | ALTITUDE | MACH NO. | AMBIENT | | FLOW | TOTAL | | THERMAL ENERGY | TOTAL | | | |
| | (ft) | - | TEMP. ($^oF$) | PRESS. (psia) | (lb/min) | TEMP. ($^oF$) | PRESS. (psia) | (Btu/hr) | TEMP. ($^oF$) | PRESS. (psia) | (Btu/hr) | |
| 1 | S.L. | 0 | 103 | 14.69 | 30.6 | 348 | 30.4 | 21,500 | 103 | 14.69 | 12,000 | Min. Thermal Energy |
| 2 | S.L. | 0 | 103 | 14.69 | 55.0 | 505 | 58.4 | 167,100 | 103 | 14.69 | 12,000 | |
| 3 | S.L. | 0 | 103 | 14.69 | 112.8 | 758 | 143.2 | 788,900 | 103 | 14.69 | 12,000 | |
| 4 | S.L. | .345 | 103 | 14.69 | 45.1 | 553 | 80.2 | 170,125 | 115 | 15.96 | 16,000 | |
| 5 | S.L. | .47 | 103 | 14.69 | 53.7 | 600 | 97.2 | 241,650 | 127 | 17.10 | 17,500 | |
| 6 | S.L. | 1.10 | 103 | 14.69 | 58.0 | 907 | 253.0 | 547,710 | 239 | 31.40 | 32,000 | |
| 7 | 15,000 | 1.30 | 45 | 8.29 | 64.2 | 891 | 199.0 | 589,621 | 214 | 22.95 | 25,000 | |
| 8 | 25,000 | .58 | 7 | 5.45 | 44.0 | 528 | 53.0 | 149,275 | 35 | 6.89 | -2,000 | |
| 9 | 25,000 | .748 | 7 | 5.45 | 51.1 | 567 | 61.3 | 203,825 | 56 | 7.94 | 1,500 | |
| 10 | 25,000 | 1.00 | 7 | 5.45 | 64.3 | 781 | 115.0 | 473,200 | 97 | 10.39 | 7,500 | |
| 11 | 25,000 | 1.60 | 7 | 5.45 | 59.4 | 907 | 179.5 | 560,300 | 242 | 23.30 | 24,500 | |
| 12 | 38,000 | .765 | -40 | 2.99 | 33.8 | 545 | 42.5 | 123,475 | 5 | 4.40 | -2,000 | |
| 13 | 38,000 | 2.00 | -40 | 2.99 | 47.4 | 937 | 143.2 | 471,000 | 289 | 23.40 | 37,000 | Max. Cooling Load & Max. Ram Air Temp. |
| 14 | 43,000 | .88 | -43.5 | 2.35 | 30.1 | 604 | 40.7 | 137,250 | 23 | 3.88 | 1,500 | |
| 15 | 45,000 | 1.90 | -42.5 | 2.14 | 40.5 | 921 | 98.1 | 392,350 | 262 | 14.30 | 29,500 | |
| 16 | 50,000 | 1.00 | -40 | 1.68 | 23.5 | 749 | 37.0 | 161,125 | 46 | 3.20 | 0 | |
| 17 | 50,000 | 1.40 | -40 | 1.68 | 30.1 | 817 | 54.9 | 239,500 | 127 | 5.37 | 10,500 | |
| 18 | 50,000 | 1.80 | -40 | 1.68 | 32.4 | 907 | 73.8 | 305,600 | 235 | 9.70 | 25,000 | |
| 19 | 51,000 | 1.60 | -40 | 1.60 | 31.4 | 877 | 64.3 | 280,450 | 178 | 6.83 | 18,000 | |

## TABLE 4-3

HEAT SOURCE AND HEAT SINK POTENTIALS
(ANA COLD DAY)

| NO. | AIRCRAFT OPERATING CONDITION: ALTITUDE (ft) | MACH NO. - | AMBIENT TEMP. (°F) | AMBIENT PRESS. (psia) | COMPRESSOR BLEED AIR: FLOW (lb/min) | TOTAL TEMP. (°F) | TOTAL PRESS. (psia) | THERMAL ENERGY (Btu/hr) | RAM AIR TOTAL TEMP. (°F) | RAM AIR TOTAL PRESS. (psia) | CABIN COOLING LOAD (Btu/hr) | REMARKS |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | S.L. | 0 | -65 | 14.69 | 87.0 | 205 | 45.1 | - | -65 | 14.69 | -11,000 | Min. Thermal Energy |
| 2 | S.L. | 0 | -65 | 14.69 | 165.2 | 373 | 114.6 | 176,900 | -65 | 14.69 | -11,000 | |
| 3 | S.L. | 0 | -65 | 14.69 | 170.6 | 540 | 168.0 | 610,475 | -65 | 14.69 | -11,000 | |
| 4 | S.L. | .345 | -65 | 14.69 | 92.4 | 317 | 95.6 | 22,900 | -50 | 15.96 | -18,000 | |
| 5 | S.L. | .47 | -65 | 14.69 | 118.1 | 351 | 114.4 | 88,350 | -42 | 17.10 | -16,500 | |
| 6 | S.L. | 1.20 | -65 | 14.69 | 165.9 | 576 | 295.0 | 684,100 | 55 | 35.63 | -3,000 | |
| 7 | 15,000 | 1.60 | -29 | 8.29 | 65.9 | 880 | 295.0 | 591,700 | 193 | 35.20 | 21,000 | |
| 8 | 25,000 | .58 | -64 | 5.45 | 54.2 | 410 | 56.0 | 87,600 | -37 | 6.89 | -12,000 | |
| 9 | 25,000 | .748 | -64 | 5.45 | 62.0 | 440 | 64.3 | 128,100 | -20 | 7.04 | -9,000 | |
| 10 | 25,000 | 1.00 | -64 | 5.45 | 82.6 | 686 | 125.8 | 482,100 | 14 | 10.39 | -3,000 | |
| 11 | 25,000 | 1.80 | -64 | 5.45 | 69.0 | 878 | 278.0 | 617,375 | 193 | 31.50 | 20,000 | |
| 12 | 38,000 | .765 | -85 | 2.99 | 40.2 | 464 | 45.6 | 97,300 | -41 | 4.31 | -9,000 | |
| 13 | 43,000 | .88 | -88 | 2.35 | 36.4 | 504 | 41.3 | 110,050 | -27 | 3.89 | -7,000 | |
| 14 | 45,000 | 2.40 | -99 | 2.14 | 53.0 | 968 | 142.0 | 554,425 | 347 | 31.12 | 36,000 | Max. Cooling Load & Ram Air Temp. |
| 15 | 50,000 | 1.00 | -123 | 1.68 | 29.5 | 606 | 39.0 | 135,400 | -29 | 3.20 | -5,000 | |
| 16 | 60,000 | 1.50 | -125 | 1.04 | 30.8 | 711 | 44.5 | 192,150 | 26 | 3.85 | -4,000 | |
| 17 | 65,000 | 1.90 | -113 | 0.82 | 31.0 | 812 | 54.5 | 243,800 | 117 | 5.48 | 7,000 | |
| 18 | 65,000 | 2.40 | -113 | 0.82 | 30.8 | 919 | 70.0 | 296,275 | 261 | 11.96 | 26,000 | |
| 19 | 67,500 | 2.20 | -106 | 0.72 | 30.0 | 889 | 61.3 | 273,550 | 199 | 7.75 | 17,000 | |

The heat rejection potential is indicated by the ram air total temperature and pressure with increases in heat rejection potential being signified by decreasing temperatures or increasing pressures. The cabin cooling loads given are for maximum cabin temperatures of 70°F at all subsonic flight conditions and 100°F maximum at all supersonic conditions. The nominal cabin temperature under normal flight conditions is 80°F. The transient cabin temperature requirements of Reference 3 allowing a transient temperature variation between 60°F and 105°F are not to be exceeded. In addition, pressure-suit ventilation flow at inlet temperatures selectible between +50°F and +120°F must be provided at any flight condition. The heating loads are for a cabin temperature of 70°F at all flight conditions. The loads include heat transfer through external and internal compartment surfaces, solar radiation loads, and personnel loads. The data shown in Tables 4-1 through 4-3 is based on information contained in References 1 and 2.

The limiting aircraft operating conditions for the Conductron environmental control system are those which result in: (1) minimum available thermal energy for system actuation, which is reflected by low bleed air temperatures and/or low bleed air flows; (2) maximum ram air temperatures, thus providing a poor heat sink for rejection of unwanted heat; and (3) maximum cabin cooling loads, which dictate system capacity and size.

Two distinct limiting aircraft operating conditions can be identified: (a) sea level, static, engine idle (Condition 1); and (b) maximum speed at an altitude between 38,000 and 45,000 feet, depending on temperature-day (Condition 13 on a Hot Day and Condition 14 on a Standard Day and a Cold Day). The first condition is characterized by the lowest compressor bleed air temperature, a low compressor bleed air flow, and no ram air for heat rejection. Therefore, ambient air must be utilized as a heat sink, requiring air circulation components. However, the bleed air temperatures are so low as to make system actuation almost impossible. Fortunately, the cabin cooling loads are relatively small, thus some very inefficient system operation may prove to be sufficient in performing the required environmental control function. Nevertheless, this condition represents very definitely a limiting condition for the Conductron system, and as such determines in part the refrigerant vapor temperature and pressure characteristics, and therefore the design requirements for the gas piston unit. The second limiting condition is identified by the highest ram air temperature and the maximum cabin cooling load. The extremely high ram air temperature renders the ram air ineffective as a heat sink, thus necessitating the use of auxiliary heat exchangers, air motors, or expansion turbines in order to reduce the ram air temperature to an acceptable level for condenser cooling. Where the ram air temperature is the controlling parameter for the condenser, the evaporator design requirements are dictated by the cabin cooling load. Since the condenser and evaporator represent the two heaviest and largest components of the Conductron system, the system weight and size will be directly dependent upon this limiting aircraft operating condition. Figure 4.1 shows the available outside air heat sink as a function of Mach number. The air heat sink available for condenser cooling is at the stagnation condition (total temperature).

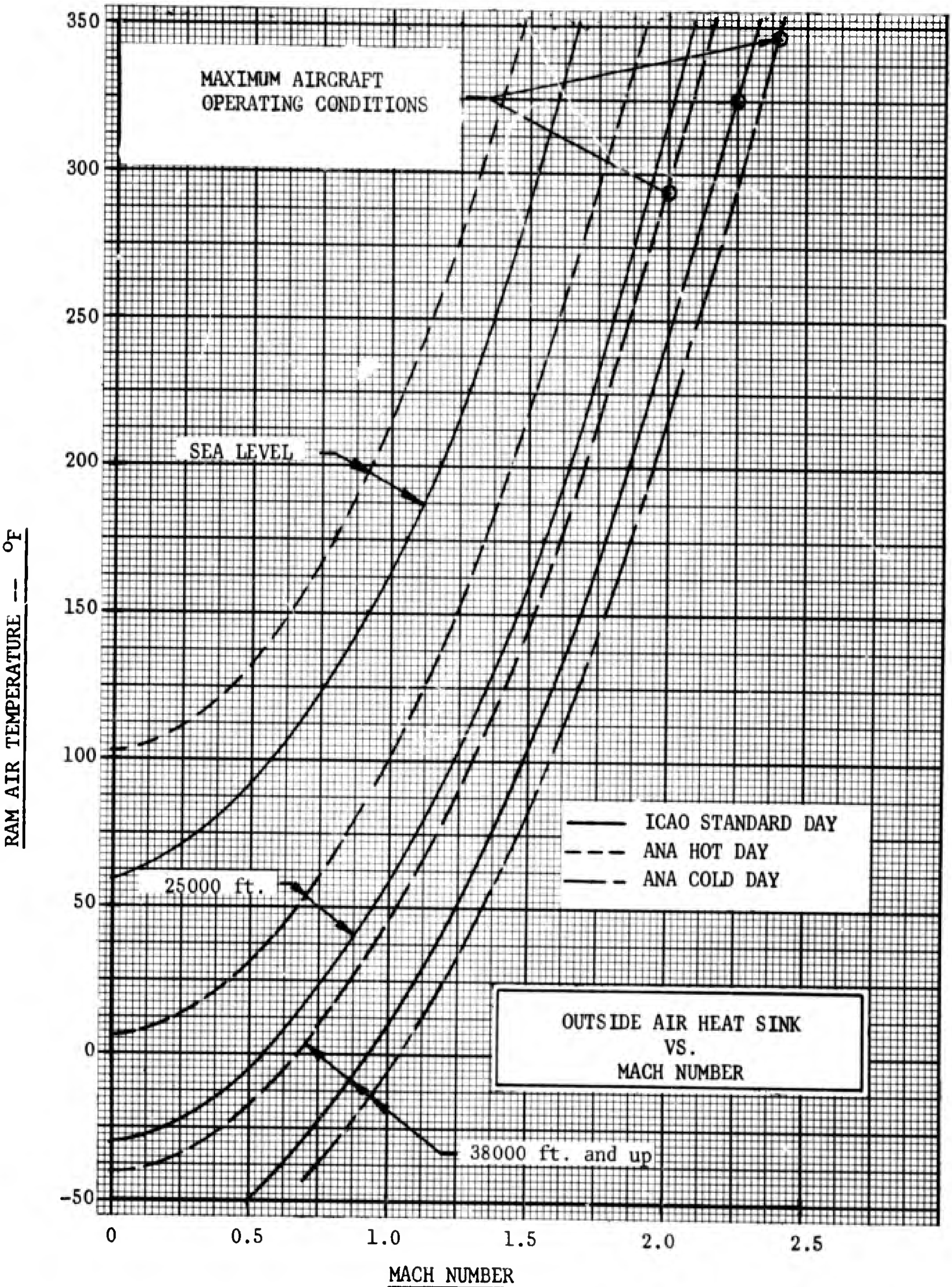

MAXIMUM AIRCRAFT
OPERATING CONDITIONS
SEA LEVEL
25000 ft.
38000 ft. and up
ICAO STANDARD DAY
ANA HOT DAY
ANA COLD DAY
OUTSIDE AIR HEAT SINK
VS.
MACH NUMBER
RAM AIR TEMPERATURE -- °F
MACH NUMBER
350
300
250
200
150
100
50
0
-50
0
0.5
1.0
1.5
2.0
2.5

FIGURE 4.1

## 4.3 Selected Design Conditions

The environmental control system performance requirements and aircraft operating conditions are the primary factors in the selection of system design conditions. It has been shown that aircraft operating conditions which result in maximum cabin cooling loads, maximum ram air temperatures, and minimum thermal energy levels for system actuation, represent the limiting conditions for the operation of the Conductron environmental control system. Since these limiting cases do not occur at the same aircraft operating condition, the component performance characteristics will not necessarily be optimum at any selected design condition.

The system design conditions must be such that the performance requirements at all other conditions are met. This suggests that the two extreme operating conditions, namely maximum speed and ground static, constitute the required system performance range. Since the maximum speed condition includes two limiting parameters, maximum cooling load, and maximum ram air temperature, a system design based on this condition should result in adequate performance characteristics for any other operating condition.

The following design condition was selected for the Conductron environmental control system:

| | |
|---|---|
| Temperature-Day: | ICAO Standard |
| Condition: | 14 |
| Flight Mach number: | 2.25 |
| Altitude: | 40,000 feet |
| Ram air temperature: | 325$^{o}$F |
| Available bleed air flow: | 44.3 lb/min |
| Bleed air temperature: | 952$^{o}$F |
| Cabin cooling load: | 38,500 BTU/hr |

The desired minimum off-design condition for the system is as follows:

| | |
|---|---|
| Temperature-Day: | ANA Hot |
| Condition: | 1 |
| Flight Mach number: | 0 |
| Altitude: | sea level |
| Ambient temperature: | 103$^{o}$F |
| Available bleed air flow: | 30.6 lb/min |
| Bleed air temperature: | 348$^{o}$F |
| Cabin cooling load: | 12,000 BTU/hr |

These system design conditions, together with certain physiological requirements for crew comfort and some overall system constraints, form the basis for component and system analyses. Some of these additional considerations are briefly stated below.

Control of cabin cooling air inlet temperatures is required for crew comfort. Cold temperatures require extensive air diffusion and mixing in order to avoid undesirable chilling of crew members. However, higher air temperatures require higher cooling flow rates, so that the availability of cooling air must be considered in the selection of cabin cooling air flow rate and inlet temperature. To avoid icing of evaporator coils, a minimum evaporating temperature of $30^{o}F$ must be maintained.

Finally, to complete the selection of system design conditions, the system weight and size constraints must be specified. Based on the existing air cycle environmental control system of the F-4 fighter aircraft (Reference 8 ) the following weight and size restrictions will be applicable to the Conductron ECS:

| | |
|---|---|
| Weight: | 126 lbs. maximum |
| Size: | 15 in. x 15 in. x 48 in. |

## 4.4 Refrigerant Selection

The selection of a working fluid (refrigerant) entails the consideration of the following characteristics of candidate refrigerants:

1) Thermodynamic properties related to energy conversion and ability to remove heat (a high latent heat of vaporization).

2) Resulting pressure and temperature levels in various system components for a given design condition.

3) Chemical stability (decomposition and degradation of the fluid as a function of temperature and time).

4) Material compatibility (effects of material-refrigerant interaction on degradation of the fluid and material as a function of temperature and time).

5) Safety (flammability and toxicity).

6) Availability and cost.

The following refrigerants are compared on the above basis: R-11; R-12; R-21; R-113; R-114; and R-C318. The performance characteristics were calculated on a 12,000 BTU/hr (1-ton) basis. The evaporating temperature and condensing temperature are taken as 40$^{o}$F and 140$^{o}$F, respectively.

Based on a "weighted" consideration of the above listed characteristics, R-11 is clearly a best choice for the Conductron heat-actuated refrigeration cycle. The strong point of R-11 is primarily its thermodynamic performance, resulting in a minimum weight system. Use of R-11 does entail operation at sub-ambient pressures (at low altitudes), which has proven no problem in sealed systems. In mechanically connected aircraft systems, special concern for air seepage in the evaporator loop should be taken; however, at the worst it means that an occasional but simple maintenance bleed-off of condenser non-condensible gases might be necessary. The performance advantages of R-11 are sufficient to warrant this possibility and, thus, its selection as the refrigerant.

Table 4-4 shows a comparison between the candidate refrigerants. The "performance rating" is based on the following desirable characteristics:

1) High coefficient of performance (COP), which is the ratio of cooling load to heat input.

2) High compression ratio (condensing pressure/evaporating pressure).

3) A positive evaporating pressure and near atmospheric pressure, minimizing leakage problems.

4) A low condensing pressure, which minimizes leakage and avoids excessive pressure levels for safety reasons.

5) Low evaporator flow rate (resulting in smaller evaporator size).

6) Low condenser flow rate (resulting in smaller condenser size).

7) Low boiler flow rate (resulting in smaller gas piston unit size).

8) Low motive-to-suction flow ratio (indicative of small system size and weight).

The "usability rating" consists of the following desirable characteristics:

1) Good chemical stability and material compatibility at operating temperatures.

2) Good safety features, including low or no toxicity, non-flammability, and no requirements for special handling.

3) Low cost and good availability.

The refrigerants are rated relative to each other, with a "1" designating the best and a "6" designating the worst refrigerant in each category.

Appendix A contains the detailed performance calculations for each refrigerant under consideration.

TABLE 4-4

REFRIGERANT RATING

| | R-11 | R-12 | R-21 | R-113 | R-114 | R-C318 |
|---|---|---|---|---|---|---|
| Coefficient of Performance (COP) | 1 | 5 | 3 | 2 | 4 | 6 |
| Pressure ratio | 5 | 1 | 4 | 6 | 3 | 2 |
| Evaporator pressure | 5 | 4 | 3 | 6 | 1 | 2 |
| Condenser pressure | 2 | 6 | 3 | 1 | 4 | 5 |
| Evaporator flow rate | 2 | 4 | 1 | 3 | 5 | 6 |
| Condenser flow rate | 1 | 5 | 2 | 3 | 4 | 6 |
| Boiler flow rate | 1 | 5 | 3 | 2 | 4 | 6 |
| Motive flow/Suction flow | 2 | 6 | 3 | 1 | 4 | 5 |
| Performance Points | 19 | 36 | 22 | 24 | 29 | 38 |
| Performance Rank | 1 | 5 | 2 | 3 | 4 | 6 |
| Stability | 6 | 3 | 5 | 4 | 2 | 1 |
| Safety | 4 | 3 | 6 | 5 | 2 | 1 |
| Cost | 1 | 2 | 4 | 3 | 5 | 6 |
| Usability Points | 11 | 8 | 15 | 12 | 9 | 8 |
| Usability Rank | 4 | 2 | 6 | 5 | 3 | 1 |
| Overall Points | 30 | 44 | 37 | 36 | 38 | 46 |
| Overall Rank | 1 | 5 | 3 | 2 | 4 | 6 |

## 5.0 COMPONENT ANALYSIS

This section presents a state-of-the-art assessment of the major system components in terms of performance, weight, and size characteristics and power requirements. The following components are analyzed: Gas Piston Unit; Condenser; Evaporator; Converter (Ejector); Air Motor; and Fan.

In order to evaluate the performance of a given system, it is necessary to: (1) identify the parameters which control the operation of each component, and (2) establish mathematical relationships between these variables over selected operating ranges. The parameters of interest are temperatures and flow rates of the primary fluid (refrigerant) and the secondary fluid (air).

Figure 5.1 shows the system variables of interest to the Conductron system and indicates the nomenclature used in this section. Table 5-1 lists the variables which affect the operation and performance of each component.

### 5.1 Heat Exchanger Characterization

The operation of any environmental control system depends on the use of one or more heat exchangers. Heat exchangers are devices which exchange the thermal energy between two fluids. In general, heat exchangers can be grouped into the following types: air-to-air, air-to-liquid, and liquid-to-liquid. The Conductron system utilizes three heat exchangers of the air-to-liquid type: boiler, condenser, and evaporator. The basic relationships presented allow the determination of heat exchanger core dimensions and weights and are, therefore, extremely useful in the evaluation of heat exchanger and system performance. (An example of a detailed heat exchanger design is presented in Appendix "C").

NOMENCLATURE:

| | |
|---|---|
| $A$ | Exchanger total heat transfer area (one side) - $ft^2$ |
| $A_c$ | Exchanger free flow area - $ft^2$ |
| $A_f$ | Exchanger total fin area - $ft^2$ |
| $A_{fr}$ | Exchanger total frontal area - $ft^2$ |
| $C$ | Flow capacity rate ($\dot{w}c_p$) - BTU/hr-$^oF$ |
| $c_p$ | Specific heat at constant pressure - BTU/lb-$^oF$ |
| $f$ | Friction factor |

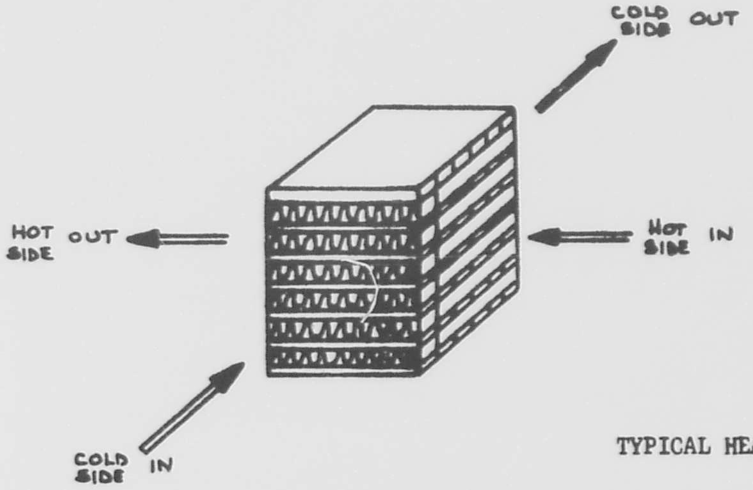
| | |
|---|---|
| $G$ | Mass velocity ($\dot{W}/A_c$) - lb/hr-ft$^2$ |
| $g$ | Acceleration of gravity - ft/sec$^2$ |
| $h$ | Convection heat transfer coefficient - BTU/hr-ft$^2$-$^o$F |
| $k$ | Thermal conductivity - BTU/hr-ft-$^o$F |
| $L$ | Exchanger flow length - ft |
| LMTD | Log-mean-temperature-difference - $^o$F |
| NTU | Number of heat transfer units |
| Pr | Prandtl number ($\mu c_p/k$) |
| $q$ | Heat transfer rate - BTU/hr |
| Re | Reynolds number ($4r_h G/\mu$) |
| $r_h$ | Hydraulic radius ($A_c L/A$) - ft |
| St | Stanton number ($h/Gc_p$) |
| $t$ | Temperature - $^o$F |
| $U$ | Overall heat transfer coefficient - BTU/hr-ft$^2$-$^o$F |
| $V$ | Velocity - ft/sec |
| $\dot{W}$ | Mass flow rate - lb/hr |

Greek symbols:

| | |
|---|---|
| $\epsilon$ | Exchanger effectiveness |
| $\eta_f$ | Fin efficiency |
| $\eta_o$ | Total surface temperature effectiveness $\equiv 1 - \frac{A_f}{A}(1-\eta_f)$ |
| $\mu$ | Viscosity - lb/hr-ft |
| $\rho$ | Density - lb/ft$^3$ |

Subscripts:

| | |
|---|---|
| c | cold side |
| h | hot side |
| i | in |
| o | out |
| min | minimum |
| max | maximum |
| ave | average |




TYPICAL HEAT EXCHANGER CORE

For any heat exchanger core, regardless of working fluids, the following energy and rate equations describe the thermal energy transfer process:

Heat lost by hot fluid:

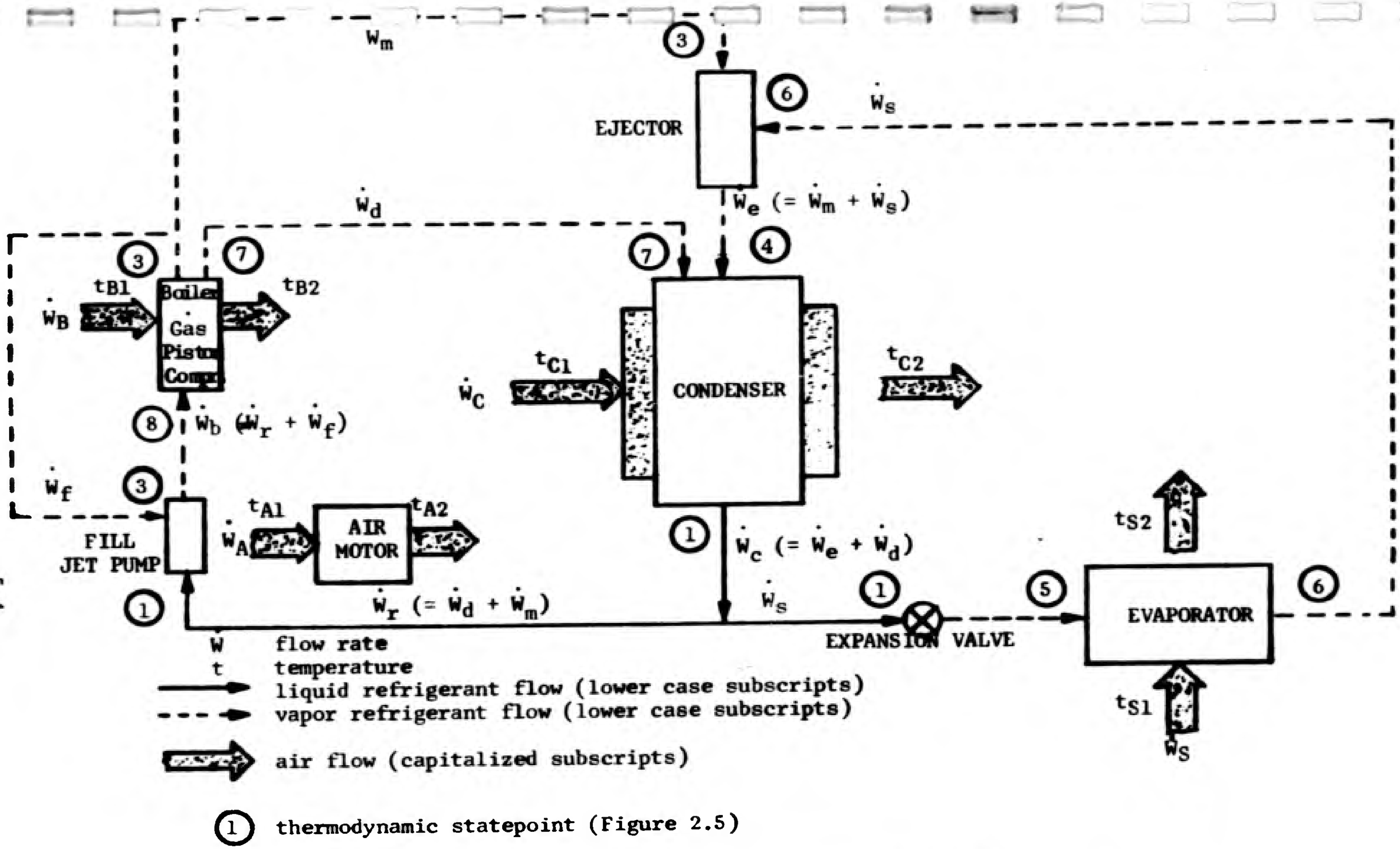
(1)............ $q_h = (\dot{w}c_p)_h\,(t_{hi} - t_{ho}) = C_h\,(t_{hi} - t_{ho})$

Heat gained by cold fluid:

(2)............ $q_c = (\dot{w}c_p)_c\,(t_{co} - t_{ci}) = C_c\,(t_{co} - t_{ci})$

Overall heat transfer rate:

(3)............ $dq/dA = U\,(t_h - t_c)$




| Refrigerant Side | | Air Side | |
|---|---|---|---|
| b | boiler | A | air motor |
| c | condenser | B | boiler |
| d | depressurization | C | condenser |
| e | ejector | S | evaporator |
| f | fill-jet pump | | |
| m | motive | | |
| r | condenser return | | |
| s | suction (evaporator) | | |

SYSTEM COMPONENT NOMENCLATURE

FIGURE 5.1

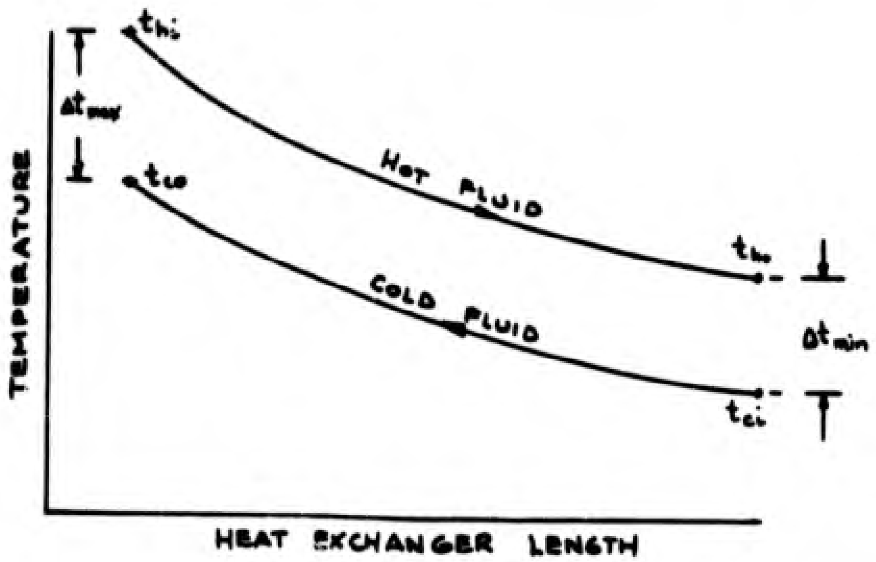
TABLE 5-1

COMPONENT PARAMETERS

| COMPONENT | PARAMETERS AFFECTING COMPONENT OPERATION | |
|---|---|---|
| | REFRIGERANT SIDE | AIR SIDE |
| Gas Piston Unit | $\dot{W}_b$ (vapor generation rate)<br>$t_b$ (boiling temperature)<br>$t_c$ (condensing temperature) | $\dot{W}_B$ (flow rate)<br>$t_{B1}$ (inlet temperature)<br>$t_{B2}$ (outlet temperature)<br>$\Delta P_B$ (air pressure drop) |
| Condenser | $\dot{W}_m$ (motive vapor flow rate)<br>$\dot{W}_d$ (depressurization flow)<br>$\dot{W}_s$ (suction flow rate)<br>$t_m$ (motive temperature)<br>$t_c$ (condensing temperature) | $\dot{W}_C$ (flow rate)<br>$t_{C1}$ (inlet temperature)<br>$t_{C2}$ (outlet temperature)<br>$\Delta P_C$ (air pressure drop) |
| Evaporator | $\dot{W}_s$ (suction flow rate)<br>$t_s$ (evaporating temperature)<br>$t_c$ (condensing temperature) | $\dot{W}_S$ (flow rate)<br>$t_{S1}$ (inlet temperature)<br>$t_{S2}$ (outlet temperature)<br>$\Delta P_S$ (air pressure drop) |
| Converter/ Ejector | $\dot{W}_m$ (motive flow rate)<br>$\dot{W}_s$ (suction flow rate)<br>$t_m$ (motive temperature)<br>$t_s$ (suction temperature)<br>$t_c$ (condensing temperature)<br>$P_m$ (motive pressure)<br>$P_s$ (suction pressure)<br>$P_c$ (condensing pressure) | |
| Air Motor | | $\dot{W}_A$ (flow rate)<br>$t_{A1}$ (inlet temperature)<br>$t_{A2}$ (outlet temperature)<br>$P_{A1}$ (inlet pressure)<br>$P_{A2}$ (outlet pressure)<br>$\mathbf{P}$ (power output) |
| Fan | | $\dot{W}$ (flow rate)<br>$\rho$ (air density)<br>$\Delta P$ (pressure head)<br>$\mathbf{P}$ (power input) |

The heat transfer rate equation can be integrated by two different methods:

1) The log-mean-temperature-difference method,
2) The heat exchanger effectiveness-NTU method.




LMTD Method: It can be shown that for counterflow and parallel flow heat exchangers, and condensers and evaporators, the total heat transfer rate through the total heat transfer area is given by:

$$(4)\ldots\ldots\ldots\ldots \quad q = UA\,(LMTD)$$

where:

$$LMTD \equiv \frac{\Delta t_{max} - \Delta t_{min}}{\ln \dfrac{\Delta t_{max}}{\Delta t_{min}}}$$

Whenever this method is used with more complex heat exchanger configurations, such as cross-flow exchangers or shell-and-tube devices with multiple passes, correction factors must be applied to the LMTD. In general, the LMTD method requires more extensive calculations, including trial-and-error solutions, particularly in the design of heat exchangers with complex arrangements. For these reasons, the effectiveness-NTU method is preferred for heat exchanger calculations.

Effectiveness-NTU Method: This method simplifies the heat exchanger design procedure by collecting various variables into a group of dimensionless parameters. The influence of all significant variables on heat exchanger performance can be visualized quite readily.

The heat exchanger effectiveness, $\epsilon$, compares the actual heat transfer rate to the maximum possible heat transfer rate, which is realizable only in a counterflow heat exchanger of infinite heat transfer area:

$$(5)\ldots\ldots\ldots\ldots \epsilon = \frac{C_h (t_{hi} - t_{ho})}{C_{min}(t_{hi} - t_{ci})} = \frac{C_c (t_{co} - t_{ci})}{C_{min}(t_{hi} - t_{ci})}$$

where: $C_{min}$ is the smaller of the $C_h$ and $C_c$ magnitudes

The number of exchanger transfer units, NTU, is a non-dimensional expression of the heat transfer size of the exchanger and is defined as:

$$(6)\ldots\ldots\ldots\ldots \mathrm{NTU} \equiv \frac{1}{C_{min}} \int_0^A U\,dA = \frac{(AU)_{ave}}{C_{min}}$$

When the NTU value is small, the effectiveness is low; and when the NTU value is large, the effectiveness approaches asymptotically the limit imposed by the flow arrangement and thermodynamic considerations.

The UA-term in the NTU expression indicates the cost of attaining a high effectiveness:

1) Increasing the area A means an increase in size, volume, and weight.
2) Increasing the U value means an increase in the power requirements to overcome the higher friction losses associated with higher film coefficients.

Neglecting tube wall resistance and fouling factors for simplicity, the overall conductance, U, is related to individual side conductance and surface geometry by:

$$(7)\ldots\ldots\ldots\ldots UA = \frac{1}{\dfrac{1}{(\eta_o h A)_h} + \dfrac{1}{(\eta_o h A)_c}}$$

or

$$(8)\ldots\ldots\ldots\ldots\quad U = \frac{1}{\dfrac{1}{(\eta_o h)_h} + \dfrac{1}{(\eta_o h)_c}\left(\dfrac{A_h}{A_c}\right)}$$

based on the total heat transfer area $A_h$ , which includes both the prime and the fin or extended area.

The capacity rate ratio, $C_{min}/C_{max}$ , where $C_{min}$ and $C_{max}$ are the smaller and the larger of the two magnitudes, respectively, is used as a parameter in plots of effectiveness versus NTU, because these capacities represent energy storage rates in each stream per unit temperature change.

Thus, the heat exchanger effectiveness is a function of:

$$\epsilon = \phi\left(NTU\,;\,C_{min}/C_{max}\,;\ \text{Flow arrangement}\right)$$

This function can be derived mathematically for each particular flow arrangement.

The most common heat exchanger flow arrangements are:

1) Counterflow: Fluids flow in a direction opposite to each other.

2) Parallel-flow: Fluids flow in the same direction.

3) Cross-flow: One fluid flows at an angle (usually 90-degree) to the other fluid.

4) Parallel-Counterflow, shell-and-tube: One fluid flows on the shell side in one or more passes, and the other fluid flows in a tube or tubes and makes two or more passes.

A comparison of the relative merits and performance characteristics of these more common types of heat exchangers shows that the poorest performance is obtained with a parallel-flow configuration. The effectiveness of the counterflow configuration is the highest, followed by the cross-flow arrangement and the tube-and-shell type heat exchanger.

In boilers (evaporators) and condensers, the fluid changing phase remains at a constant temperature throughout the heat exchanger, and the heat capacity rate is by definition equal to infinity.

Therefore, $C_{max} = \infty$ ; $C_{min}/C_{max} = 0$

and

(9)............ $\epsilon = 1 - e^{-NTU}$

In addition, the effectiveness can be expressed in terms of temperatures only:

For boilers or evaporators:

(10)........... $\epsilon = \dfrac{(t_{hi} - t_{ho})}{(t_{hi} - t_c)}$

For condensers:

(11)........... $\epsilon = \dfrac{t_{co} - t_{ci}}{t_h - t_{ci}}$

It is evident that the performance of boilers, evaporators, or condensers is independent of the direction of evaporating or condensing fluid flow. Therefore, from performance considerations, there is no advantage in using a counterflow heat exchanger over a cross-flow or parallel-flow heat exchanger. The choice of flow configuration is usually dictated by header and ducting considerations. Because of the convenient header configurations, cross-flow heat exchangers are widely utilized in airborne systems.

Figure 5.2 shows the performance characteristics of cross-flow heat exchangers in terms of effectiveness, NTU, and heat capacity ratio. The curve $C_{min}/C_{max} = 0$ represents the performance characteristics of boiling and condensing heat exchangers and is applicable to any flow arrangement.

A complete heat exchanger characterization must consider the pressure losses associated with various heat exchanger cores. The total pressure loss in a heat exchanger can be divided into three parts:

## CROSS-FLOW HEAT EXCHANGER PERFORMANCE

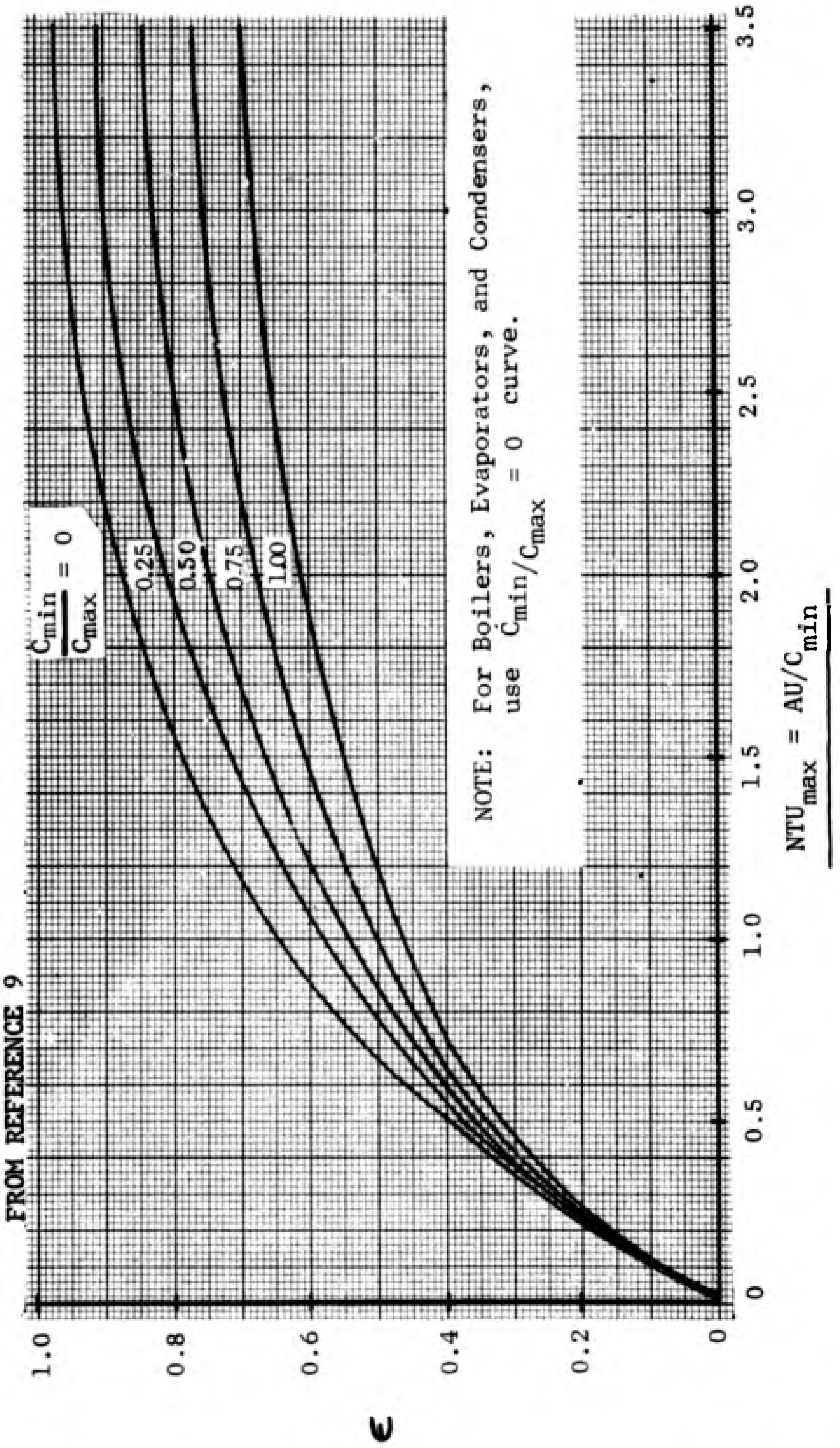



FIGURE 5.2

(1) Entrance (contraction) losses; (2) Core (friction) losses; and (3) Exit (expansion) losses. Entrance and exit losses provide usually only a small contribution to the overall pressure loss through a heat exchanger core, since the core friction term controls the magnitude of the pressure drop due to the large value of heat transfer surface area to free-flow area. Therefore, the entrance and exit effects may normally be neglected.

The friction pressure loss in a heat exchanger core may be computed from the following equation:

$$(12)\ldots\ldots\ldots\ldots \Delta P = f \rho \frac{V^2}{2g} \frac{A}{A_c}$$

Compact heat exchangers are almost exclusively employed for various airborne applications. They are characterized by very large area compactness (heat transfer area per unit volume) and flexibility in design, since the two fluid sides are independent of each other and the most suitable surface can be chosen for each of the fluids. Tubular surfaces represent the simplest form of compact heat transfer surfaces. Finned tube surfaces lend themselves to application where gas-to-liquid heat transfer is desired, since the gas side surface area can be made many times the liquid surface area. The most compact heat exchangers are obtained with plate-fin surfaces, which lend themselves to air-to-air as well as liquid-to-air applications.

Heat exchanger design data is presented for a multitude of various heat transfer surfaces in Reference 10 . This basic form for the presentation of design information consists of heat transfer and pressure loss data as a function of Reynolds Number. The heat transfer performance is summarized by means of the Colburn parameter ( $j = St \times Pr^{2/3}$). The pressure loss data is summarized by the Fanning friction factor.

The calculation of a heat exchanger size to accomplish a specific requirement is a trial-and-error iterative procedure. First, the required heat transfer size of the heat exchanger is determined from:

$$(13)\ldots\ldots\ldots\ldots (UA)_{req'd} = NTU\,(C_{min})$$

Secondly, a heat exchanger size is assumed; and the heat transfer size of this exchanger is computed from:

$$(14)\ldots\ldots\ldots\ldots(UA)_{available} = \frac{1}{\frac{1}{(\eta_o h)_h A_h} + \frac{1}{(\eta_o h)_c A_c}}$$

The data required for the computation of the UA value for the particular assumed size is readily obtained from the given data. The value of (UA) available is then compared with the value of (UA) required. When the two values agree, the selected heat exchanger meets the required heat transfer performance. The pressure data is then checked. If the two UA values do not agree, the heat exchanger dimensions are altered until the two values check. This trial-and-error procedure converges rather rapidly and usually not more than four iterations are sufficient.

If the pressure drop in a particular application tends to be high, the flow velocities can be reduced by increasing the number of flow passages in the heat exchanger. This will also decrease the heat transfer rate per unit of surface area, but the reduction in heat transfer rate will be considerably less than the pressure-drop reduction. The loss of heat transfer rate is then made up by increasing the surface area (lengthening the tubes), which in turn also increases the pressure drop, but only in the same proportion as the heat transfer surface area is increased.

The design and development of compact heat exchangers has been highly advanced in recent years, due to the ever-increasing applications for this type of equipment. Since substantial advances in heat transfer surface designs are not very likely, heat exchanger performance improvements will have to be realized by trade-offs between weight, size, and power requirements.

## 5.2 Gas Piston Unit

The gas piston unit is the main component of the power cycle of the Conductron system. It consists of a gas piston compressor/boiler, whose primary function is the generation of refrigerant vapor at the flow rate, temperature and pressure levels required for system operation, and a fill-jet pump, which "pumps" the condenser return flow into the gas piston compressor/boiler to sustain system operation.

Figure 5.3 is a schematic representation of a gas piston unit showing the variables which affect the operation and performance of this unit. Also shown are the temperature profiles for the primary (refrigerant) and secondary (air) fluids through the gas piston compressor/boiler.

Heat Balance:

$$\dot{W}_B c_p (t_{B1} - t_{B2}) = \dot{W}_b (h_3 - h_8)$$

$$\dot{W}_b = \frac{\dot{W}_B c_p (t_{B1} - t_{B2})}{(h_3 - h_8)}$$

Since the fill-jet pump motive flow, $\dot{W}_f$ , is only 7% of the condenser return flow, $\dot{W}_r$ , the enthalpy at (8) will not be appreciable higher than at (1) .

Thus, assuming $h_8 \approx h_1$ , the refrigerant inlet conditions to the boiler are essentially the condenser outlet conditions. For the case of no condenser subcooling, the boiler inlet enthalpy is a function of the condensing temperature, $h_8 = f(t_c)$. Assuming no superheat, the boiler exit enthalpy is a function of the boiling temperature, $h_3 = f(t_b)$.

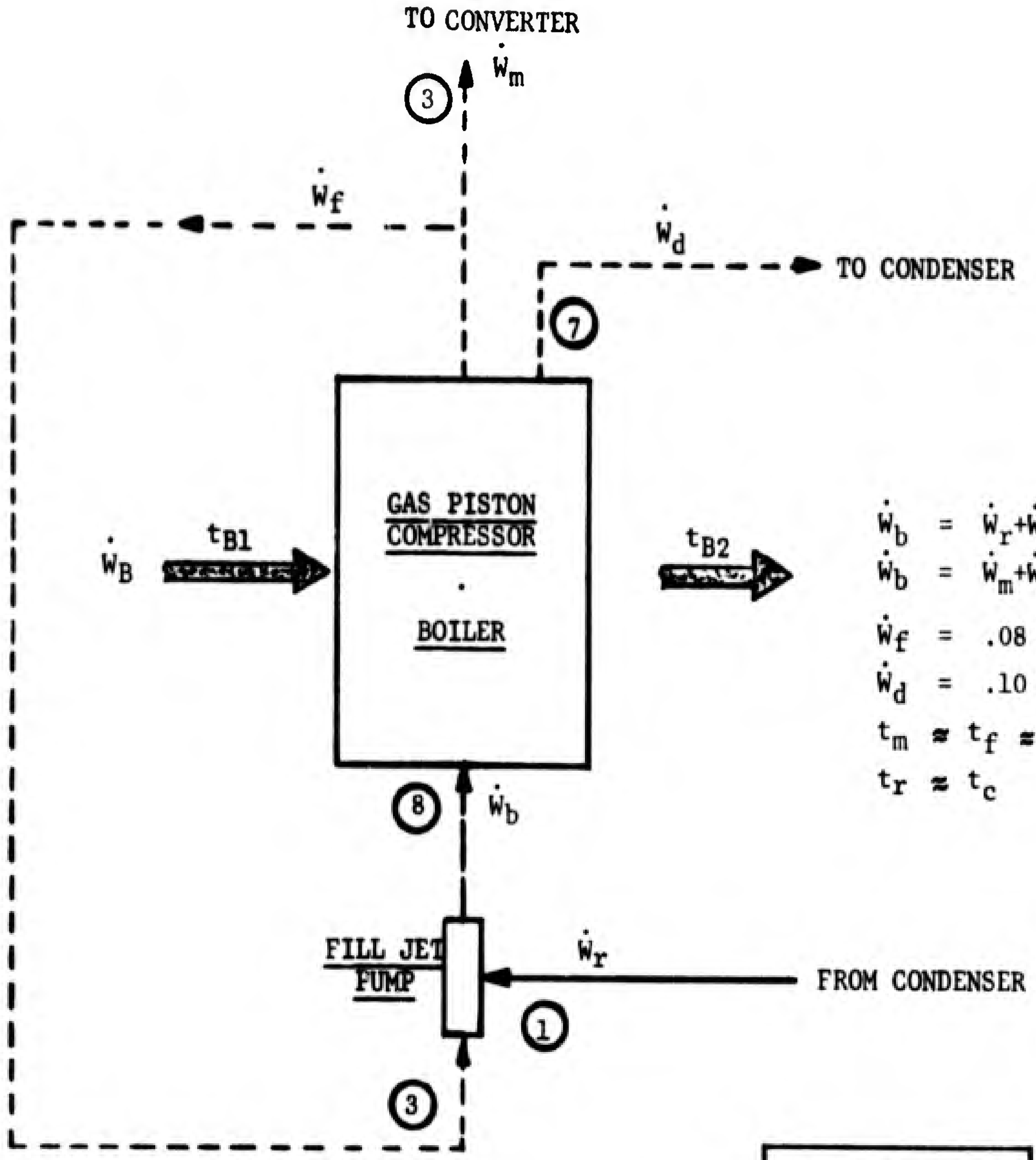
The refrigerant boil-off rate, $\dot{W}_b$ , can be expressed as:

$$\dot{W}_b = \frac{Q_B}{(h_3 - h_1)}$$

where $Q_B$ is the heat input to the boiler from the compressor bleed air, and $h_3$ and $h_1$ are the boiler inlet and outlet enthalpies as determined by the boiling and condensing temperatures, respectively.

The actual heat input into the boiler is dependent to a large degree upon the actual design of the boiler and all of the applicable heat exchanger characterizations.

Figures 5.4 through 5.8 show the relationships between the refrigerant boil-off rate, the heat input into the boiler, and condensing temperature for various boiling temperatures. These curves indicate that for a given heat input the refrigerant boil-off rate increases with an increase in condensing temperature and a decrease in boiling temperature.




$$\dot{W}_b = \dot{W}_r + \dot{W}_f$$
$$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f$$
$$\dot{W}_f = .08\ \dot{W}_m$$
$$\dot{W}_d = .10\ (\dot{W}_m + \dot{W}_f)$$
$$t_m \approx t_f \approx t_d$$
$$t_r \approx t_c$$

GAS PISTON UNIT CHARACTERISTICS

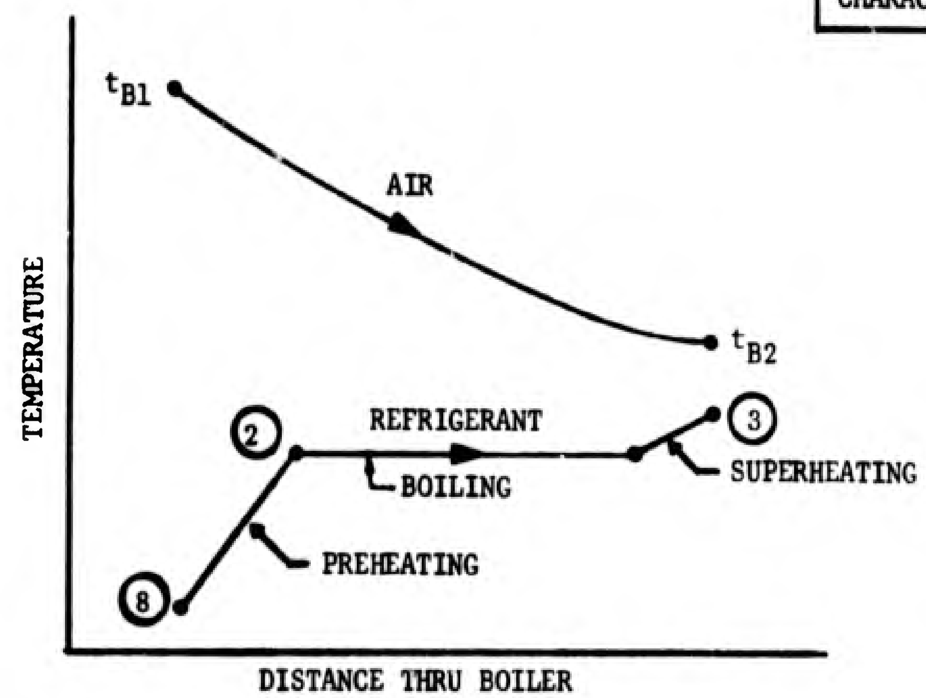





FIGURE 5.3

REFRIGERANT BOIL-OFF RATE
VS.
REQUIRED HEAT INPUT

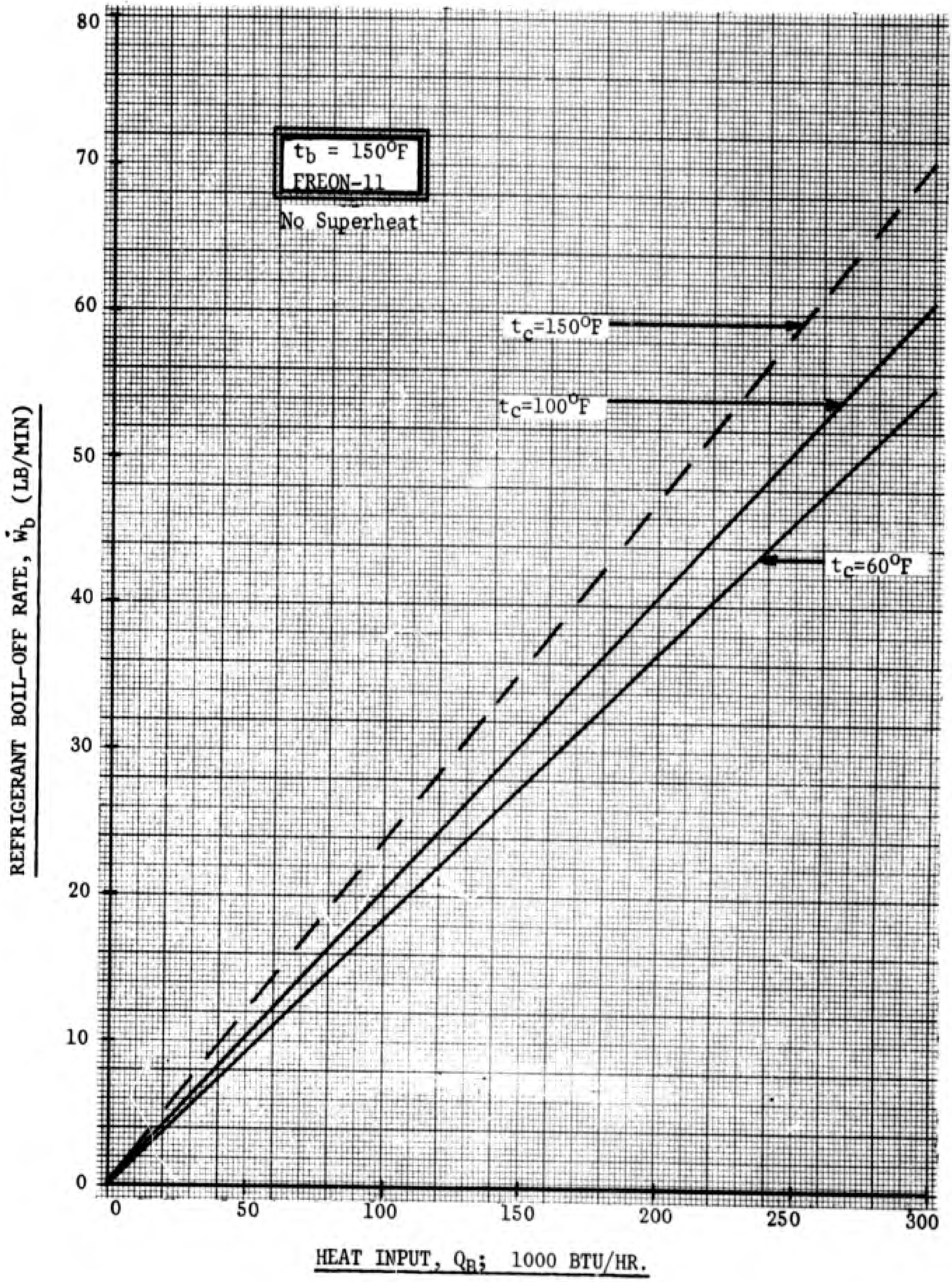
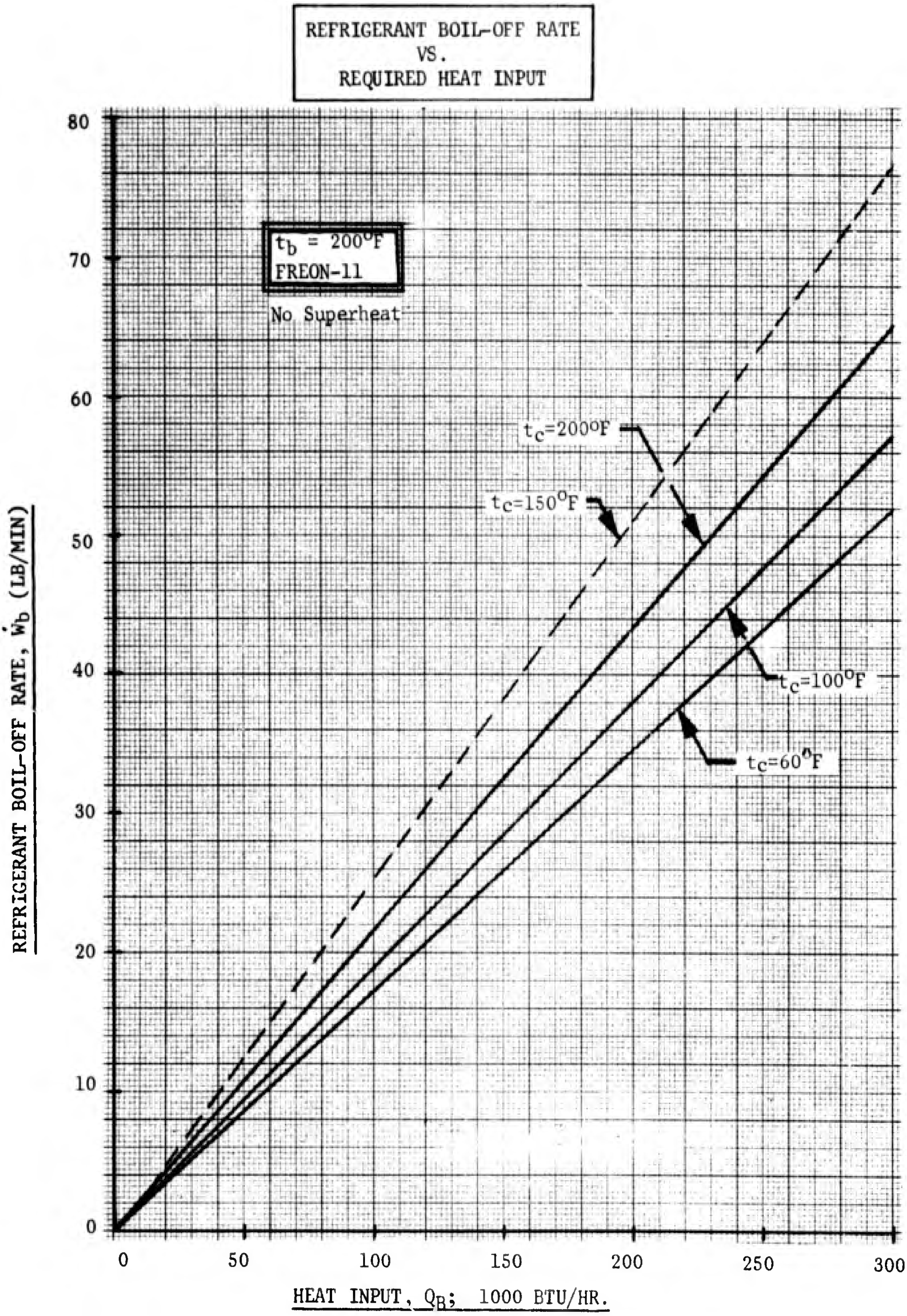



FIGURE 5.4



REFRIGERANT BOIL-OFF RATE
VS.
REQUIRED HEAT INPUT
$t_b$ = 200°F
FREON-11
No Superheat
$t_c$=200°F
$t_c$=150°F
$t_c$=100°F
$t_c$=60°F
REFRIGERANT BOIL-OFF RATE, $\dot{w}_b$ (LB/MIN)
HEAT INPUT, $Q_B$; 1000 BTU/HR.
0
10
20
30
40
50
60
70
80
0
50
100
150
200
250
300


FIGURE 5.5

REFRIGERANT BOIL-OFF RATE
VS.
REQUIRED HEAT INPUT

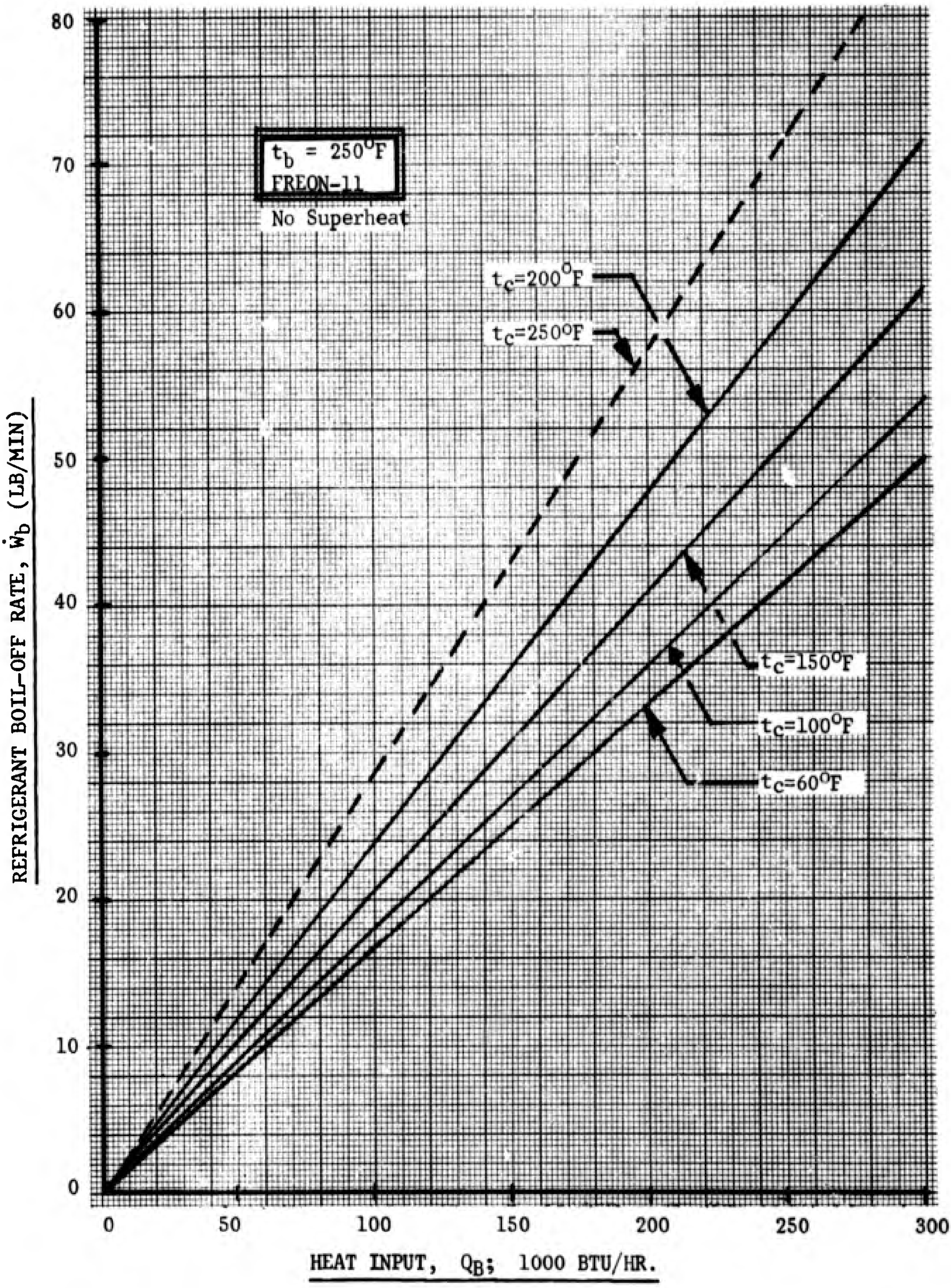





FIGURE 5.6

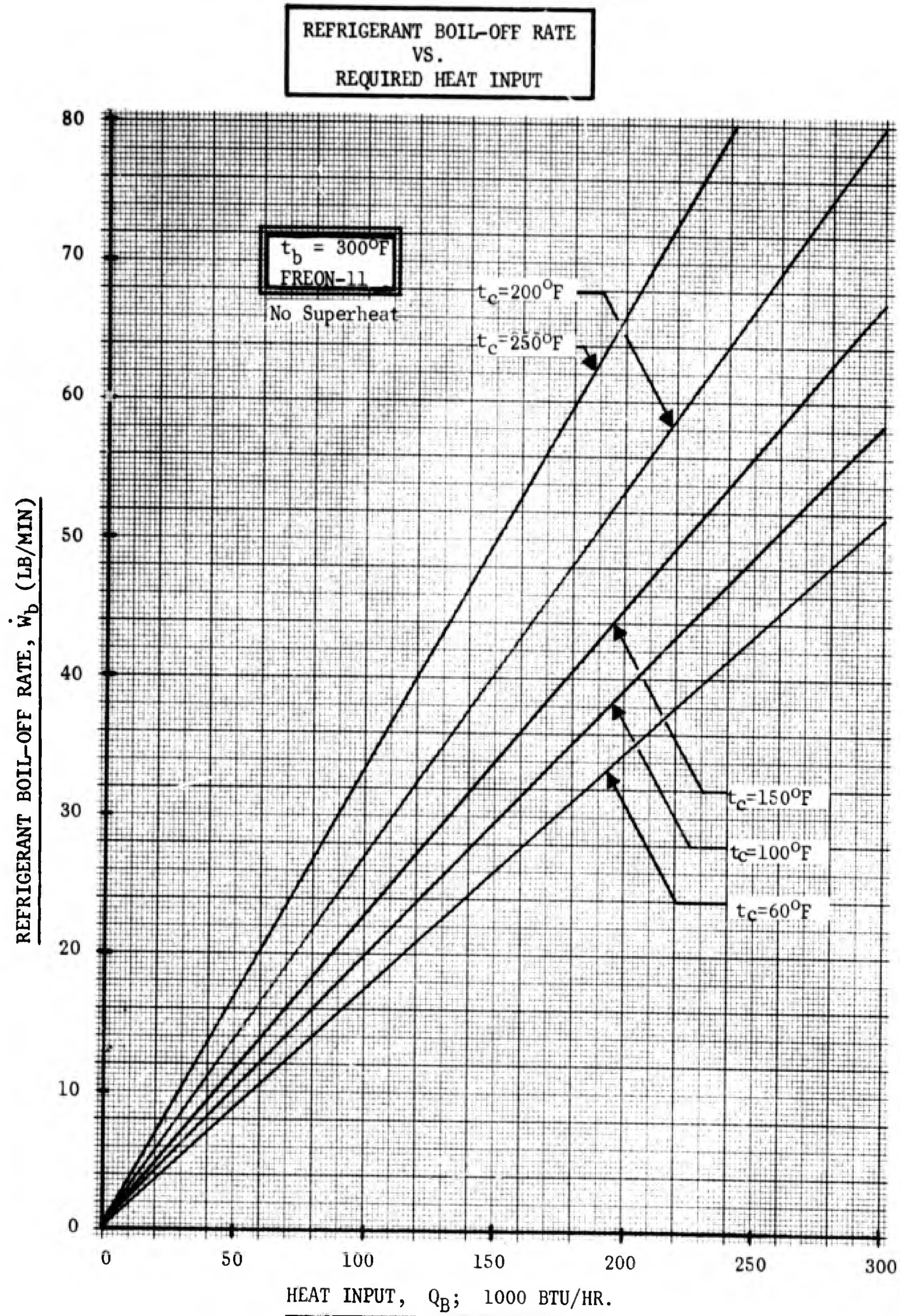

REFRIGERANT BOIL-OFF RATE
VS.
REQUIRED HEAT INPUT
$t_b$ = 300°F
FREON-11
No Superheat
$t_c$=200°F
$t_c$=250°F
$t_c$=150°F
$t_c$=100°F
$t_c$=60°F
REFRIGERANT BOIL-OFF RATE, $\dot{W}_D$ (LB/MIN)
HEAT INPUT, $Q_B$; 1000 BTU/HR.
0
10
20
30
40
50
60
70
80
0
50
100
150
200
250
300



FIGURE 5.7

REFRIGERANT BOIL-OFF RATE
VS.
REQUIRED HEAT INPUT

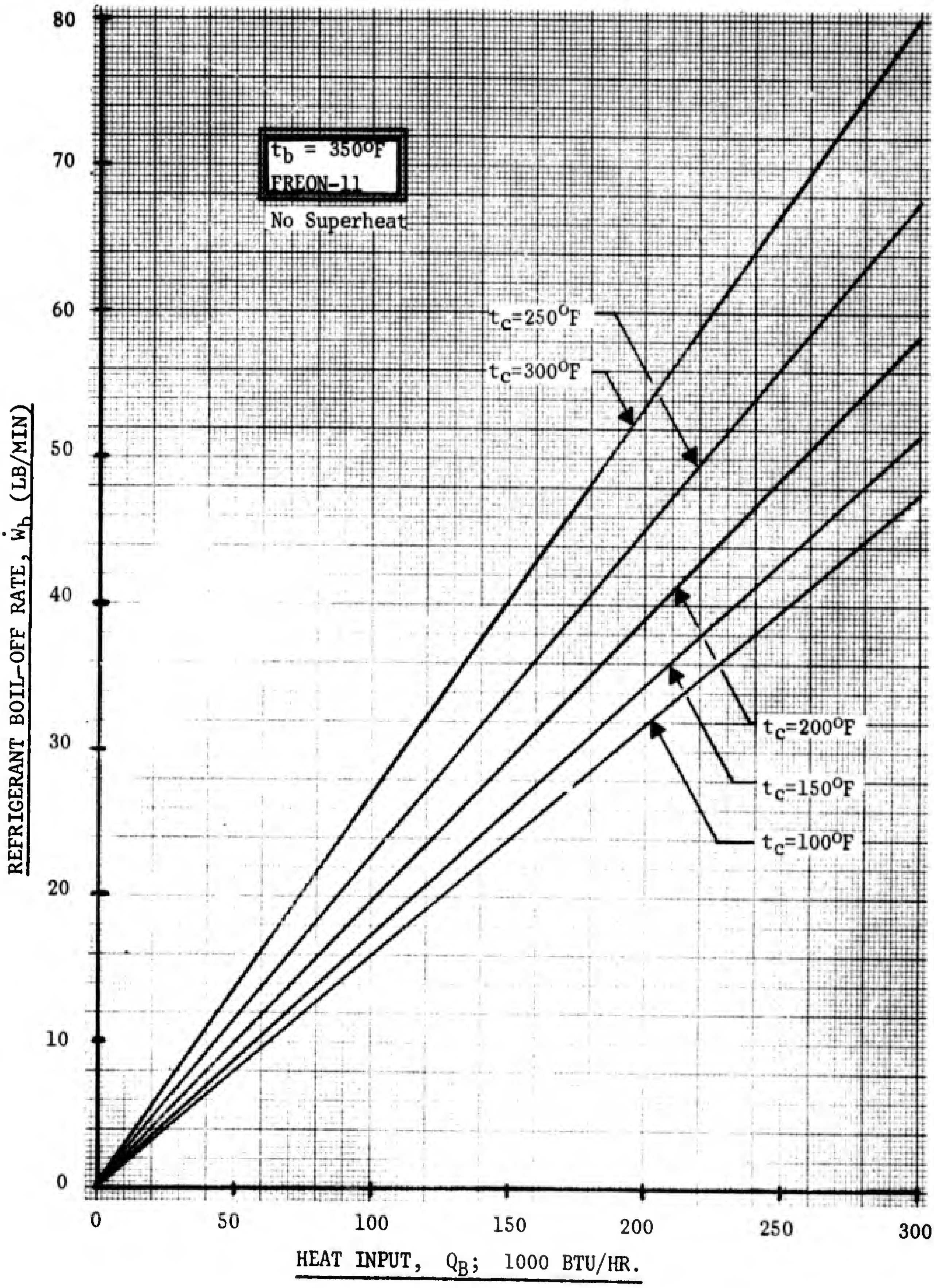





FIGURE 5.8

## 5.3 Condenser

The condensing heat exchanger is the system component where the unwanted system thermal energy is rejected to a suitable heat sink. In the basic Conductron system, the condenser is used to reject the heat associated with condensing the motive flow and the depressurization flow of the power cycle and the suction flow of the refrigeration cycle. With the exception of the converter (ejector), it is the only component common to both loops of the Conductron system.

Figure 5.9 is a schematic representation of the condenser showing the variables affecting the operation and performance of this component. The temperature profiles of the primary fluid (refrigerant) and secondary fluid (air) through the condenser are also shown.

Heat Balance:

$$\dot{W}_c c_p (t_{c2} - t_{c1}) = \dot{W}_d (h_7 - h_1) + \dot{W}_e (h_4 - h_1)$$

defining:

ejector flow ratio, $x = \dot{W}_m / \dot{W}_s$

and

depressurization flow ratio, $y = \dot{W}_d / \dot{W}_m$

$$\dot{W}_e = (1 + x)\dot{W}_s \quad \text{and} \quad \dot{W}_d = xy\dot{W}_s$$

and since $h_7 = h_3$,

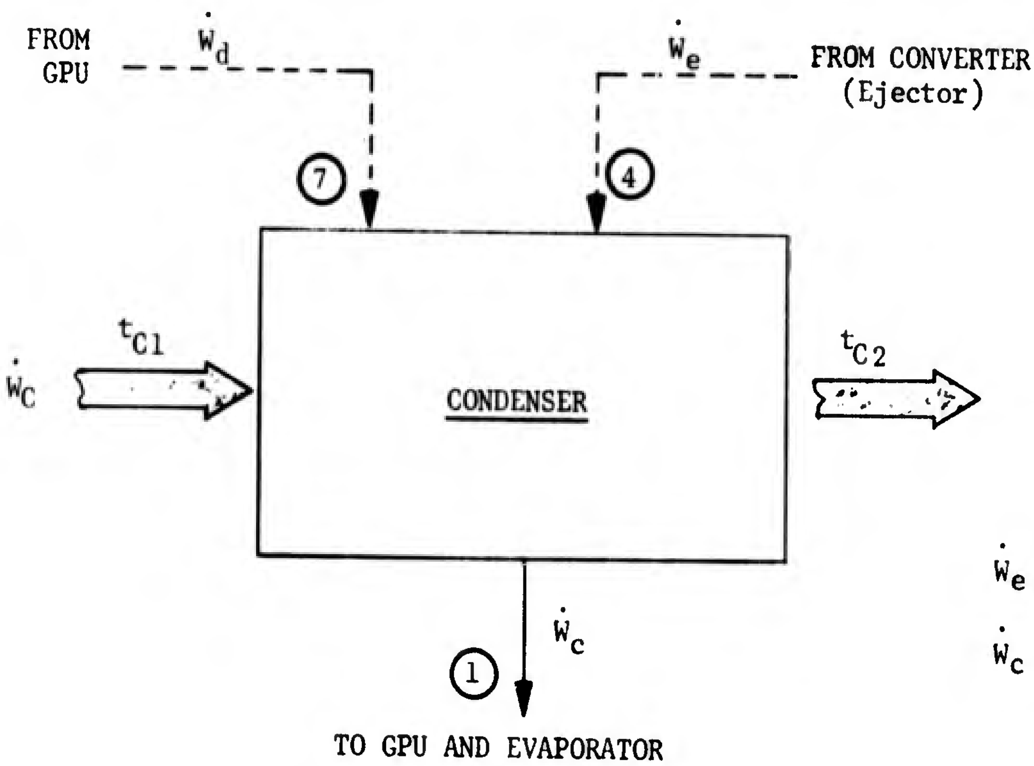
$$\dot{W}_c c_p (t_{c2} - t_{c1}) = xy\dot{W}_s (h_3 - h_1) + (1 + x)\dot{W}_s (h_4 - h_1)$$

or

$$\frac{\dot{W}_c}{\dot{W}_s} = \frac{xy(h_3 - h_1) + (1 + x)(h_4 - h_1)}{c_p (t_{c2} - t_{c1})}$$

also: $$\frac{\dot{W}_c}{\dot{W}_s} = \frac{\dot{W}_e + \dot{W}_d}{\dot{W}_s} = (1 + x) + xy = 1 + (1 + y)x$$

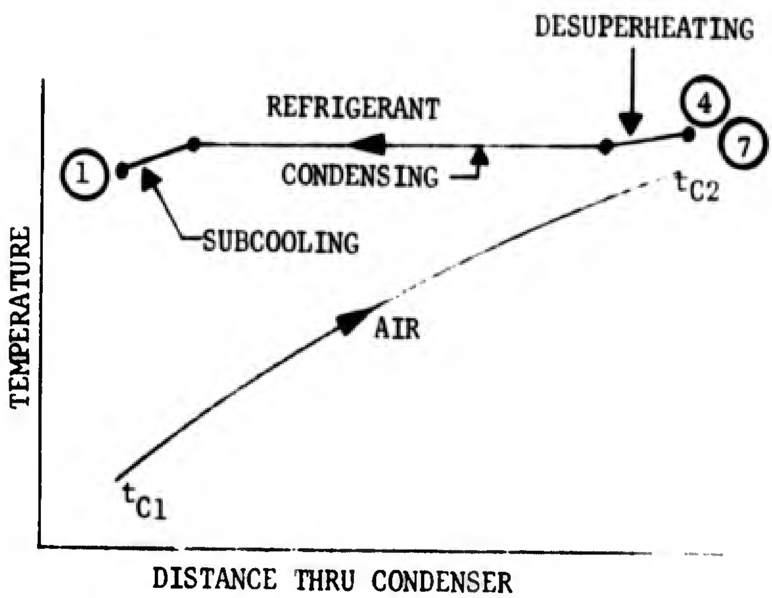
Figures 5.10 through 5.14 show the condenser air flow to evaporating (suction) flow ratio ($\dot{W}_c/\dot{W}_s$) as a function of air temperature drop across the condenser and ejector flow ratio ($\dot{W}_m/\dot{W}_s$) for various condensing temperatures.




$$\dot{W}_e = \dot{W}_m + \dot{W}_s$$

$$\dot{W}_c = \dot{W}_e + \dot{W}_d$$

$$= \dot{W}_m + \dot{W}_s + \dot{W}_d$$




CONDENSER CHARACTERISTICS



FIGURE 5.9

CONDENSER AIR FLOW/EVAPORATING FLOW
VS.
CONDENSER AIR TEMPERATURE DROP AND EJECTOR FLOW RATIO

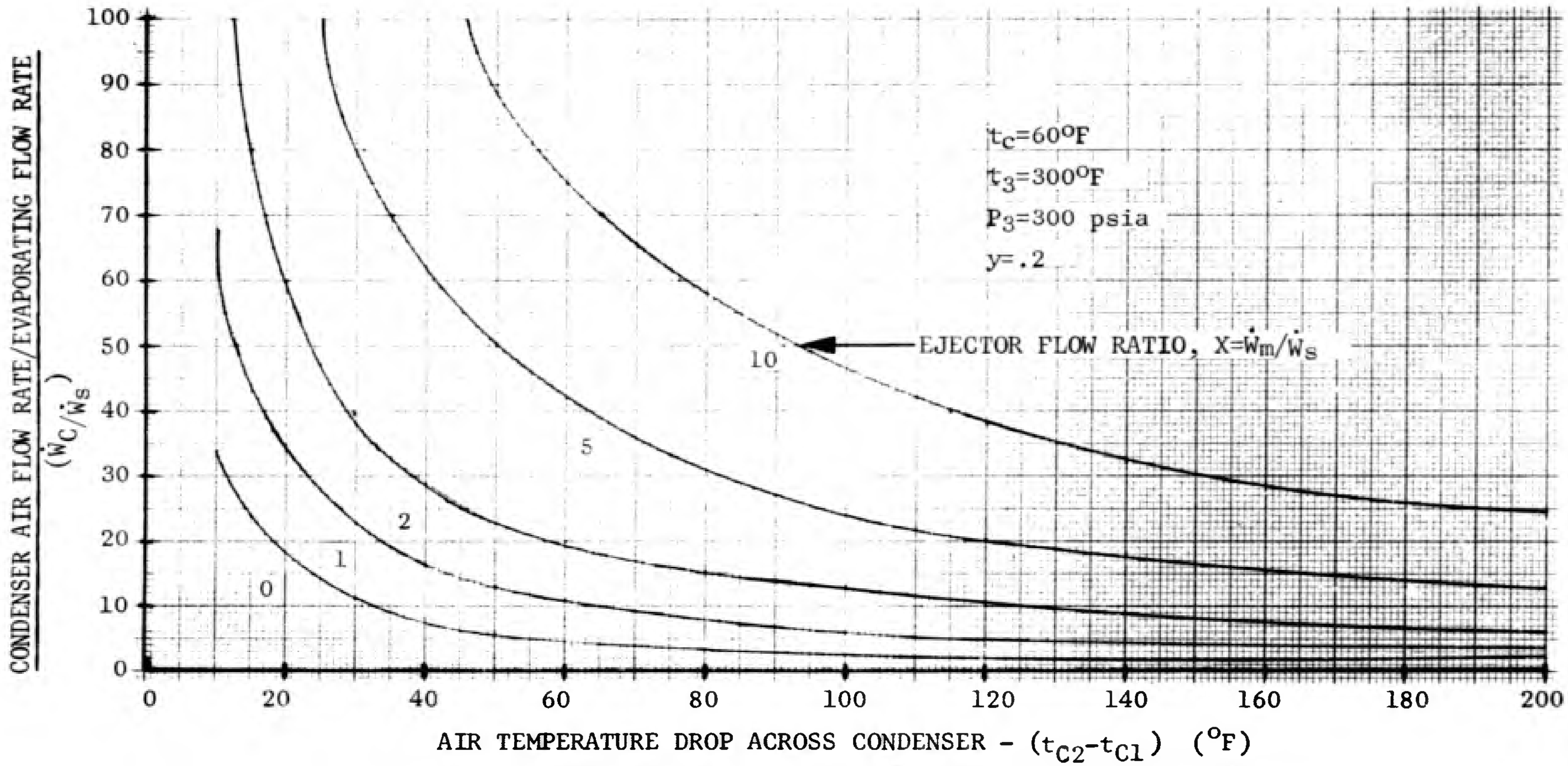



FIGURE 5.10

CONDENSER AIR FLOW/EVAPORATING FLOW
VS.
CONDENSER AIR TEMPERATURE DROP AND EJECTOR FLOW RATIO

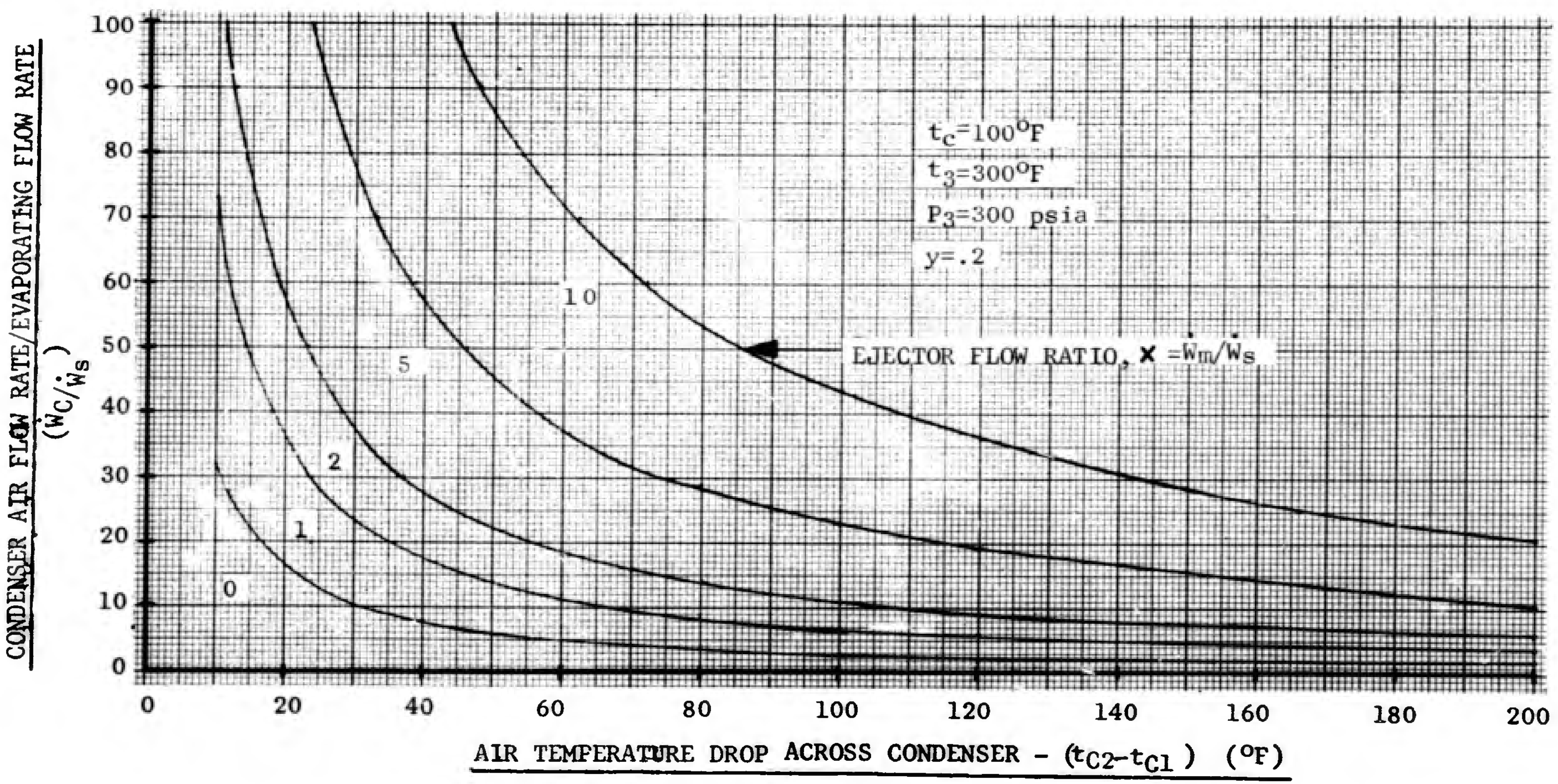





FIGURE 5.11

**CONDENSER AIR FLOW/EVAPORATING FLOW**
**VS.**
**CONDENSER AIR TEMPERATURE DROP AND EJECTOR FLOW RATIO**

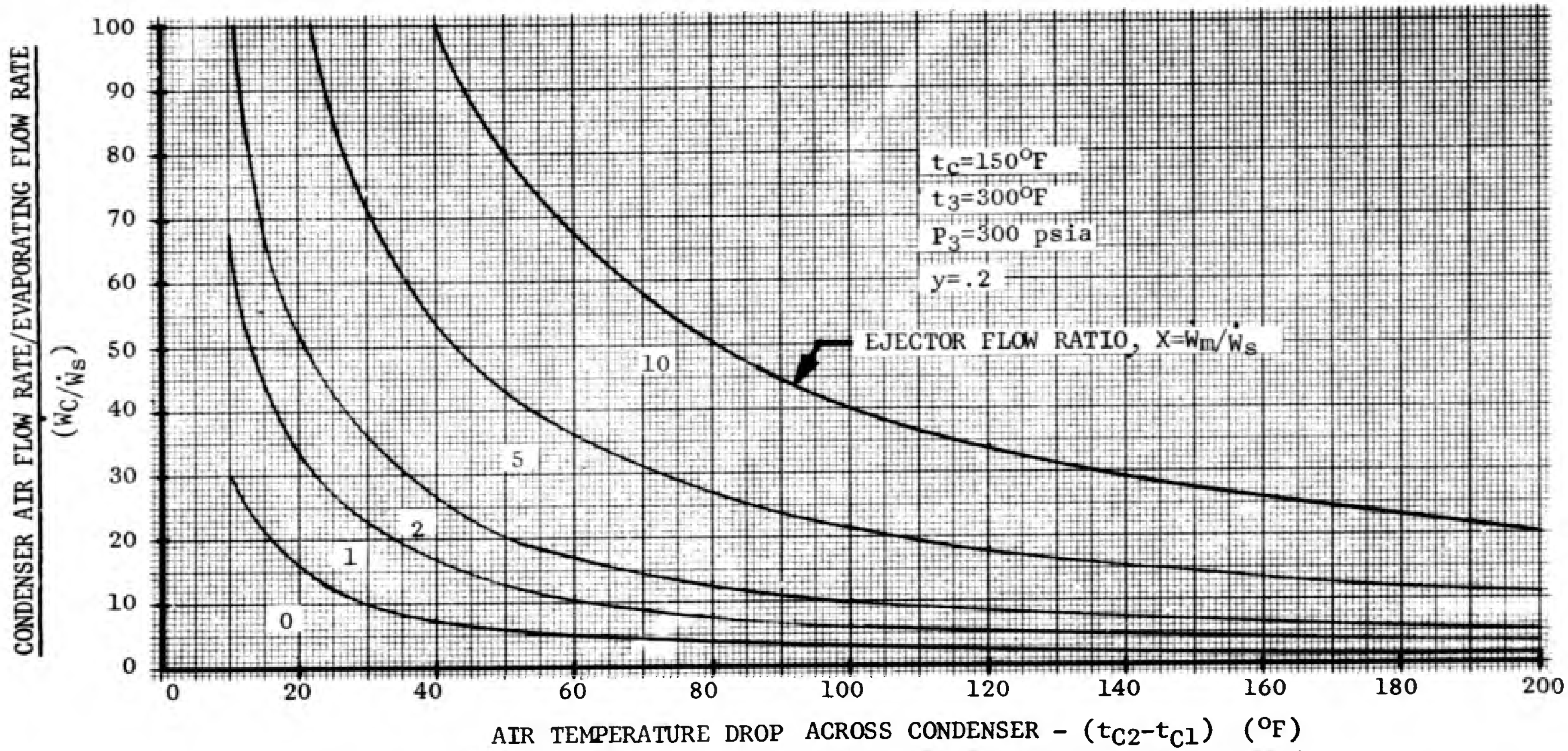





FIGURE 5.12

CONDENSER AIR FLOW/EVAPORATING FLOW
VS.
CONDENSER AIR TEMPERATURE DROP AND EJECTOR FLOW RATIO

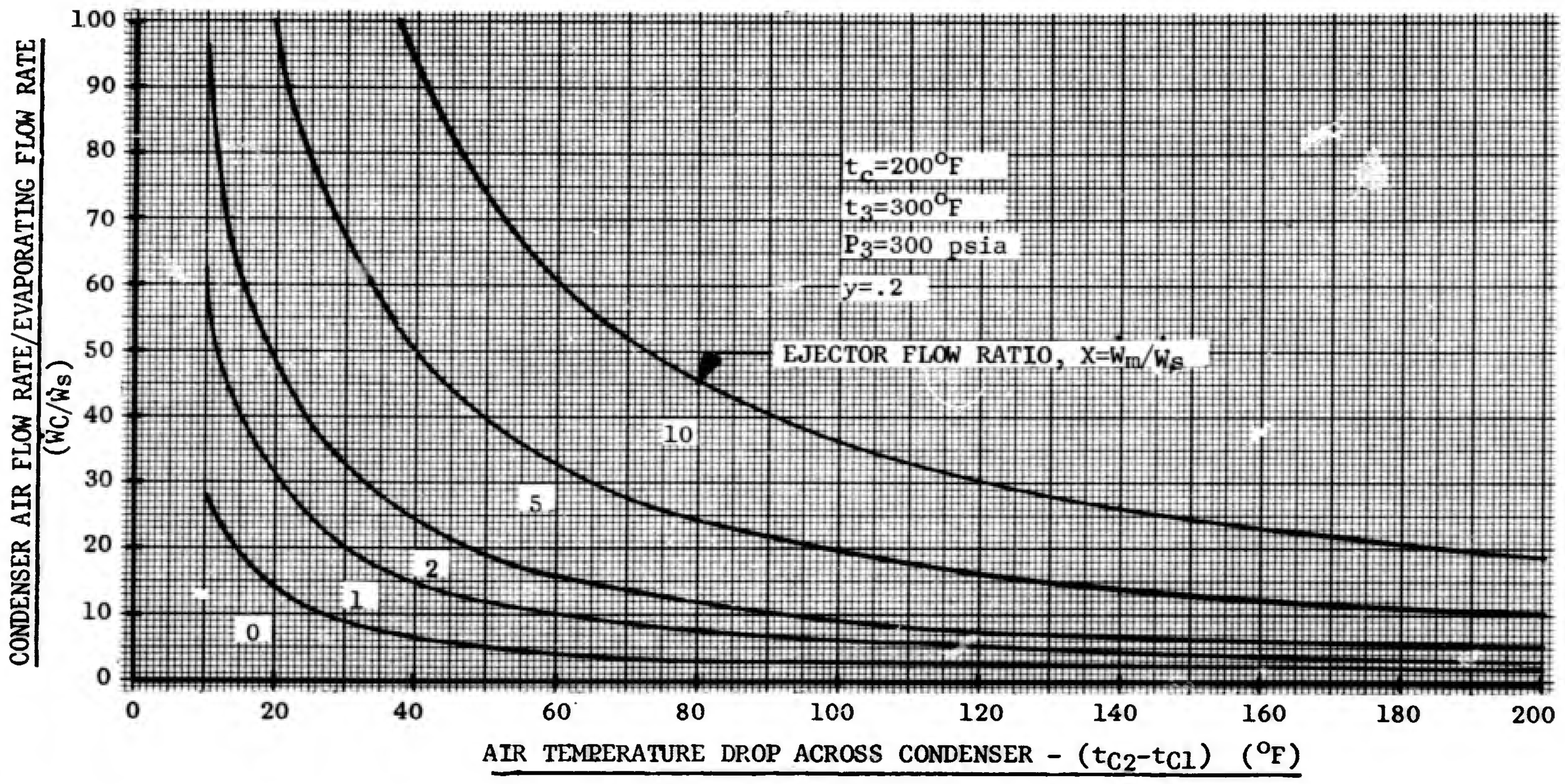





FIGURE 5.13

CONDENSER AIR FLOW/EVAPORATING FLOW
VS.
CONDENSER AIR TEMPERATURE DROP AND EJECTOR FLOW RATIO

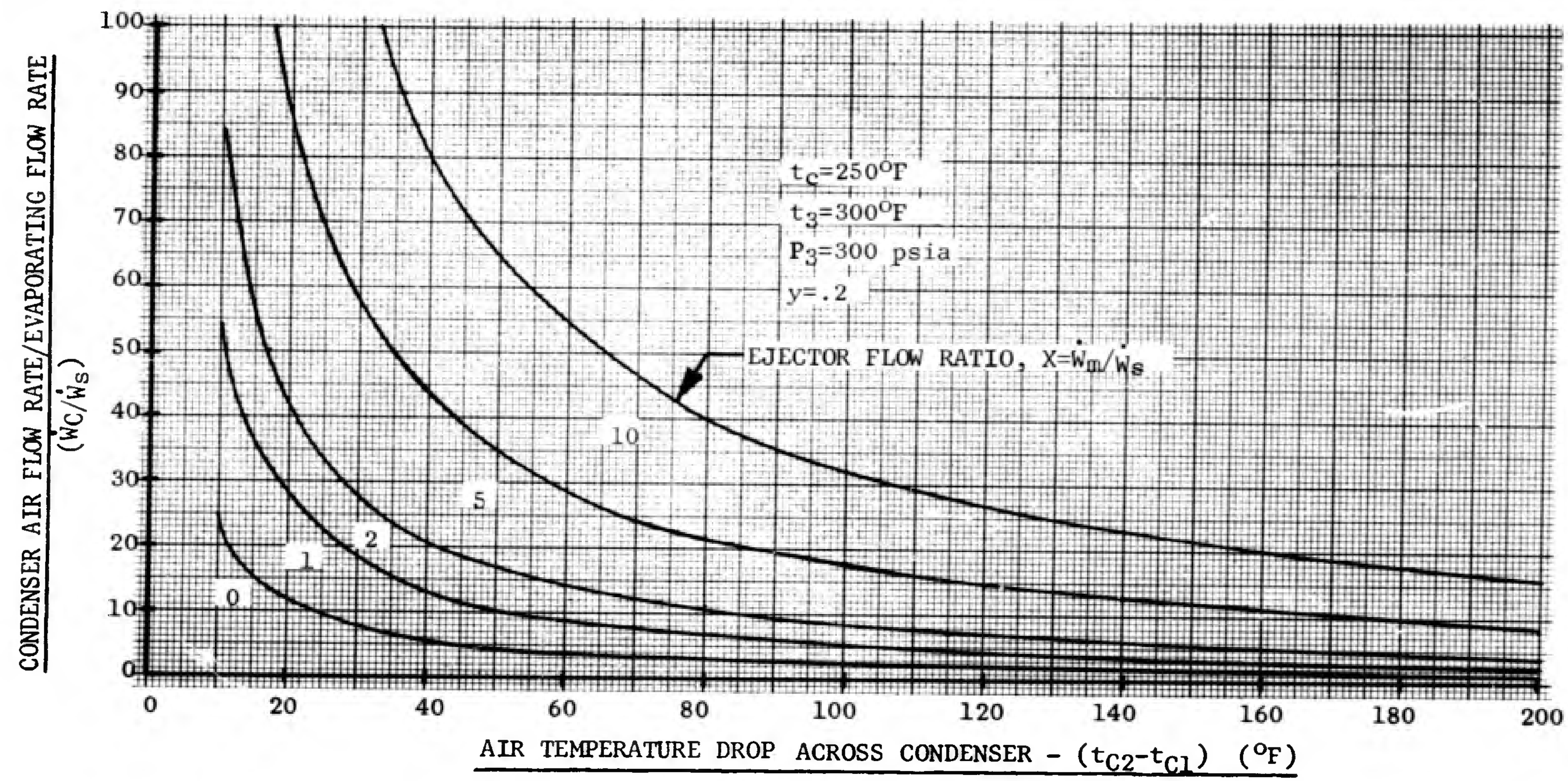





FIGURE 5.14

These curves are based on boiler outlet conditions of saturated vapor at 300 psia (300°F) and a depressurization flow ratio $y = 0.2$. In addition, no condenser subcooling is assumed, thus the condenser outlet enthalpy is a function of the condensing temperature, $h_1 = f(t_c)$. The ejector outlet enthalpy, $h_4$, was taken as the enthalpy corresponding to the condensing pressure and a superheat of 15°F. The curves show that for a given air temperature drop across the condenser and fixed boiler outlet conditions (pressure and temperature) the condenser air flow to evaporating flow ratio increases with a decrease in condensing temperature and an increase in converter flow ratio, respectively. Therefore, for a given evaporator flow rate, changes in condenser cooling flow requirements are a function of condensing temperature and/or converter motive flow.

Figure 5.15 shows the effect of depressurization flow ratio at a condensing temperature of 150°F and an ejector flow ratio of 5. The condenser inlet and outlet conditions were assumed to be those of the previous figures. It can be seen that the condenser cooling flow requirements increase with increasing depressurization flow ratios for any given condenser air temperature drop.

Figure 5.16 is a plot of the condensing flow to evaporating flow ratio ($\dot{W}_c / \dot{W}_4$) as a function of ejector flow ratio, $X$, for various depressurization flow ratios, $y$. The condensing flow rate increases with increasing ejector flow ratio and/or depressurization flow ratio.

CONDENSER AIR FLOW/EVAPORATING FLOW
VS.
CONDENSER AIR TEMPERATURE DROP AND DEPRESSURIZATION FLOW RATIO

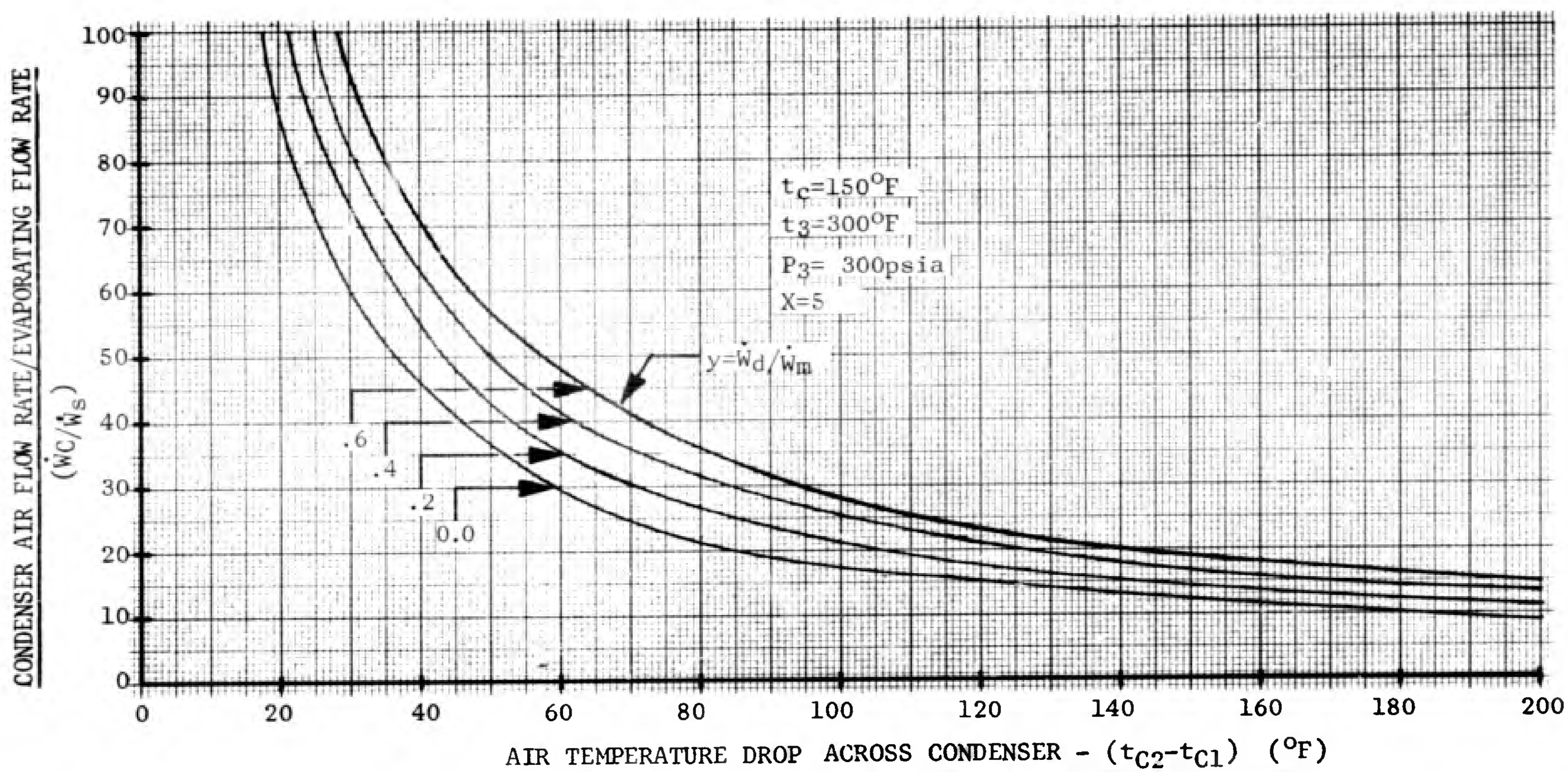





FIGURE 5.15

CONDENSING FLOW/EVAPORATING FLOW
VS.
EJECTOR FLOW RATIO AND DEPRESSURIZATION FLOW RATIO

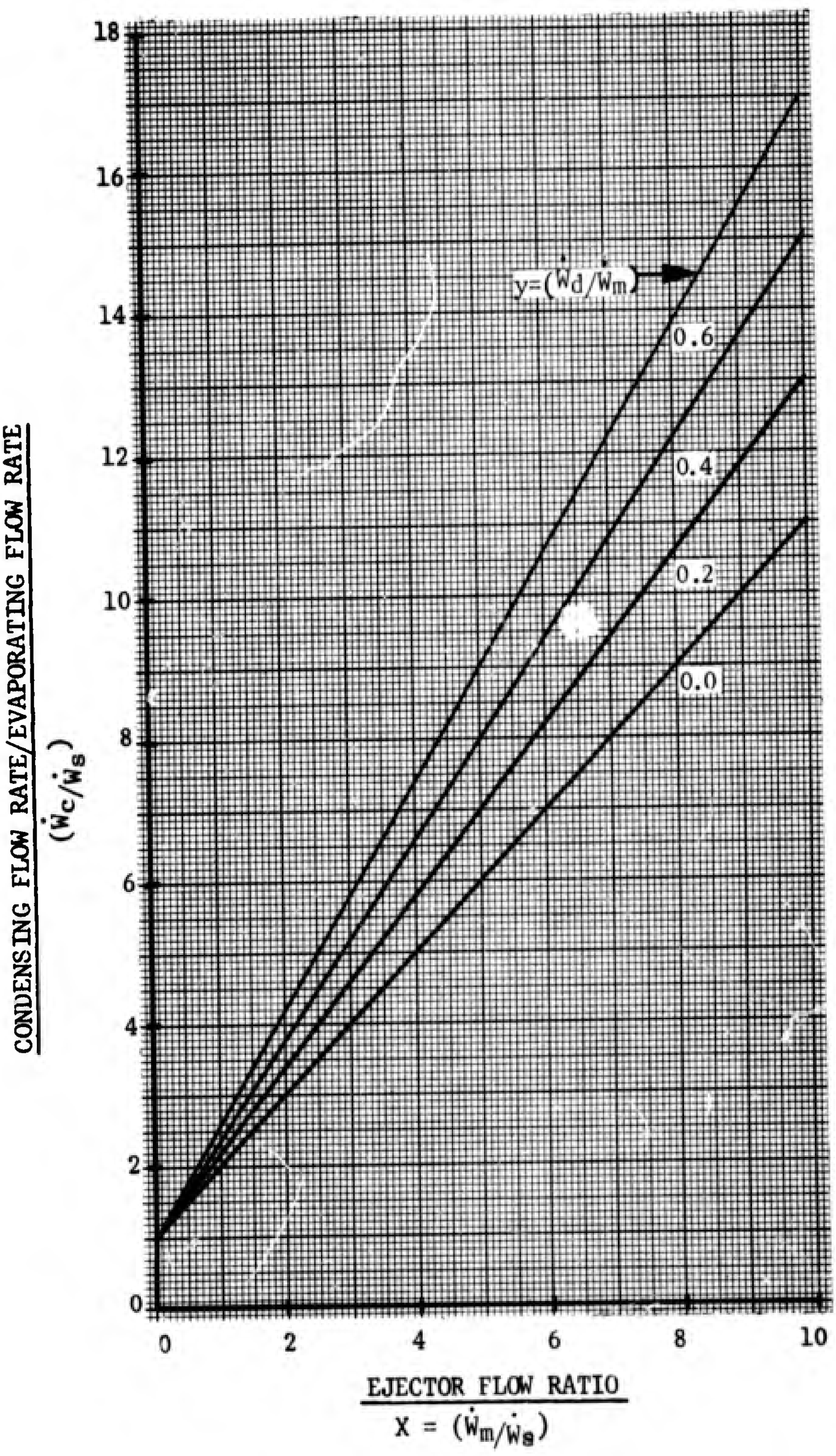





FIGURE 5.16

## 5.4 Evaporator

The evaporating heat exchanger is used in the Conductron system to either accomplish the complete or partial removal of the cabin cooling load (closed cycle) or serve as in intermediate step in the reduction of the temperature level of the bleed air used for cabin cooling (open cycle). This component is part of the refrigeration cycle of the Conductron system.

Figure 5.17 is a schematic representation of the evaporator, showing the variables affecting the operation and performance of this component. Also presented are the temperature profiles of the primary fluid (refrigerant) and secondary fluid (air) through the evaporator.

Heat Balance:

$$\dot{W}_s c_p (t_{s1} - t_{s2}) = \dot{W}_3 (h_6 - h_5)$$

$$\frac{\dot{W}_s}{\dot{W}_3} = \frac{(h_6 - h_5)}{c_p (t_{s1} - t_{s2})}$$

but since $h_5 = h_1$ , the enthalpy at the evaporator inlet can be expressed as a function of the condensing temperature, $h_5 = f(t_c)$.

Therefore,

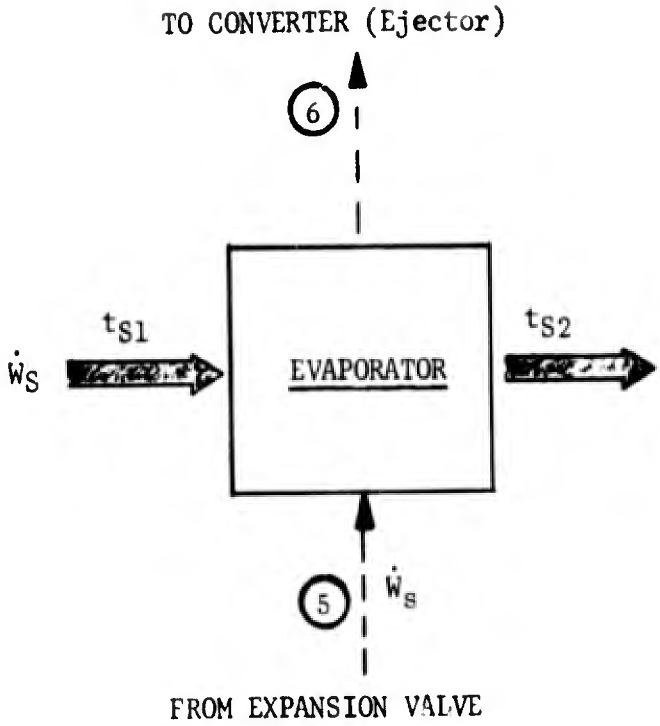
$$\frac{\dot{W}_s}{\dot{W}_3} = \frac{(h_6 - h_1)}{c_p (t_{s1} - t_{s2})}$$

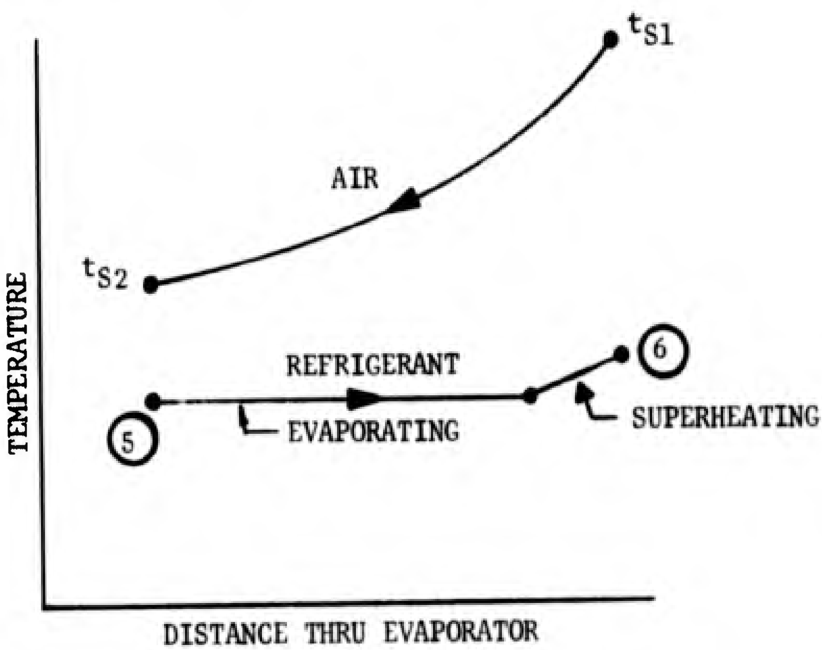
Also,

$$\dot{W}_3 = \frac{Q_c}{(h_6 - h_1)}$$

where $Q_c$ is the cabin cooling load.

For a closed system, the required evaporating flow is a function of cabin cooling load, evaporating temperature, and condensing temperature.




EVAPORATOR CHARACTERISTICS



FIGURE 5.17

Figure 5.18 is a plot of the evaporator air flow to evaporating flow ratio as a function of air temperature drop across the evaporator and various condensing temperatures for an evaporating temperature of 40°F. A 5°F superheat is assumed for the evaporator outlet. The figure shows that for a given air temperature drop across the evaporator and a fixed evaporating flow rate, the evaporator air flow decreases with increasing condensing temperatures.

Figure 5.19 shows the effect of evaporating temperature on the flow rate ratio $\dot{W}_\ell/\dot{W}_1$ for various condensing temperatures at a selected evaporator air temperature difference of 20°F. The evaporator air flow to evaporating flow ratio increases with increasing evaporating temperatures and decreasing condensing temperatures. For a given evaporator cooling load higher condensing temperatures tend to reduce the required evaporator air flow.

Figure 5.20 represents the evaporator performance characteristics in terms of evaporating flow rate per 12,000 BTU/hr. (1-ton) cooling load and evaporating temperature for a range of condensing temperatures. This figure shows that high evaporating temperatures and low condensing temperatures result in the lowest refrigerant flow requirements for a given cooling load.

## 5.5 Converter/Ejector

The converter is a device which utilizes a high pressure (motive) gas to raise the pressure of a lower pressure (suction) gas to a common intermediate pressure level. In the Conductron system, the converter is used to raise the low pressure refrigerant vapor leaving the evaporator to the condensing pressure. This can be accomplished by a number of devices, such as ejectors, reciprocating boosters, and turbo-compressors; but primary interest is centered on those which utilize kinetic energy effects and eliminate the need for moving parts, resulting in components of relatively low weight, mechanical simplicity, and high reliability. For these reasons, only conventional ejectors and new devices based on the energy and momentum exchange principle are being considered for the Conductron system. A classical analysis of the thermodynamic process in conventional ejectors is presented in Appendix B.

Figure 5.21 is a schematic representation of a converter. The variables affecting the performance of such a device are identified, and the ideal thermodynamic process of converters is shown on a pressure-enthalpy diagram.

EVAPORATOR AIR FLOW/EVAPORATING FLOW
VS.
EVAPORATOR AIR TEMPERATURE DROP
AND CONDENSING TEMPERATURE

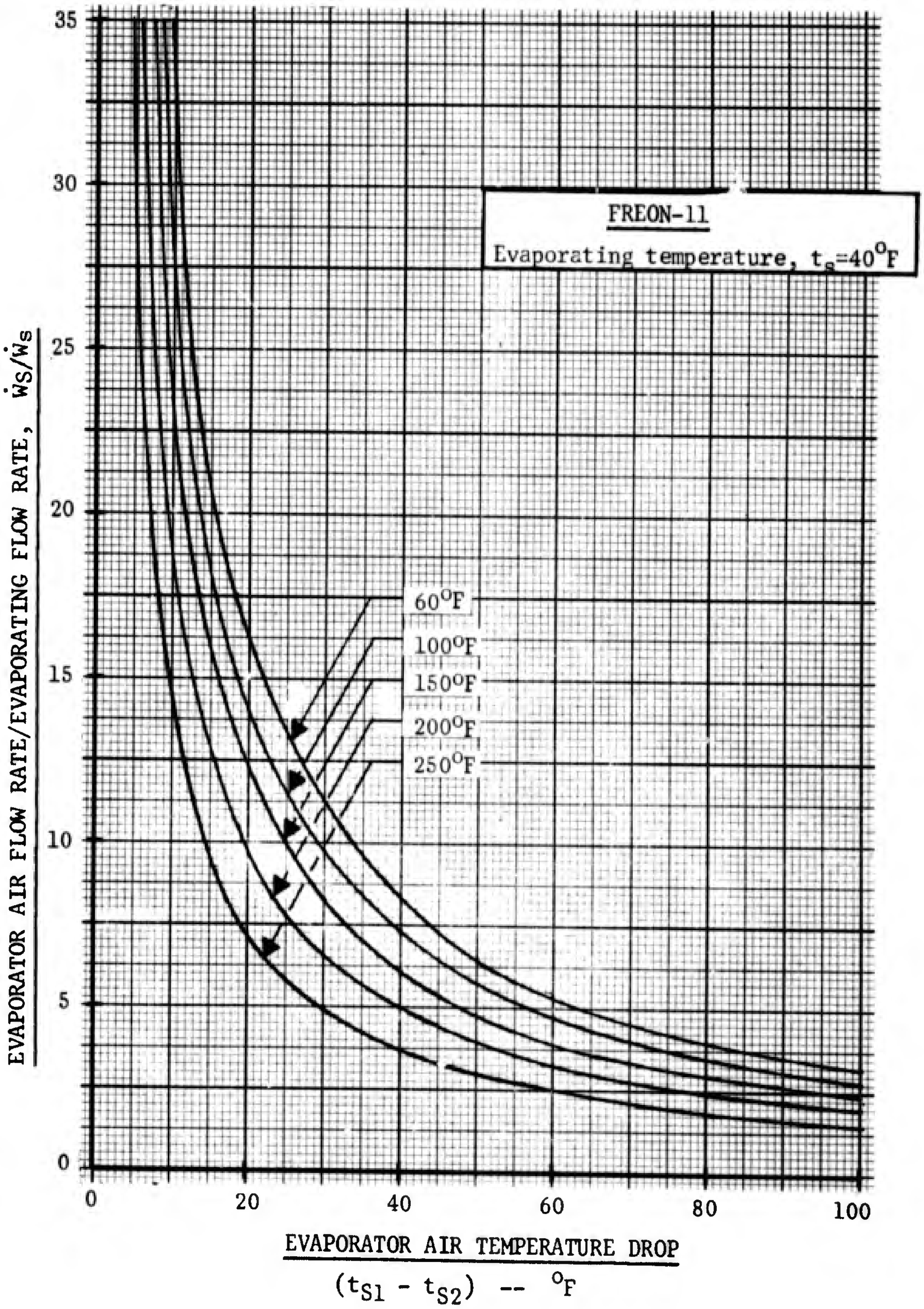





FIGURE 5.18

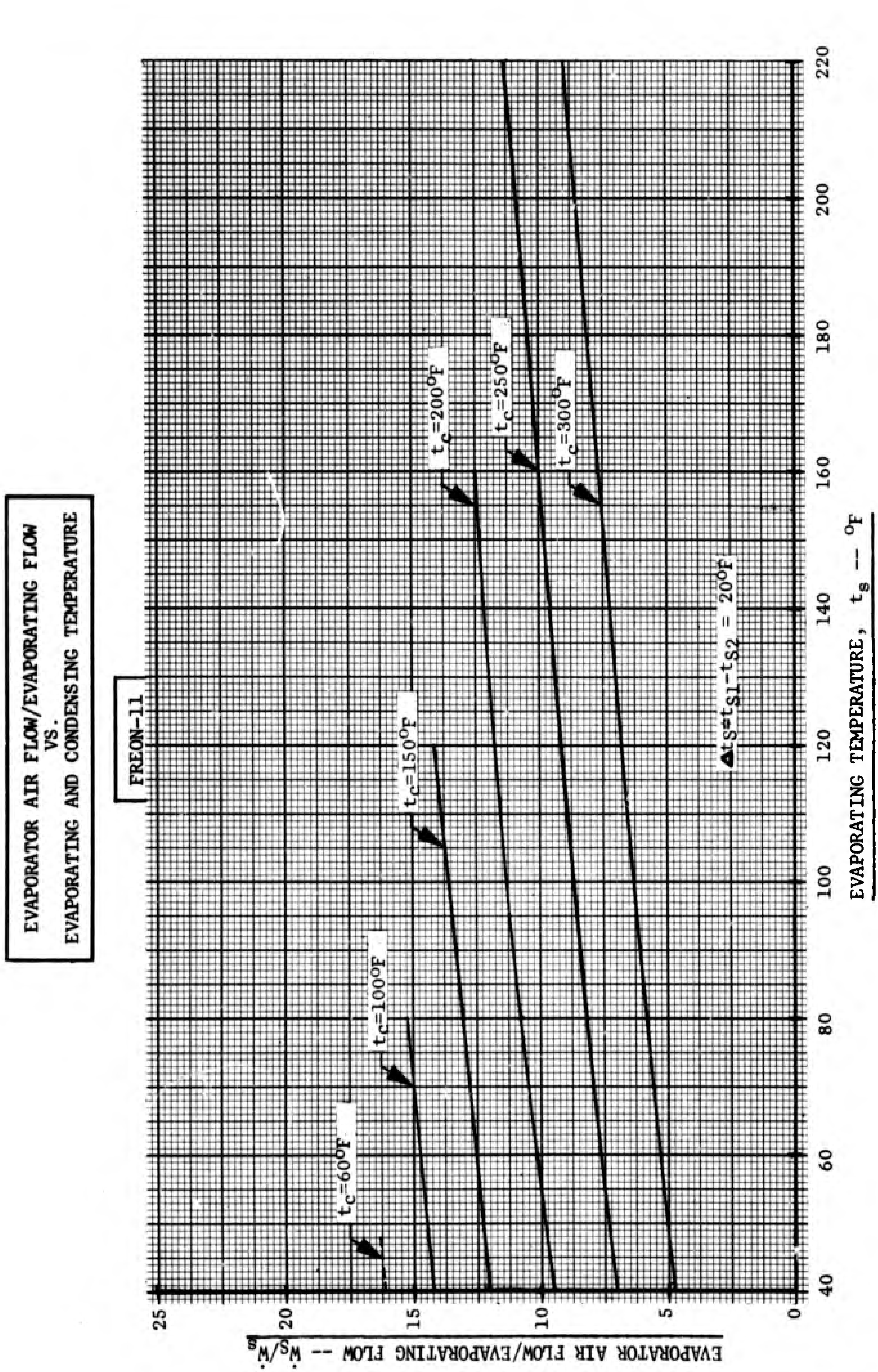
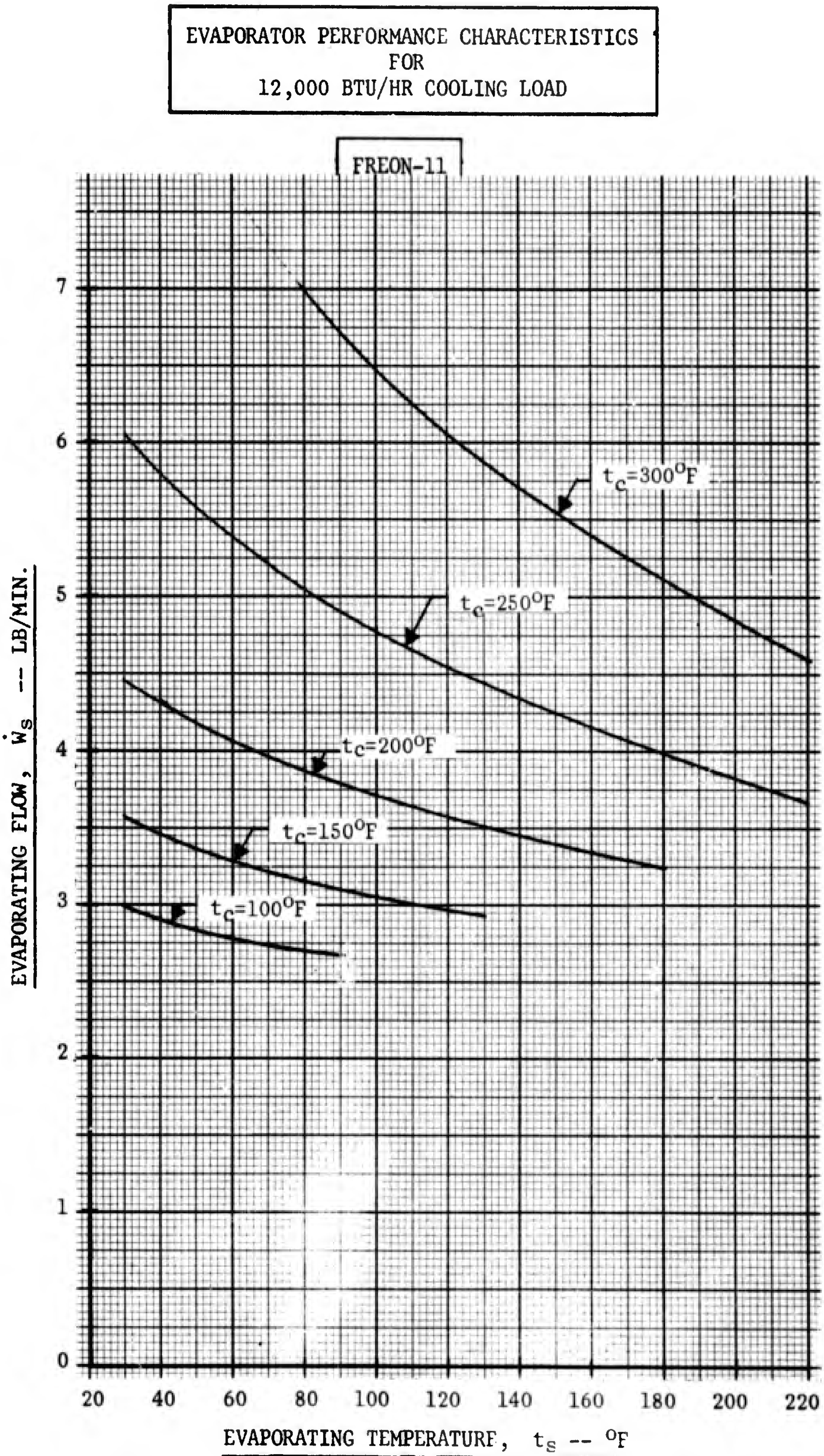

EVAPORATOR AIR FLOW/EVAPORATING FLOW
VS.
EVAPORATING AND CONDENSING TEMPERATURE
FREON-11
$t_c=60^oF$
$t_c=100^oF$
$t_c=150^oF$
$t_c=200^oF$
$t_c=250^oF$
$t_c=300^oF$
$\Delta t_S = t_{S1} - t_{S2} = 20^oF$
EVAPORATOR AIR FLOW/EVAPORATING FLOW -- $\dot{W}_S/\dot{W}_s$
EVAPORATING TEMPERATURE, $t_s$ -- $^oF$
0
5
10
15
20
25
40
60
80
100
120
140
160
180
200
220



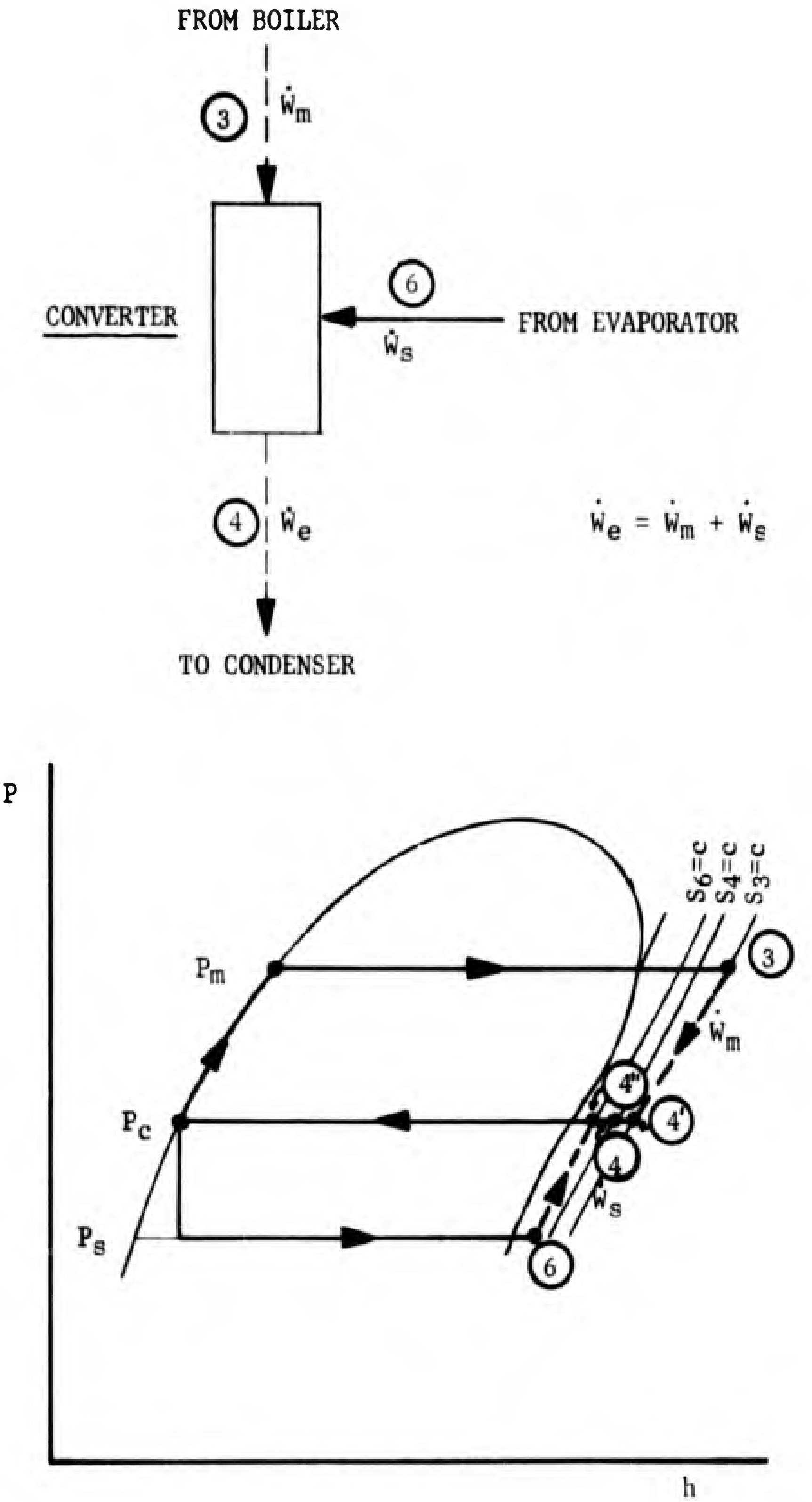
FIGURE 5.19



EVAPORATOR PERFORMANCE CHARACTERISTICS
FOR
12,000 BTU/HR COOLING LOAD
FREON-11
$t_c$=300°F
$t_c$=250°F
$t_c$=200°F
$t_c$=150°F
$t_c$=100°F
EVAPORATING FLOW, $\dot{W}_s$ -- LB/MIN.
0
1
2
3
4
5
6
7
20
40
60
80
100
120
140
160
180
200
220
EVAPORATING TEMPERATURE, $t_s$ -- °F




FIGURE 5.20




CONVERTER CHARACTERISTICS

## Energy Balance:

For ideal converter operation:

energy used for compression = energy required for compression

(1)............ $$\dot{W}_m\left[h(P_3,t_3)-h(P_4,s_3)\right] = \dot{W}_s\left[h(P_4,s_6)-h(P_6,t_6)\right]$$

Where: $\dot{W}_m$ = motive flow (lb/hr)

$\dot{W}_s$ = suction flow (lb/hr)

$h$ = enthalpy (BTU/lb)

$P$ = pressure (psia)

$t$ = temperature ($^{o}$F)

$s$ = entropy (BTU/lb-$^{o}$F)

$h(P_3,t_3)$= enthalpy at statepoint 3

$h(P_4,s_3)$= enthalpy at statepoint 4'

$h(P_4,s_6)$= enthalpy at statepoint 4"

$h(P_6,t_6)$= enthalpy at statepoint 6

Defining a converter entrainment ratio, $\lambda$, as the ratio of suction flow to motive flow, and a converter flow ratio, $X$, as the ratio of motive flow to suction flow:

$$\lambda \equiv \dot{W}_s/\dot{W}_m \quad \text{and} \quad X \equiv \dot{W}_m/\dot{W}_s$$

and using equation (1) results in:

(2)............ $$\lambda_i = \frac{[h(P_3,t_3)-h(P_4,s_3)]}{[h(P_4,s_6)-h(P_6,t_6)]}$$

and

(3)............ $$X_i = \frac{[h(P_4,s_6)-h(P_6,t_6)]}{[h(P_3,t_3)-h(P_4,s_3)]}$$

Where: $\lambda_i$ = ideal entrainment ratio

$X_i$ = ideal flow ratio

These ideal ratios can never be realized in practice because of losses due to fluid wall friction, heat transfer, turbulence, imperfect mixing, and existance of shocks. Of these deviations from the ideal performance, the mixing losses are generally considered to have the largest magnitude. Most converter analyses, therefore, are based on some analytical description of the mixing process involved.

The <u>converter efficiency</u>, $\eta_c$, can be defined as:

$$(4)\ldots\ldots\ldots\ldots \eta_c \equiv \frac{\lambda_a}{\lambda_i} = \frac{(\dot{W}_s/\dot{W}_m)_{actual}}{(\dot{W}_s/\dot{W}_m)_{ideal}}$$

or

$$(5)\ldots\ldots\ldots\ldots \eta_c \equiv \frac{X_i}{X_a} = \frac{(\dot{W}_m/\dot{W}_s)_{ideal}}{(\dot{W}_m/\dot{W}_s)_{actual}}$$

The converter efficiency is seen to be the ratio of ideal to actual motive flow for any given suction flow.

Ideal converter performance is presented in Figures 5.22 through 5.25. Figure 5.22 shows the ideal converter entrainment ratio as a function of evaporating temperature and condensing temperature for a fixed motive flow condition. Also shown are lines of constant compression ratio (condensing pressure/evaporating pressure). Increasing the evaporating temperature or lowering of the condensing temperature results in higher entrainment ratios. For a desired compression ratio, lower evaporating and lower condensing temperatures yield higher entrainment ratios.

Figure 5.23 is a plot of ideal converter entrainment ratio as a function of compression ratio and condensing temperature for the same motive flow conditions. Increasing compression ratios and condensing temperatures are seen to lower the converter entrainment ratio and, thus, the converter performance.

IDEAL CONVERTER ENTRAINMENT RATIO
VS.
EVAPORATING TEMPERATURE
AND
CONDENSING TEMPERATURE

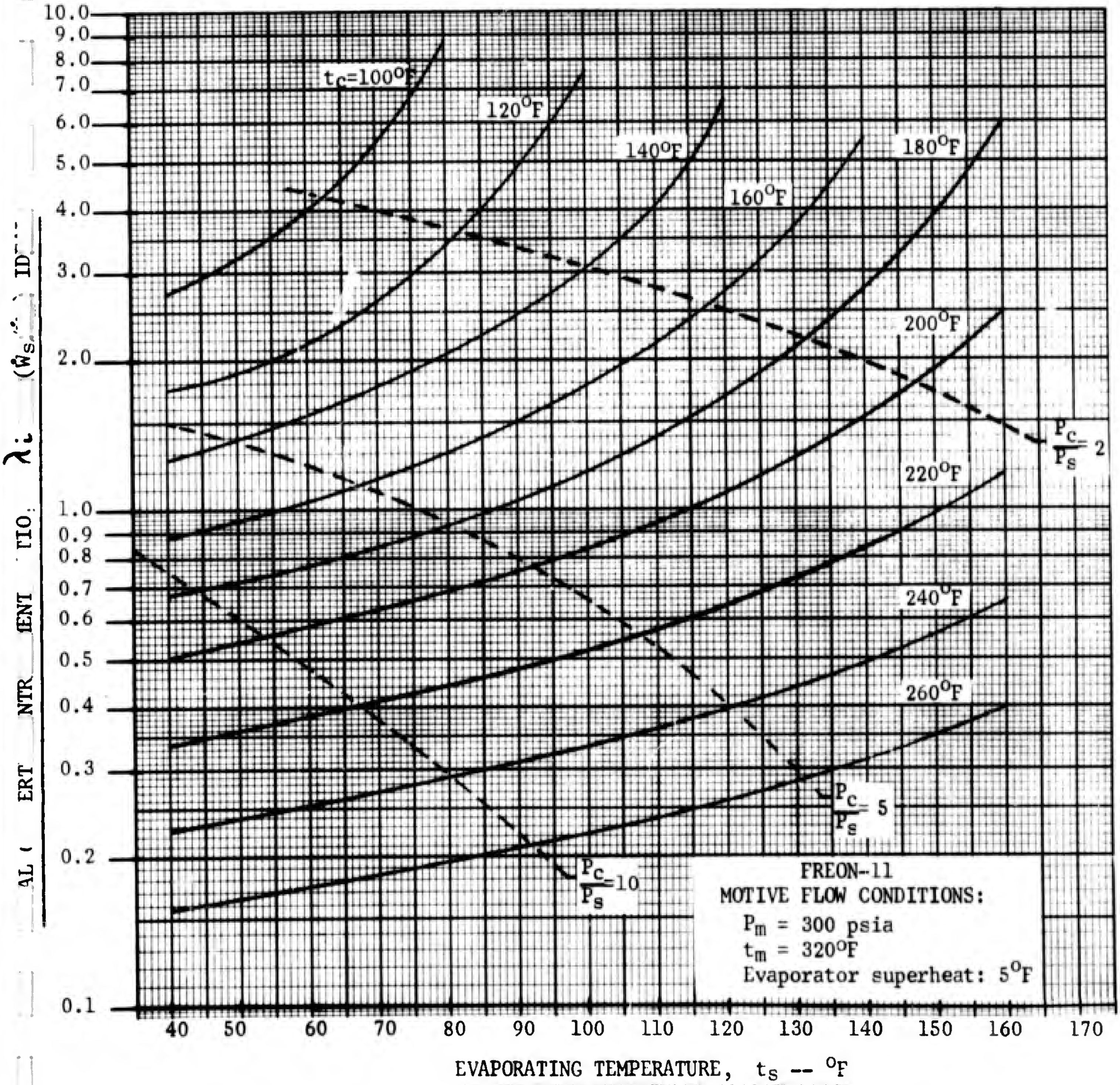





FIGURE 5.22

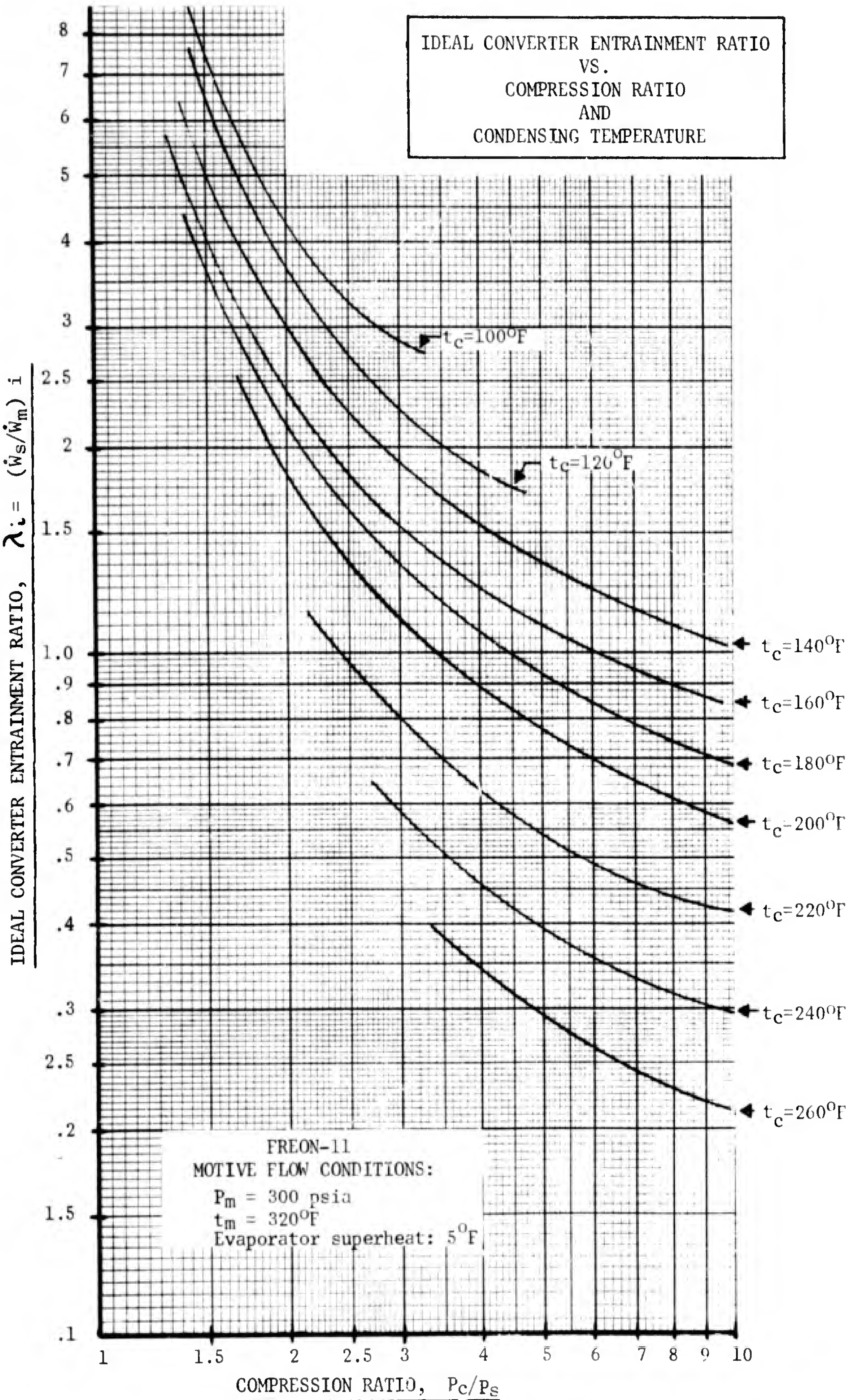

IDEAL CONVERTER ENTRAINMENT RATIO
VS.
COMPRESSION RATIO
AND
CONDENSING TEMPERATURE
IDEAL CONVERTER ENTRAINMENT RATIO, $\lambda_i = (\dot{W}_s/\dot{W}_m)_i$
COMPRESSION RATIO, $P_c/P_s$
$t_c$=100°F
$t_c$=120°F
$t_c$=140°F
$t_c$=160°F
$t_c$=180°F
$t_c$=200°F
$t_c$=220°F
$t_c$=240°F
$t_c$=260°F
FREON-11
MOTIVE FLOW CONDITIONS:
$P_m$ = 300 psia
$t_m$ = 320°F
Evaporator superheat: 5°F



FIGURE 5.23

Converter efficiency as a function of actual converter flow ratio and compression ratio is shown in Figure 5.24 for fixed motive and suction flow conditions. This figure shows that in order to achieve the same converter efficiency at higher compression ratios, higher converter flow ratios are required. It also implies that for a given evaporator load and desired compression ratio, higher efficiency converters require less motive flow.

The effect of motive pressure and superheat on ideal converter performance is indicated in Figure 5.25. The figure shows that the ideal converter entrainment ratio increases with increasing motive pressure. For Freon-11, the maximum entrainment ratio is obtained at a motive pressure of approximately 420 psia and a corresponding saturation temperature of 340°F. At any given motive pressure, higher superheats will result in increased ideal entrainment ratios. This indicates that superheating of the motive flow is undesirable, since it tends to lower the converter efficiency. It can be also shown that the converter performance is directly related to the change in specific volume of the working fluid between the motive and discharge flows, with larger specific volume changes corresponding to higher entrainment ratios. For any working fluid with a fixed condensing temperature and pressure, higher motive pressures will result in higher entrainment ratios, because specific volume decreases with pressure, thus increasing the difference in specific volumes between the converter discharge and inlet. The effect of superheat is seen to decrease converter performance due to the increase in specific volume with temperature (at constant pressure) with the associated decrease in the difference in specific volumes between converter discharge and inlet. This functional relationship between converter performance and the specific volume change is very useful for direct comparisons and selections of working fluids based on converter performance alone.

The performance characteristics of the ideal converter are compared with those of an ideal ejector in Figure 5.26. The ideal ejector performance is based on the Stodola-Hook ejector analysis method (constant pressure mixing) as presented in Appendix B. Figure 5.26 shows that the ejector requires approximately three times the motive flow that an ideal converter would require for the same suction flow.

Figure 5.27 shows the ideal ejector efficiency as a function of compression ratio for a motive pressure and temperature of 300 psia and 320°F, respectively; and a suction pressure and temperature of 7.0 psia and 45°F, respectively. The ideal ejector efficiency was defined as the ratio of ideal converter flow ratio to ideal ejector flow ratio. It can be seen from Figure 5.27 that the ideal ejector efficiency increases with compression ratio, and that it varies between 26% and 36% for compression ratios between 2 and 10. Although the ideal ejector efficiency increases with compression ratio, actual ejector performance decreases with increasing compression ratios because momentum losses in the ejector increase rapidly with compression ratio.

CONVERTER EFFICIENCY
VS.
ACTUAL CONVERTER FLOW RATIO
AND
COMPRESSION RATIO

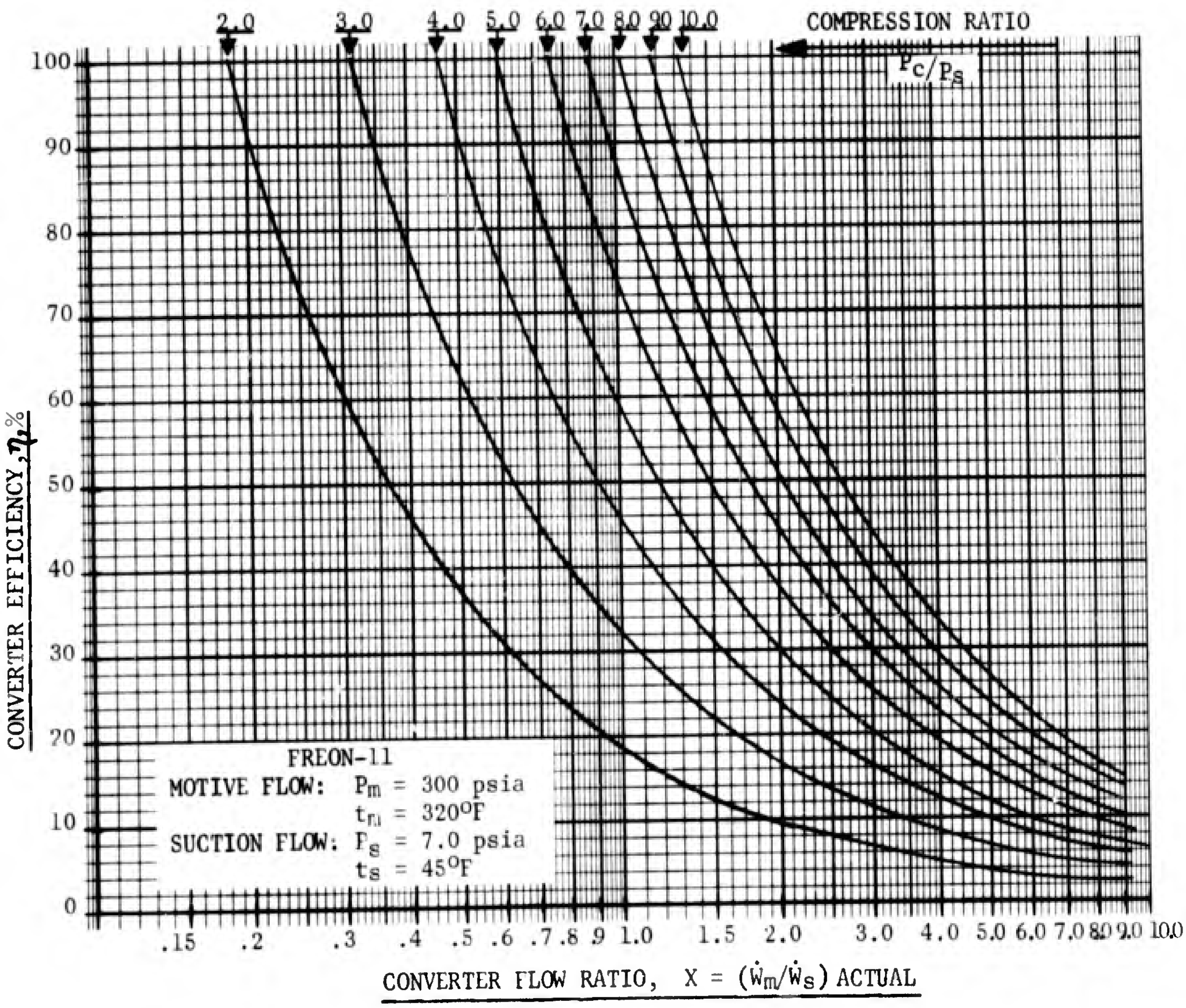





FIGURE 5.24

EFFECT OF MOTIVE PRESSURE AND SUPERHEAT ON IDEAL CONVERTER PERFORMANCE

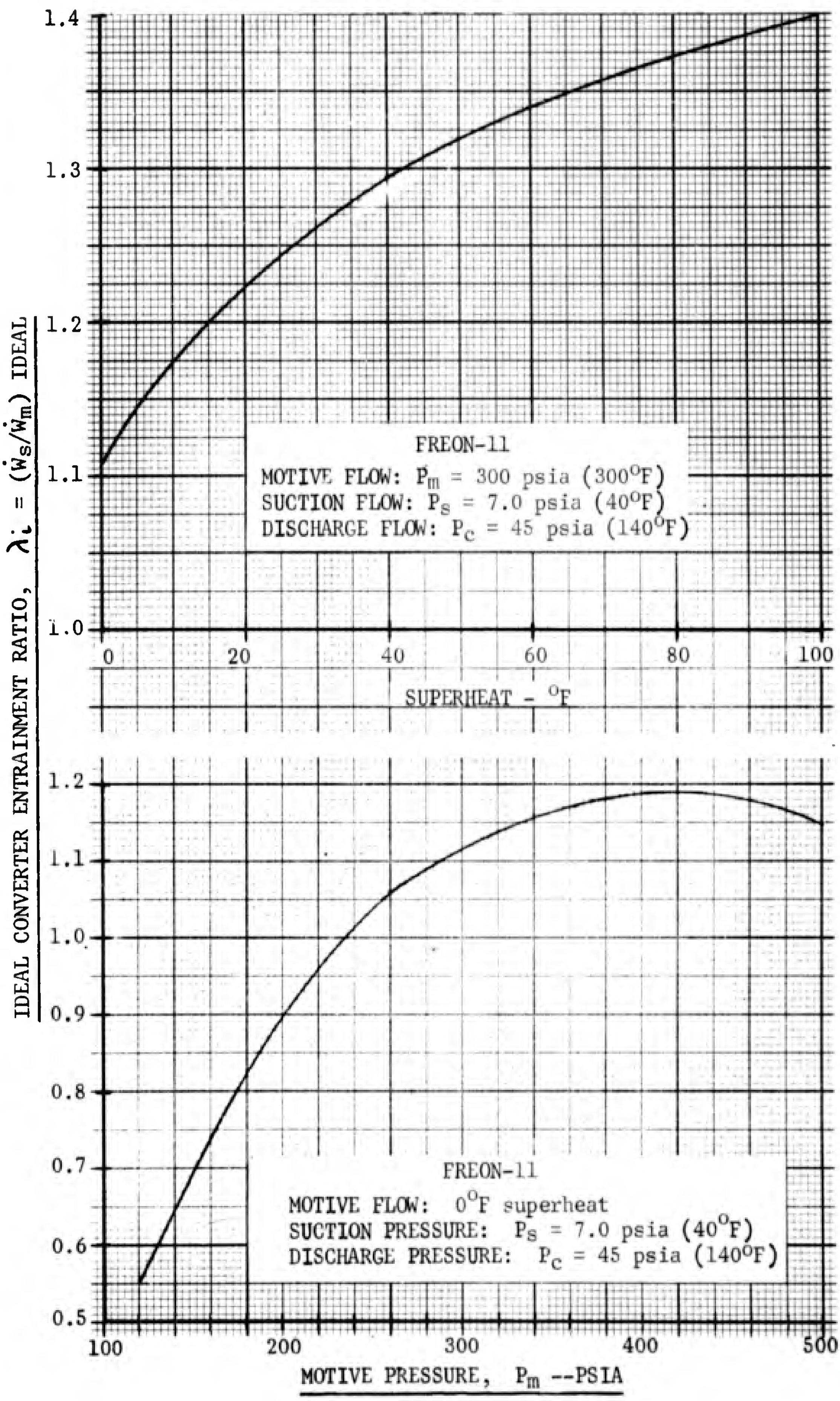





FIGURE 5.25

PERFORMANCE CHARACTERISTIC OF IDEAL CONVERTER AND IDEAL EJECTOR

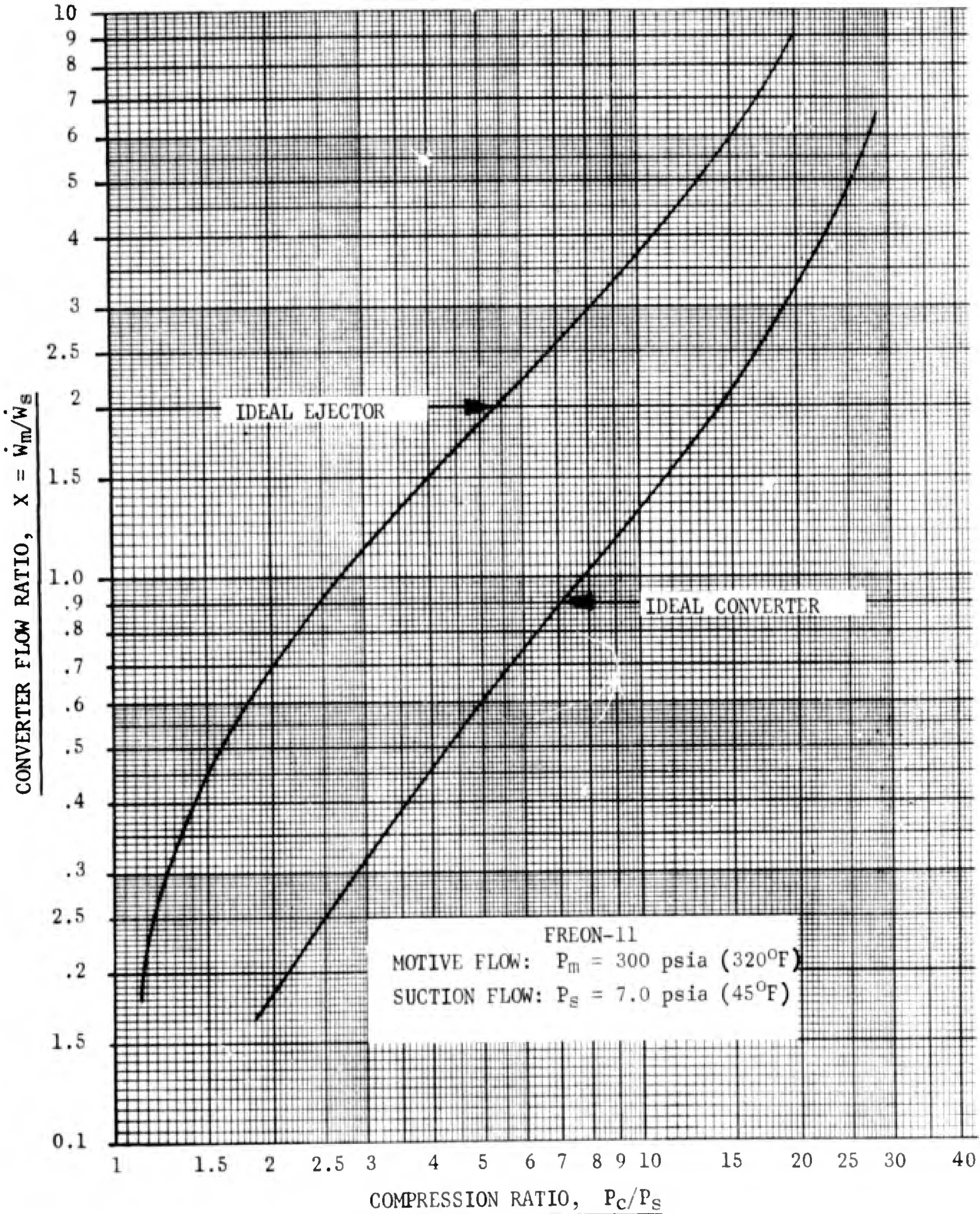





FIGURE 5.26

IDEAL EJECTOR EFFICIENCY
VS.
COMPRESSION RATIO

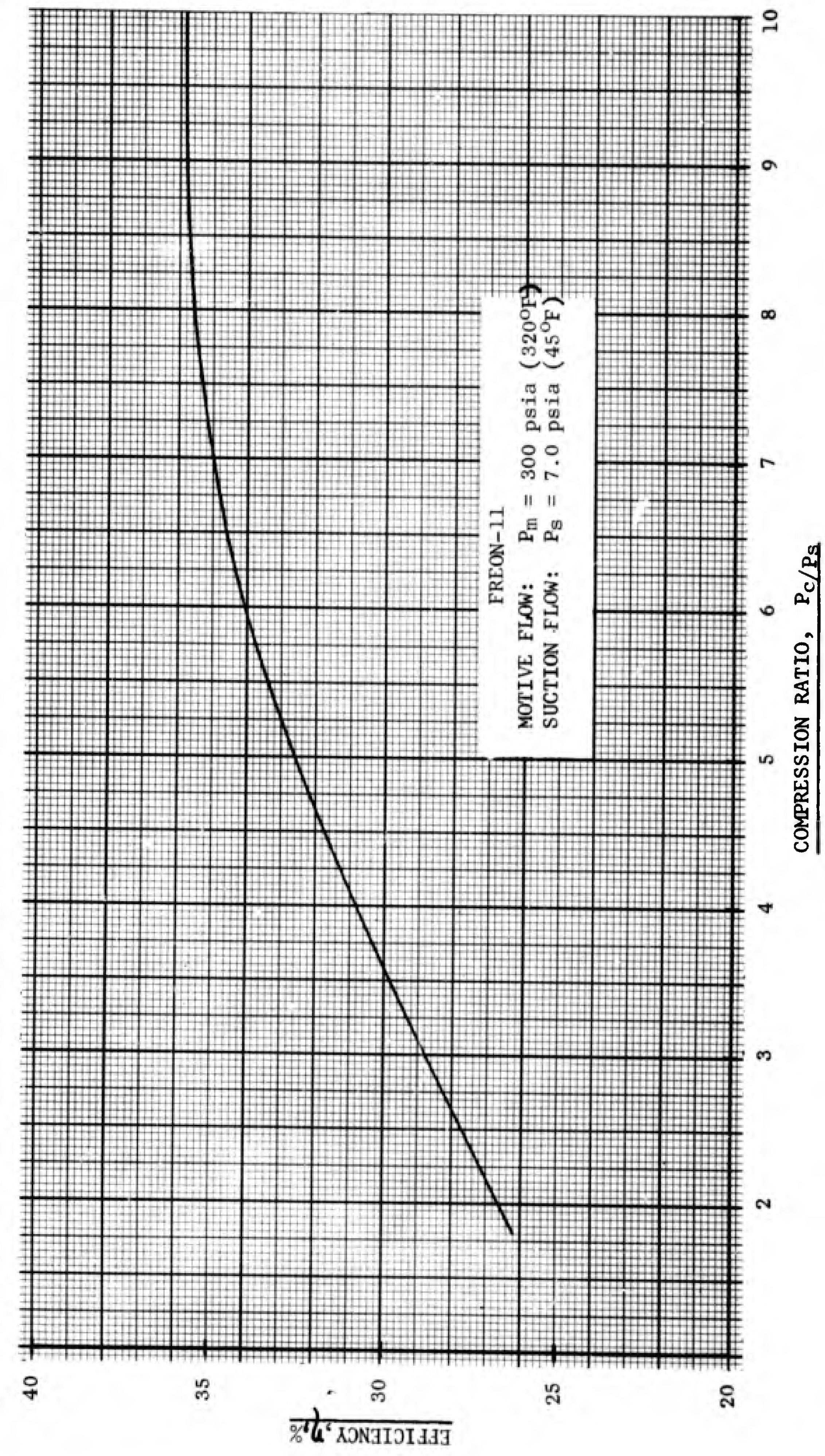





FIGURE 5.27

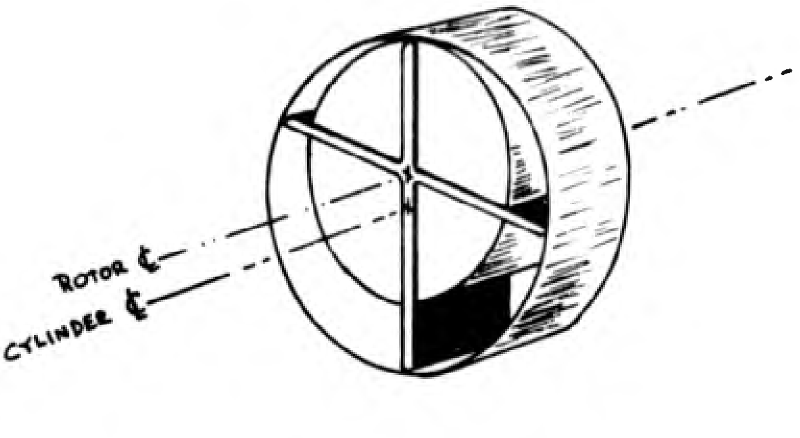
The state-of-the-art in ejector design is characterized by recent developments at Conductron where single-stage ejector efficiencies of 28.5% have been attained. With continued development, ejector efficiencies around 35% are anticipated. Advanced work in new converter devices has indicated a sound basis for anticipating significant performance improvements compared to ejectors. Conservative analyses indicate performance up to 60% efficiency.

## 5.6 Air (Vane) Motor

The air or vane motor serves a dual purpose in the Conductron environmental control system mechanizations considered in this analysis: (a) it provides the power to drive condenser cooling fans and evaporator fans; and (b) it is used as an expander for compressor bleed air, providing the desired cooling air temperatures, and ram air, yielding a usable condenser heat sink at high ram air temperatures.

Both of these functions can be accomplished by various expansion devices; but since it is highly desirable to eliminate any relatively unreliable high-speed machines, low speed rotary-vane expanders were chosen for this application.

Any rotary-vane machine operates by rolling a cylindrical rotor around inside a cylinder. The rotor can be eccentric or concentric with the shaft, but the cylinder is usually mounted eccentric to the shaft center. Blades, inserted into the rotor and contacting the cylinder wall (or vice versa), trap volumes of air which shrink or expand as the rotor rolls. Figure 5.28 is a sketch of a typical rotary-vane expander.




SKETCH OF A TYPICAL ROTARY-VANE MOTOR FIGURE 5.28

Figure 5.29 shows the parameters of interest in the evaluation of the performance characteristics of air (vane)motors. The basic thermodynamic process of any expander is indicated on an enthalpy-entropy diagram showing an ideal and an actual expansion process.

Assuming isentropic (ideal) expansion,

$$(1)\ldots\ldots\ldots\ldots \quad \frac{T_1}{T_{2i}} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$$

or

$$(2)\ldots\ldots\ldots\ldots \quad t_1 - t_{2i} = t_1\left[1 - \frac{1}{\left(P_1/P_2\right)^{\frac{\gamma-1}{\gamma}}}\right]$$

where: $T$ = temperature ($^{o}R$)

$t$ = temperature ($^{o}F$)

$P$ = pressure (psia)

$\gamma$ = $c_p/c_v$ = 1.395 for air

$c_p$ = specific heat at constant pressure (BTU/lb-$^{o}F$)

$c_v$ = specific heat at constant volume (BTU/lb-$^{o}F$)

subscripts refer to Figure 5.29

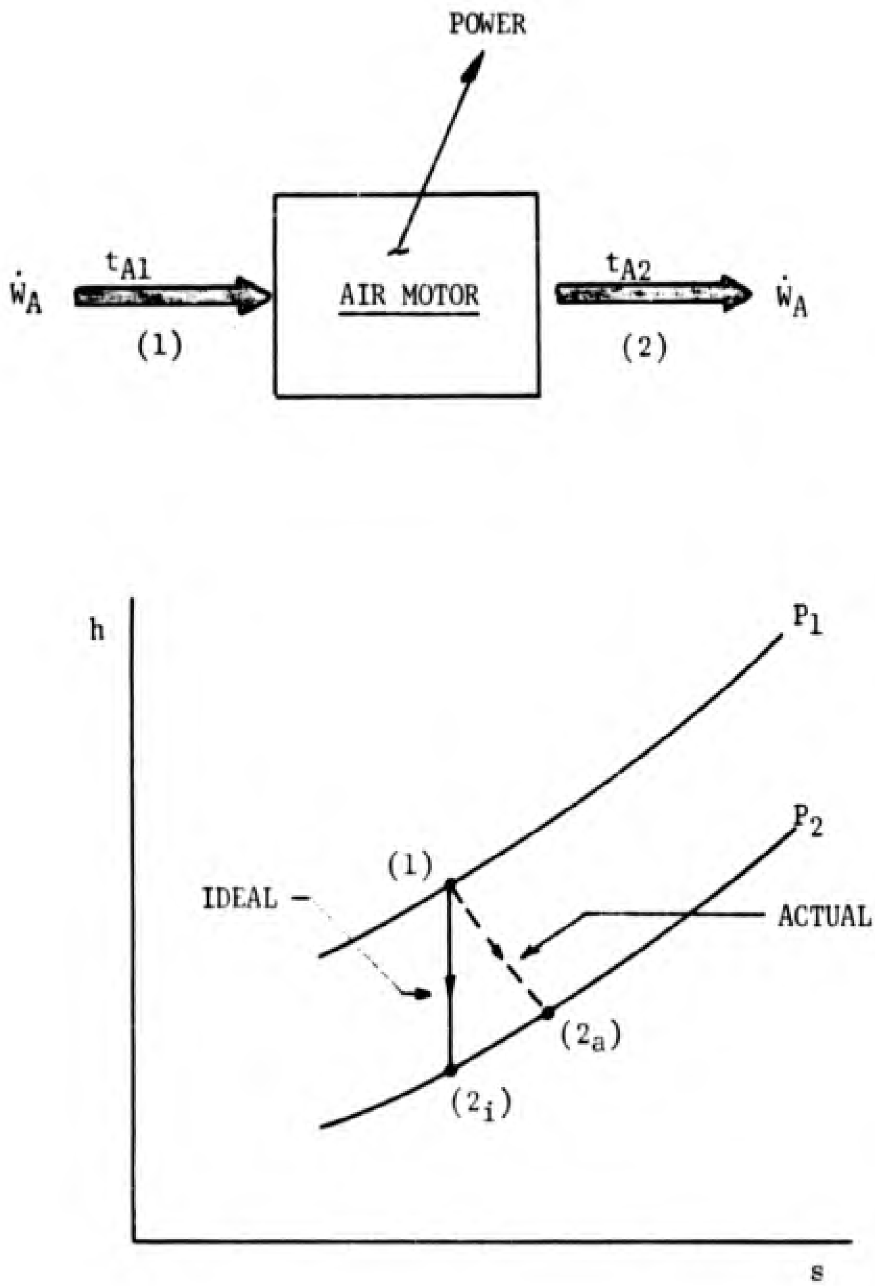
(NOTE: $t_1 = t_{A1}$ and $t_2 = t_{A2}$ )

Defining adiabatic efficiency as,

$$(3)\ldots\ldots\ldots\ldots \quad \eta_a \equiv \frac{W_a}{W_i}$$

where: $W_a$ = work of actual adiabatic expansion

$W_i$ = work of reversible adiabatic expansion between the same initial state and the same final total pressure




AIR (VANE) MOTOR CHARACTERISTICS

and applying the first law of thermodynamics:

$$(4)\ldots\ldots\ldots\ldots \quad \eta_a = \frac{h_1 - h_2 + \dfrac{V_1^2 - V_2^2}{2gJ}}{h_1 - h_{2i} + \dfrac{V_1^2 - V_{2i}^2}{2gJ}}$$

where: $h$ = enthalpy (BTU/lb)

$V$ = velocity (ft/sec)

$g$ = gravitational acceleration (ft/sec$^2$)

$J$ = 778 ft-lb/BTU

Assuming no change in kinetic energy,

$$(5)\ldots\ldots\ldots\ldots \quad \eta_a = \frac{h_1 - h_2}{h_1 - h_{2i}}$$

and ideal gas behavior,

$$(6)\ldots\ldots\ldots\ldots \quad \eta_a = \frac{t_1 - t_2}{t_1 - t_{2i}}$$

or

$$(7)\ldots\ldots\ldots\ldots \quad t_1 - t_2 = \eta_a (t_1 - t_{2i})$$

and substitution in Equation (2) results in:

$$(8)\ldots\ldots\ldots\ldots \quad t_1 - t_2 = \eta_a t_1 \left[ 1 - \frac{1}{(P_1/P_2)^{\frac{k-1}{k}}} \right]$$

The power produced by the expansion process can be shown to be:

(9)............ $$\mathbb{P} = \frac{\gamma R}{(\gamma-1)J}\dot{W}_A\,(t_{A1} - t_{A2})$$

where: $\mathbb{P}$ = power (BTU/hr)

$\dot{W}_A$ = air flow (lb/hr)

$R$ = gas constant ($ft\text{-}lb/lb\text{-}^{o}R$)

for air: $\gamma$ = 1.395 and $R$ = 53.35 ($ft\text{-}lb/lb_m\text{-}^{o}R$)

Therefore:

(10)........... $$\mathbb{P} = 0.242\,\dot{W}_A\,(t_{A1} - t_{A2}) \quad , \text{BTU/hr}$$

The performance capability of expanders is shown graphically in Figures 5.30 through 5.37. The outlet temperature is plotted as a function of the pressure ratio across the expander and the air inlet temperature for various adiabatic efficiencies. For any given air inlet temperature, higher pressure ratios and higher efficiencies produce lower outlet temperatures.

The power produced by the air expander is shown in Figures 5.38 through 5.40. In Figure 5.38, the air motor power is plotted as a function of temperature drop and mass flow rate. The air motor power increases with increasing temperature drops and higher flow rates. Figure 5.39 shows the air motor power in terms of volume flow rate and inlet pressure for an inlet temperature of $50^{o}F$ and an adiabatic efficiency of 0.30. As expected, power increases with flow rate and inlet pressure. Figure 5.40 illustrates the performance characteristics of a typical commercial air motor. The power output is shown as a function of volume flow rate and inlet pressure.

The state-of-the-art in rotary-vane motor design is characterized by Figure 5.40. Present vane motors are designed for industrial use where intermittent duty and reversible rotation is desirable. Very little or no development effort has been devoted to rotary-vane motors in general and to their potential as an expansion device in particular.

## OUTLET TEMPERATURE VS. PRESSURE RATIO AND INLET TEMPERATURE

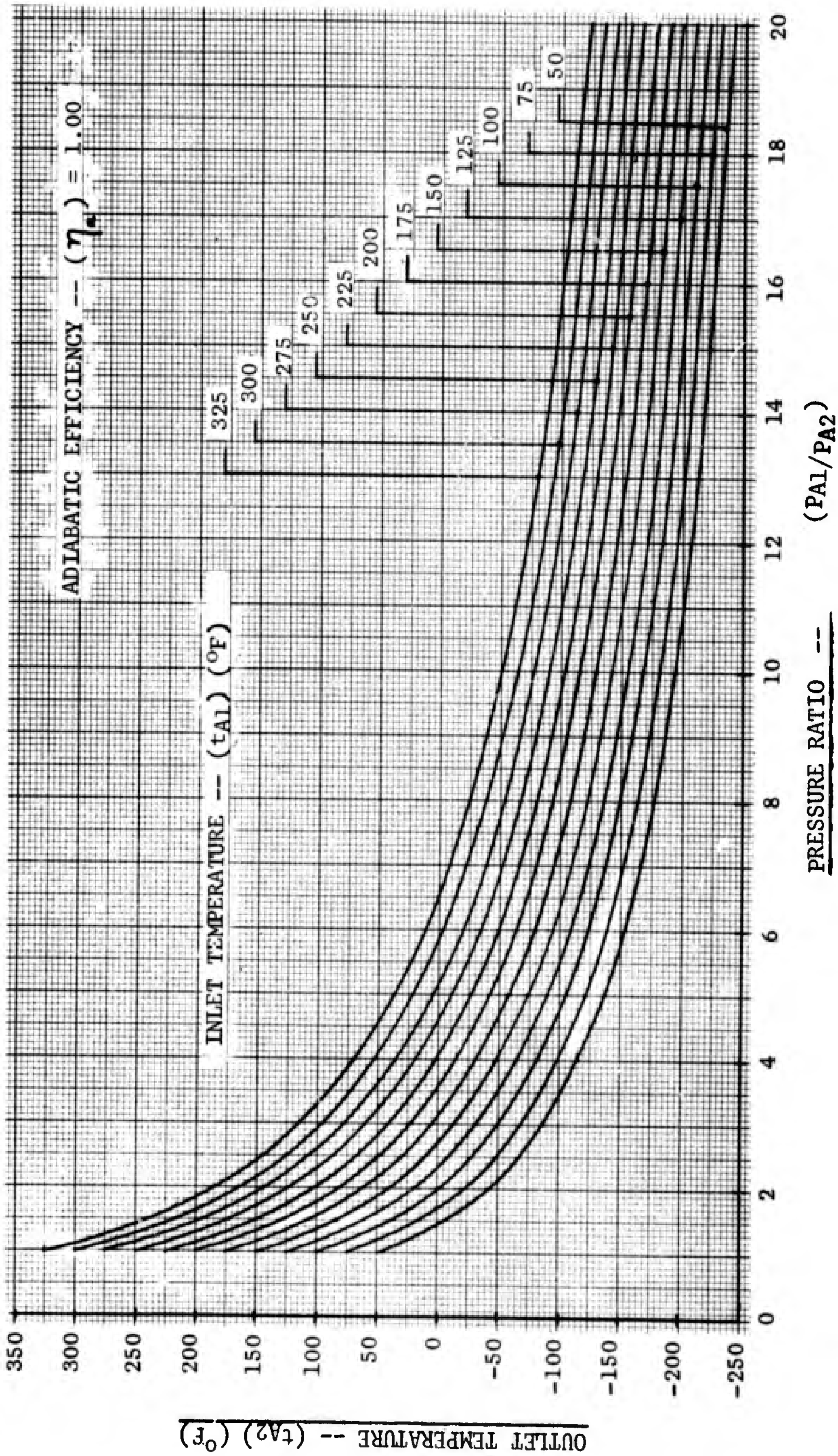





FIGURE 5.30

OUTLET TEMPERATURE
VS.
PRESSURE RATIO AND INLET TEMPERATURE

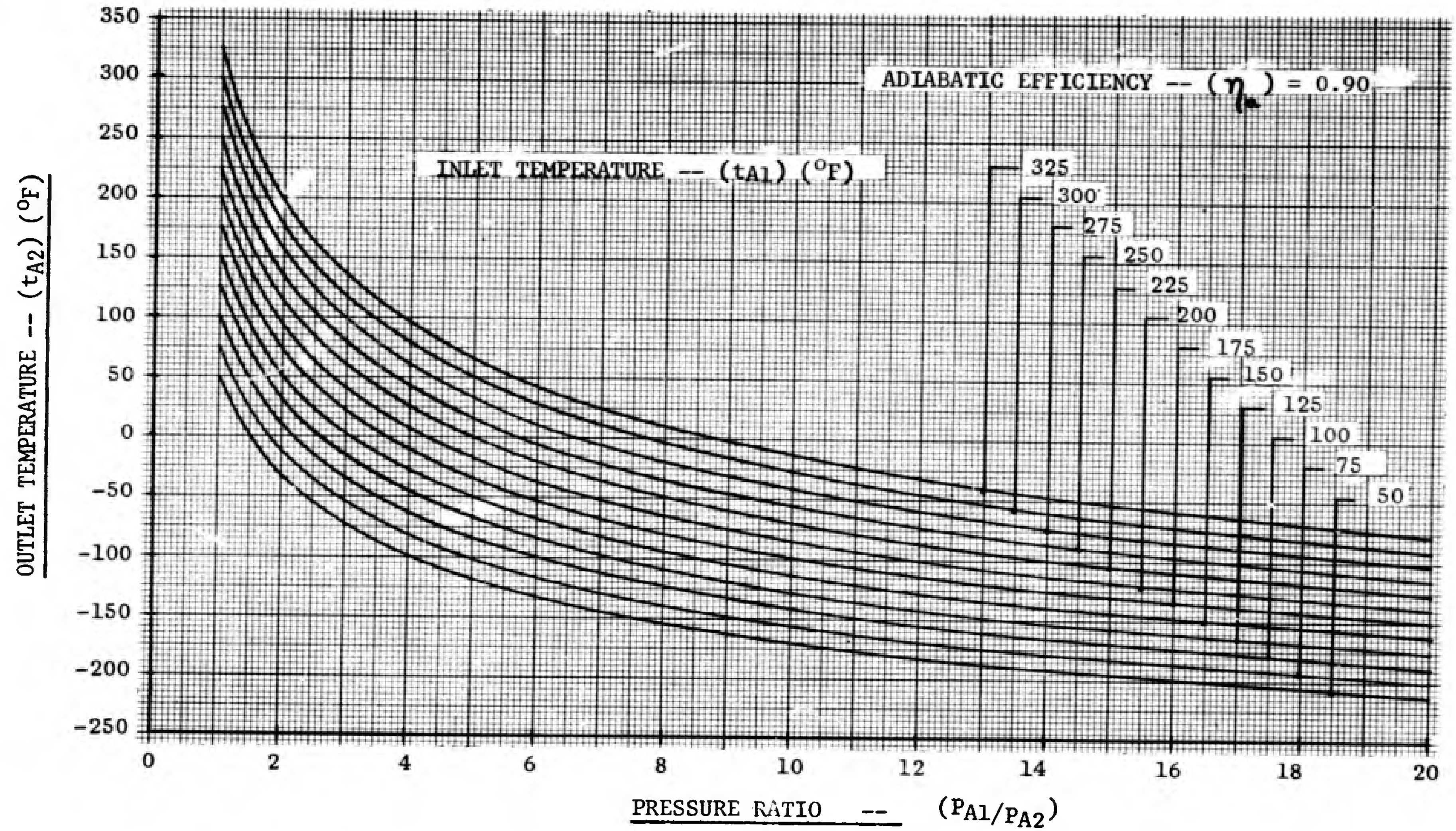





FIGURE 5.31

OUTLET TEMPERATURE
VS.
PRESSURE RATIO AND INLET TEMPERATURE

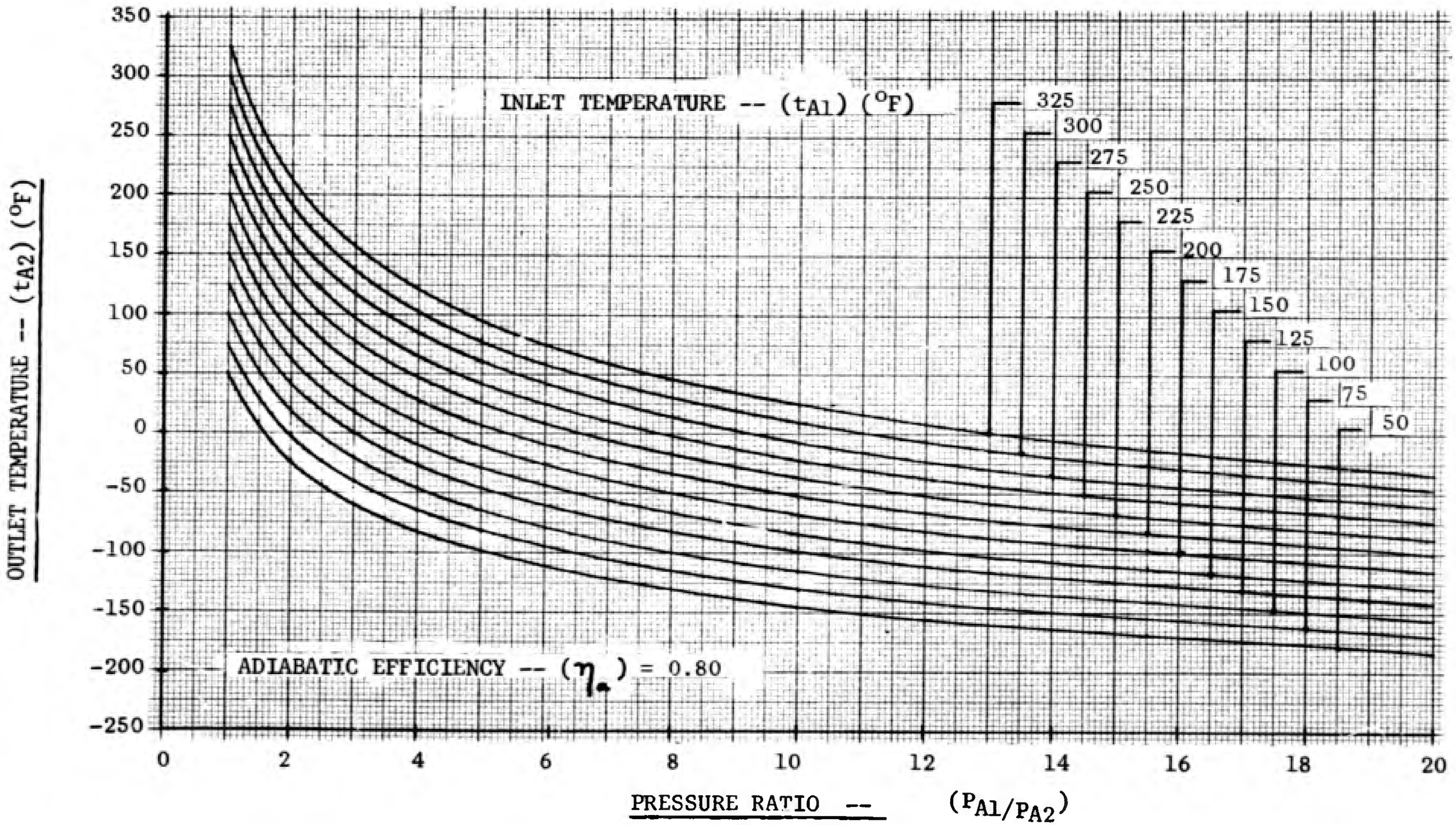





FIGURE 5.32

OUTLET TEMPERATURE
VS.
PRESSURE RATIO AND INLET TEMPERATURE

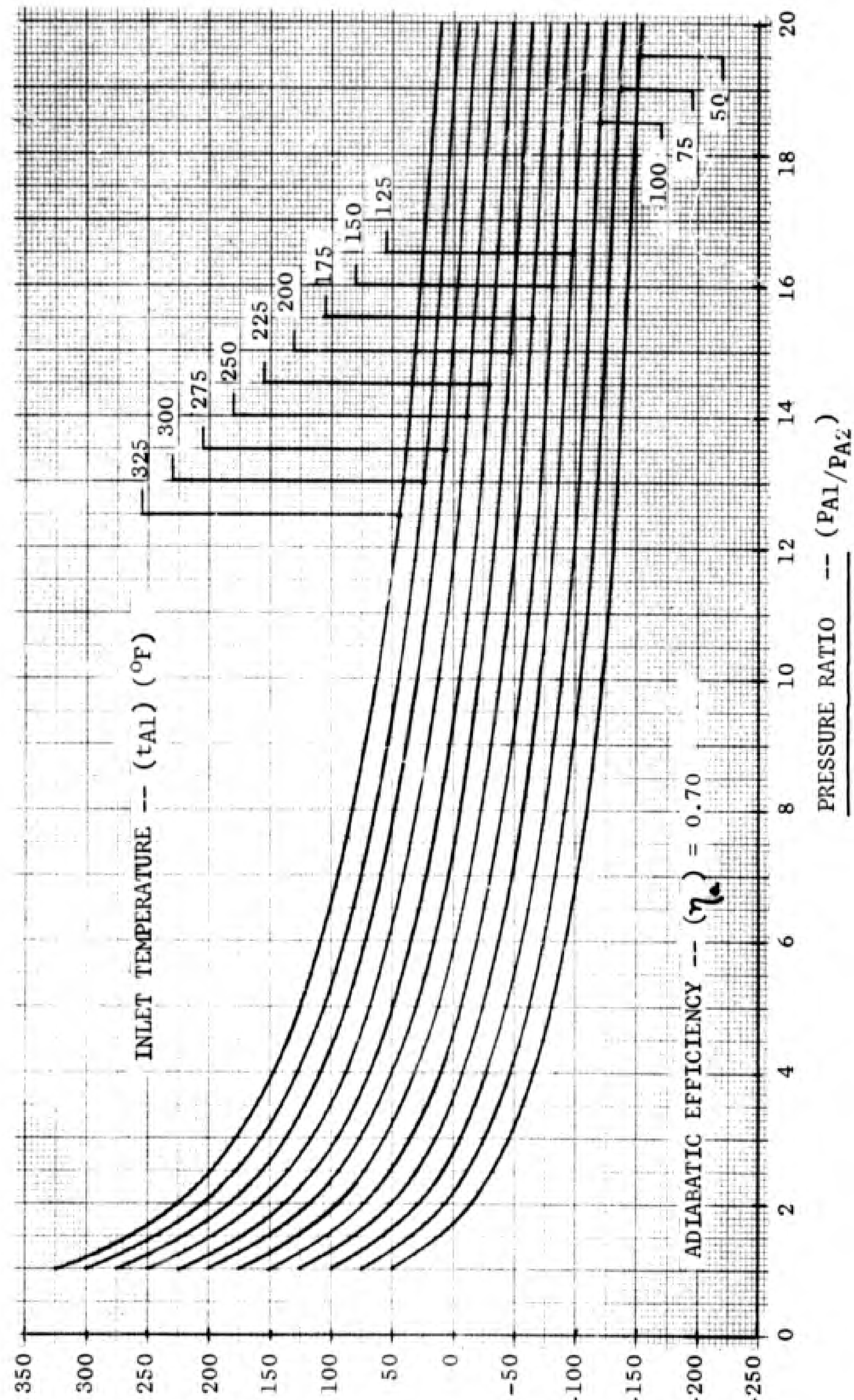



FIGURE 5.33

OUTLET TEMPERATURE
VS.
PRESSURE RATIO AND INLET TEMPERATURE

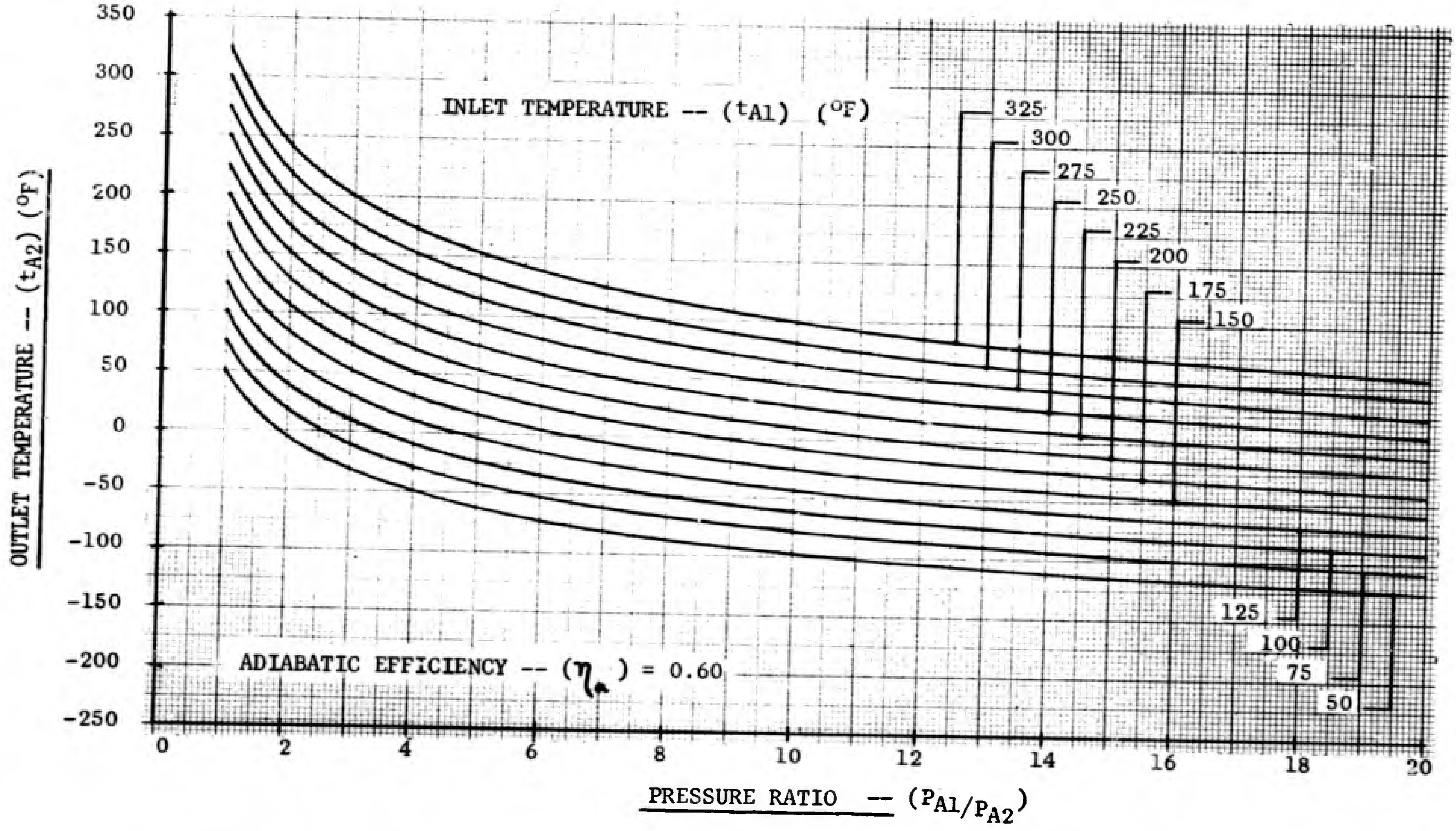





FIGURE 5.34

**OUTLET TEMPERATURE VS. PRESSURE RATIO AND INLET TEMPERATURE**

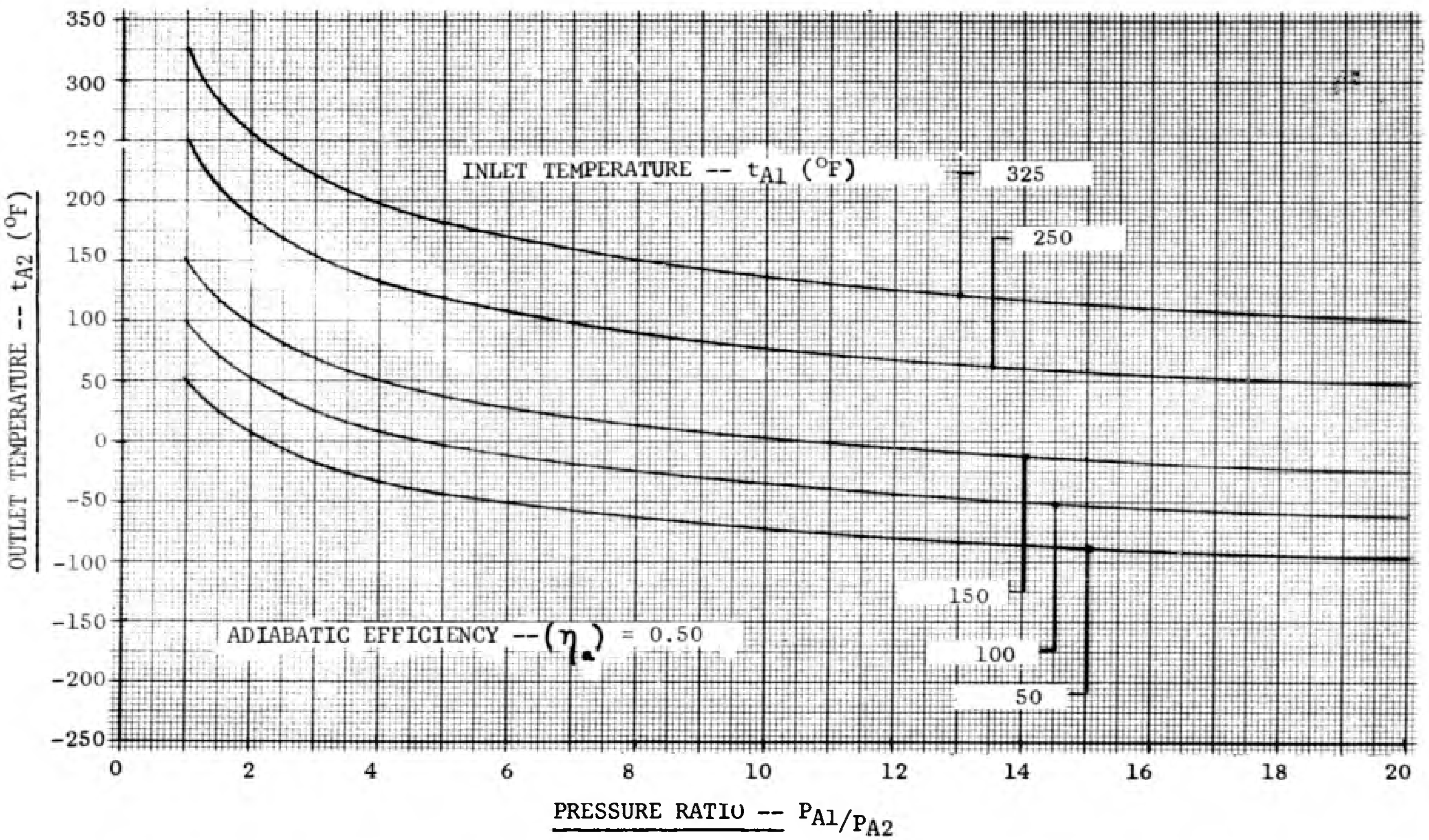
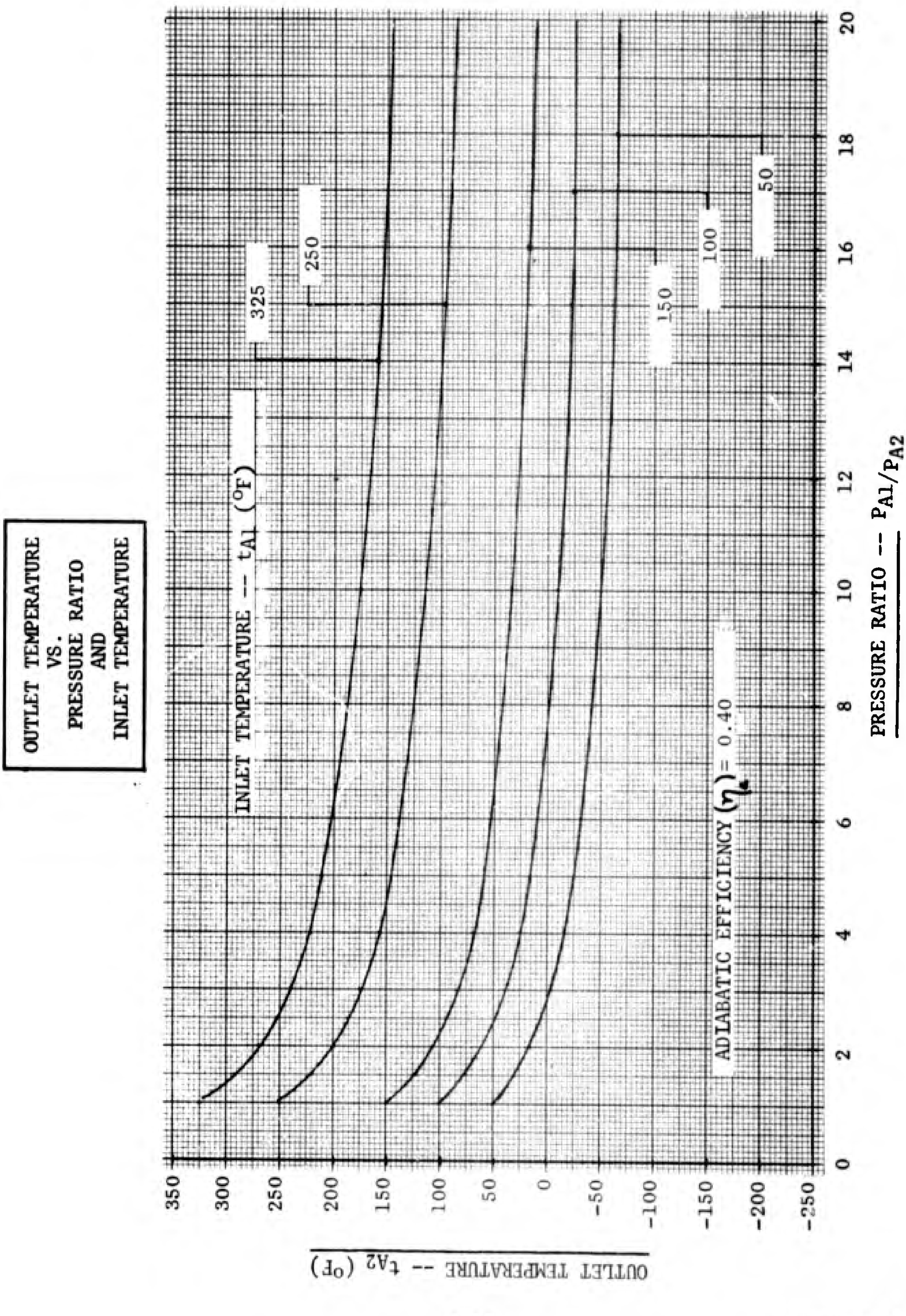



FIGURE 5.35



OUTLET TEMPERATURE
VS.
PRESSURE RATIO
AND
INLET TEMPERATURE
INLET TEMPERATURE -- tA1 (°F)
325
250
150
100
50
ADIABATIC EFFICIENCY (η) = 0.40
PRESSURE RATIO -- PA1/PA2
0
2
4
6
8
10
12
14
16
18
20
OUTLET TEMPERATURE -- tA2 (°F)
350
300
250
200
150
100
50
0
-50
-100
-150
-200
-250




FIGURE 5.36

**OUTLET TEMPERATURE**
**VS.**
**PRESSURE RATIO**
**AND**
**INLET TEMPERATURE**

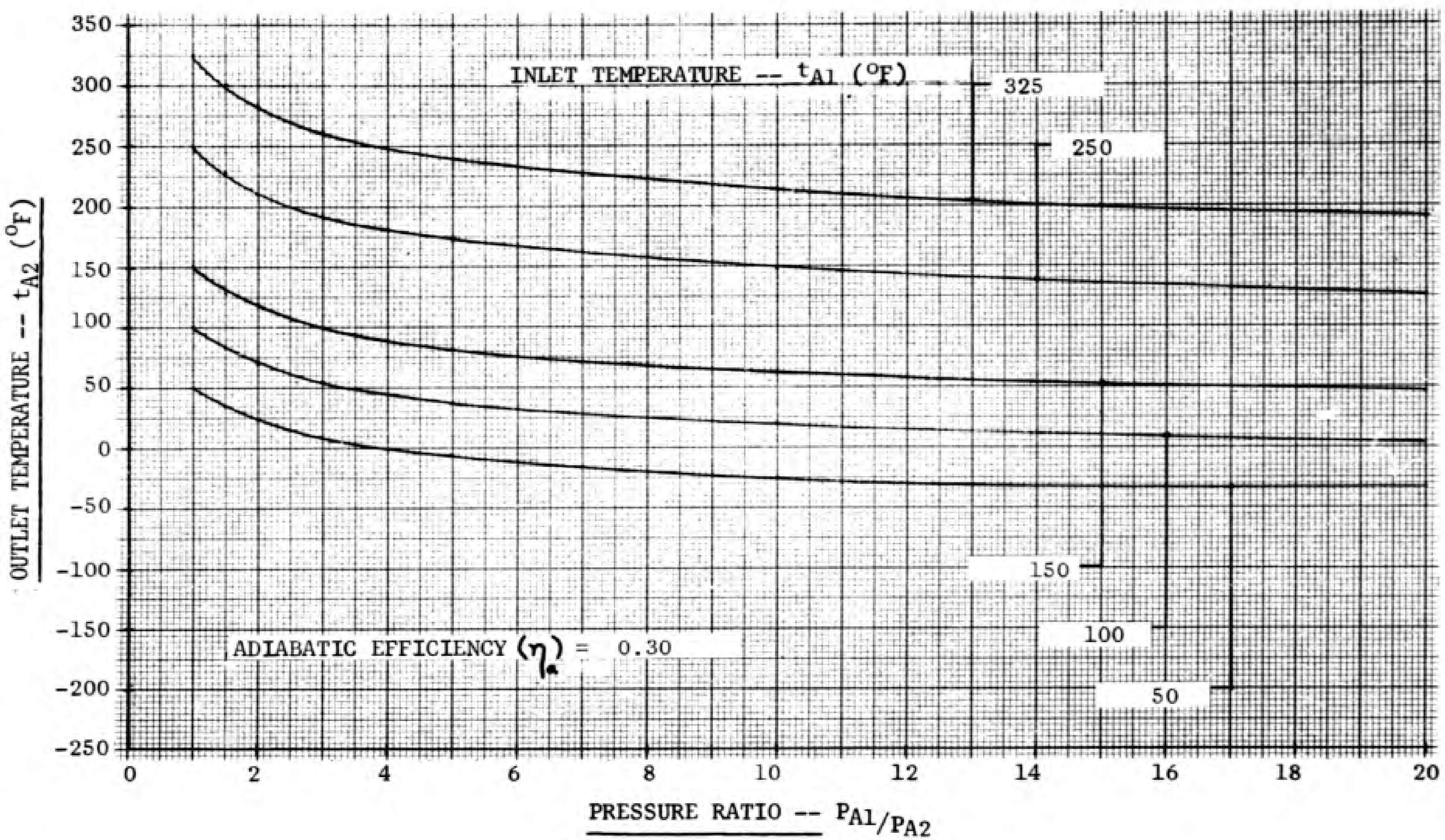





FIGURE 5.37

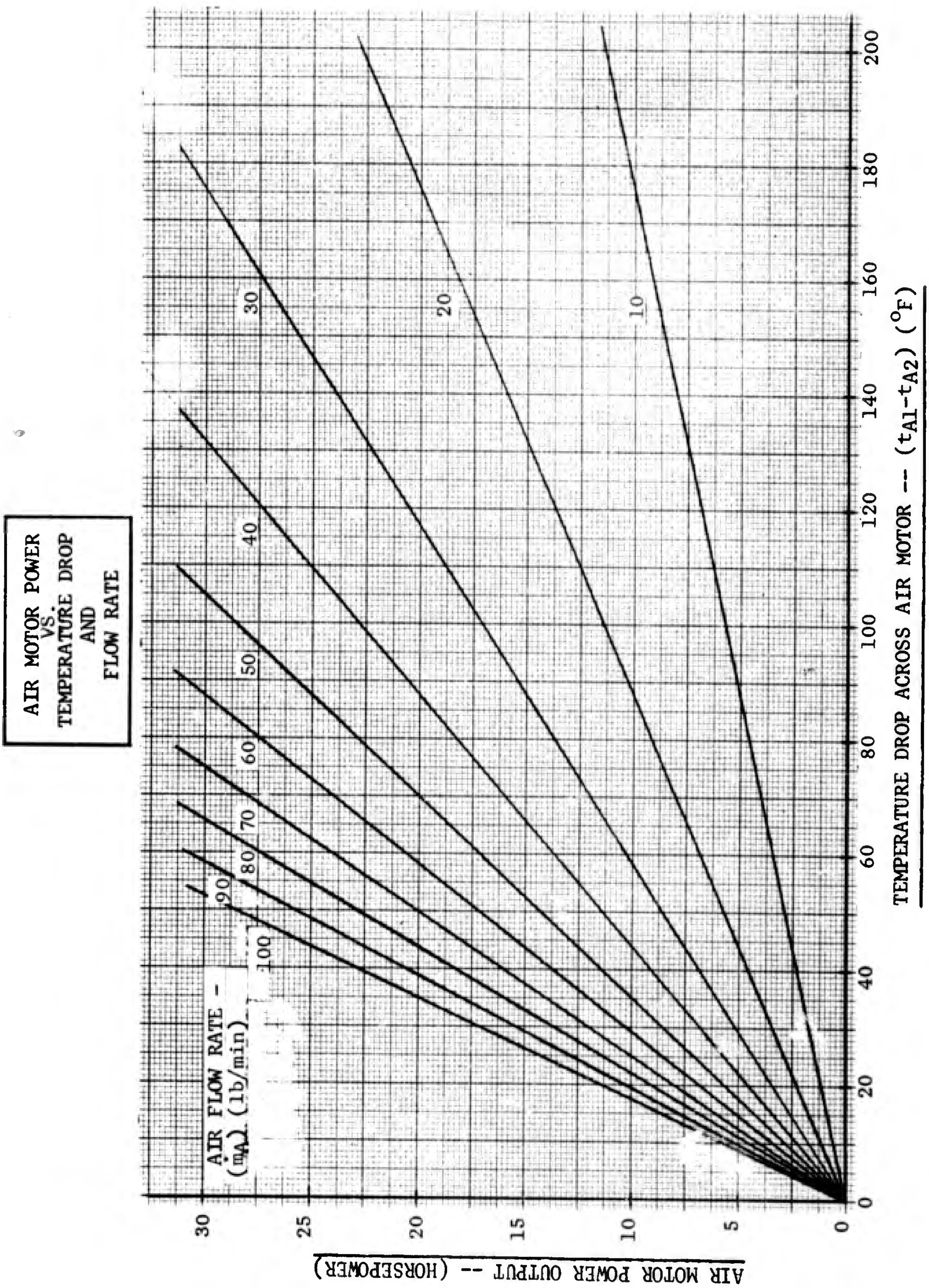

AIR MOTOR POWER
VS.
TEMPERATURE DROP
AND
FLOW RATE
AIR FLOW RATE -
($\dot{m}_A$) (lb/min)
10
20
30
40
50
60
70
80
90
100
TEMPERATURE DROP ACROSS AIR MOTOR -- ($t_{A1}-t_{A2}$) ($^oF$)
0
20
40
60
80
100
120
140
160
180
200
AIR MOTOR POWER OUTPUT -- (HORSEPOWER)
0
5
10
15
20
25
30



FIGURE 5.38

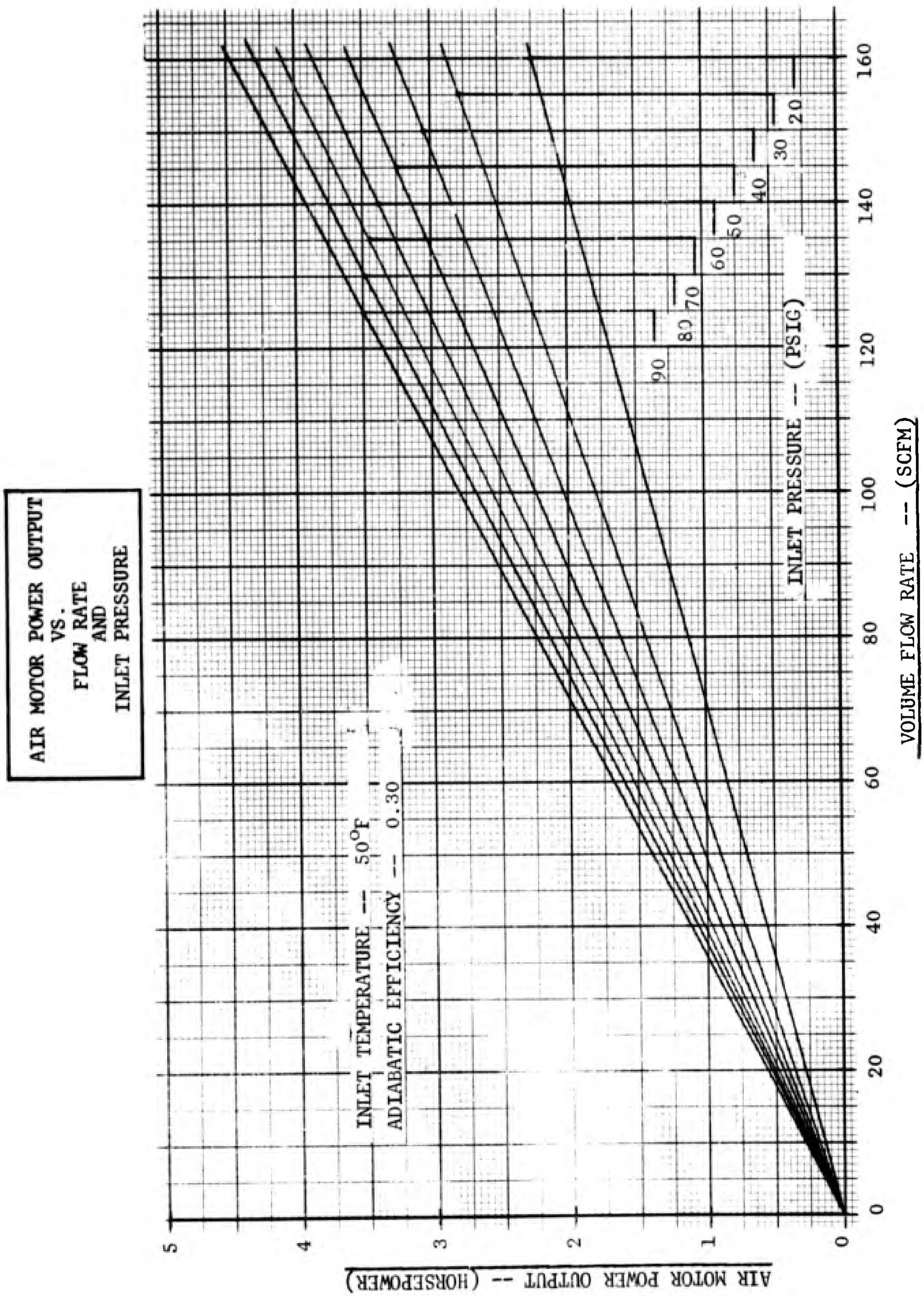

AIR MOTOR POWER OUTPUT
VS.
FLOW RATE
AND
INLET PRESSURE
INLET TEMPERATURE -- 50°F
ADIABATIC EFFICIENCY -- 0.30
INLET PRESSURE -- (PSIG)
90
80
70
60
50
40
30
20
VOLUME FLOW RATE -- (SCFM)
0
20
40
60
80
100
120
140
160
AIR MOTOR POWER OUTPUT -- (HORSEPOWER)
0
1
2
3
4
5



FIGURE 5.39

TYPICAL AIR MOTOR
PERFORMANCE CHARACTERISTICS

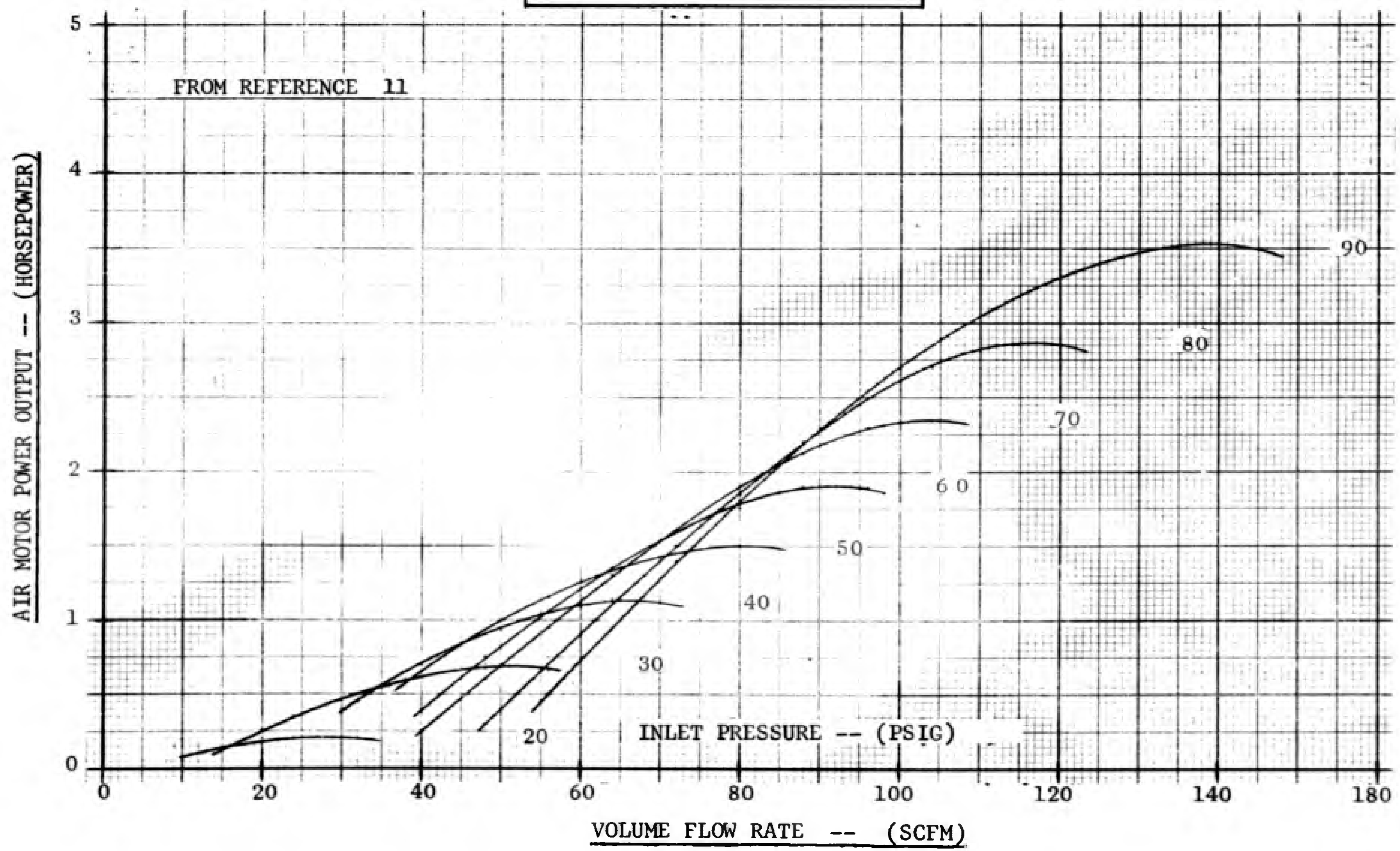





FIGURE 5.40

Since the primary sources of power loss in a rotary-vane expander are due to friction and throttling, the efficiency of air motors can be greatly increased by modifications of the inlet and outlet ports to take advantage of the maximum expansion ratio available. In addition to port design, the number of blades affects both the displacement and the efficiency of rotary expanders. The peak displacement per revolution is obtained with four blades (Reference 12 ). However, the blade-tip friction is a function of the number of blades. While blade-tip friction is not as significant as viscous shear (between the rotor and the end plates), it should be minimized. It can be shown (Reference 12 ) that a two-bladed expander represents about the best compromise between low friction and high displacement. It would have nearly the mechanical efficiency of a one-bladed machine, and would have almost as much displacement as a four-bladed unit. Reference 12 shows an estimated efficiency of 70% for rotary-vane motors. A comprehensive study of air motors is presented in Reference 13.

## 5.7 Fans

The Conductron environmental control system utilizes fans in two areas: (a) the cabin cooling loop of closed systems, where a fan recirculates the cabin air over the evaporator; and (b) the condenser heat sink air circuit, where a fan supplies the required air flow during static aircraft conditions. These fans can be driven in many ways, including electric motors, engine shaft drives, and rotary-vane motors.

Fan performance can be expressed in terms of the fan flow rate, the pressure head produced, and the power required to drive the fan. These variables are a function of fan type (centrifugal, axial), fan size (impeller diameter), fan speed, and gas density. These four independent variables specify the fan design.

The static or mechanical efficiency of a fan is the ratio of air horsepower to the shaft horsepower:

$$\eta_s = \frac{\dot{Q}\ \Delta P}{229.2\ \mathbb{P}}$$

where: $\dot{Q}$ = volume flow (ft$^3$/min)

$\Delta P$ = pressure rise (lb/in$^2$)

$\mathbb{P}$ = fan input power (HP)

Therefore, the fan power input is given by:

$$\mathbb{P} = \frac{\dot{Q}\ \Delta P}{229.2\ \eta_s} \quad \text{(Horsepower)}$$

or

$$\mathbb{P} = \frac{\dot{m}\,\Delta P}{229.2\,\eta_s\,\rho} \qquad \text{(Horsepower)}$$

where: $\dot{m}$ = mass flow rate (lb/min)

$\rho$ = air density (lb/ft$^3$)

Figures 5.41 through 5.44 represent typical fan performance characteristics. The required fan input power is plotted as a function of volume flow rate and pressure rise of the fan for various static efficiencies. For a fan with a given static efficiency, the fan input power increases with volume flow rate and pressure rise, and decreases with increasing static efficiencies.

Figure 5.45 shows the fan pressure rise as a function of volume flow rate attainable for a fan of fixed fan input power and various static efficiencies. The fan pressure rise decreases with volume flow rate and a decrease in static efficiency.

FAN INPUT POWER
VS.
FLOW RATE AND PRESSURE RISE

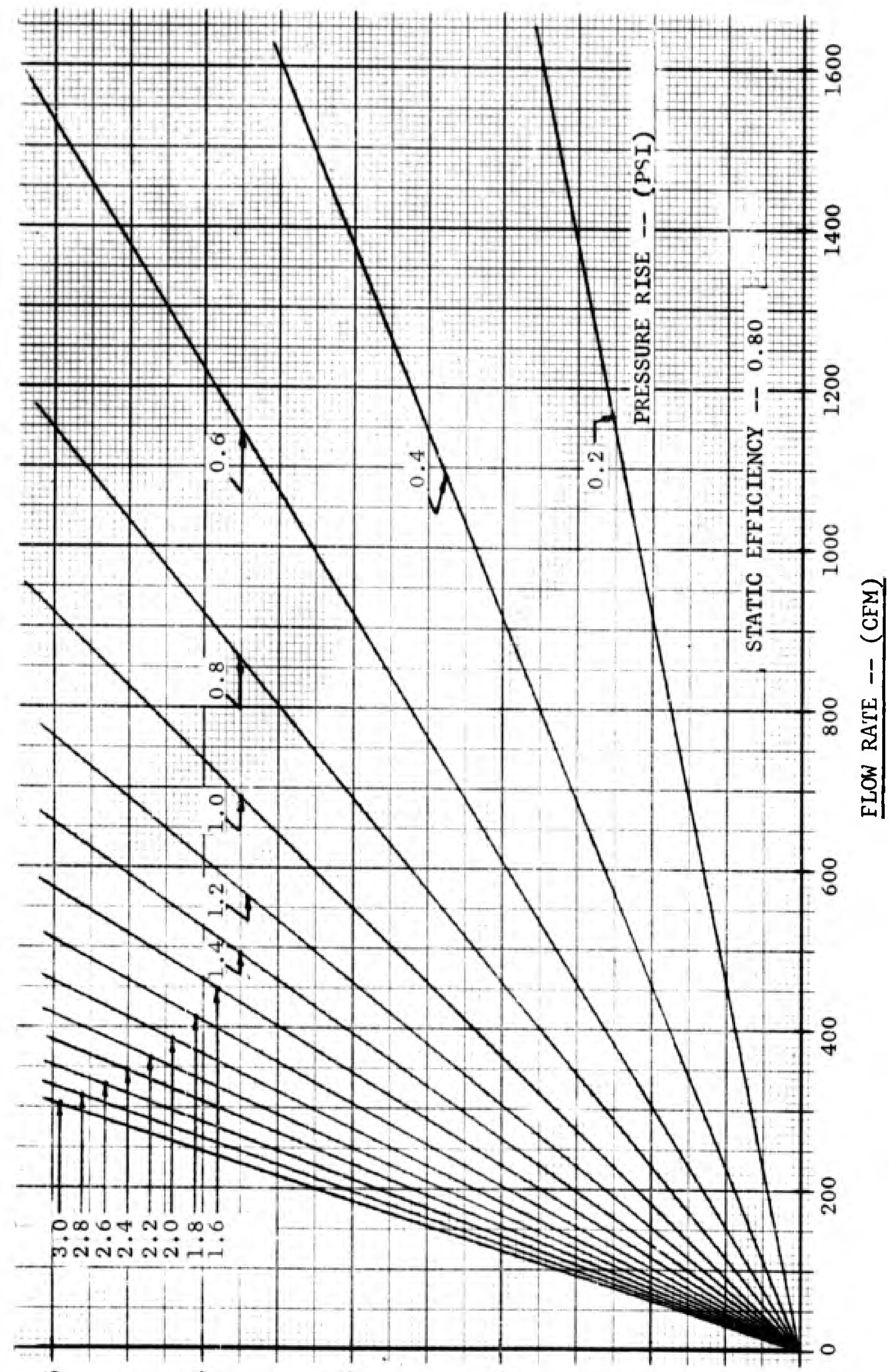





FIGURE 5.41

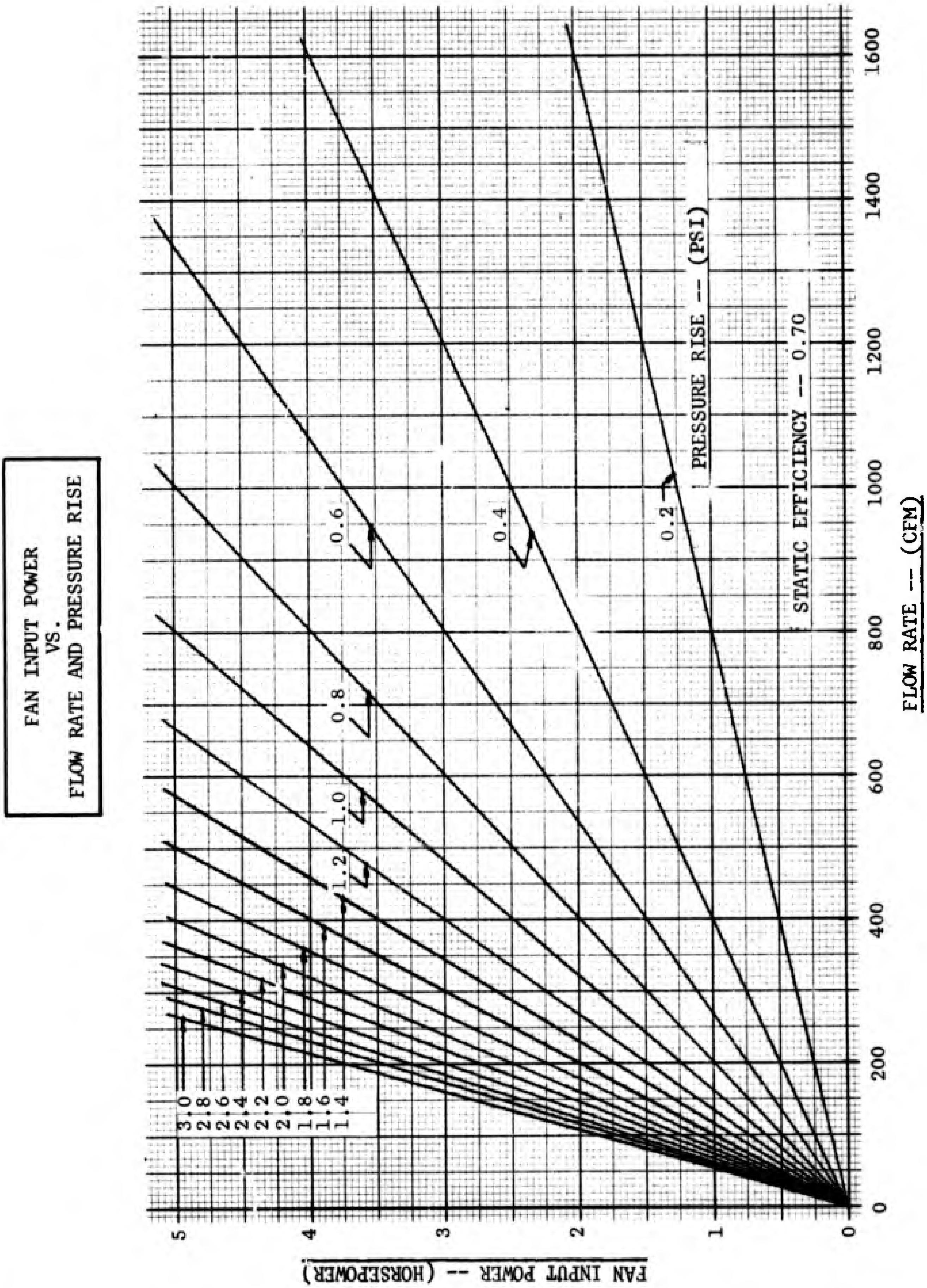

FAN INPUT POWER
VS.
FLOW RATE AND PRESSURE RISE
PRESSURE RISE -- (PSI)
STATIC EFFICIENCY -- 0.70
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0
FAN INPUT POWER -- (HORSEPOWER)
0
1
2
3
4
5
FLOW RATE -- (CFM)
0
200
400
600
800
1000
1200
1400
1600



FIGURE 5.42

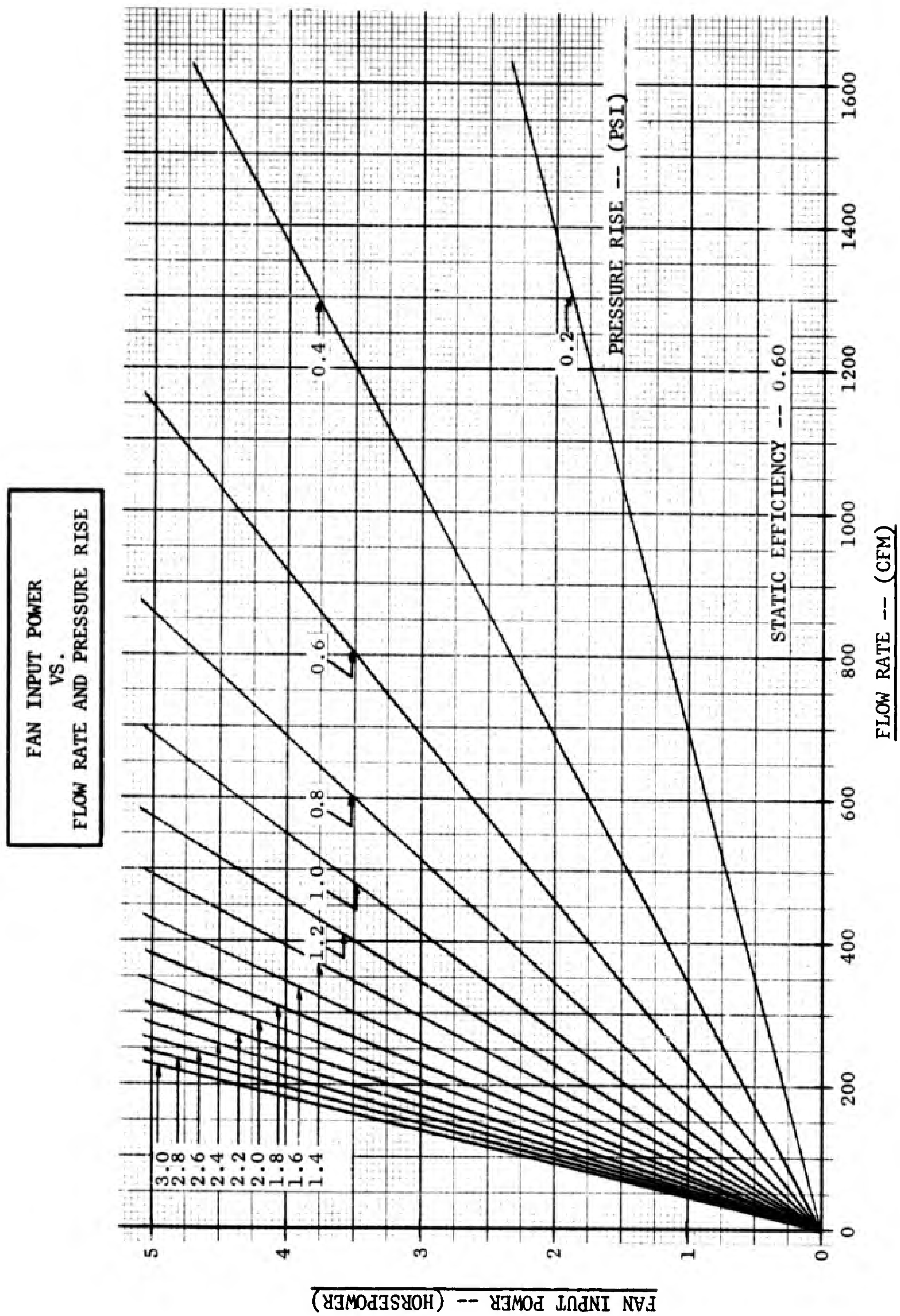

FAN INPUT POWER
VS.
FLOW RATE AND PRESSURE RISE
FAN INPUT POWER -- (HORSEPOWER)
FLOW RATE -- (CFM)
PRESSURE RISE -- (PSI)
STATIC EFFICIENCY -- 0.60
0
200
400
600
800
1000
1200
1400
1600
0
1
2
3
4
5
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2.2
2.4
2.6
2.8
3.0



FIGURE 5.43

FAN INPUT POWER
VS.
FLOW RATE AND PRESSURE RISE

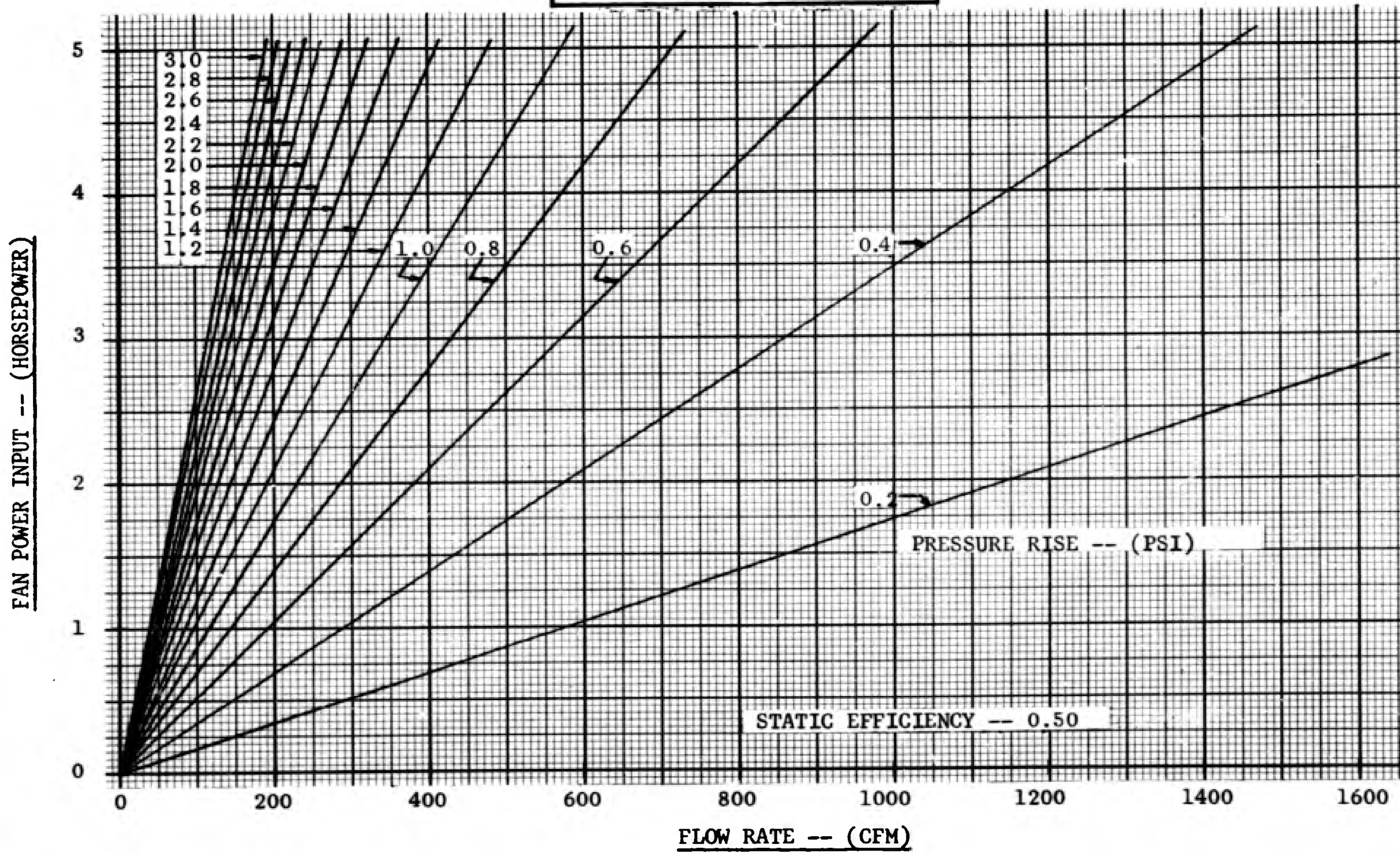



FIGURE 5.44

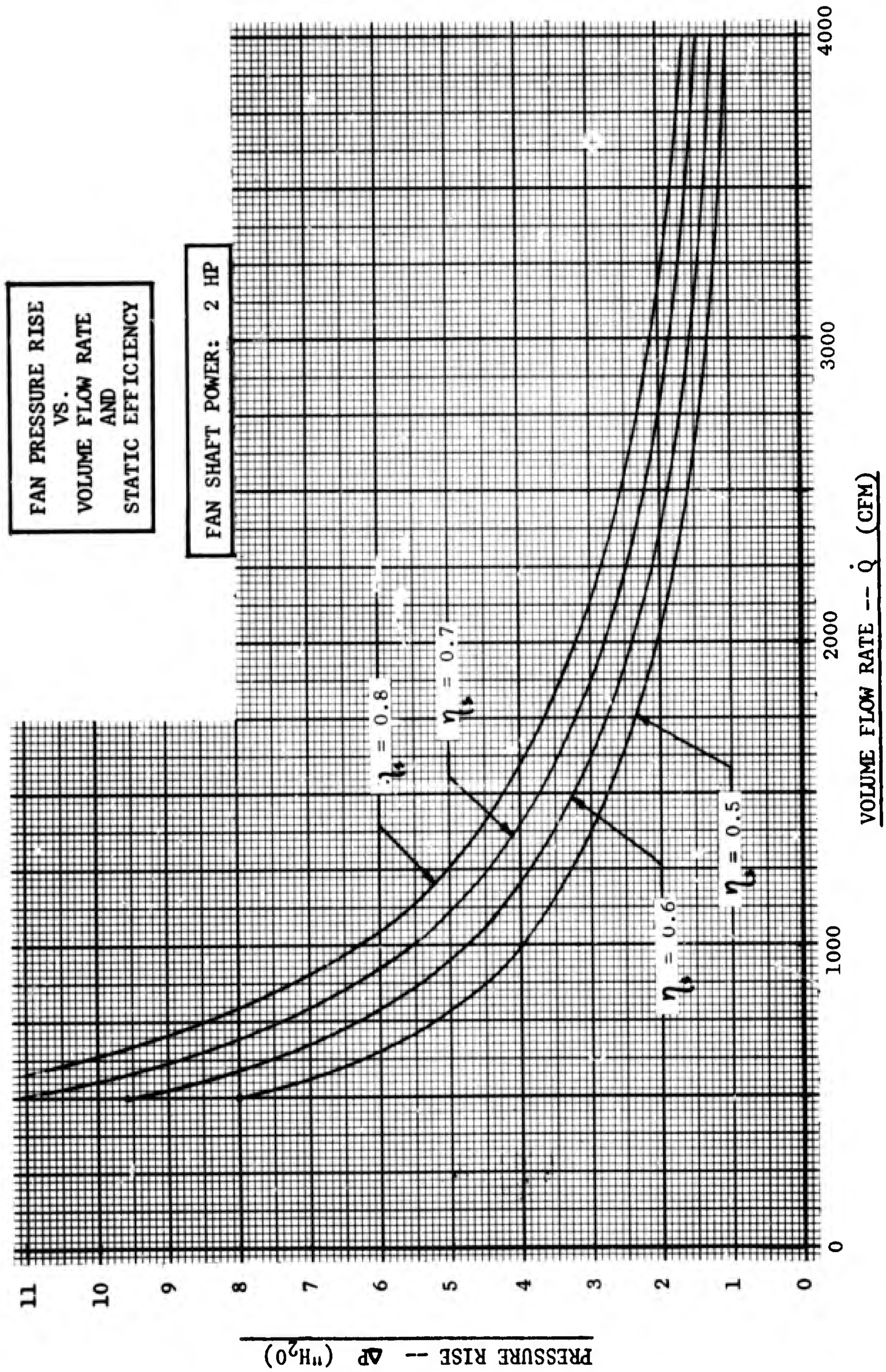

FAN PRESSURE RISE
VS.
VOLUME FLOW RATE
AND
STATIC EFFICIENCY
FAN SHAFT POWER: 2 HP
$\eta_s = 0.8$
$\eta_s = 0.7$
$\eta_s = 0.6$
$\eta_s = 0.5$
0
1000
2000
3000
4000
VOLUME FLOW RATE -- $\dot{Q}$ (CFM)
0
1
2
3
4
5
6
7
8
9
10
11
PRESSURE RISE -- $\Delta P$ ("$H_2O$)



FIGURE 5.45

## 6.0 SYSTEM ANALYSIS

This section presents the system analysis of the Conductron environmental control system for the F-4 fighter aircraft. The analyses are based on the performance requirements and constraints and the component characteristics developed in preceding sections.

Four system mechanizations are analyzed in detail, and their respective performance capabilities are compared with various performance requirements of the F-4 fighter aircraft environmental control system. System operation and system control features are discussed, and estimates of component weights and sizes are presented for each system allowing system comparisons on a weight and size basis. Complete component design specifications are tabulated for each system, and schematic thermal flow diagrams of each of the four environmental control systems for the F-4 fighter aircraft are included.

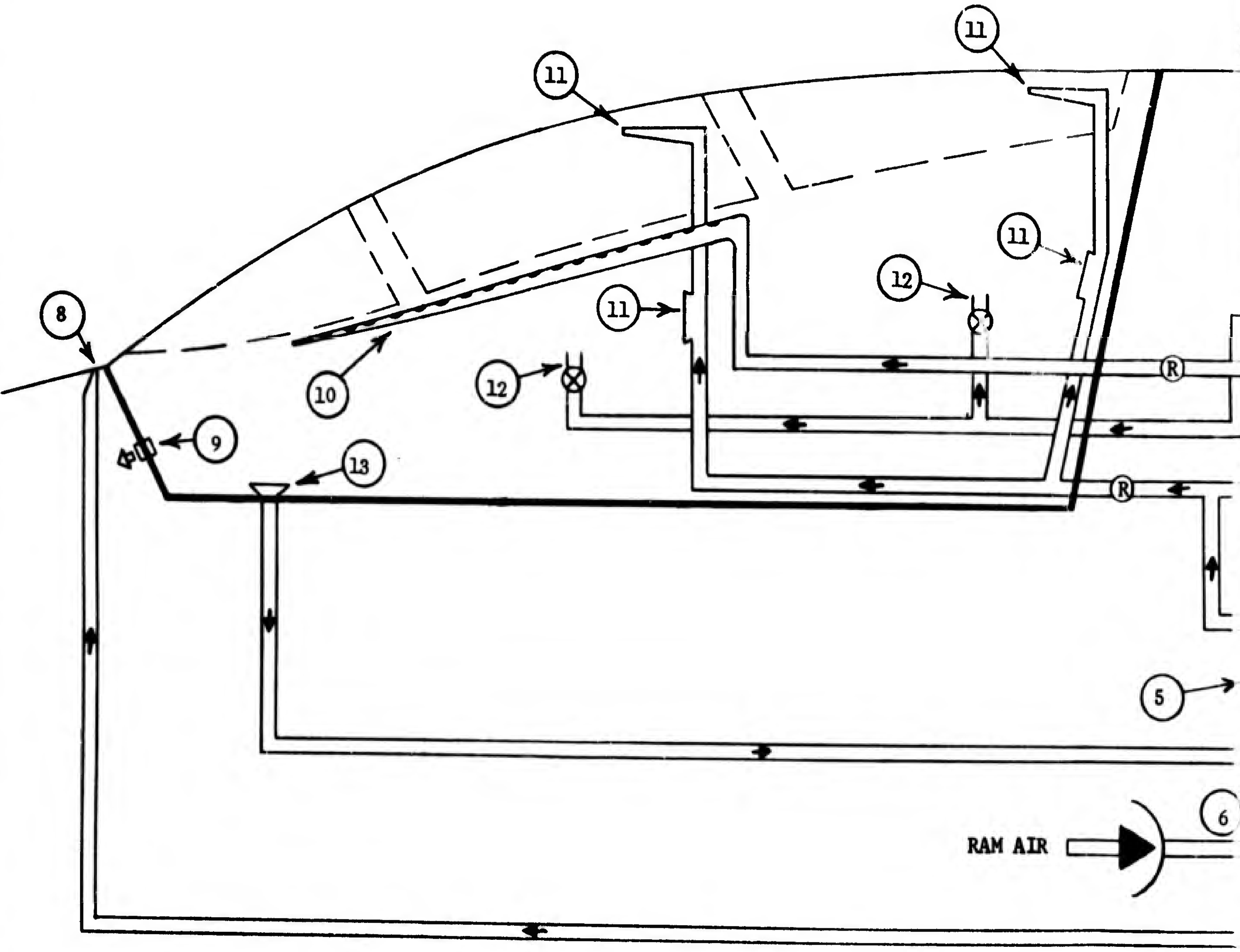
### 6.1 Candidate Systems

The versatility of the Conductron environmental control system lends itself to many different component arrangements. Based upon the available information about the operational and installation requirements of the F-4 fighter aircraft, the following four methods of system mechanization were selected for detailed analysis. These four system mechanizations are identified as System "A", System "B-1", System "B-2", and System "C".

The four Conductron environmental control system mechanizations for the Navy F-4 fighter aircraft are shown in Figures 6.1, 6.2, and 6.3. These schematic flow diagrams show the entire environmental control system, including the mechanization for all cabin ECS functions (cabin cooling, cabin pressurization, pressure-suit ventilation, and rain removal), with all necessary valving for temperature modulation, flow regulation, flow diverting, and flow shut-off. Since Systems "B-1" and "B-2" are identical from a mechanization point of view, the same schematic flow diagram is used to describe these systems (Figure 6.2).

In System "A" (Figure 6.1), compressor bleed air is used as the heat source for system actuation. Heat extraction is accomplished by a compact, finned tube, air-to-liquid heat exchanger. Heat rejection to ram (or ambient) air is by means of a compact condenser/heat exchanger (air-to-liquid) of plate-fin type construction. The ram air is passed through an air expander* where it is cooled by essentially adiabatic expansion prior to entering the condenser. Cabin temperature control is achieved by recirculation of cabin air through a compact evaporator/heat exchanger (air-to-liquid). The cabin air is recirculated by means of a fan which is driven by the ram air expander or the compressor bleed air expander. Cabin pressurization and pressure-suit temperature control are

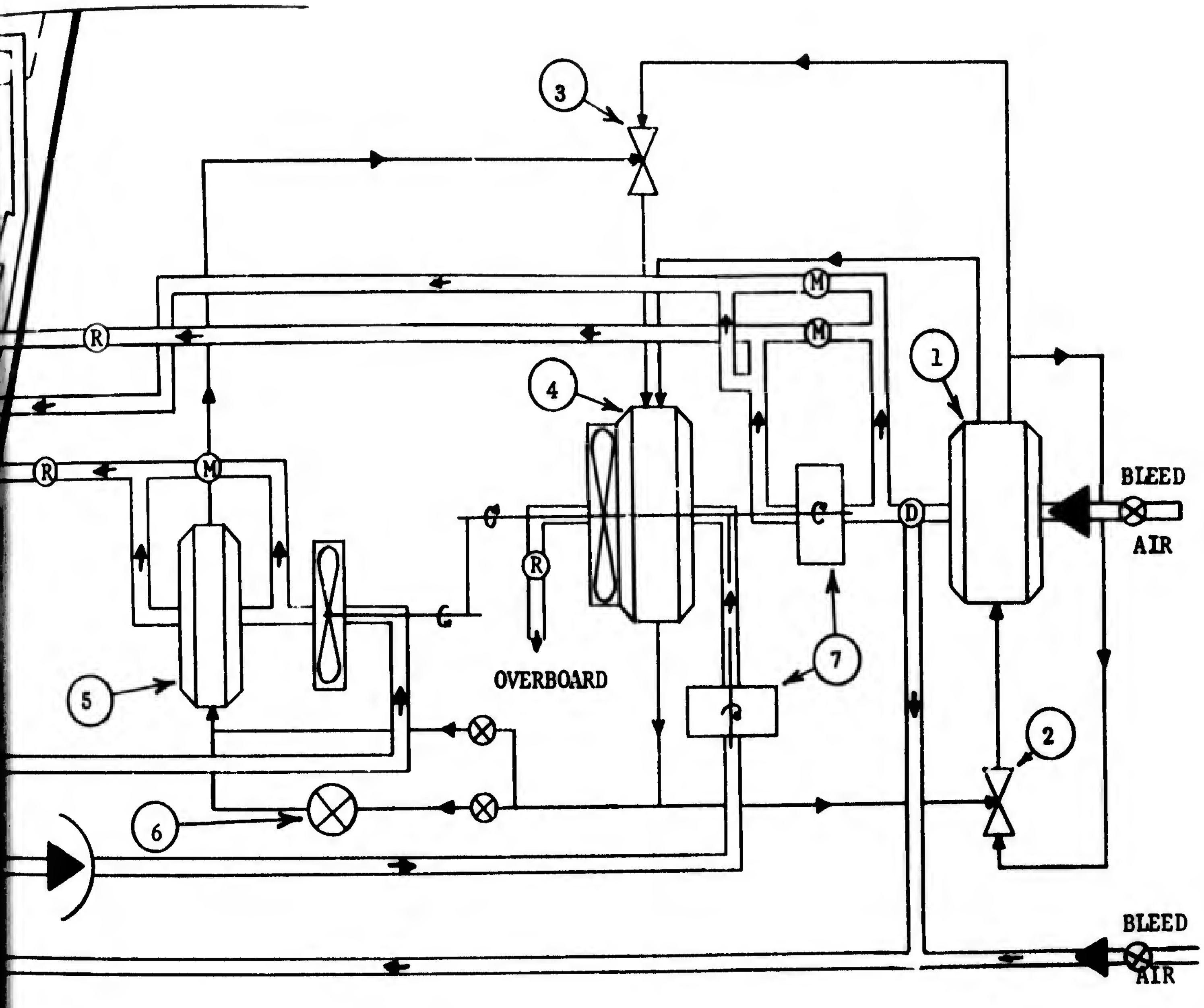
*An air expander with an adiabatic efficiency of 0.80 is required. Rotary-Vane motors may be developed to yield this performance. Off-the-shelf expansion turbines can easily meet this requirement.




1. Gas Piston Compressor / Boiler
2. Fill Jet Pump
3. Converter
4. Condenser with Fan
5. Evaporator with Fan
6. Thermostatic Expansion Valve
7. Air Expander
8. Rain Removal Nozzle
9. Cabin Pressurization Control Vent
10. Defogging Nozzle
11. Cabin Cooling Air Inlet Diffuser
12. Pressure-Suit Connection
13. Cabin Cooling Air Return

⊗ Shut-off Valve
D Diverting Valve
M Modulating Valv
R Regulating Valv

A




hut-off Valve

iverting Valve

odulating Valve

egulating Valve

NAVY F-4 FIGHTER AIRCRAFT

CABIN ENVIRONMENTAL CONTROL

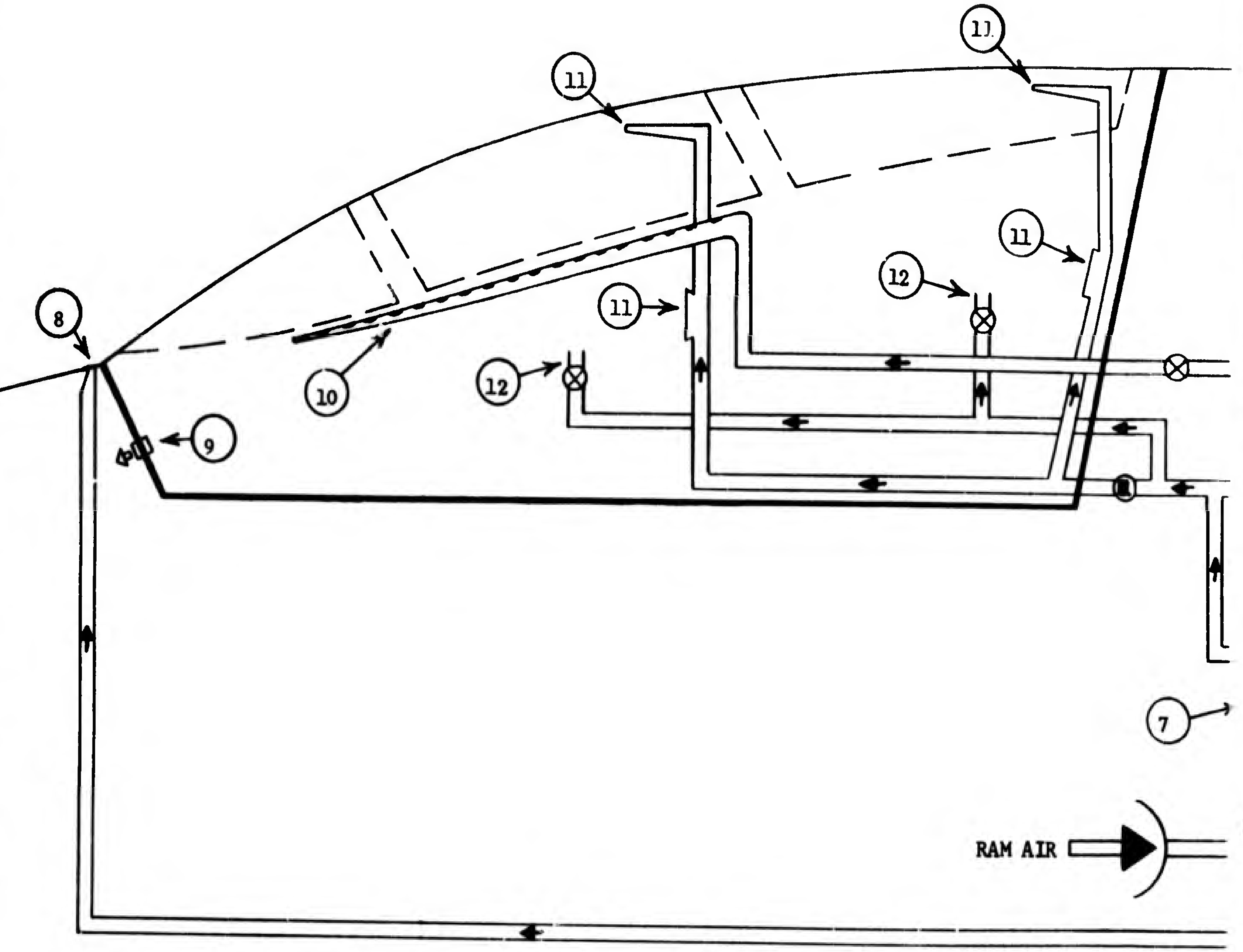
- SCHEMATIC FLOW DIAGRAM -

SYSTEM "A"



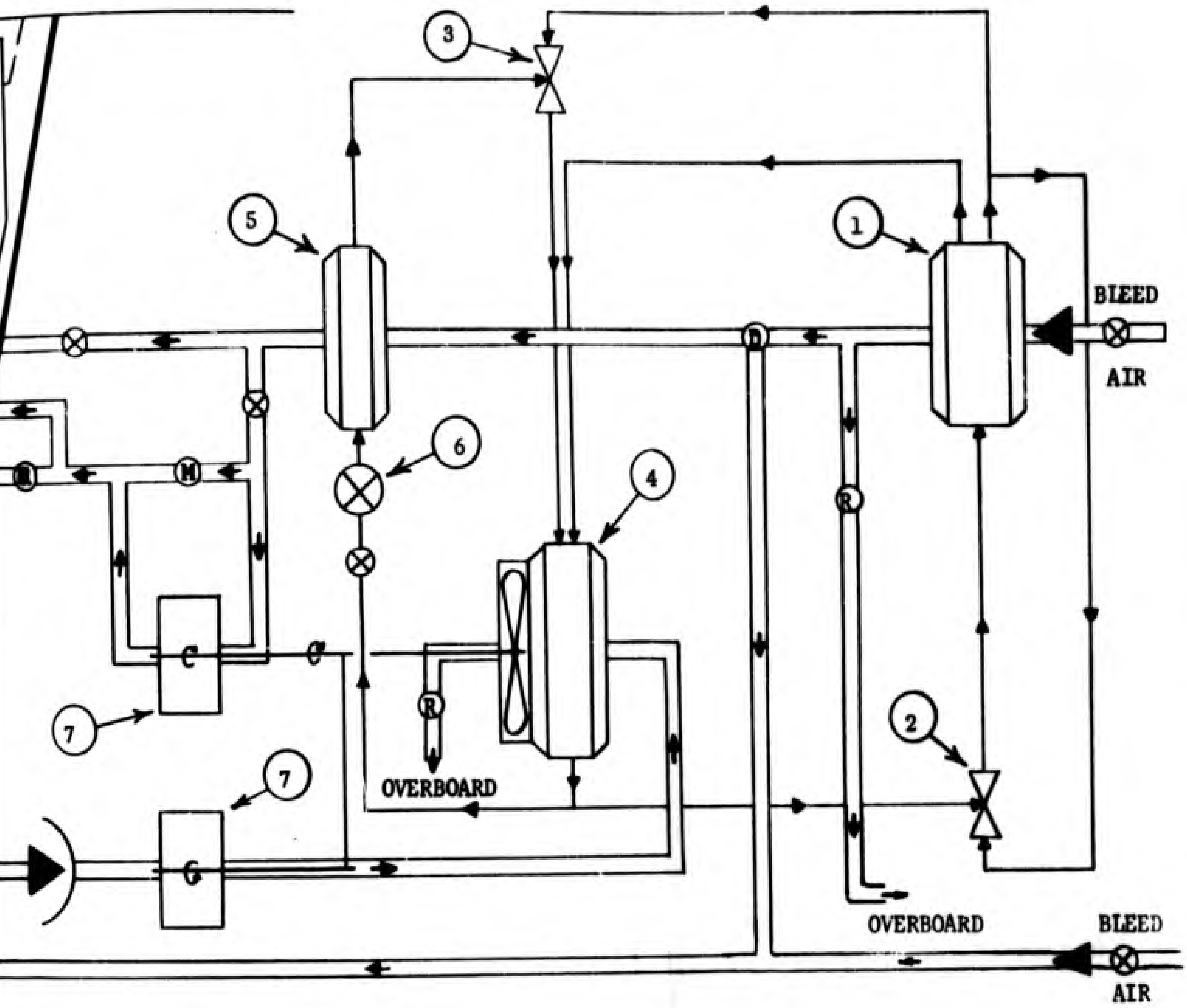
FIGURE 6.1

B




| | | | | | |
|---|---|---|---|---|---|
| 1. | Gas Piston Compressor / Boiler | 7. | Air Expander | ⊗ | Shut |
| 2. | Fill Jet Pump | 8. | Rain Removal Nozzle | D | Dive |
| 3. | Converter | 9. | Cabin Pressurization Control Vent | M | Modu |
| 4. | Condenser with Fan | 10. | Defogging Nozzle | R | Regu |
| 5. | Evaporator | 11. | Cabin Cooling Air Inlet Diffuser | | |
| 6. | Thermostatic Expansion Valve | 12. | Pressure-Suit Connection | | |

A

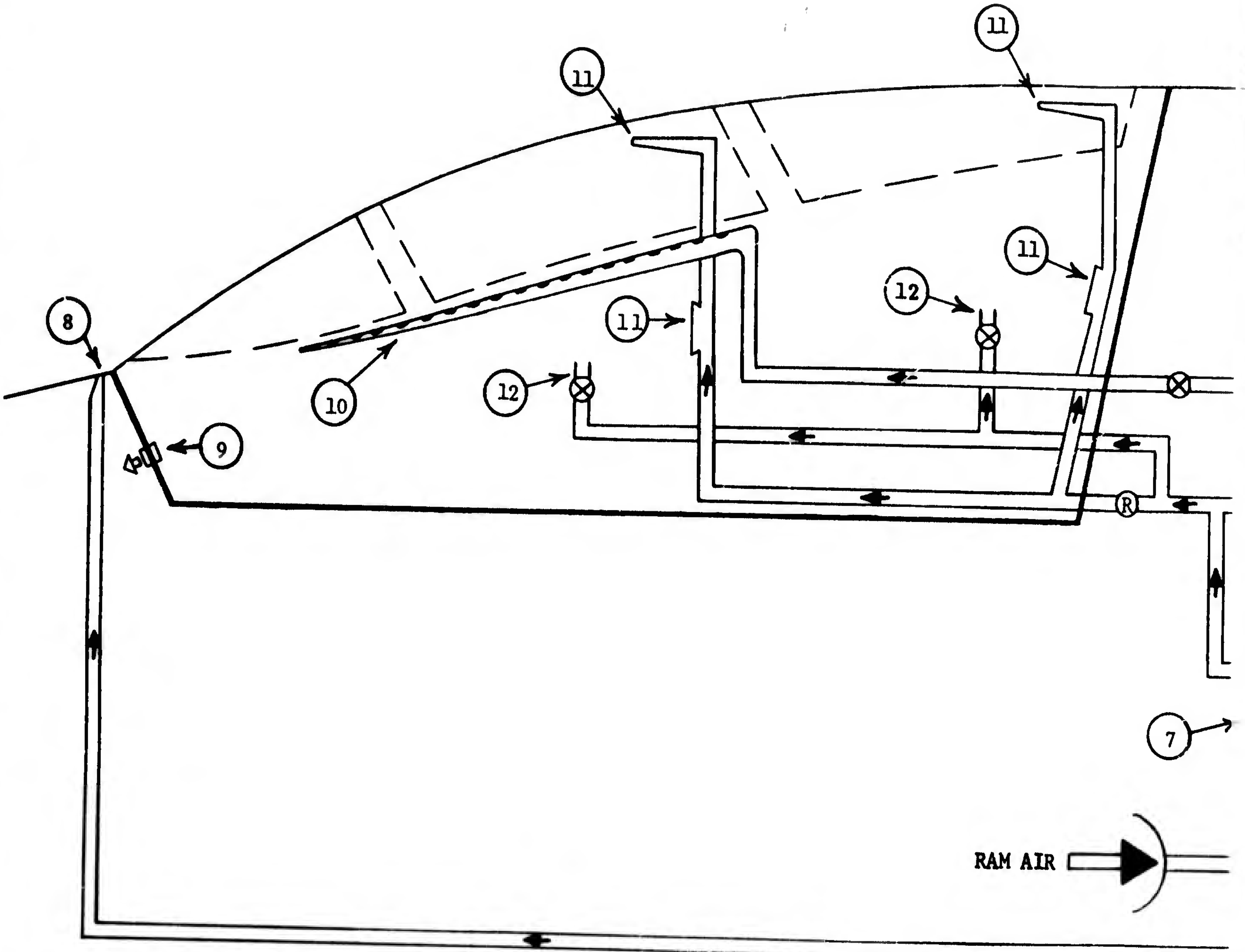



- Ⓧ Shut-off Valve
- Ⓓ Diverting Valve
- Ⓜ Modulating Valve
- Ⓡ Regulating Valve

NAVY F-4 FIGHTER AIRCRAFT
CABIN ENVIRONMENTAL CONTROL
- SCHEMATIC FLOW DIAGRAM -
SYSTEM "B-1" AND "B-2"

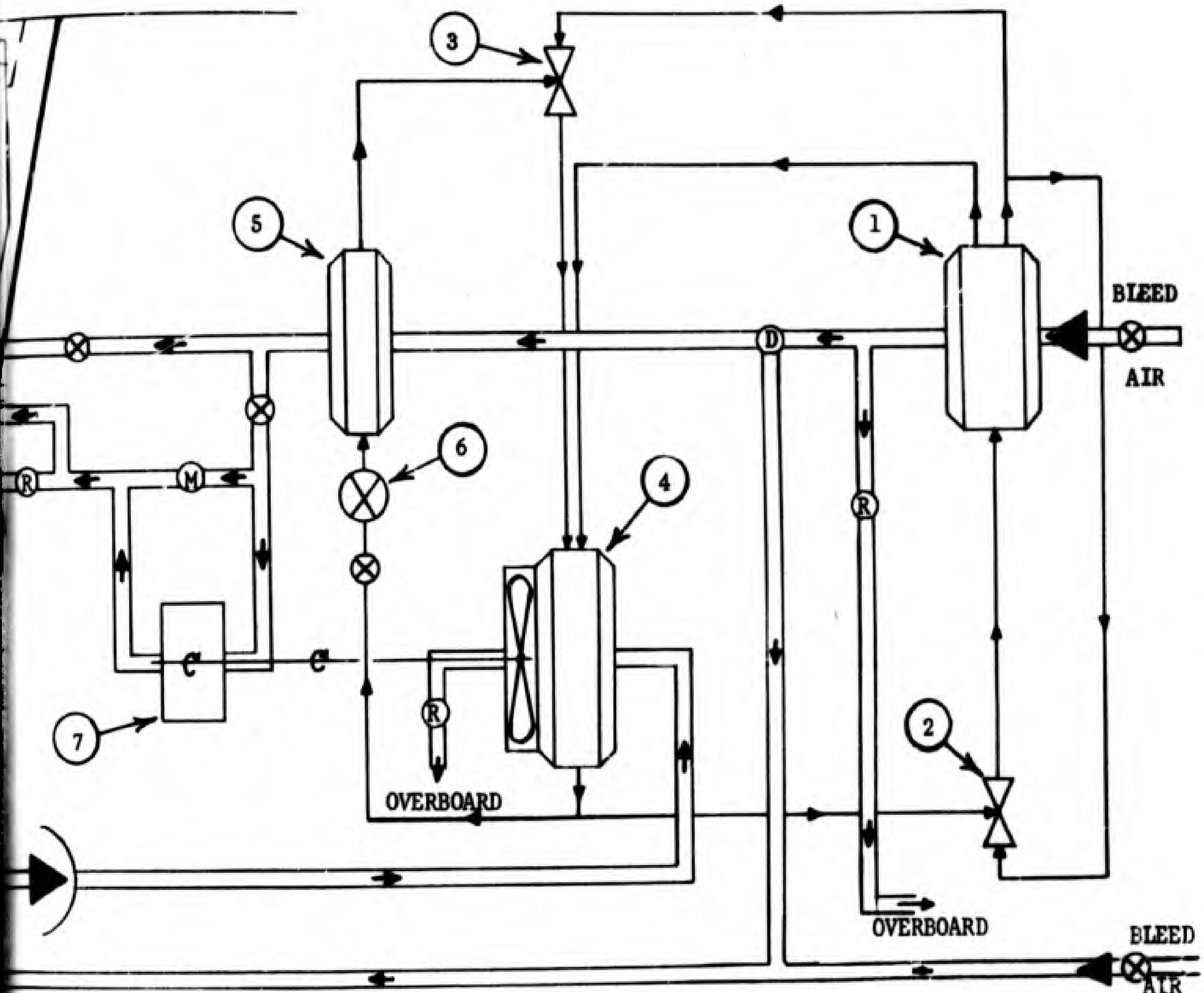


FIGURE 6.2




| | | | |
|---|---|---|---|
| 1. Gas Piston Compressor / Boiler | 7. Air Expander | Ⓧ Shut-off Valve |
| 2. Fill Jet Pump | 8. Rain Removal Nozzle | Ⓓ Diverting Valve |
| 3. Converter | 9. Cabin Pressurization Control Vent | Ⓜ Modulating Valve |
| 4. Condenser with Fan | 10. Defogging Nozzle | Ⓡ Regulating Valve |
| 5. Evaporator | 11. Cabin Cooling Air Inlet Diffuser | |
| 6. Thermostatic Expansion Valve | 12. Pressure-Suit Connection | |

A




-off Valve

rting Valve

lating Valve

lating Valve

NAVY F-4 FIGHTER AIRCRAFT

CABIN ENVIRONMENTAL CONTROL

– SCHEMATIC FLOW DIAGRAM –

SYSTEM "C"



FIGURE 6.3

B

provided by compressor bleed air. The compressor bleed air leaving the gas piston unit is passed through a bleed air expander where it is cooled prior to entering the cabin and/or the pressure-suit.

System "B-1" (Figure 6.2) uses the same mechanization for system actuation and heat rejection as System "A". Cabin and pressure-suit temperature control and pressurization are accomplished by passing the compressor bleed air leaving the gas piston unit through a compact evaporator/heat exchanger (air-to-liquid) and then through a bleed air expander where it is further cooled prior to entry into the cabin and/or pressure-suit. After absorption of the cooling load, the bleed air is dumped overboard (open system). The ram air and bleed air expanders used with this system are of the lowest possible efficiency that still results in adequate system performance at most supersonic flight conditions. This mechanization of the Conductron environmental control system is one which can utilize rotary vane motors (air motors) requiring only moderate development efforts. Since these expanders are low speed machines, the reliability and maintainability of System "B-1" should be superior to the other mechanizations.

System "B-2" (Figure 6.2) is identical to System "B-1", except that a higher efficiency bleed air expander is used, resulting in a system with performance characteristics which meet all aircraft operating conditions.

System "C" (Figure 6.3) is another open-loop system, which utilizes compressor bleed air as the heat source for system actuation and ram air (ambient air) for condenser cooling. Cabin cooling and pressure-suit temperature control are accomplished by compressor bleed air, which passes from the gas piston unit through an evaporator/heat exchanger and bleed air expander before it enters the cabin (or the pressure-suit). It differs from System "B-1" in that no ram air expander is used. The condenser heat sink is therefore always at the ram air temperature. At high supersonic flight conditions, high condensing temperatures with proportionately high pressures throughout the system will exist. This system will not operate at the extreme flight conditions, because the ram air temperatures (and, therefore, the condensing temperatures) approach the critical temperature of Refrigerant-11.

The significant differences between the four system mechanizations considered in this section are summarized below:

| | SYSTEM | | | |
|---|---|---|---|---|
| | "A" | "B-1" | "B-2" | "C" |
| Cabin cooling loop | closed | open | open | open |
| Ram air expander efficiency | 0.7 | 0.5 | 0.5 | - |
| Bleed air expander efficiency | 0.8 | 0.5 | 0.7 | 0.8 |

TABLE 6-1: SYSTEM IDENTIFICATION

The thermodynamic operating conditions selected for the analysis of the four mechanizations of the Conductron system are indicated on the pressure-enthalpy diagrams of Figures 6.4, 6.5, and 6.6.

System "A" (Figure 6.4) generates refrigerant vapor at a pressure of 300 psia and a temperature of 300°F with 10°F superheat. Condensation takes place at 45 psia (140°F). The evaporator conditions are 6.35 psia pressure and 35°F temperature with 5°F superheat.

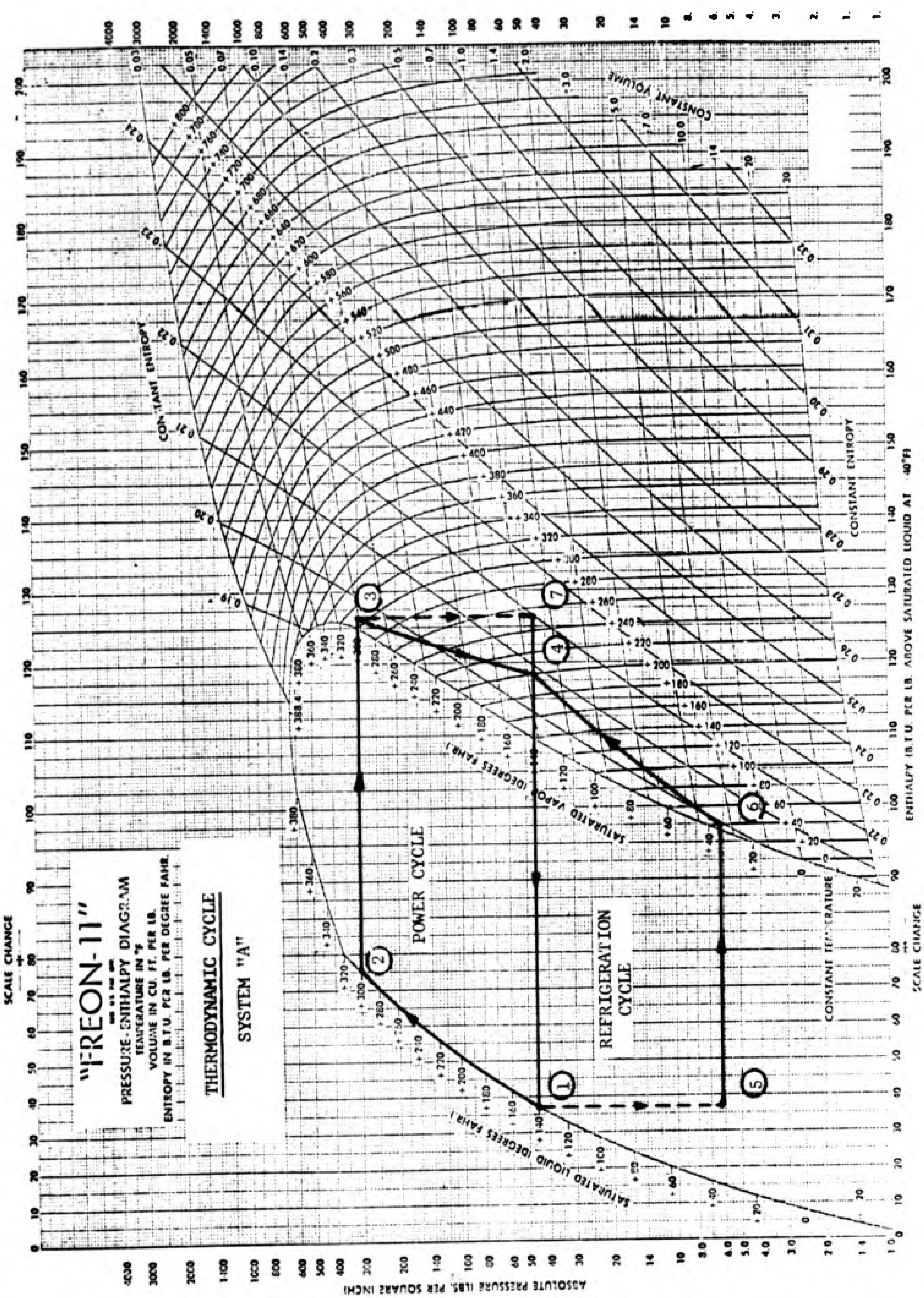
Systems "B-1" and "B-2" have identical operating conditions (Figure 6.5). System actuation is at a refrigerant pressure of 300 psia and a temperature of 300°F with 10°F superheat. The condenser operates at 130.5 psia pressure and a temperature of 220°F. The evaporating pressure and temperature is 52 psia and 150°F (5°F superheat), respectively.

System "C" (Figure 6.6) has a refrigerant vapor motive pressure of 509 psia with a temperature of 360°F and 10°F superheat. The condenser conditions are 248 psia pressure and 280°F temperature. The evaporator operates at an evaporating pressure of 130 psia and an evaporating temperature of 220°F and 5°F superheat.

## 6.2 System Performance

The performance capabilities of the four selected mechanizations of the Conductron environmental control system were analyzed in detail. The results are presented in terms of aircraft operating conditions; namely, Mach number and altitude, since the cabin cooling load is a function of Mach number for any given altitude. (Figures 6.7 through 6.12). Also plotted on each figure is a curve representing the required cooling capacity, thus allowing a direct comparison between the required cooling load and the system cooling capability. All of these figures are based on the requirement of maintaining a cabin temperature of 100°F.

Figures 6.7 and 6.9 indicate that System "A" and System "B-2" will meet or exceed the cooling requirements at all aircraft operating conditions. Figure 6.8 shows that System "B-1" can meet the required cabin cooling loads up to an altitude of 37,000 feet (on a ICAO Standard Day). Above this altitude at maximum flight speed, the cabin temperature may reach 115°F. Figure 6.10 shows that System "C" can meet the design cooling loads up to an altitude of 26,000 feet (on a ICAO Standard Day). Above this altitude at maximum flight speed, the cabin temperature is estimated to reach 145°F. Figures 6.11 and 6.12 present a comparison between the cooling capacity of System "C" and the required cooling capabilities for the F-4 fighter aircraft on a ANA Hot Day and a ANA Cold Day, respectively.



"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN °F
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
SYSTEM "A"
POWER CYCLE
REFRIGERATION CYCLE
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT TEMPERATURE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
SCALE CHANGE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)




FIGURE 6.4

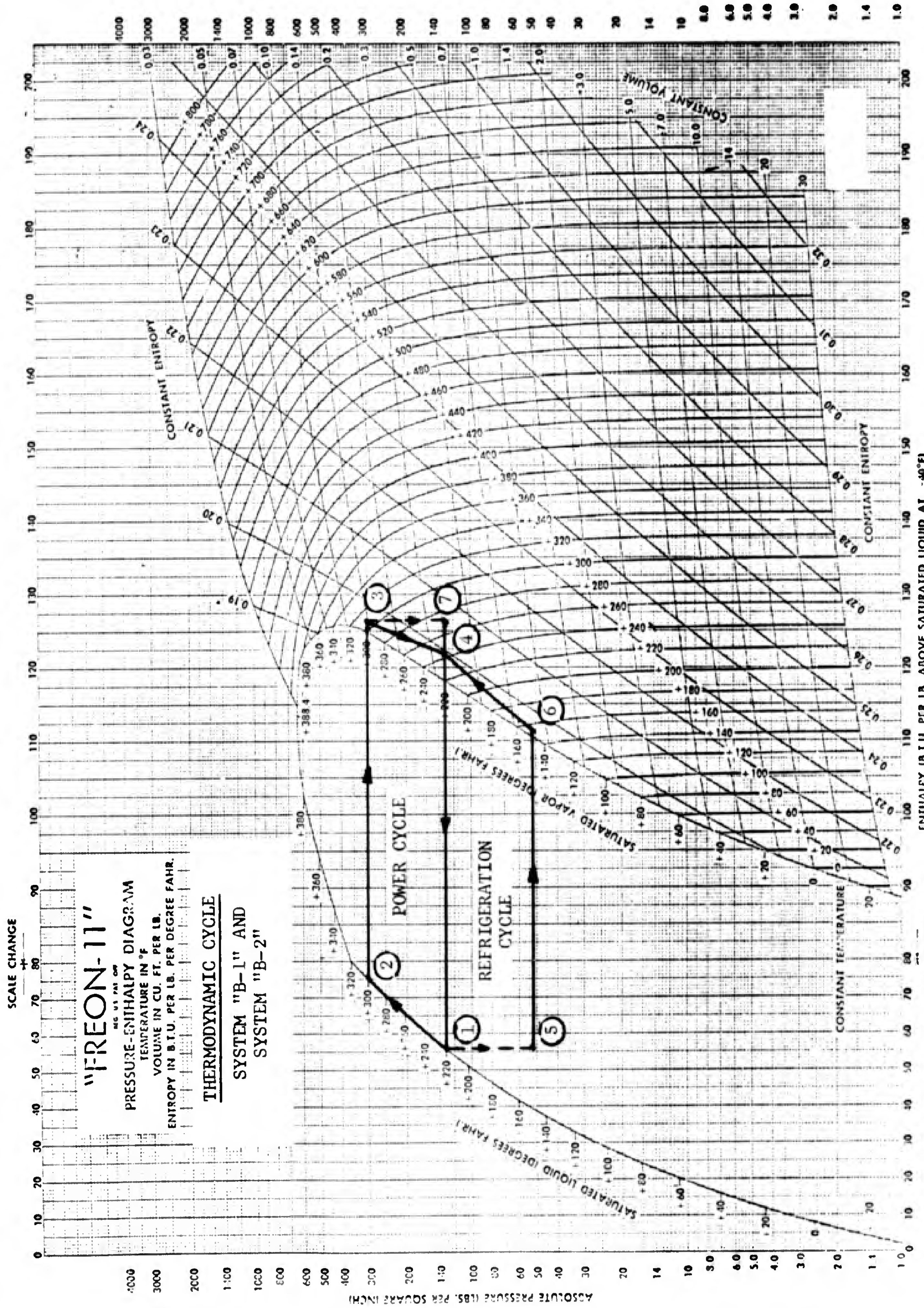

"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN °F
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
SYSTEM "B-1" AND SYSTEM "B-2"
POWER CYCLE
REFRIGERATION CYCLE
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT TEMPERATURE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
SCALE CHANGE



FIGURE 6.5

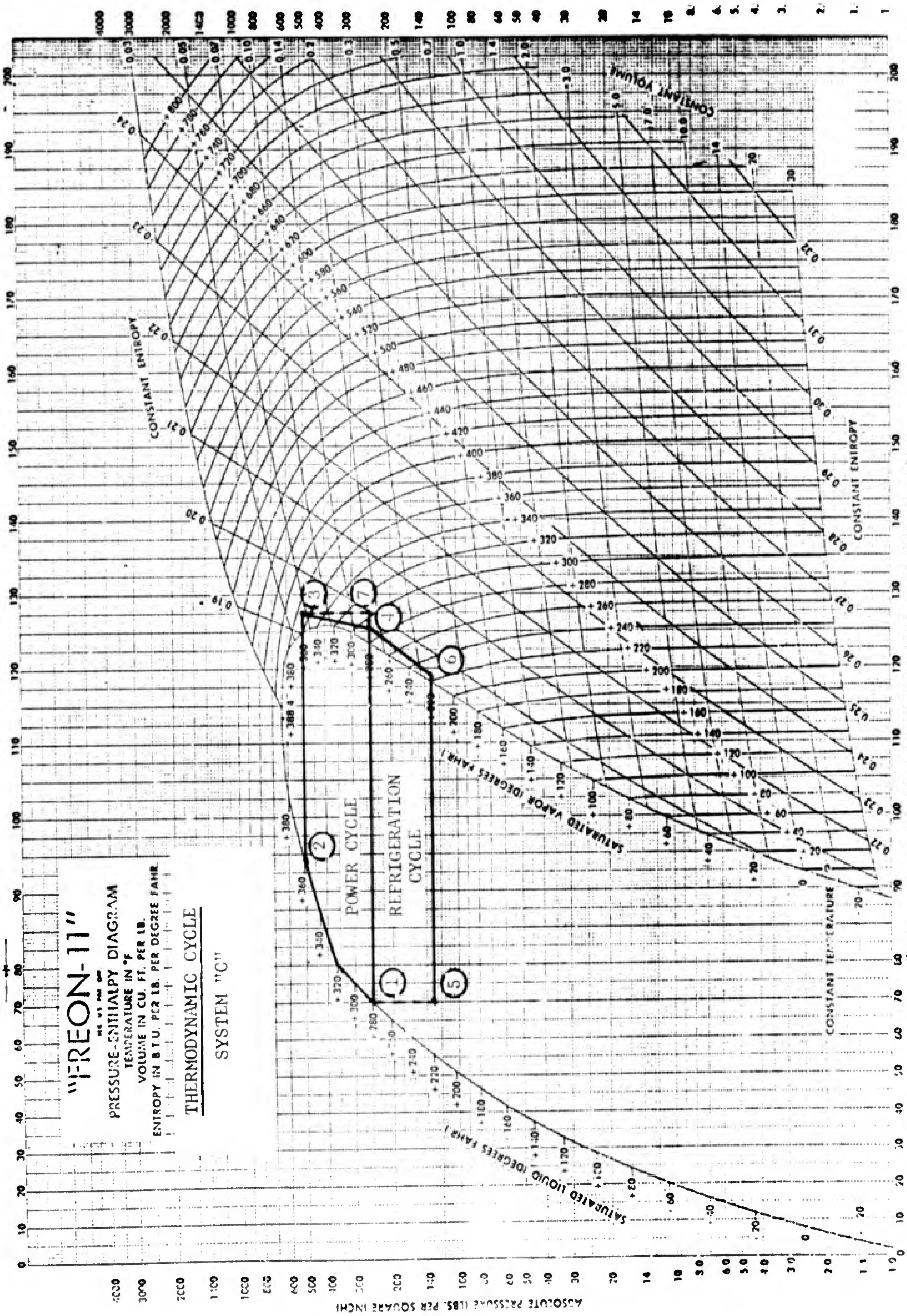

"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN °F
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
SYSTEM "C"
POWER CYCLE
REFRIGERATION CYCLE
SCALE CHANGE
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT TEMPERATURE
SATURATED VAPOR (DEGREES FAHR.)
SATURATED LIQUID (DEGREES FAHR.)
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)



FIGURE 6.6

SYSTEM PERFORMANCE
SYSTEM "A"
ICAO STANDARD DAY

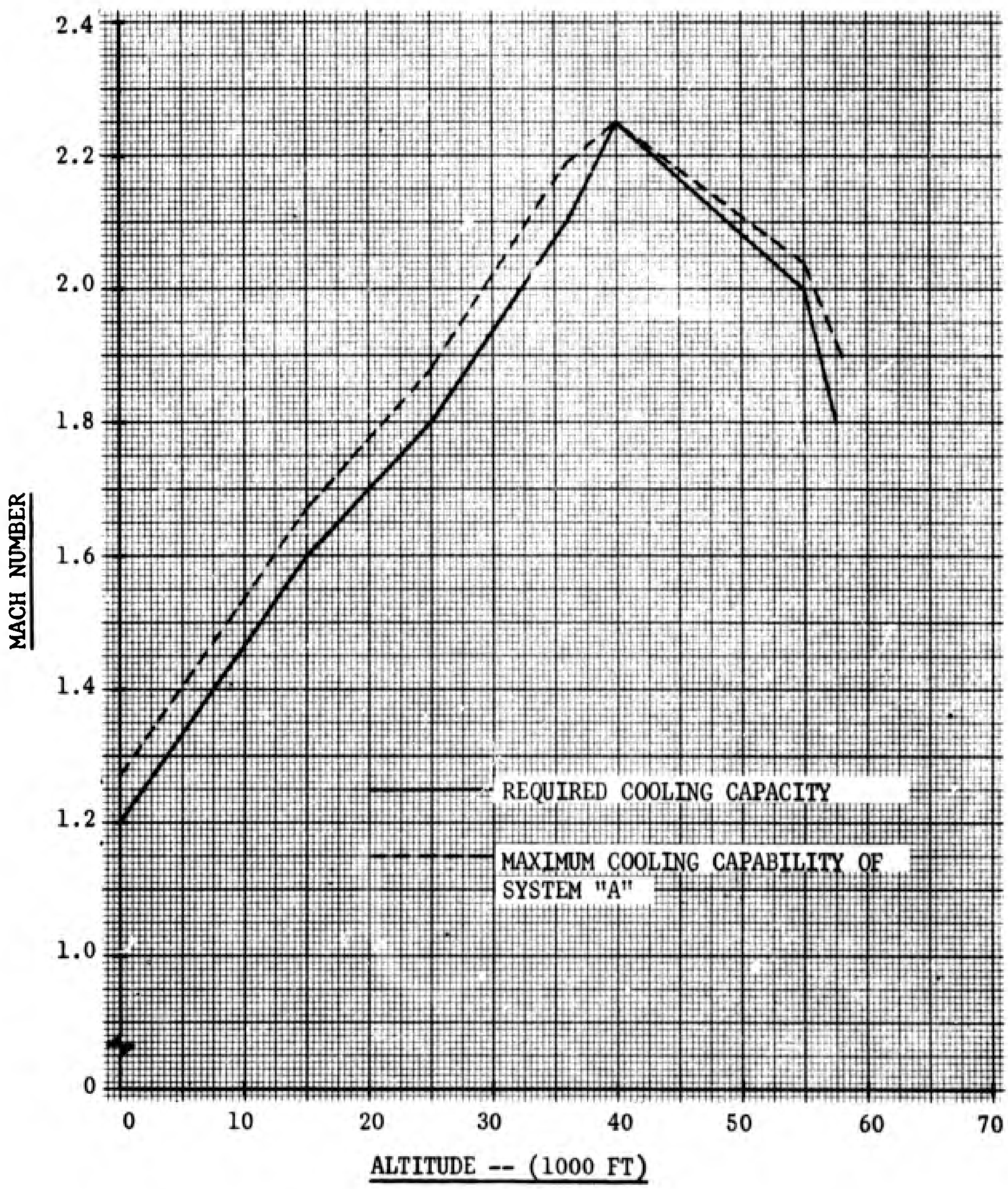





FIGURE 6.7

SYSTEM PERFORMANCE
SYSTEM "B-1"
ICAO STANDARD DAY

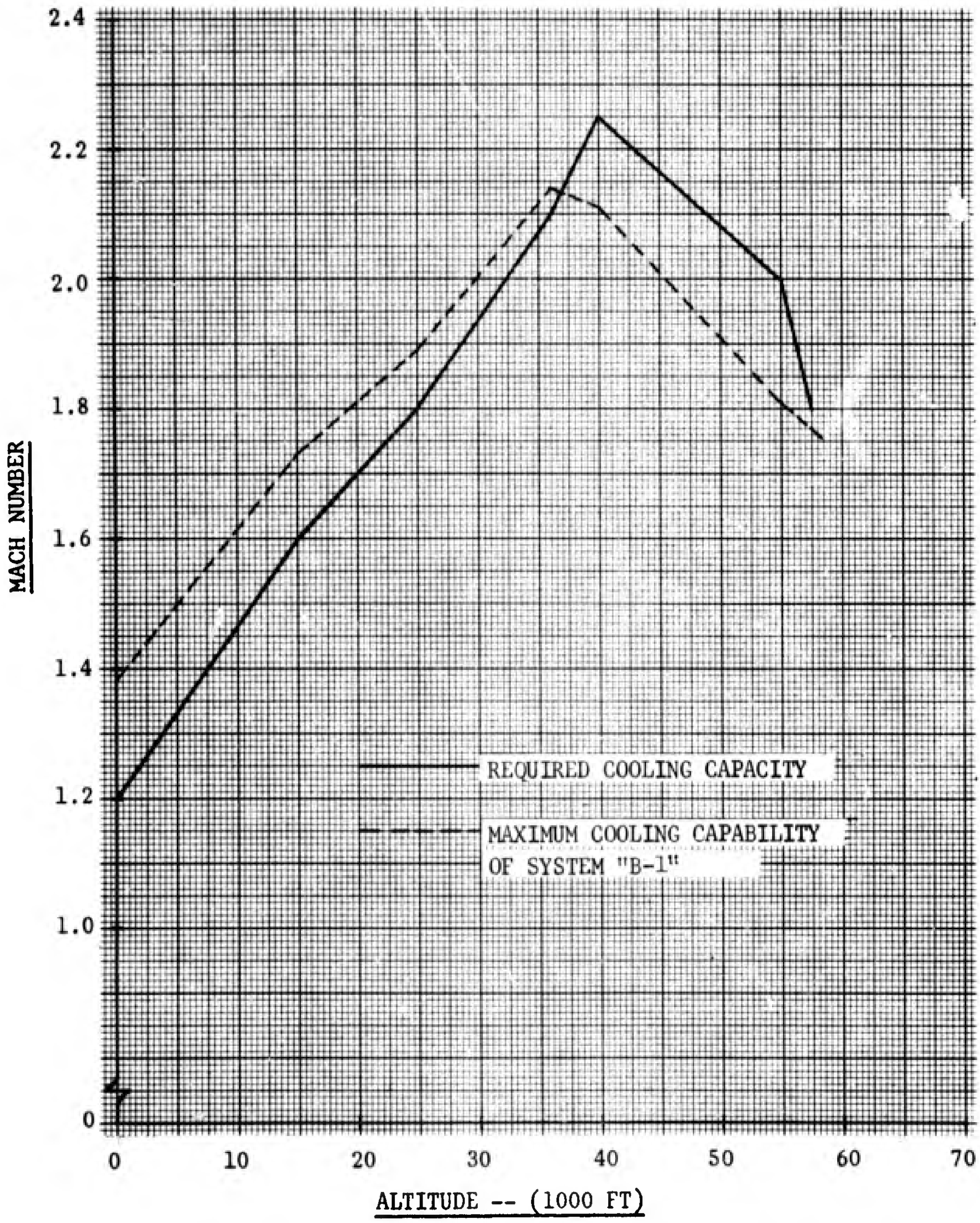





FIGURE 6.8

SYSTEM PERFORMANCE
SYSTEM "B-2"
ICAO STANDARD DAY

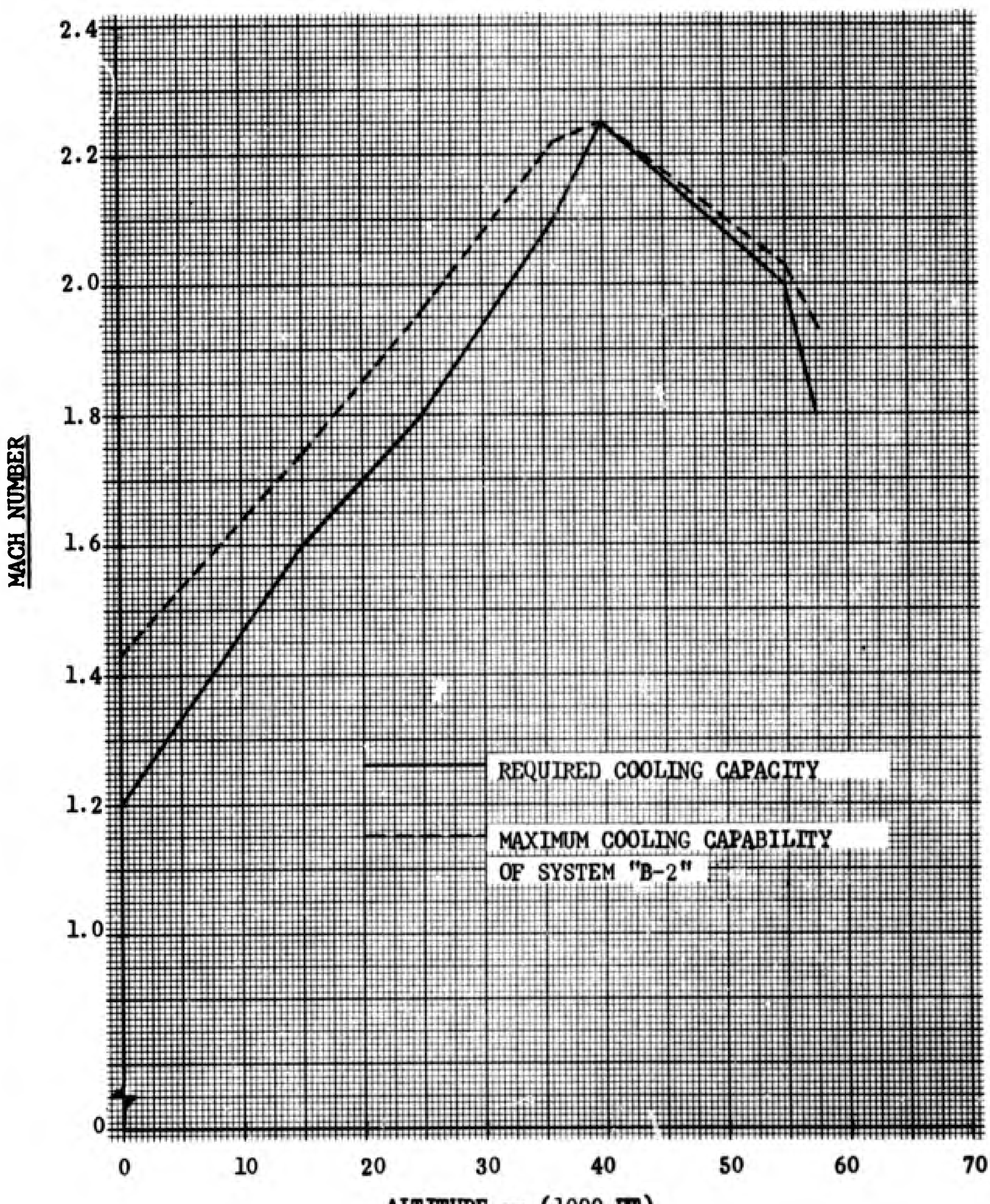





FIGURE 6.9

SYSTEM PERFORMANCE
SYSTEM "C"
ICAO STANDARD DAY

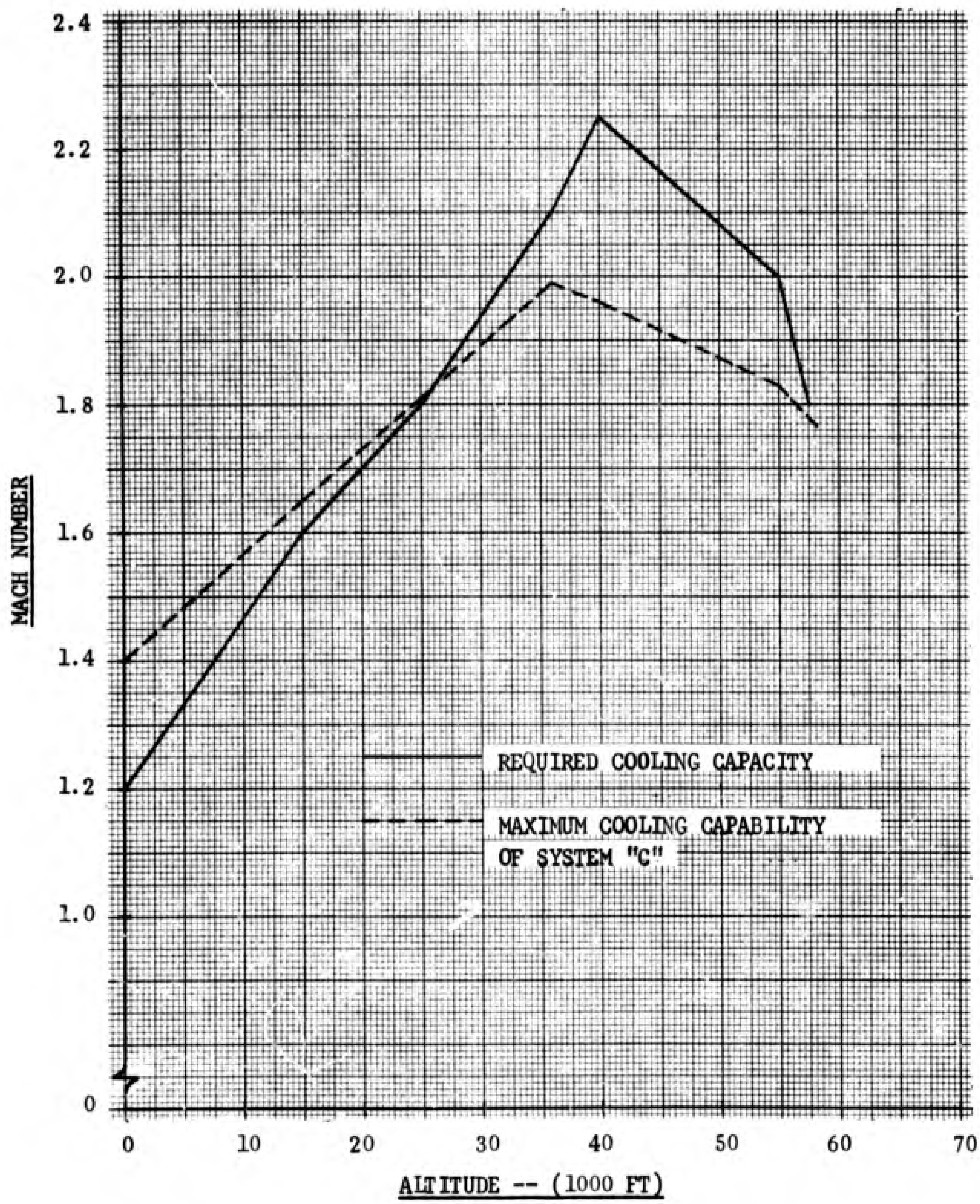





FIGURE 6.10

SYSTEM PERFORMANCE
SYSTEM "C"
ANA HOT DAY

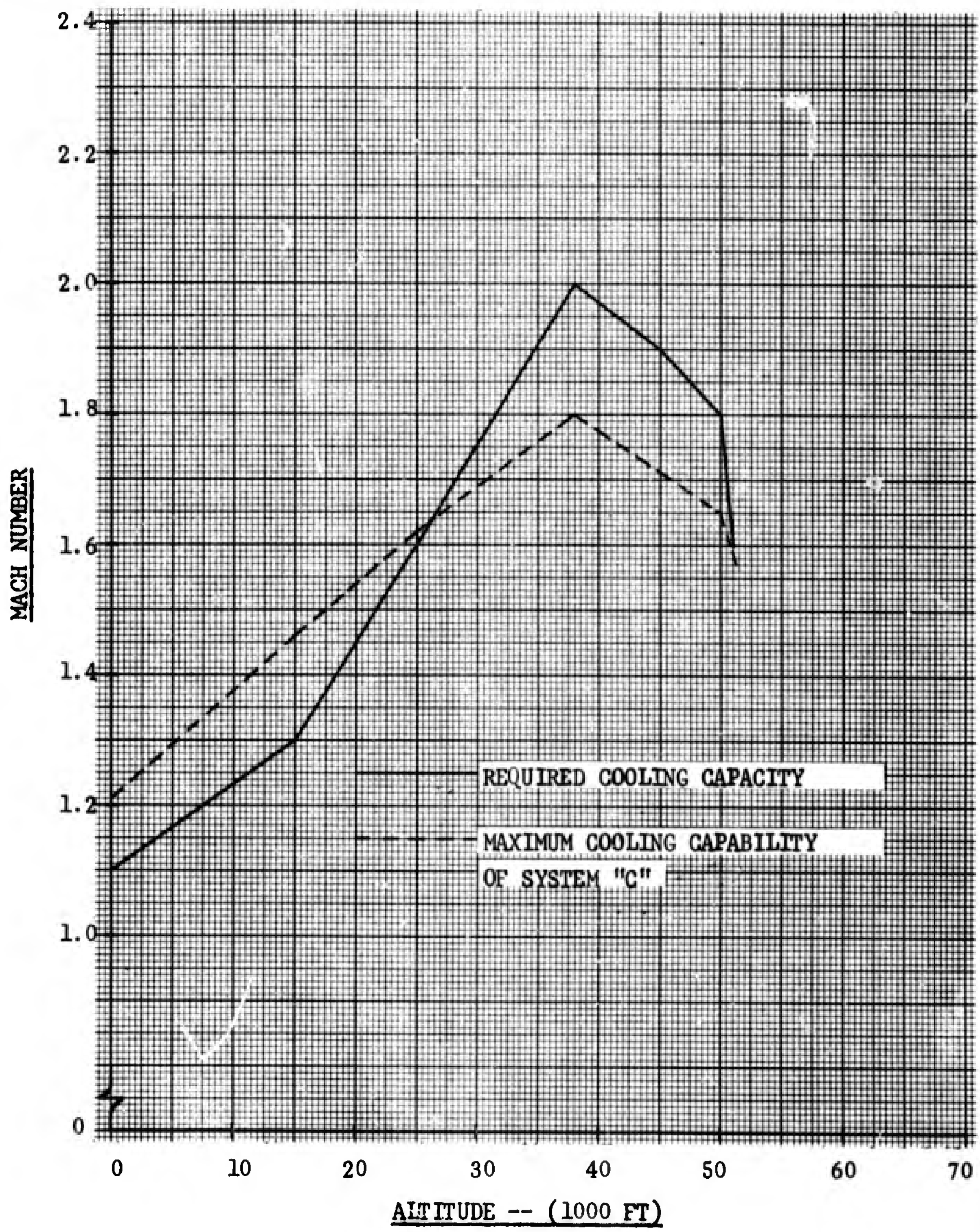





FIGURE 6.11

SYSTEM PERFORMANCE
SYSTEM "C"
ANA COLD DAY

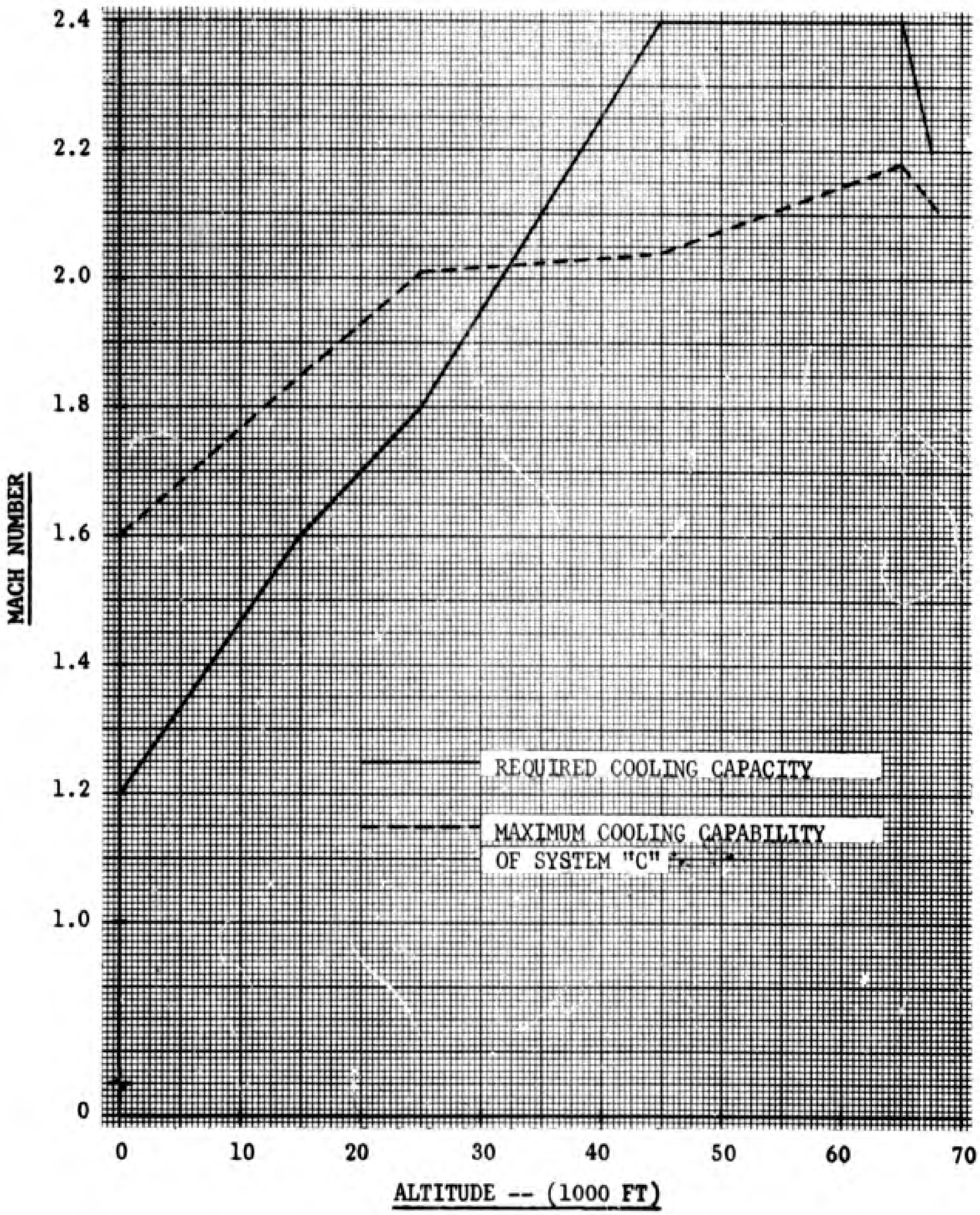





FIGURE 6.12

The cooling capacity of Systems "B-1" and "C" is limited at high altitudes by the ram air mass flow rate that can be passed through the existing ram air scoop of the Navy F-4 fighter aircraft. Suitable modifications of the ram air scoop should increase the cooling capacity of these systems without impairing aircraft performance. Although these two systems can not meet the maximum flight conditions, they are clearly capable of meeting the cooling loads encountered at "normal" flight speeds. Since maximum flight speed conditions are maintained for relatively short periods of time only, it may be permissible to allow the cabin temperature to exceed the 100°F design temperature during these flight conditions. In addition, crew comfort can be achieved at these conditions by the pressure-suit ventilation system. All four system mechanizations are capable of providing the required pressure-suit ventilation flow rates, temperatures, and pressures at all aircraft operating conditions.

The system analyses show that the Conductron environmental control system requires considerably less compressor bleed air than the air cycle system presently used in the Navy F-4 fighter aircraft. The compressor bleed air requirements for the four system mechanizations are shown below as a percentage of the air cycle bleed air requirements:

| | |
|---|---|
| System "A" | 24% |
| System "B-1" | 61% |
| System "B-2" | 37% |
| System "C" | 41% |

The performance of environmental control systems can be assessed by a comparison of their Coefficients-of-Performance (C.O.P.), which is generally defined as the ratio of the cooling effect produced to the total energy input required. Two COP definitions are applicable and meaningful to the Conductron system: (1) System COP, which measures the performance of the entire environmental control system (including the bleed air expander) and is defined as the system cooling load capacity divided by the system heat input; and (2) Thermodynamic Cycle COP, which measures the performance effectiveness of the Conductron thermodynamic cycle only. It is defined by the ratio of evaporator cooling load to the gas piston unit heat input. Table 6-2 shows both Coefficients-of-Performance for the four Conductron system mechanizations analyzed. It can be seen that the thermodynamic performance of the Conductron system is relatively insensitive to differences in mechanization, as well as the associated changes in system temperature and pressure levels. However, the overall system performance is very much dependent upon the selected system mechanization. The system effectiveness increases as evaporator temperature decreases and/or bleed air expander efficiency increases.

TABLE 6-2

COEFFICIENTS OF PERFORMANCE

| SYSTEM MECHANIZATION | SYSTEM C.O.P. | THERMODYNAMIC C.O.P. |
|---|---|---|
| "A" | 0.420 | 0.211 |
| "B-1" | 0.140 | 0.224 |
| "B-2" | 0.267 | 0.224 |
| "C" | 0.204 | 0.218 |

This system analysis was performed using standard air conditioning practice, where the design cabin temperature is interpreted to mean the maximum or outlet temperature in the cabin. This is a good assumption for large rooms, where with proper ducting design, mixing and entrainment phenomena at the cooling air inlet result in substantially no or very small temperature gradients throughout the major portion of the space being conditioned. However, in small enclosures, such as a fighter aircraft cabin, such mixing and entrainment may be difficult to attain, resulting in an "effective" cabin temperature that may be approximated by the arithmetic mean between the cabin inlet and outlet temperatures. Such an assumption would reduce the cooling air flow requirement for a given cooling load by a factor of 2. For a system designed to the conventional definition of cabin temperature ( = cabin outlet temperature), the crew will be exposed to a potential temperature gradient between the cooling air inlet temperature and the cabin temperature. Therefore, it may be possible to design and position the inlet diffusers in such a manner so that a physiologically acceptable comfort condition is established for the crew during the maximum flight speed conditions where the design cabin temperature is exceeded (Systems "B-1" and "C").

Lacking data on the ram air scoop pressure recovery characteristics of the Navy F-4 fighter aircraft, Figure 1B-19 of Reference 9 was used for the analyses. Data on the F-4 fighter aircraft pressure recovery factor as a function of ram air flow rate for a Mach number of 2.20 (Reference 14) indicates that the ram air scoop characteristics of the F-4 fighter aircraft are better than those based on Reference 9. If this is true for all flight Mach numbers, the system capabilities shown in Figures 6.7 through 6.12 will be slightly higher. According to Reference 4, the maximum condenser cooling capacity is achieved at 89% of the maximum ram air flow rate. This flow rate factor of 0.89 was maintained constant throughout the performance analysis. A complete typical system performance analysis is presented in Appendix "D"; and a detailed analysis of system performance capability is given in Appendix "E".

The system analyses were based on state-of-the-art performance characteristics for all components, except the converter and rotary-vane motor (air motor). Development of ejectors and other converter concepts is being actively pursued at Conductron. Ejector efficiencies of 28.5% have been demonstrated, and converter efficiencies of up to 60% have been predicted for some of the unique converter concepts under development. On this basis, a 35% converter efficiency, as assumed throughout the system analysis, appears to be a reasonable projection of the state-of-the-art, and one which can be realized within a short development time.

Rotary-vane motors at present are designed for industrial and shop usage, where intermittent duty cycles are common and reversible rotation is desirable. Little or no effort has been expended in developing air motors for use as an expansion device. The potential of air motors for such applications has been realized previously, and analytical predictions of performance capability and design considerations were carried out (References 12 and 13). It can be shown that the efficiency of present air motors can be greatly increased by modification of inlet and outlet ports, so that full advantage is taken of the maximum expansion ratio available. According to Reference 12, adiabatic efficiencies of 70% are feasible. Thus, the efficiencies of 50% assumed for the air motors of Systems "B-1" and "B-2" should be attainable with a minimum of development effort.

The gas piston unit (GPU) is not considered to be a critical component in the same sense as the converter or air motor. The only part of the GPU that requires some special attention for airborne applications is the gas piston compressor. However, this component is receiving continuous attention at Conductron for ground applications, resulting in increasingly simpler and more reliable units. The adoption of these devices to airborne applications does not appear to represent more than the addition of simple control features to compensate for certain adverse conditions, particularly vibration and orientation. The development of an air-worthy GPU is therefore considered to be entirely within the state-of-the-art of hydraulic valve design and control.

For all system mechanizations under consideration, it has been assumed that the compressor bleed is contaminant free or that contaminant concentrations are at very low levels. Under this assumption, contaminants produced by the cabin occupants or equipment may present a problem in a closed system if not contaminant removal is provided. In the case of a semi-closed system (System "A"), no contaminant removal equipment is needed if the make-up rate dictated by cabin leakage is in excess of the contaminant concentration built-up rate. With the relatively high leak rates of the F-4 cabin, no air purifiers are needed. The required ventilation rate for the F-4 fighter aircraft (Reference 3) is 24 CFM, which is 1/10 of the leak rate.

## 6.3 System Operation and Control

The operation of the Conductron environmental control system over the wide range of aircraft operating conditions in both the cooling and the heating modes requires certain control features which are based on simple, proven techniques and can be achieved with off-the-shelf equipment.

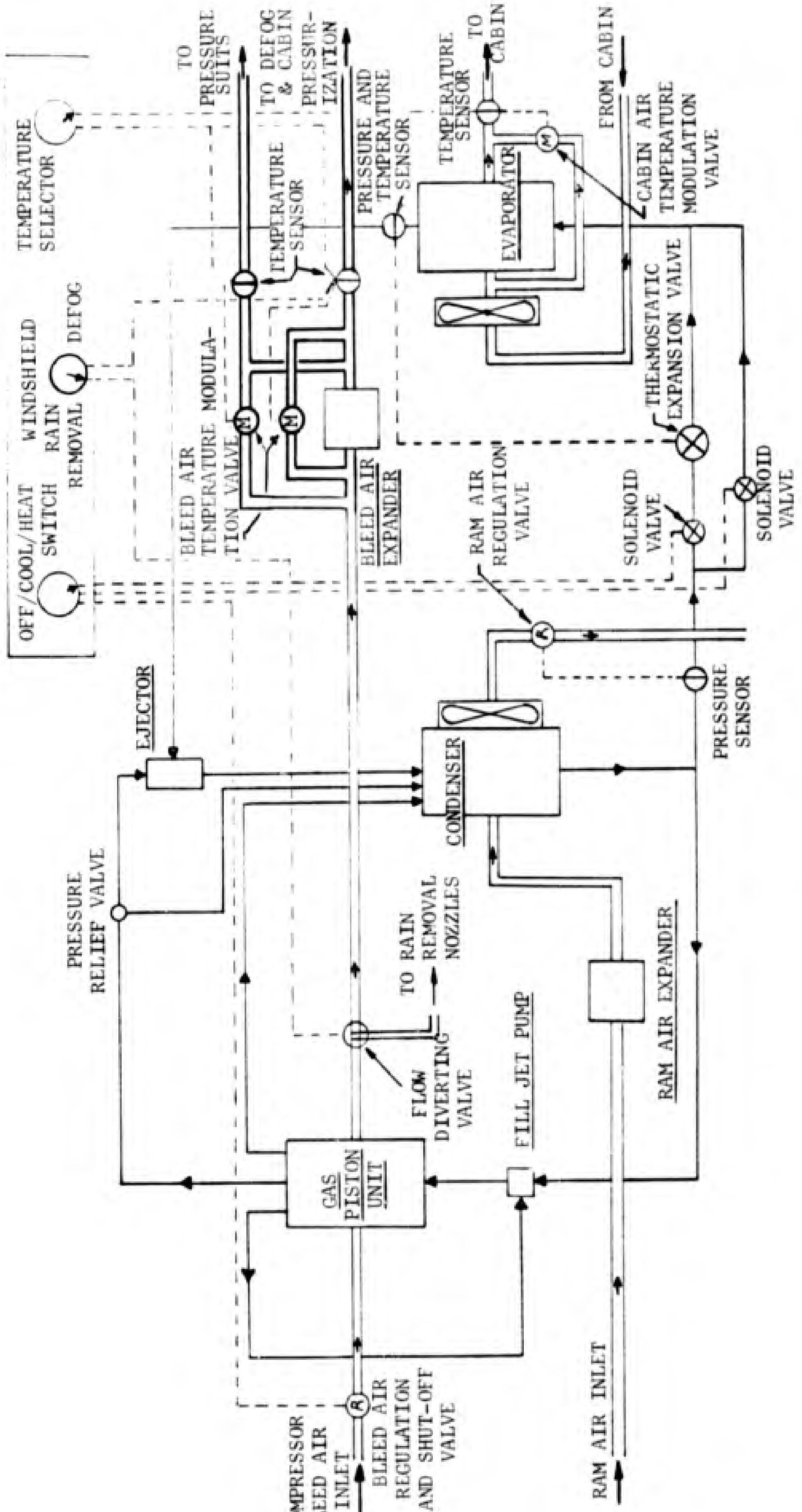
Figures 6.13, 6.14, and 6.15 are schematic diagrams for System "A", Systems "B-1" and "B-2", and System "C", respectively, showing the location and type of control required for system operation. A brief description of the operational and control characteristics of the four Conductron system mechanizations follows:

For maximum cooling effect, all temperature modulating valves of the Conductron system are in a closed position. In the case of System "A", the main cabin cooling load is handled by recirculation of cabin air through the evaporator where the cabin air is cooled. The cool air is then returned to the cabin through the cabin air distribution system. Cabin pressurization, ventilation, and pressure-suit cooling is accomplished with compressor bleed air. The bleed air temperature is first reduced in the gas piston unit and is then further cooled by means of an air expander prior to being ducted to the cabin and/or pressure-suit distribution system. The cabin pressurization flow also serves to handle part of the cabin cooling load.

For Systems "B-1", "B-2", and "C", compressor bleed air is used for cabin cooling and pressurization, and pressure-suit control. The bleed air is first cooled in the gas piston unit and undergoes further cooling by passing over the evaporator and through an air expander before it is distributed to the cabin and/or pressure-suit loops.

Modulated operation of the Conductron environmental control system to meet the varying load demands of the cabin and pressure-suit is accomplished by the bleed air expander bypass modulating valve. This valve bypasses some of the hot compressor bleed air around the bleed air expander and mixes it with the bleed air expander discharge air stream. System "A" uses an additional evaporator bypass valve, which takes some of the recirculated cabin air around the evaporator and mixes it with the main flow downstream of the evaporator. These temperature-mixing valves are positioned in response to the cabin and pressure-suit temperature control signals.

The operation of the Conductron environmental control system in the <u>cooling mode</u> requires two basic control features: namely, regulation of the evaporator cooling capacity and of the minimum condenser pressure. This is accomplished in the following manner:

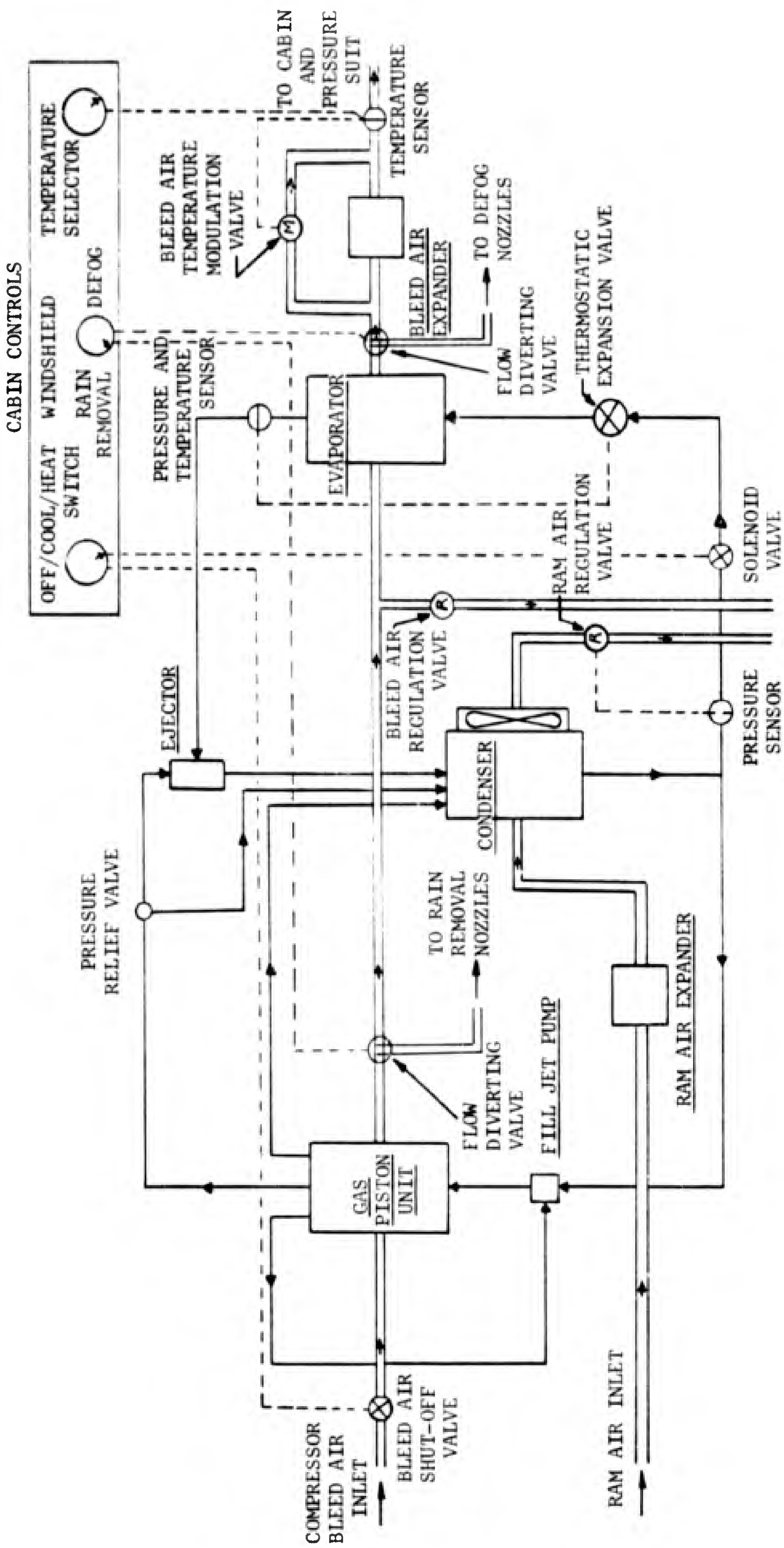



CONTROL SCHEMATIC

SYSTEM "A"



FIGURE 6.13

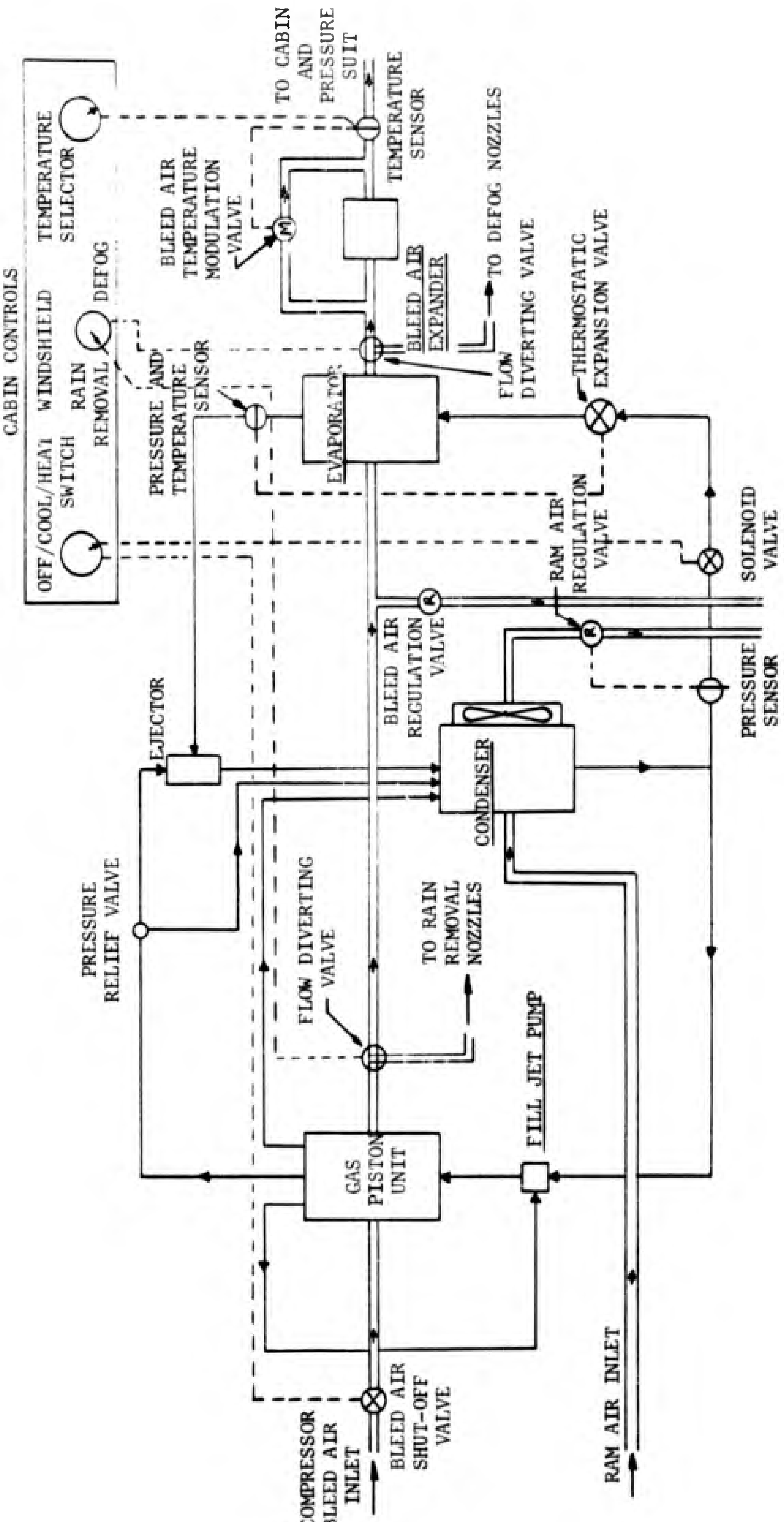



CONTROL SCHEMATIC

SYSTEMS "B-1" AND "B-2"



FIGURE 6.14




CONTROL SCHEMATIC
SYSTEM "C"

FIGURE 6.15

A solenoid valve, refrigerant filter, and a thermostatic expansion valve are placed in the refrigerant line upstream of the evaporator. The solenoid valve is actuated by a simple on-off switch. A temperature sensor in the evaporator outlet air stream limits the evaporating temperature to levels above the freezing point of water in order to avoid icing of the evaporator air side passages. This icing protection is required on System "A" only as the other mechanizations operate at temperatures well above the freezing temperature of water. The thermostatic expansion valve senses the evaporator refrigerant outlet pressure and temperature, and modulates the refrigerant flow rate through the evaporator so that a constant superheat condition is maintained. A pressure regulator in the condenser refrigerant outlet line will maintain the condenser pressure above the minimum level required to pass the amount of refrigerant through the expansion valve that is needed for the maximum cooling load condition. This pressure regulation can be accomplished by regulation of the condenser air flow. Since the Conductron system is designed for minimum available heat energy conditions to the GPU, any increase in compressor bleed air temperature and/or flow rate will result in an increase in motive vapor pressure and flow rate. This, in turn, will tend to increase the condensing pressure and the heat rejection capability of the condenser/heat exchanger; thus, the power loop will be essentially self-regulating. A pressure relief valve, vented to the condenser, is located in the converter motive vapor flow line to provide overload protection whenever excessive pressure conditions exist. In the advent of system malfunction, the compressor bleed air supply will be shut off and ram air admitted directly to the cabin.

Operation of the Conductron environmental control system in the <u>heating mode</u> is accomplished in the following way:

For maximum heating requirements, the temperature mixing valves are in a modulated position. Duct temperature limiters limit the air temperature entering the cabin and the pressure-suits to maximum values of 230°F and 120°F, respectively. System "A" also requires provisions for refrigerant bypass of the expansion valve, allowing liquid refrigerant to enter the evaporator at condenser temperatures, where heating of the recirculated cabin air takes place. A solenoid valve, actuated by a manually controlled heat/cool switch, is placed in the expansion valve bypass line. Air temperature control is accomplished by a temperature sensor in the evaporator outlet air (cabin inlet air) stream, which actuates the cabin air temperature modulation valve, controlling the amount of air bypassing the evaporator. The control loop on the condenser refrigerant outlet side serves to keep the condensing temperature at an adequate level for heating by regulation of the condenser air flow. Since the converter is more than adequate as a liquid jet pump for the required flow rates, the system will continue to operate in a normal manner.

For Systems "B-1", "B-2", and "C", heating is simply accomplished by bypassing the bleed air expander. Cabin air temperature control is accomplished by a temperature sensor located at the inlet to the cabin, which actuates a bleed air temperature modulation valve, thus controlling the amount of bleed air bypassing the bleed air expander. The remainder of the system controls operate in the same manner as in the cooling mode. The condenser pressure control loop assures that the evaporating temperature remains at a sufficiently high level to provide the necessary air temperatures for cabin heating.

The windshield rain removal function is accomplished in all Conductron system mechanizations by compressor bleed air leaving the gas piston unit. This provides air flow at a temperature of 350°F to 450°F for rain removal purposes. In order to provide higher mass flow rates, additional compressor bleed air can be mixed with this cooler bleed air, resulting in higher temperatures (450°F - 700°F). Windshield overtemperature protection must be provided. System "A" will continue to provide cabin cooling during periods of rain removal, since bleed air is not the source of cabin cooling air as in the other three mechanizations.

During defogging of the cabin transparencies using System "A", the temperature of the bleed air used for cabin pressurization and ventilation is automatically controlled to a temperature level of approximately 150°F. With Systems "B-1", "B-2", and "C", part of the bleed air leaving the evaporator is simply diverted to the defogging nozzles, resulting in defogging air temperatures of approximately 160°F. All of the system mechanizations will continue to provide cabin and pressure-suit cooling during defogging periods, since the defogging flow requirements are considerably less than the available bleed air flow rates.

Each system mechanization employs a simple flow limiting venturi in the compressor bleed air supply line to regulate the compressor bleed air flow. For System "A", this venturi will simultaneously regulate the bleed air flow to the cabin (cabin pressurization, pressure-suit ventilation, and defogging). The cabin air flow for Systems "B-1", "B-2", and "C" can be regulated by the bleed air expander nozzle and/or an orifice in the bleed air expander hot air bypass line.

## 6.4 System Size and Weight

Estimates on the total weight of the four Conductron environmental control system mechanizations are summarized in Table 6-3, and a detailed breakdown of component sizes and weights is given in Table 6-4. Two of the system mechanizations result in total weights, which are below the 126 lbs of the air cycle system presently installed in the F-4 fighter aircraft. If the ground cooling requirements are met by auxiliary ground equipment rather than the aircraft environmental control system, then the total system weight of each Conductron system mechanization can be reduced by the elimination of the condenser cooling fans. This weight savings is reflected in Table 6-3.

TABLE 6-3

TOTAL WEIGHT ESTIMATES

| | WITH GROUND COOLING CAPABILITY | WITHOUT GROUND COOLING CAPABILITY |
|---|---|---|
| System "A" | 124.8 lbs | 112.8 lbs |
| System "B-1" | 171.1 lbs | 159.1 lbs |
| System "B-2" | 134.4 lbs | 130.2 lbs |
| System "C" | 105.6 lbs | 101.4 lbs |

TABLE 6-4

WEIGHT AND SIZE COMPARISON

| | SYSTEM "A" | | SYSTEM "B-1" | | SYSTEM "B-2" | | SYSTEM "C" | |
|---|---|---|---|---|---|---|---|---|
| | Size | Wt. (lbs) | Size | Wt. (lbs) | Size | Wt. (lbs) | Size | Wt. (lbs) |
| GAS PISTON UNIT | | | | | | | | |
| Heat Exchanger | 4½" x 4½" x 6" | 7.0 | 5½" x 5½"x 11½" | 18.3 | 5" x 5" x 8½" | 11.0 | 5" x 5" x 7¼" | 9.7 |
| Gas Piston Compressor | 4¼ dia x 4¼" | 5.2 | 6½ dia x 6¼" | 13.1 | 5½ dia x 5½" | 9.1 | 5 " dia x 5 " | 9.5 |
| Fill Jet Pump | ¾" dia x 3" | 0.7 | 1¼" dia x 4½" | 1.0 | 1" dia x 4" | 0.9 | ⅞" dia x 3½" | 0.8 |
| Brackets, etc. | -- | 0.7 | -- | 1.8 | -- | 1.1 | -- | 1.0 |
| | | 13.6 | | 34.2 | | 22.1 | | 21.0 |
| CONDENSER | | | | | | | | |
| Heat Exchanger | 30" x 13" x 5" | 29.0 | 48" x 15" x 5¼" | 57.3 | 38" x 15" x 5" | 42.4 | 45" x 15" x 5" | 50.1 |
| Fan | 16¼" dia x 3¾" | 12.0 | 16¼" dia x 3¾" | 12.0 | 8¼" dia x 2½" | 4.2 | 8¼" dia x 2½" | 4.2 |
| Brackets, etc. | -- | 2.9 | -- | 5.7 | -- | 4.2 | -- | 5.0 |
| | | 43.9 | | 75.0 | | 50.8 | | 59.3 |
| EVAPORATOR | | | | | | | | |
| Heat Exchanger | 8" x 7" x 5" | 5.8 | 8" x 8" x 5½" | 7.6 | 7" x 6" x 5" | 4.5 | 5" x 4" x 6½" | 2.8 |
| Fan | 7" dia x 2¼" | 4.4 | ——— | - | ——— | - | ——— | - |
| Brackets, etc. | -- | 0.6 | -- | 0.8 | -- | 0.5 | -- | 0.3 |
| | | 10.8 | | 8.4 | | 5.0 | | 3.1 |
| CONVERTER | 1¾" dia x 12" | 3.1 | 2¼" dia x 16½" | 3.7 | 2" dia x 13½" | 3.3 | 2" dia x 15" | 3.5 |
| BLEED AIR EXPANDER | 5½" dia x 6¾" | 4.4 | 7" dia x 7¼" | 6.0 | 5" dia x 7" | 3.7 | 5" dia x 7" | 3.7 |
| RAM AIR EXPANDER | 9" dia x 9¼" | 32.0 | 12" dia x 11 " | 28.8 | 12" dia x 14" | 34.5 | -- | - |
| CONTROLS, VALVES, ETC | -- | 17.0 | -- | 15.0 | -- | 15.0 | -- | 15.0 |
| TOTAL | | 124.8 | | 171.1 | | 134.4 | | 105.6 |

The heat exchanger weights shown are based on state-of-the-art compact heat exchanger designs using plate-fin construction. An approximation of manifold weight equal to 25% of the core weight is included. Bracket weights are taken as 10% of the total heat exchanger weight.

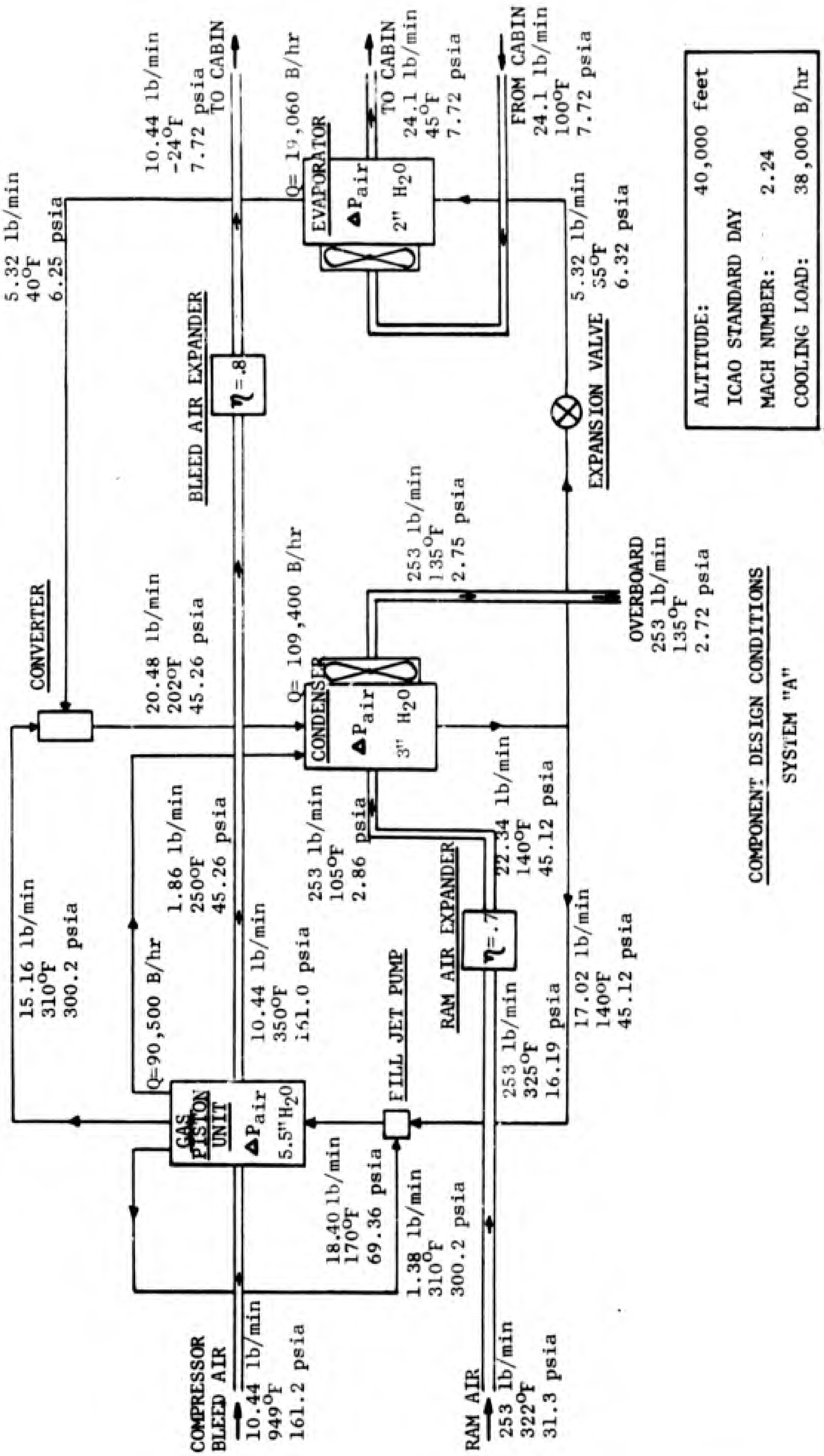
Compressor bleed air expander weights are calculated on the basis of 0.2 lbs per horsepower. This approximation is based on state-of-the-art air expansion turbine weight characteristics (Reference 15). Ram air expander weights are based on the approximation of 0.1 lbs per horsepower as given by Reference 15. This lower weight factor is made possible by the lower structural requirements due to the relatively low ram air pressures. Bleed air and ram air rotary-vane motor weights are based on weight factors of 0.30 lbs per horsepower and 0.15 lbs per horsepower, respectively. These approximations are based on presently available air motors, corrected for magnesium rather than steel construction, including projected improvements in air motor efficiency (from 30% to 50%).

The fan weights are based on off-the-shelf aircraft fan designs. Fill-jet pump, converter, controls, and valve weights are based on in-house prototype and breadboard models. The GPC (Gas Piston Compressor) weight is based on a cycling rate of 10 cycles/minute and an allowance of 20% for the volume of the float mechanism and unusable expansion volume. These assumptions are derived from prototype GPC designs. Air motors of aluminum construction can also be used at a weight penalty of approximately 1.5.

## 6.5 Component Design Specifications

The system heat balances, refrigerant and air flow rates, thermodynamic statepoints, and other component characteristics of the four Conductron system mechanizations at their respective maximum operating points at 40,000-ft altitude on an ICAO Standard Day are shown in Figures 6.16 through 6.19. Refrigerant pressure drops through the heat exchangers are included. Other refrigerant line losses are considered to be negligible, due to the compact packaging requirements of a F-4 fighter aircraft installation.

Figure 6.16 shows that System "A" can deliver 38,000 BTU/hr of cooling capacity on an ICAO Standard Day at 40,000-ft altitude and a maximum flight Mach number of 2.24. System "B-1" (Figure 6.17) produces 32,500 BTU/hr of cooling at 40,000-ft altitude and a maximum Mach number of 2.11 (ICAO Standard Day). The maximum operating point for System "B-2" (Figure 6.18) is identical to System "A" (38,000 BTU/hr cooling capacity; 40,000-ft altitude; Mach 2.24; ICAO Standard Day). System "C" (Figure 6.19) can produce 25,000 BTU/hr of cooling at a maximum flight Mach number of 1.96 (design altitude: 40,000 ft - ICAO Standard Day). In addition to an indication of the relative performance capabilities of each Conductron system mechanization, Figures 6.16 through 6.19 show the required component characteristics for each system. These component design specifications are summarized in Table 6-5 for each candidate system.

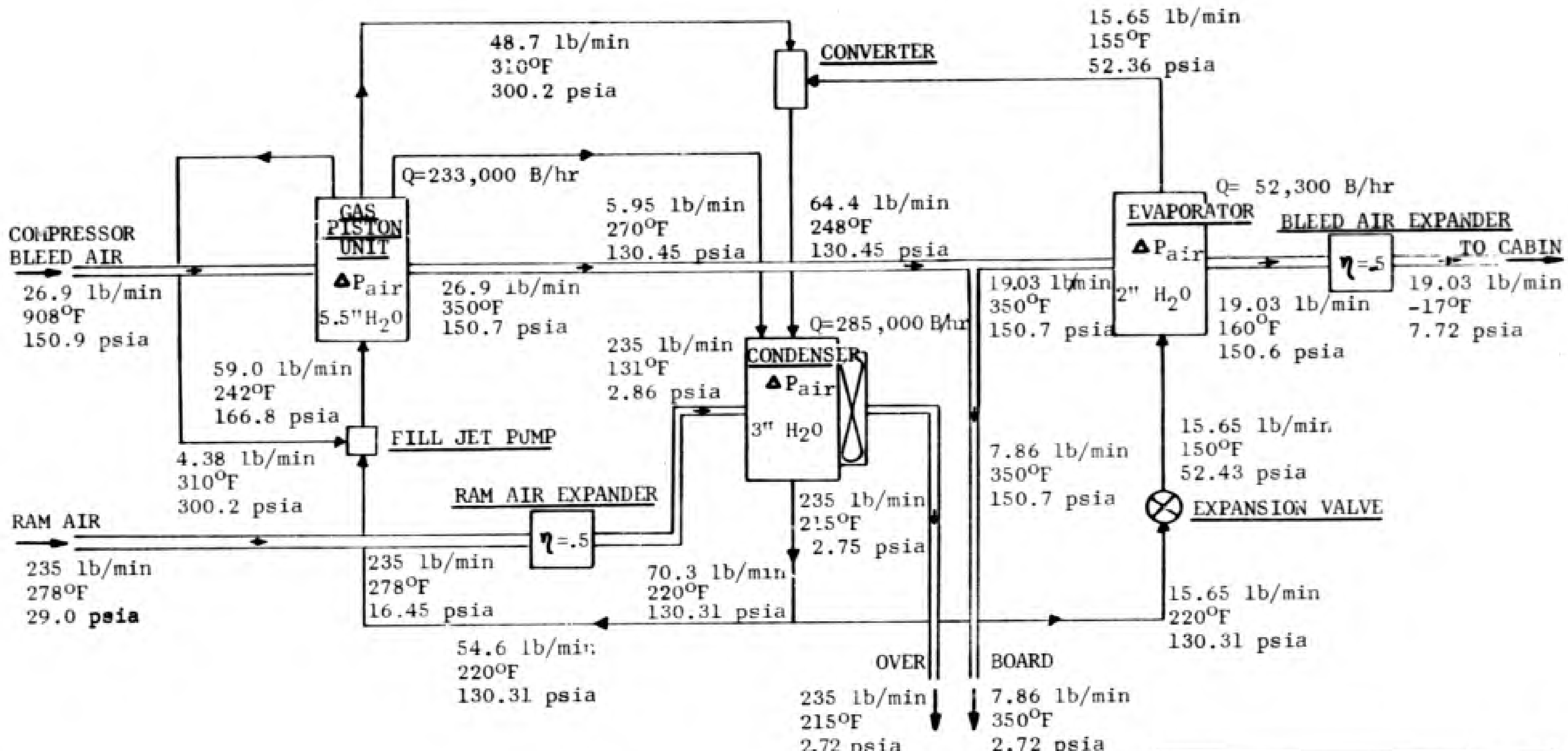



COMPONENT DESIGN CONDITIONS

SYSTEM "A"



FIGURE 6.16

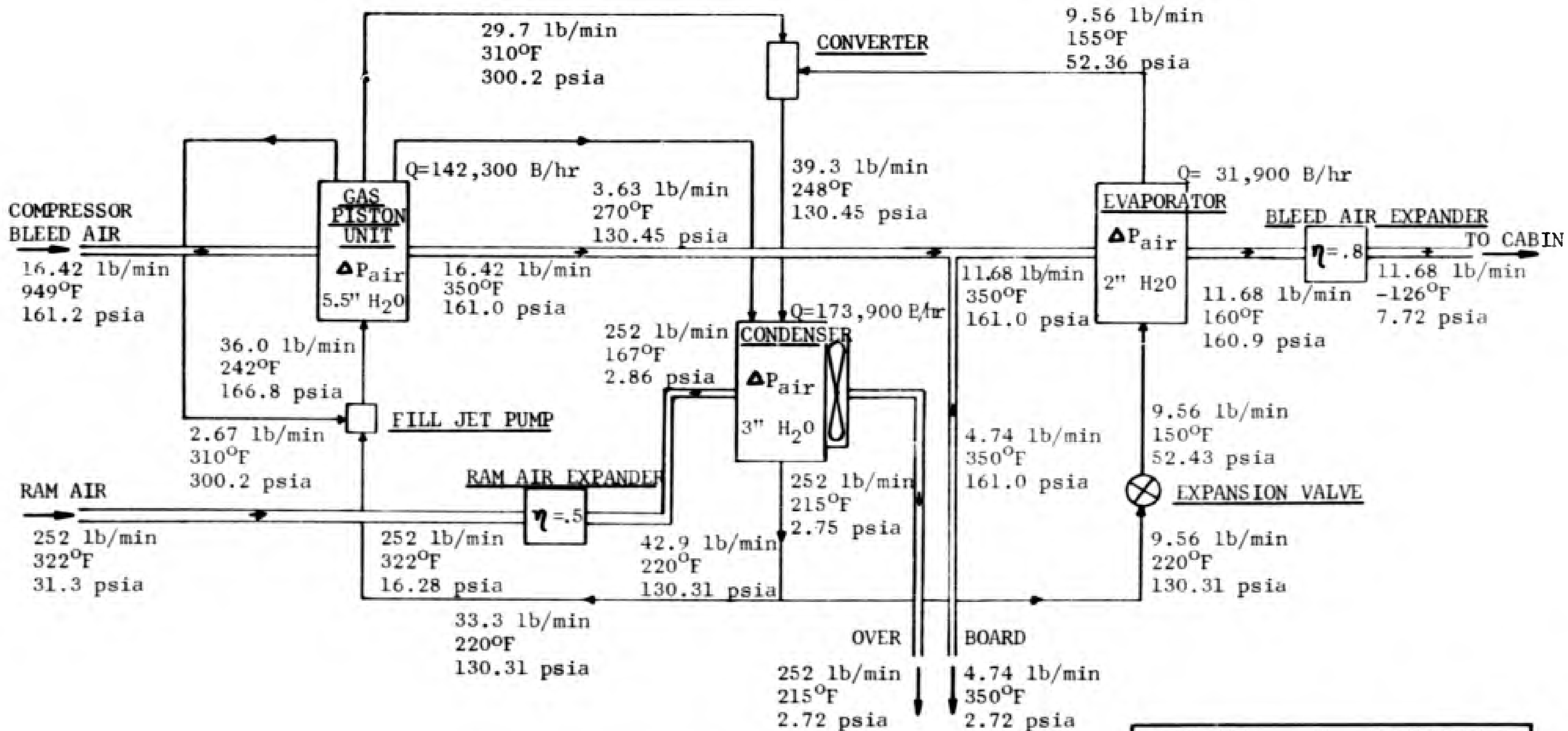



| | |
|---|---|
| ALTITUDE: | 40,000 feet |
| ICAO STANDARD DAY | |
| MACH NUMBER: | 2.11 |
| COOLING LOAD: | 32,500 B/hr |

COMPONENT DESIGN CONDITIONS

SYSTEM "B-1"

FIGURE 6.17



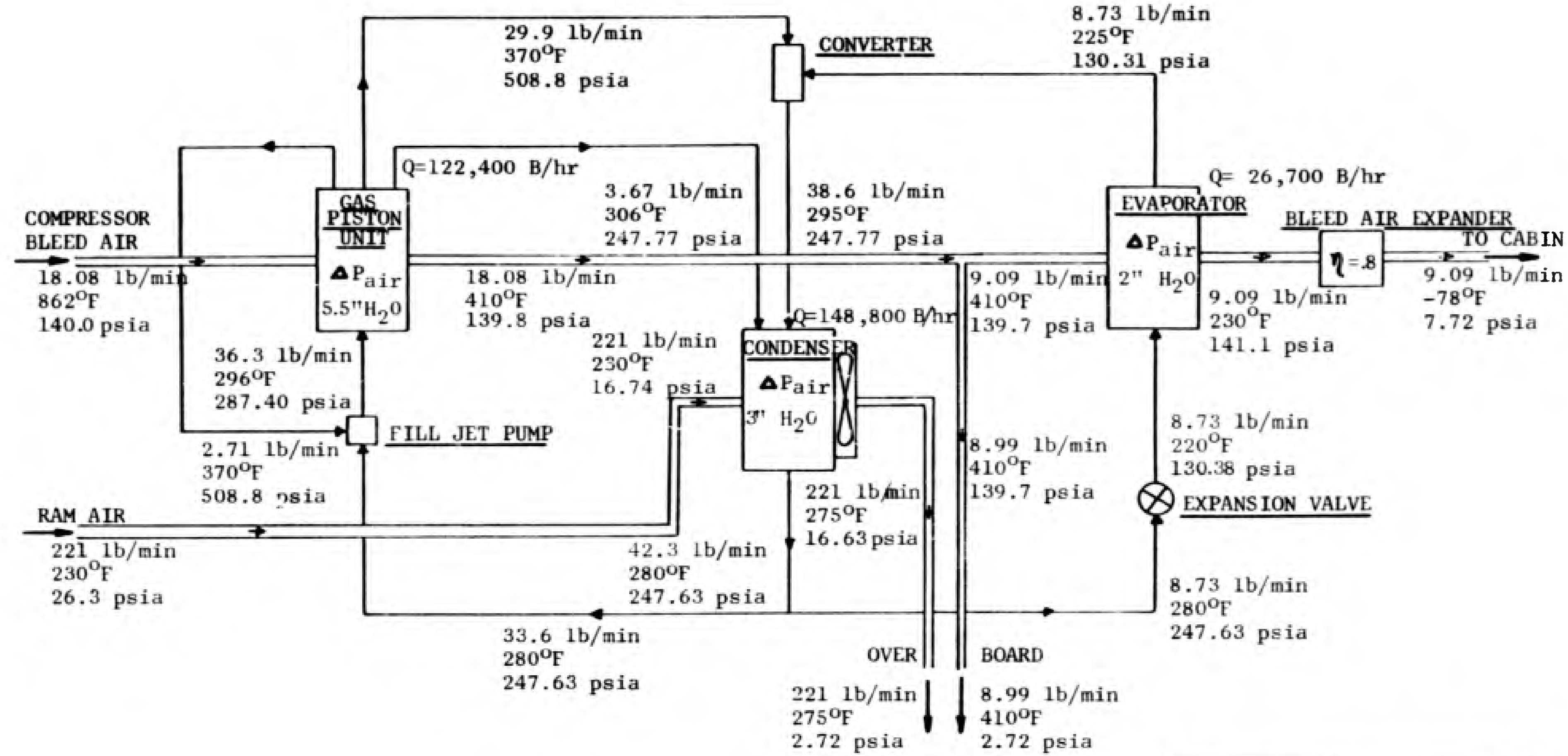

| | |
|---|---|
| ALTITUDE: | 40,000 feet |
| ICAO STANDARD DAY | |
| MACH NUMBER: | 2.24 |
| COOLING LOAD: | 38,000 B/hr |

COMPONENT DESIGN CONDITIONS

SYSTEM "B-2"



FIGURE 6.18




ALTITUDE: 40,000 feet
ICAO STANDARD DAY
MACH NUMBER: 1.96
COOLING LOAD: 25,000 B/hr

COMPONENT DESIGN CONDITIONS

SYSTEM "C"



FIGURE 6.19

TABLE 6-5

COMPONENT DESIGN SPECIFICATIONS

| | SYSTEM "A" | SYSTEM "B-1" | SYSTEM "B-2" | SYSTEM "C" |
|---|---|---|---|---|
| GAS PISTON UNIT | | | | |
| Heat Load, (BTU/hr) | 90,500 | 237,000 | 142,300 | 124,400 |
| Effectiveness, | .937 | .933 | .937 | .919 |
| Weight, (lbs) | 12.2 | 31.4 | 20.1 | 19.2 |
| Air Side | | | | |
| Flow Rate, (lbs/min) | 10.44 | 26.9 | 16.42 | 18.08 |
| Inlet Temperature, (°F) | 949 | 908 | 949 | 862 |
| Outlet Temperature, (°F) | 350 | 350 | 350 | 410 |
| Pressure Drop, (in $H_2O$) | 5.5 | 5.5 | 5.5 | 5.5 |
| Refrigerant Side | | | | |
| Flow Rate, (lbs/min) | 18.40 | 59.0 | 36.0 | 36.3 |
| Inlet Temperature, (°F) | 170 | 242 | 242 | 296 |
| Inlet Pressure, (psi) | 69.36 | 166.8 | 166.8 | 287.4 |
| Outlet Temperature, (°F) | 310 | 310 | 310 | 370 |
| Outlet Pressure, (psi) | 300.2 | 300.2 | 300.2 | 508.8 |
| CONDENSER | | | | |
| Heat Load, (BTU/hr) | 109,600 | 290,000 | 173,900 | 151,300 |
| Effectiveness , | .857 | .944 | .906 | .900 |
| Weight, (lbs) | 29.0 | 57.3 | 42.4 | 50.1 |
| Air Side | | | | |
| Flow Rate, (lbs/min) | 253 | 235 | 252 | 221 |
| Inlet Temperature, (°F) | 105 | 131 | 167 | 230 |
| Outlet Temperature, (°F) | 135 | 215 | 215 | 275 |
| Pressure Drop, (in $H_2O$) | 3.0 | 3.0 | 3.0 | 3.0 |
| Refrigerant Side | | | | |
| Flow Rate, (lbs/min) | 22.3 | 70.3 | 42.9 | 42.3 |
| Inlet Temperature, (°F) | 206 | 250 | 250 | 296 |
| Inlet Pressure, (psi) | 45.26 | 130.5 | 130.5 | 247.8 |
| Outlet Temperature, (°F) | 140 | 220 | 220 | 280 |
| Outlet Pressure, (psi) | 45.12 | 130.3 | 130.3 | 247.6 |
| CONDENSER FAN | | | | |
| Heat Load, (BTU/hr) | 68,800 | 121,000 | 212,000 | 121,000 |
| Inlet Temperature, (°F) | 103 | 103 | 103 | 103 |
| Inlet Pressure, (psi) | 14.69 | 14.69 | 14.69 | 14.69 |
| Outlet Temperature, (°F) | 125 | 195 | 165 | 195 |
| Pressure Head, (in $H_2O$) | 3.0 | 3.0 | 3.0 | 3.0 |
| Weight, (lbs) | 12.0 | 4.2 | 12.0 | 4.2 |

TABLE 6-5

(Continued)

COMPONENT DESIGN SPECIFICATIONS

| | SYSTEM "A" | SYSTEM "B-1" | SYSTEM "B-2" | SYSTEM "C" |
|---|---|---|---|---|
| EVAPORATOR | | | | |
| Heat Load, (BTU/hr) | 19,100 | 53,100 | 31,900 | 18,830 |
| Effectiveness, | .917 | .974 | .974 | .973 |
| Weight, (lbs) | 5.8 | 7.6 | 4.5 | 2.8 |
| Air Side | | | | |
| Flow Rate, (lbs/min) | 24.1 | 19.03 | 11.68 | 9.09 |
| Inlet Temperature, (°F) | 100 | 350 | 350 | 410 |
| Outlet Temperature, (°F) | 45 | 160 | 160 | 230 |
| Pressure Drop, (in $H_2O$) | 2.0 | 2.0 | 2.0 | 2.0 |
| Refrigerant Side | | | | |
| Flow Rate, (lbs/min) | 5.32 | 15.65 | 9.56 | 8.73 |
| Inlet Temperature, (°F) | 35 | 150 | 150 | 220 |
| Inlet Pressure, (psi) | 6.32 | 52.43 | 52.43 | 130.38 |
| Outlet Temperature, (°F) | 40 | 155 | 155 | 225 |
| Outlet Pressure, (psi) | 6.25 | 52.36 | 52.36 | 130.31 |
| EVAPORATOR FAN | | | | |
| Heat Load, (BTU/hr) | 12,000 | NOT APPLICABLE | | |
| Inlet Temperature, (°F) | 70 | | | |
| Inlet Pressure, (psia) | 14.69 | | | |
| Outlet Temperature, (°F) | 45 | | | |
| Pressure Head, (in $H_2O$) | 2.0 | | | |
| Weight, (lbs) | 3.4 | | | |
| CONVERTER | | | | |
| Converter Efficiency, | .35 | .35 | .35 | .35 |
| Motive Flow Rate, (lbs/min) | 15.16 | 48.7 | 29.7 | 29.9 |
| Motive Pressure, (psia) | 300.2 | 300.2 | 300.2 | 508.8 |
| Motive Temperature, (°F) | 310 | 310 | 310 | 370 |
| Suction Flow Rate, (lbs/min) | 5.32 | 15.65 | 9.56 | 8.73 |
| Suction Pressure, (psia) | 6.25 | 52.36 | 52.36 | 130.31 |
| Suction Temperature, (°F) | 40 | 155 | 155 | 225 |
| Exhaust Flow Rate, (lbs/min) | 20.48 | 64.4 | 39.3 | 38.6 |
| Exhaust Pressure, (psia) | 45.36 | 130.45 | 130.45 | 247.8 |
| Exhaust Temperature, (°F) | 202 | 248 | 248 | 295 |
| Weight, (lbs) | 3.1 | 3.7 | 3.3 | 3.5 |

TABLE 6-5

(Continued)

COMPONENT DESIGN SPECIFICATIONS

| | SYSTEM "A" | SYSTEM "B-1" | SYSTEM "B-2" | SYSTEM "C" |
|---|---|---|---|---|
| FILL JET PUMP | | | | |
| Adiabatic Efficiency, | .98 | .98 | .98 | .98 |
| Motive Flow Rate, (lbs/min) | 1.38 | 4.38 | 2.67 | 2.71 |
| Motive Pressure, (psia) | 300.2 | 300.2 | 300.2 | 508.8 |
| Motive Temperature, (°F) | 310 | 310 | 310 | 370 |
| Suction Flow Rate, (lbs/min) | 17.02 | 54.6 | 33.3 | 33.6 |
| Suction Pressure, (psia) | 45.12 | 130.31 | 130.31 | 247.63 |
| Suction Temperature, (°F) | 140 | 220 | 220 | 280 |
| Exhaust Flow Rate, (lbs/min) | 18.40 | 59.0 | 360 | 36.3 |
| Exhaust Pressure, (psia) | 69.36 | 166.8 | 166.8 | 287.4 |
| Exhaust Temperature, (°F) | 170 | 242 | 242 | 296 |
| Weight, (lbs) | 0.7 | 1.1 | 0.9 | 0.8 |
| THERMOSTATIC EXPANSION VALVE | | | | |
| Flow Rate, (lbs/min) | 5.32 | 15.65 | 9.56 | 8.73 |
| Inlet Temperature, (°F) | 140 | 220 | 220 | 280 |
| Inlet Pressure, (psia) | 45.12 | 130.31 | 130.31 | 247.63 |
| Outlet Temperature, (°F) | 35 | 150 | 150 | 220 |
| Outlet Pressure, (psia) | 6.32 | 52.43 | 52.43 | 130.38 |
| Weight, (lbs) | 1.2 | 2.4 | 1.8 | 2.0 |
| COMPRESSOR BLEED AIR EXPANDER | | | | |
| Flow Rate, (lbs/min) | 10.44 | 19.03 | 11.68 | 9.09 |
| Inlet Temperature, (°F) | 350 | 160 | 160 | 230 |
| Inlet Pressure, (psia) | 161.0 | 150.6 | 160.9 | 141.1 |
| Outlet Temperature, (°F) | -24 | -17 | -126 | -78 |
| Outlet Pressure, (psia) | 7.72 | 7.72 | 7.72 | 7.72 |
| Adiabatic Efficiency, | .80 | .50 | .80 | .80 |
| Weight, (lbs) | 4.4 | 6.0 | 3.7 | 3.7 |
| RAM AIR EXPANDER | | | | |
| Flow Rate, (lbs/min) | 253 | 235 | 252 | NOT APPLICABLE |
| Inlet Temperature, (°F) | 325 | 278 | 325 | |
| Inlet Pressure, (psia) | 16.19 | 16.45 | 16.28 | |
| Outlet Temperature, (°F) | 105 | 131 | 167 | |
| Outlet Pressure, (psia) | 2.86 | 2.86 | 2.86 | |
| Adiabatic Efficiency, | .70 | .50 | .50 | |
| Weight, (lbs) | 32.0 | 28.8 | 34.5 | |

## 7.0 CONCLUSIONS and RECOMMENDATIONS

This report has defined the advantages as well as limitations of the heat-actuated Conductron environmental control system in relation to its applicability as the cabin air conditioning system for the Navy F-4 fighter aircraft. The general conclusion from this study is that it is feasible to adopt the Conductron system to the requirements of the Navy F-4 fighter aircraft. Furthermore, it is shown that such a system is superior in performance and results in lower weight penalties than the air cycle systems presently used in the F-4 fighter aircraft. Based on the results of this study, a recommendation can be made for the initiation of a Phase II effort for the evaluation of a breadboard-type laboratory system as a logical continuation of a step-by-step development of the Conductron system for the Navy F-4 fighter aircraft. The following discussion summarizes specific conclusions gained from this study and presents a brief discussion of the recommendations made.

Based on the results of this feasibility study, the following <u>conclusions</u> can be reached:

1) The Conductron environmental control concept can be adapted to the Navy F-4 fighter aircraft requirements without exceeding the existing weight, size, and power constraints.

2) A Conductron environmental control system incorporating air expansion turbines can meet the air conditioning requirements of the Navy F-4 fighter aircraft at all operating conditions; such a system would require state-of-the-art improvement of converters only.

3) A system excluding high-speed turbine machinery and using rotary-vane motors can meet most performance requirements except the maximum cooling load conditions; this system would require state-of-the-art improvement of converter and air motor performance characteristics.

4) The projections for state-of-the-art improvements of critical components are realistic. The timely development of these components within the framework of an airborne system development plan should be predictable with a high degree of confidence.

5) All Conductron system mechanizations result in significant reductions of compressor bleed air requirements as compared with the present Navy F-4 air cycle system. The penalty imposed on aircraft performance characteristics by the installation of air conditioning

equipment is lowered by the Conductron ECS, since compressor bleed air extraction is directly related to a loss of engine thrust.

6) The performance capability of the Conductron ECS is superior to air cycle systems. In addition to adequate environmental control at extreme flight conditions, ground cooling can be provided.

7) The weight of the Conductron ECS is comparable to the air cycle system weight; with certain system mechanizations weights lower than those of present air cycle systems can be realized.

8) The Conductron ECS can be packaged in such a way as to fit the existing air conditioning system installation constraints of the F-4 fighter aircraft.

9) No electrical or other external power supplies are required for the operation of the Conductron system.

10) The Conductron concept for environmental control is adaptable to other high-performance Navy aircraft, although the specific results of this analysis are applicable only to the Navy F-4 fighter aircraft.

11) Since this study considered the feasibility of the Conductron system to the environmental control of an aircraft with the most severe operational and performance requirements, air conditioning requirements of any other aircraft, particularly with subsonic performance characteristics, should be met more easily.

12) Excellent reliability and maintainability characteristics of the Conductron ECS can be projected, based on experience gained so far in the development of military and commercial ground air conditioning systems.

13) Development of the unique Conductron concept for aircraft applications is recommended.

The following specific recommendations can be made, based on the results of this feasibility study:

1) Phase II (Laboratory Test Unit) of the step-by-step evaluation and development of the Conductron environmental control concept should be initiated.

2) This program should consist of the design, fabrication, and testing of a breadboard-type laboratory unit of the F-4 fighter aircraft. This breadboard unit would utilize off-the-shelf and state-of-the-art components.

3) The objective of the Phase II program would be to evaluate the present capabilities of state-of-the-art components, and to demonstrate system operation at aircraft conditions simulating ground-static and sea-level flight.

4) In addition, it would be desirable to demonstrate the air-worthiness of certain system components, particularly the Gas Piston Unit. Simulation of aircraft vibration, shock, and attitude is feasible and desirable.

5) Development of air motors (rotary-vane motors) for use as air expanders should be undertaken on an exploratory basis in order to establish the real potential of these low speed machines.

6) Completion of such a Phase II Experimental Program would result in the required data on component and system performance which would be the basis for the design, fabrication, and testing of a full-scale field or flight test unit of a Conductron F-4 environmental control system.

REFERENCES


1. "Cabin and Pressure-Suit Air Conditioning System for F4H-1 Fighter Aircraft", McDonnell Specification 32-83006.

2. "F-4B Refrigeration System Design Operating Conditions and Requirements", McDonnell Specification Control Drawing 32-83062, Sheets 62m, 62n, and 62p.

3. "General Requirements for Environmental Systems of Pressurized Aircraft", MIL-E-18927D (Wep), 29 December 1961.

4. "F-4 Cooling Equipment Design Requirements", letter from McDonnell Aircraft, 27 December 1965.

5. Perry, J. A., Jr., "Critical Flow Through Sharp-Edged Orifices", Transactions of the ASME, p. 757 - 764, October 1949.

6. "ICAO Standard Atmosphere", National Aeronautics and Space Administration, U. S. Air Force, U. S. Weather Bureau; - U. S. Government Printing Office, 1962.

7. "Climatic Extremes for Military Equipment", MIL-STD-210A, 2 August 1957.

8. Personal communication with D. H. Lanahan of McDonnell Aircraft Corporation, 11 February 1966.

9. "Aerospace Applied Thermodynamics Manual", Society of Automotive Engineers, Inc., Committee A-9, February 1960.

10. Kays, W. M., and London, A.L., "Compact Heat Exchangers", Second Edition, McGraw-Hill, 1964.

11. "Leiman Rotary Air Motor — Model 14-300 — Performance Curves", Leiman Bros., Inc.

12. Cress, H. A.; et al, "The Design and Development of a Gas-Fueled Double-Loop Refrigeration System", Battelle Memorial Institute, 5 February 1957.

13. Teichmann, O. E., "Analysis and Design of Air Motors", Product Engineering, p. 167 - 178, February 1957.

14. Personal communication with L. Gunther of McDonnell Aircraft Corporation, 30 June 1966.

15. Personal communication with The Garrett Corporation, AiResearch Manufacturing Division personnel, July 1966.

REFERENCES (Cont'd)


16. "ASHRAE Guide and Data Book - Fundamentals and Equipment", American Society of Heating, Refrigerating, and Air Conditioning Engineers, 1965.

17. Turnblade, R. C., and Carlson, G. R., "Some Results of the Conductron Converter Development Program" (Proprietary Information), Conductron Internal Progress Report, 15 July 1966.

18. Kreith, F., "Principles of Heat Transfer", Second Edition, National Textbook Company, 1965.

19. "Thermodynamic Properties of Freon-11", E. I. DuPont De Nemours and Company.

## APPENDIX "A"

## REFRIGERANT COMPARISON

Contained in this section are the calculations upon which the refrigerant comparison is based.

For this analysis, the evaporator temperature was assumed constant at 40°F with 10°F superheat, and the condenser temperature was assumed constant at 140°F. The boiler temperature is unique for each refrigerant and was chosen with respect to each refrigerant's thermodynamic limitations. The ideal ejector performance is based on the ejector analysis described in Appendix "B".

NOMENCLATURE:

| | | |
|---|---|---|
| h | enthalpy | BTU/lbm |
| P | pressure | psia |
| Q | heat load | BTU/hr |
| t | temperature | °F |
| $\dot{W}$ | flow rate | lb/min |
| $\lambda$ | entrainment ratio | $\dot{W}_s/\dot{W}_m$ |
| COP | Coefficient of Performance | $Q_s/Q_b$ |

Subscripts:

| | |
|---|---|
| 1 | boiler inlet |
| 3 | boiler outlet |
| 5 | evaporator inlet |
| 6 | evaporator outlet |
| b | boiler |
| c | condenser |
| d | depressurization flow |
| f | fill-jet pump flow |
| s | evaporator |

R-11

$h_6 = 98.5$ ($10^oF$ S.H.)
$h_5 = 37.0$
$\Delta h_s = 61.5$ BTU/lb

$\dot{W}_s = 200/\Delta h_s$
$= 200/61.5$
$\dot{W}_s = 3.25$ lb/min

$P_s = 7.0$ psia
$P_c = 45.0$ psia

$P_c/P_s = 45/7 = 6.42$

at $t_b = 300^oF$
$P_b = 300$ psia

Ideal ejector performance from Figure B-2,

$\dot{W}_m/\dot{W}_s = 2.65$
$\dot{W}_m = (\dot{W}_m/\dot{W}_s)\dot{W}_s$

$\lambda_i = .377$

$\dot{W}_m = 2.65(3.25) = 8.61$ lb/min

$\dot{W}_d = .10\ \dot{W}_m = .1(8.61) = 0.86$ lb/min

$\dot{W}_f = .05\ \dot{W}_m = .05(8.61) = 0.43$ lb/min

$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = 9.90$ lb/min

$h_3 = 126.5$ ($10^oF$ S.H.)
$h_1 = 37.0$
$\Delta h_b = 89.5$ BTU/lb

$Q_b = \dot{W}_b\ \Delta h_b = 9.90(89.5)(60) = 53{,}163$ BTU/hr

$COP \approx \frac{12{,}000}{53{,}163} \approx 0.225$

$Q_c = Q_b + Q_s - \dot{W}_f(h_3 - h_1)$

$= 53{,}163 + 12{,}000 - (.43)(60)(89.5)$

$= 53{,}163 + 12{,}000 - 2{,}309$

$Q_c = 62{,}854$ BTU/hr

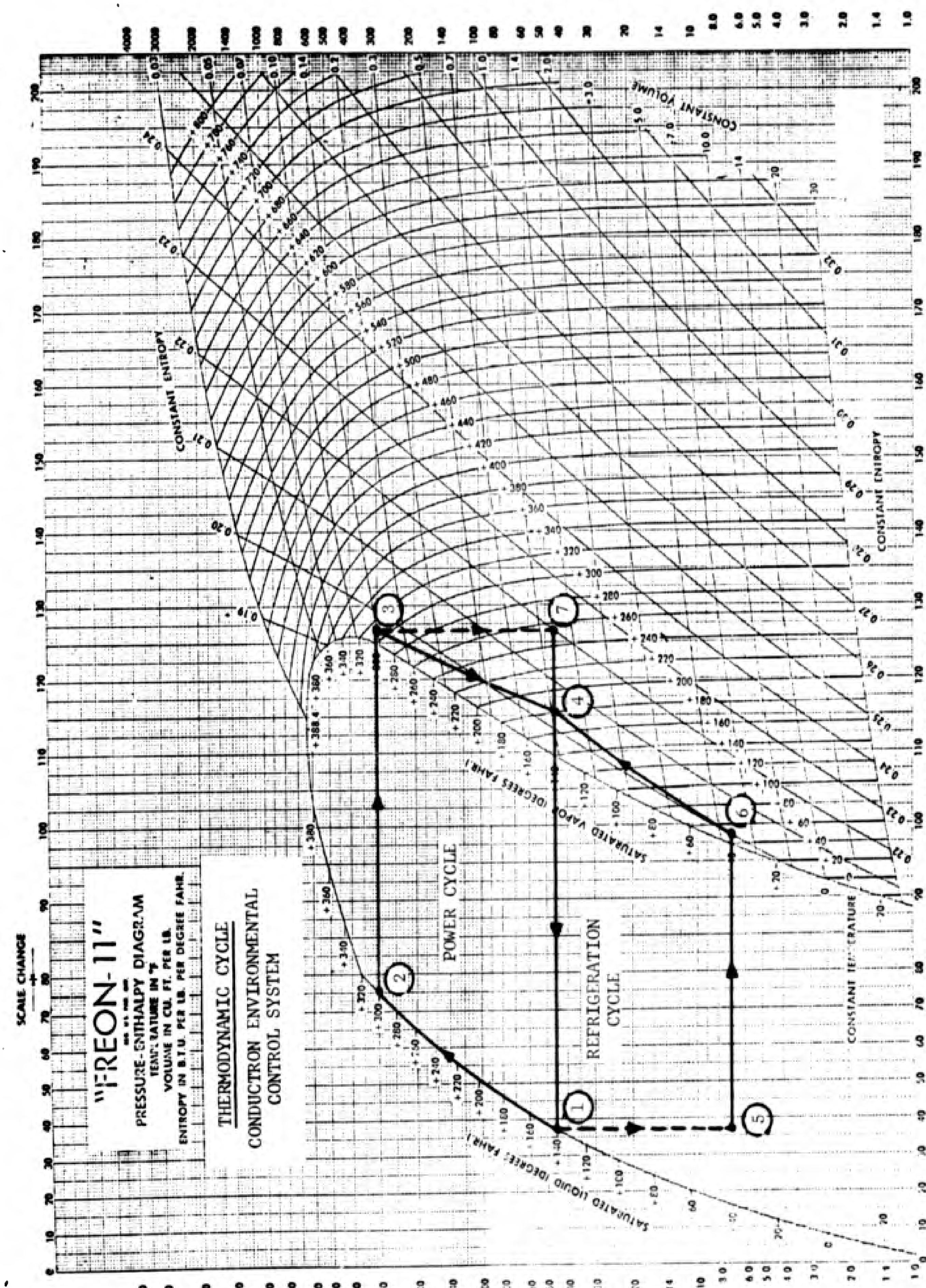

"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
CONDUCTRON ENVIRONMENTAL
CONTROL SYSTEM
POWER CYCLE
REFRIGERATION
CYCLE
CONSTANT ENTROPY
CONSTANT VOLUME
SATURATED VAPOR (DEGREES FAHR.)
SATURATED LIQUID (DEGREES FAHR.)
CONSTANT TEMPERATURE
SCALE CHANGE
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)



FIGURE A.1

R-12

$h_6 = 84.5$ ($10^oF$ S.H.)

$h_5 = 41.0$

$\Delta h_s = 43.5$ BTU/lb

$\dot{W}_s = 200/\Delta h_s$

$= 200/43.5$

$\dot{W}_s = 4.59$ lb/min

$P_s = 52$ psia

$P_c = 220$ psia

$P_c/P_s = 220/52 = 4.23$

at $t_b = 200^oF$

$P_b = 430$ psia

Ideal ejector performance from Figure B-2,

$\dot{W}_m/\dot{W}_s = 7.7$

$\lambda_i = .129$

$\dot{W}_m = 7.7(4.59) = 35.34$ lb/min

$\dot{W}_d = .10\ \dot{W}_m = 3.53$

$\dot{W}_f = .05\ \dot{W}_m = 1.77$

$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = 40.64$ lb/min

$h_3 = 95.5$ ($10^oF$ S.H.)

$h_1 = 41.0$

$\Delta h_b = 54.5$ BTU/lb

$Q_b = \dot{W}_b\,\Delta h_b = 40.64(54.5)(60) = 132,870$ BTU/hr

$$COP \approx \frac{12,000}{132,870} \approx .0903$$

$Q_c = Q_b + Q_s - \dot{W}_f(h_3-h_1)$

$= 132,870 + 12,000 - (1.77)(60)54.5$

$= 132,870 + 12,000 - 5,788$

$Q_c = 139,082$ BTU/hr

# PRESSURE-ENTHALPY DIAGRAM

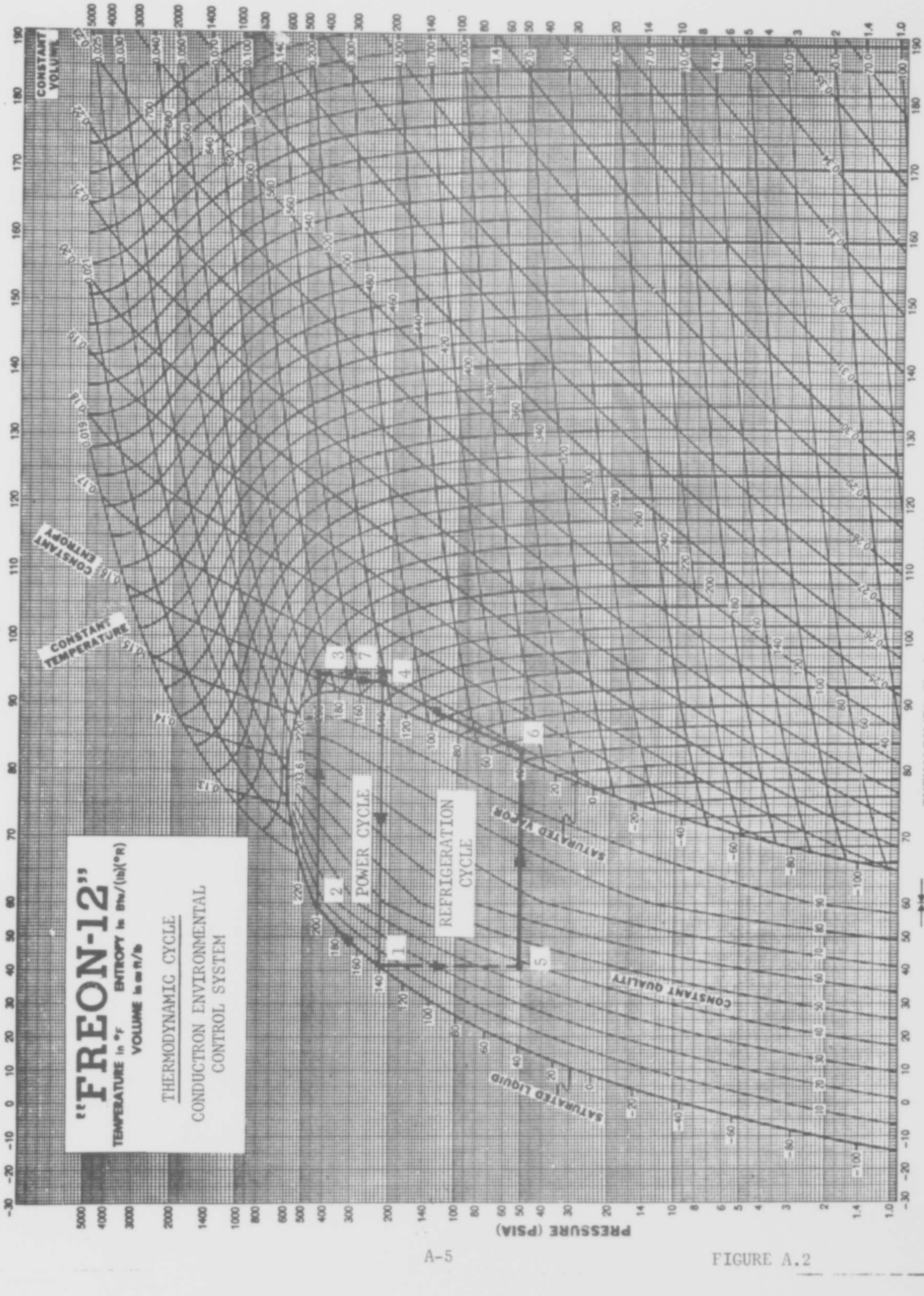





FIGURE A.2

R-21

$$h_6 = 126.5 \quad (10^oF \text{ S.H.})$$
$$h_5 = 45.0$$
$$\Delta h_s = 81.5 \quad \text{BTU/lb}$$

$$\dot{W}_s = 200/\Delta h_s$$
$$= 200/81.5$$
$$\dot{W}_s = 2.45 \text{ lb/min}$$

$$P_s = 13.0 \text{ psia}$$
$$P_c = 76.0 \text{ psia}$$

$$P_c/P_s = 76/13 = 5.85$$
$$\text{at } t_b = 250^oF$$
$$P_b = 300 \text{ psia}$$

Ideal ejector performance from Figure B-2,

$$\dot{W}_m/\dot{W}_s = 3.9$$

$$\lambda_i = .256$$

$$\dot{W}_m = 3.9(2.45) = 9.55 \text{ lb/min}$$

$$\dot{W}_d = .1\ \dot{W}_m = .96$$
$$\dot{W}_f = .05\ \dot{W}_m = .48$$

$$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = 10.99 \text{ lb/min}$$

$$h_3 = 146.0$$
$$h_1 = 45.0$$
$$\Delta h_b = 101.0 \quad \text{BTU/lb}$$

$$Q_b = \dot{W}_b\,\Delta h_b = 10.99(101)(60) = 66{,}594 \text{ BTU/hr}$$

$$\text{COP} \approx \frac{12{,}000}{66{,}594} \approx 0.180$$

$$Q_c = Q_b + Q_s - \dot{W}_f(h_3-h_1)$$
$$= 66{,}594 + 12{,}000 - .48(60)(101)$$
$$= 66{,}594 + 12{,}000 - 2{,}908$$
$$Q_c = 75{,}686 \quad \text{BTU/hr}$$

THERMODYNAMIC CYCLE

CONDUCTRON ENVIRONMENTAL
CONTROL SYSTEM

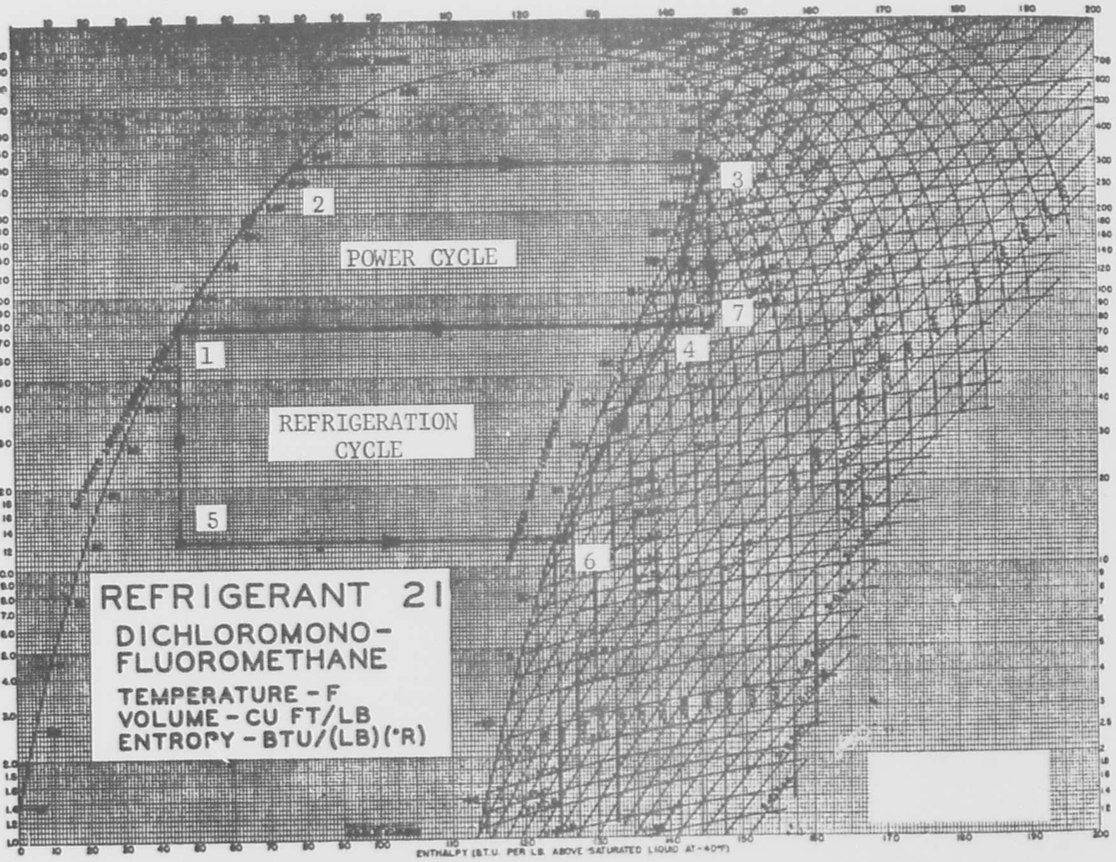





FIGURE A.3

R-113

$$h_6 = 85.0 \quad (10^{o}F \text{ S.H.})$$
$$h_5 = 37.0$$
$$\Delta h_s = 48.0 \quad BTU/lb$$

$$\dot{W}_s = 200/\Delta h_s$$
$$= 200/48.0$$
$$\dot{W}_s = 4.17 \text{ lb/min}$$

$$P_s = 2.65 \text{ psia}$$
$$P_c = 21.93 \text{ psia}$$

$$P_c/P_s = 21.93/2.65 = 8.27$$
$$\text{at} \quad t_b = 360^{o}F$$
$$P_b = 309 \text{ psia}$$

Ideal ejector performance from Figure B-2,

$$\dot{W}_m/\dot{W}_s = 2.1$$

$$\lambda_i = .476$$

$$\dot{W}_m = (\dot{W}_m/\dot{W}_s)\dot{W}_s$$
$$\dot{W}_m = 2.1(4.17) = 8.75 \text{ lb/min}$$
$$\dot{W}_d = .1\ \dot{W}_m = .88$$
$$W_f = .05\ \dot{W}_m = .44$$
$$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = 10.07 \text{ lb/min}$$

$$h_3 = 125.5$$
$$h_1 = 37$$
$$\Delta h_b = 88.5 \quad BTU/lb$$

$$Q_b = \dot{W}_b \Delta h_b = 10.07(88.5)(60) = 53{,}471 \text{ BTU/hr}$$

$$COP = \frac{12{,}000}{53{,}471} \approx 0.224$$

$$Q_c = Q_b + Q_s - \dot{W}_f(h_3 - h_1)$$
$$= 53{,}471 + 12{,}000 - (.44)(60)(88.5)$$
$$= 53{,}471 + 12{,}000 - 2{,}336$$
$$Q_c = 63{,}135 \quad BTU/hr$$

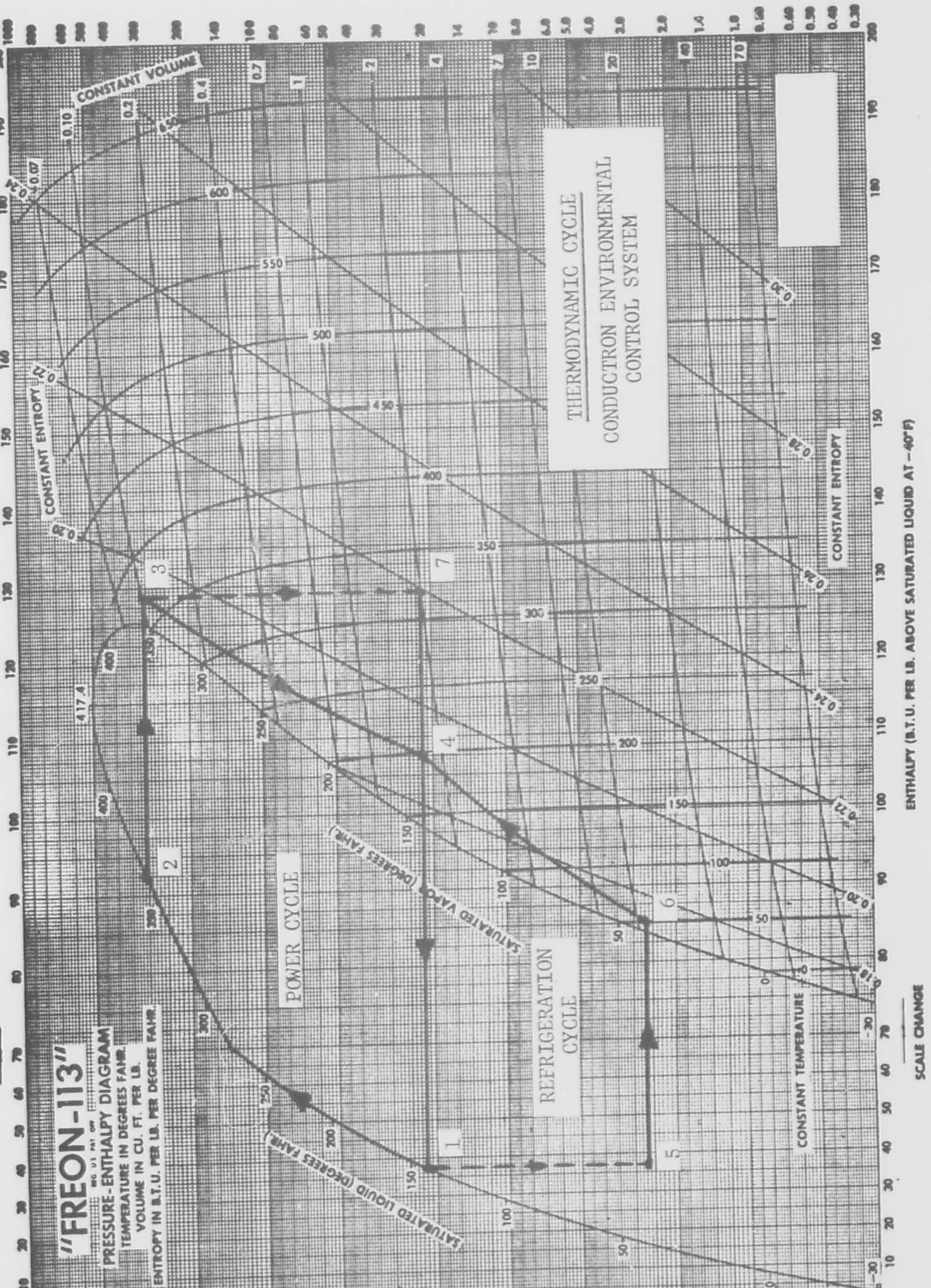

"FREON-113"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN DEGREES FAHR.
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
CONDUCTRON ENVIRONMENTAL
CONTROL SYSTEM
POWER CYCLE
REFRIGERATION CYCLE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT VOLUME
CONSTANT ENTROPY
CONSTANT TEMPERATURE
SCALE CHANGE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)



FIGURE A.4

R-114

$$h_6 = 78.5 \quad (10^{o}F \text{ S.H.})$$
$$h_5 = 41.0$$
$$\Delta h_s = 37.5 \quad \text{BTU/lb}$$

$$\dot{W}_s = 200/\Delta h_s$$
$$= 200/37.5$$
$$\dot{W}_s = 5.33 \text{ lb/min}$$

$$P_s = 15.22 \text{ psia}$$
$$P_c = 85.0 \text{ psia}$$
$$P_c/P_s = 85/15.22 = 5.58$$
$$\text{at} \quad t_b = 230^{o}F$$
$$P_b = 250 \text{ psia}$$

Ideal ejector performance from Figure B-2,

$$\dot{W}_m/\dot{W}_s = 4.8 \qquad \lambda_i = .208$$
$$\dot{W}_m = 4.8(5.33) = 25.58 \text{ lb/min}$$

$$\dot{W}_d = .1\ (\dot{W}_m) = 2.56$$
$$\dot{W}_f = .05\ (\dot{W}_m) = 1.28$$
$$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = 29.42 \text{ lb/min}$$

$$h_3 = 113.5$$
$$h_1 = 41.0$$
$$\Delta h_b = 72.5 \quad \text{BTU/lb}$$

$$Q_b = \dot{W}_b \Delta h_b = 29.42\ (72.5)(60) = 127{,}974 \text{ BTU/hr}$$

$$\text{COP} \approx \frac{12{,}000}{127{,}974} \approx .0938$$

$$Q_c = Q_b + Q_s - \dot{W}_f(h_3 - h_1)$$
$$= 127{,}974 + 12{,}000 - 1.28(60)(72.5)$$
$$= 127{,}974 + 12{,}000 - 5{,}568$$
$$Q_c = 134{,}406 \quad \text{BTU/hr}$$

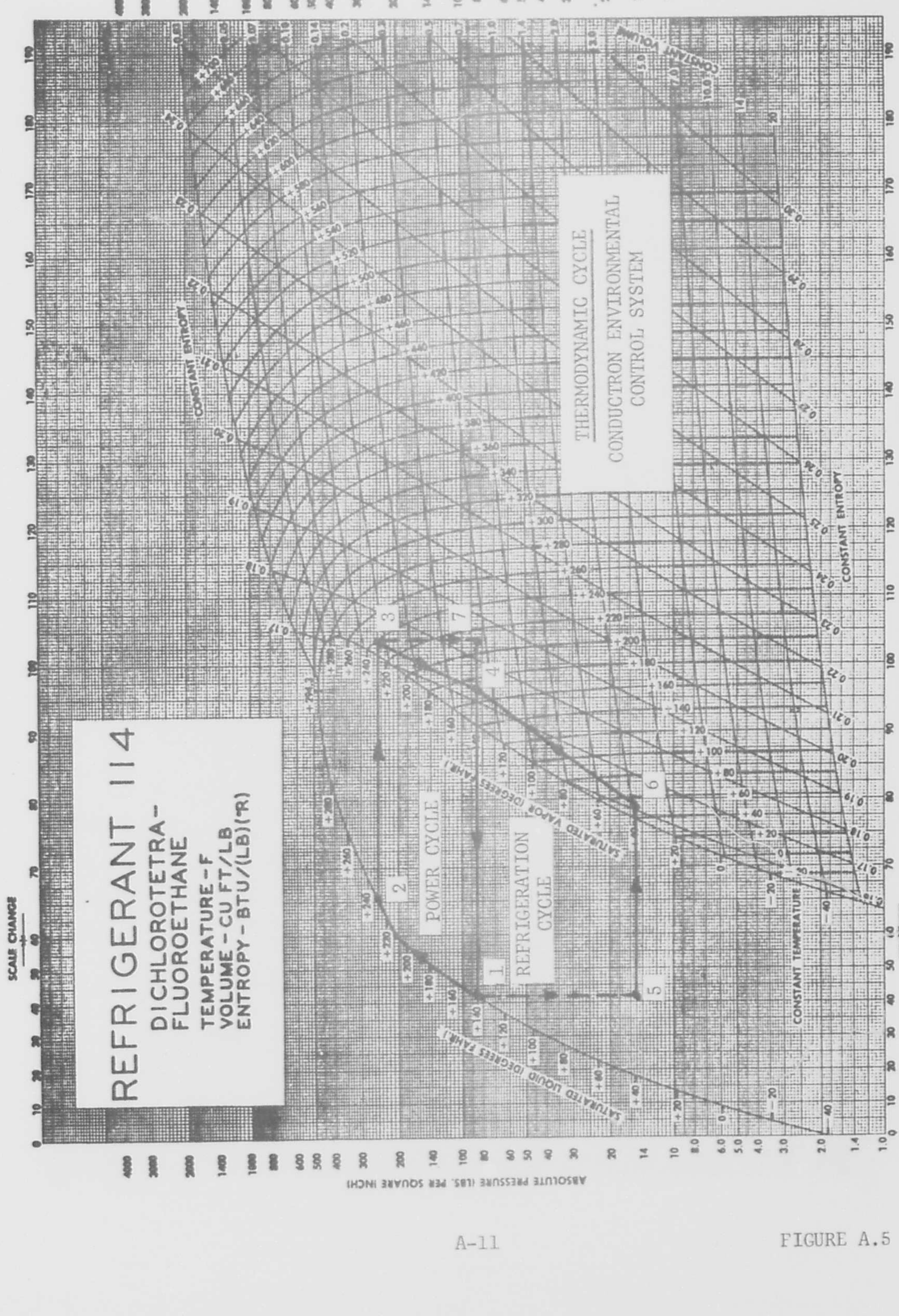

REFRIGERANT 114
DICHLOROTETRA-FLUOROETHANE
TEMPERATURE - F
VOLUME - CU FT/LB
ENTROPY - BTU/(LB)(°R)
THERMODYNAMIC CYCLE
CONDUCTRON ENVIRONMENTAL CONTROL SYSTEM
POWER CYCLE
REFRIGERATION CYCLE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT ENTROPY
CONSTANT TEMPERATURE
CONSTANT VOLUME
SCALE CHANGE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)



FIGURE A.5

$h_6 = 71.5$ (10°F S.H.)
$h_5 = 48.0$

$\Delta h_s = 23.5$ BTU/lb

$\dot{W}_s = 200/\Delta h_s$
$= 200/23.5$
$\dot{W}_s = 8.51$ lb/min

$P_s = 22$ psia
$P_c = 120$ psia

$P_c/P_s = 120/22 = 5.45$

at $t_b = 212°F$
$P_b = 300$ psia

Ideal ejector performance from Figure B-2,

$\dot{W}_m/\dot{W}_s = 6.2$

$\lambda_i = .162$

$\dot{W}_m = 6.2(8.51) = 52.76$ lb/min

$\dot{W}_d = .1\ \dot{W}_m = 5.28$
$\dot{W}_f = .05\ \dot{W}_m = 2.64$

$\dot{W}_b = \dot{W}_m + \dot{W}_d + \dot{W}_f = \underline{60.68 \text{ lb/min}}$

$h_3 = 97.5$
$h_1 = 48.0$

$\Delta h_b = 49.5$ BTU/lb

$Q_b = \dot{W}_b \Delta h_b = 60.68(49.5)(60) = \underline{180{,}222 \text{ BTU/hr}}$

$\text{COP} \approx \dfrac{12{,}000}{180{,}222} \approx 0.0665$

$Q_c = Q_b + Q_s - \dot{W}_f(h_3 - h_1)$
$= 180{,}222 + 12{,}000 - (2.64)(60)49.5$
$= 180{,}222 + 12{,}000 - 7{,}840$
$Q_c = 184{,}382$ BTU/hr

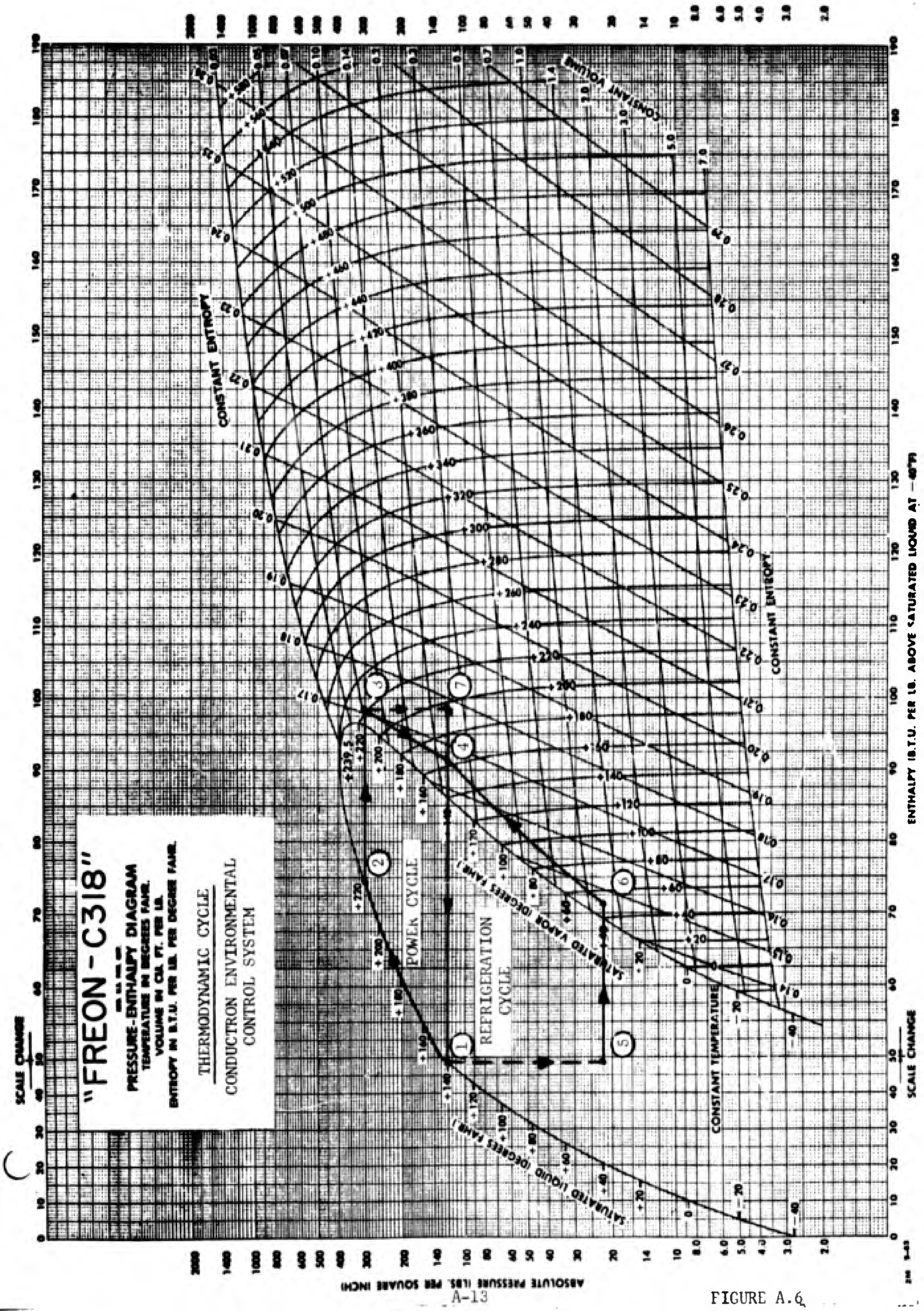

"FREON-C318"
PRESSURE-ENTHALPY DIAGRAM
TEMPERATURE IN DEGREES FAHR.
VOLUME IN CU. FT. PER LB.
ENTROPY IN B.T.U. PER LB. PER DEGREE FAHR.
THERMODYNAMIC CYCLE
CONDUCTRON ENVIRONMENTAL CONTROL SYSTEM
POWER CYCLE
REFRIGERATION CYCLE
SATURATED LIQUID (DEGREES FAHR.)
SATURATED VAPOR (DEGREES FAHR.)
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT TEMPERATURE
SCALE CHANGE
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (B.T.U. PER LB. ABOVE SATURATED LIQUID AT -40°F)



FIGURE A.6

TABLE A-1

REFRIGERANT COMPARISON

| ITEM | R-11 | R-12 | R-21 | R-113 | R-114 | R-C318 |
|---|---|---|---|---|---|---|
| Evaporator pressure at 40°F | 7.0 psia | 52.0 psia | 13.0 psia | 2.65 psia | 15.2 psia | 22.0 psia |
| Evaporator flow rate (lb/min) | 3.25 | 4.59 | 2.45 | 4.17 | 5.33 | 8.51 |
| Pressure ratio ($P_c/P_s$) | (6.42) | (4.23) | (5.85) | (8.27) | (5.58) | (5.45) |
| Condenser pressure at 140°F | 45.0 psia | 220 psia | 76.0 psia | 22.0 psia | 85.0 psia | 120.0 psia |
| Condenser flow rate (lb/min) | 12.72 | 43.46 | 12.96 | 13.80 | 33.47 | 66.55 |
| Heat rejection (BTU/hr) | 62,854 | 139,082 | 75,686 | 63,135 | 134,406 | 184,382 |
| Relative weight increase | 1.00 | 2.22 | 1.21 | 1.005 | 2.14 | 2.94 |
| Boiling pressure | 300 psia | 430 psia | 300 psia | 309 psia | 250 psia | 300 psia |
| Boiling temperature (°F) | 300 | 200 | 250 | 360 | 230 | 212 |
| Boiler flow rate (lb/min) | 9.90 | 40.64 | 10.99 | 10.07 | 29.42 | 60.68 |
| Heat input (BTU/hr) | 53,163 | 132,870 | 66,594 | 53,471 | 127,974 | 180,222 |
| Relative weight increase | 1.00 | 2.49 | 1.25 | 1.005 | 2.40 | 3.39 |
| Ideal ejector entrainment ratio | .377 | .129 | .256 | .476 | .208 | .162 |
| Coefficient of Performance | 0.225 | 0.0903 | 0.180 | 0.224 | 0.0938 | 0.0665 |
| Performance | A | C | B | A | C | D |
| Chemical stability | D | B | D | C | A | A |
| Safety | B | A | C | C | A | A |
| Cost | A | B | C | B | C | D |

NOTE: A = excellent; B = good; C = average; D = poor

COOLING LOAD: 12,000 BTU/hr

TABLE A-2

IDEAL EJECTOR PERFORMANCE

| ITEM | | REFRIGERANT NUMBER | | | | |
|---|---|---|---|---|---|---|
| | 11 | 12 | 21 | 113 | 114 | C318 |
| Specific volume at boiler ($ft^3/lb$) | .14 | .085 | .22 | .1 | .13 | .075 |
| Specific volume at condenser ($ft^3/lb$) | 1.0 | .2 | .8 | 1.6 | .4 | .22 |
| Change in specific volume | .86 | .115 | .58 | 1.5 | .27 | .145 |
| Rank according to change in specific volume (Highest = 1) | 2 | 6 | 3 | 1 | 4 | 5 |
| Rank according to theoretical ejector performance (Lowest $\dot{W}_m/\dot{W}_s$ = 1) | 2 | 6 | 3 | 1 | 4 | 5 |

These results indicate a direct relationship between the change in specific volume from boiler to condenser and theoretical ejector performance.

Stated mathematically:

$$\dot{W}_m/\dot{W}_s = \phi \left[ \frac{1}{v_{\text{CONDENSER}} - v_{\text{BOILER}}} \right]$$

The refrigerant thermodynamic data used in this Appendix was taken from Reference 16.

APPENDIX "B"

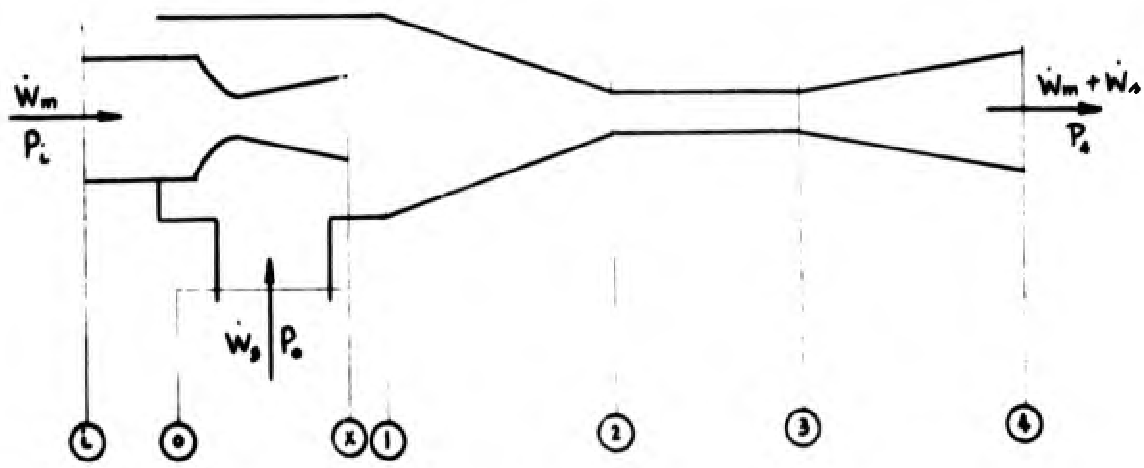
## EJECTOR ANALYSIS

The method of ejector analysis used in this report is the Stodola-Hook ejector analysis. The Stodola-Hook analysis has long been used as the basic ejector model upon which the majority of experimental data has been correlated. This analysis assumes that the motive gas is accelerated and expanded adiabatically to the suction pressure. The motive gas and suction gas are then mixed at constant pressure with conservation of momentum. Then the mixture is decelerated and compressed adiabatically.

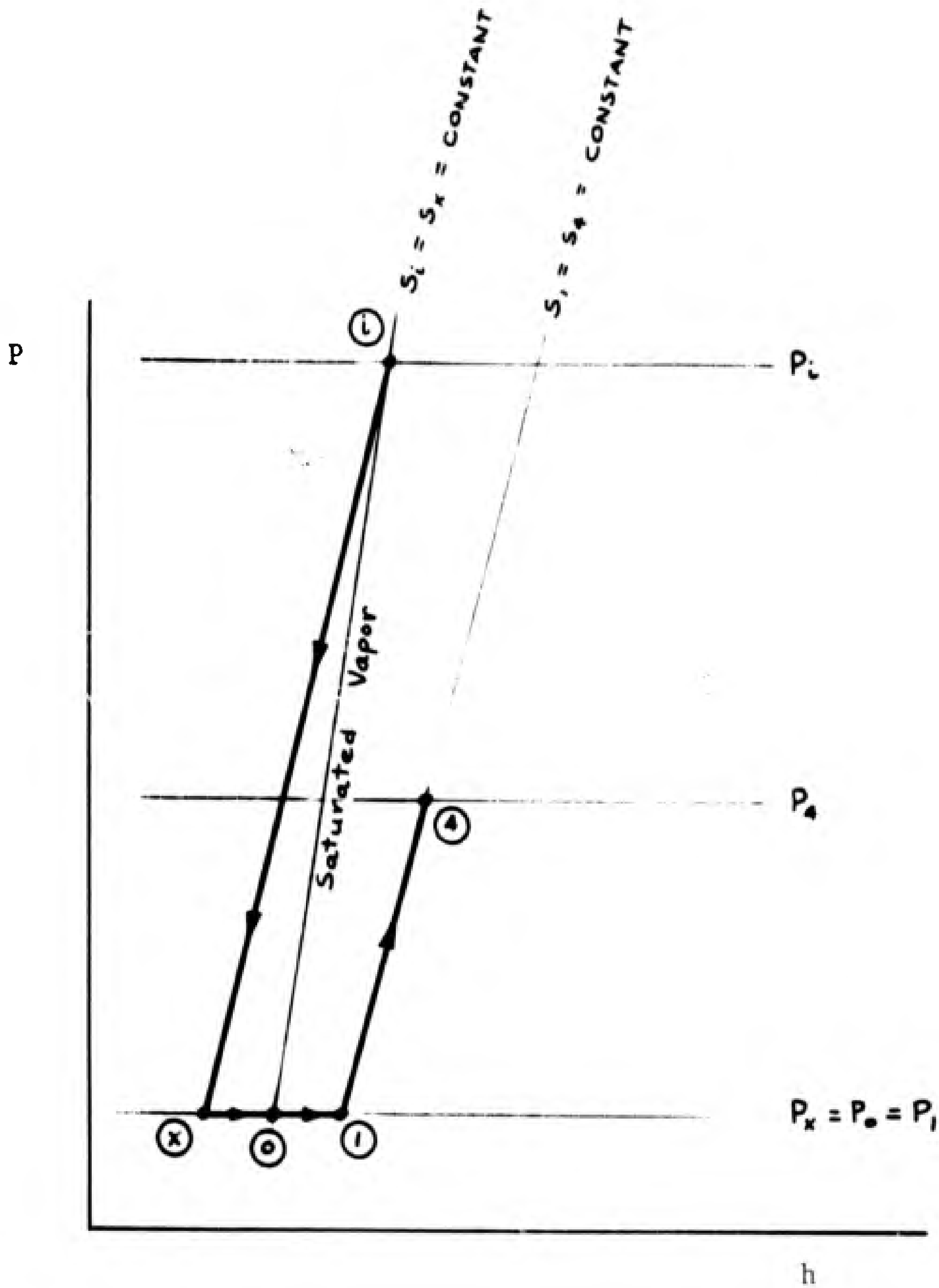
The following is a derivation of the working equations for this ejector model. The end product of this derivation is a procedure for determining an ideal entrainment ratio for given ejector design conditions.

Figure B.2 shows the ideal ejector performance in terms of ejector flow ratio (motive/suction) and compression ratio (condensing pressure/ evaporating pressure) obtainable with various refrigerants at optimum motive temperatures.

EJECTOR NOMENCLATURE:




| | |
|---|---|
| i | Motive gas inlet |
| o | Suction gas inlet |
| x | Motive nozzle exit |
| 1 | State after mixing and before compression |
| 2 | State after compression and before shock diffuser |
| 3 | State after shock diffuser and before subsonic diffuser |
| 4 | State after subsonic diffuser |




Typical Pressure-Enthalpy Diagram
for Ejector Thermodynamic Process
(Constant Pressure Mixing)



FIGURE B.1

IDEAL EJECTOR PERFORMANCE

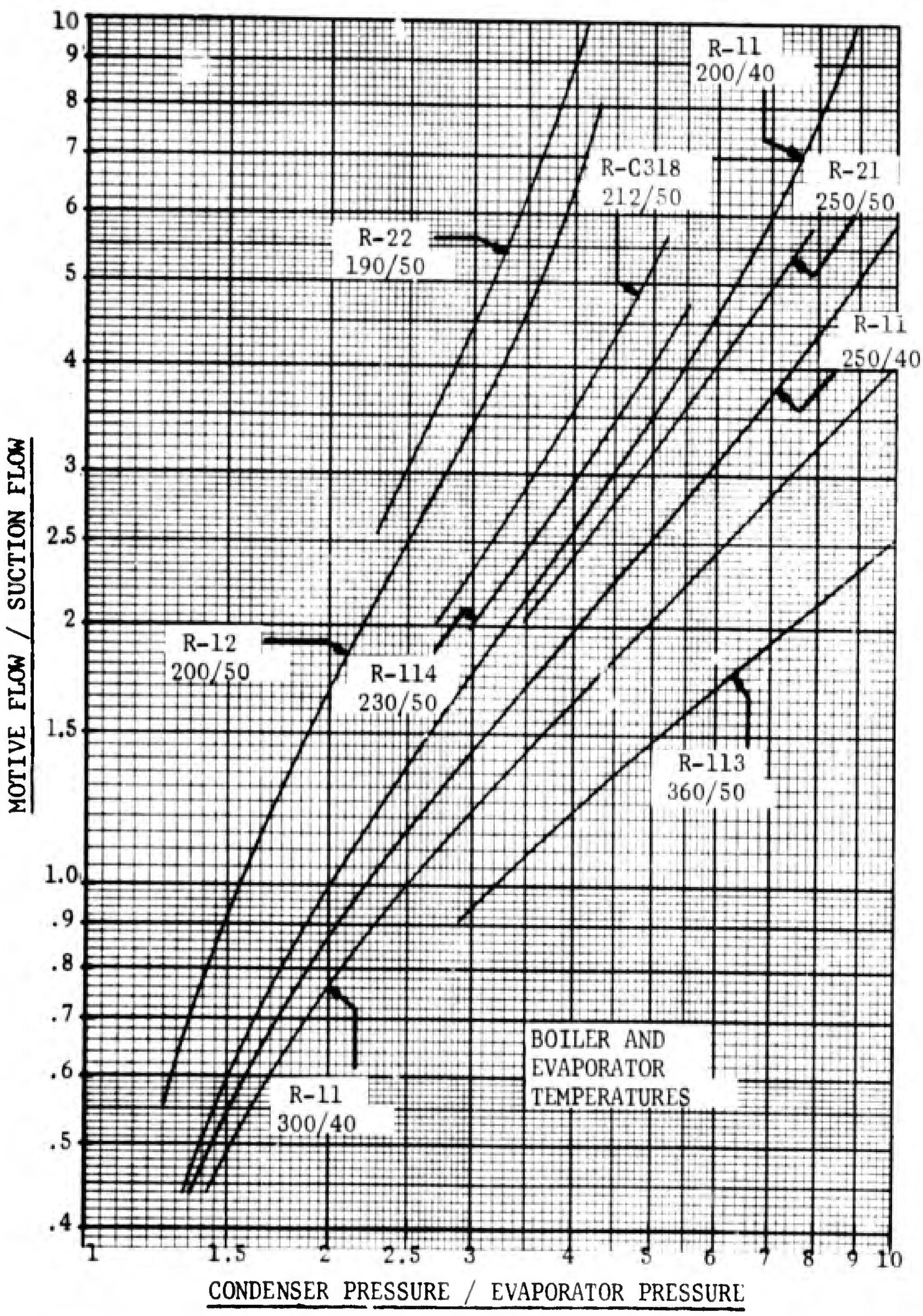





FIGURE B.2

Assumptions:

1. The motive and suction velocities are negligible. $V_i = V_o = 0$.
2. The velocity at the end of the diffusion nozzle is negligible. $V_4 = 0$.
3. Isentropic process from statepoint (i) to statepoint (x).
4. Isentropic process from statepoint (1) to statepoint (4).
5. Constant pressure process from statepoints (x) and (o) to statepoint (1) with conservation of momentum.

For an isentropic process, assuming no change in potential energy:

(1)............ $$h_T = h + \frac{V^2}{2gJ} \qquad \left( h_T = \text{constant} \right)$$

where:
$h_T$ = total enthalpy (BTU/lb)
$h$ = enthalpy (BTU/lb)
$V$ = velocity (ft/sec)
$g$ = 32.2 ft/sec$^2$
$J$ = 778 ft-lb/BTU

Therefore, for the isentropic expansion across the motive nozzle:

(2)............ $$h_i + \frac{V_i^2}{2gJ} = h_x + \frac{V_x^2}{2gJ}$$

Since $V_i = 0$

(3)............ $$\Delta h_{i-x} = \frac{V_x^2}{2gJ}$$

For the isentropic compression from statepoint (1) to statepoint (4):

(4)............ $$h_1 + \frac{V_1^2}{2gJ} = h_4 + \frac{V_4^2}{2gJ}$$

Since $V_4 = 0$

(5a)............ $$\Delta h_{1-4} = \frac{-V_1^2}{2gJ}$$

or

(5b)............ $$\Delta h_{4-1} = \frac{V_1^2}{2gJ}$$

Applying conservation of momentum from statepoints (x) and (o) to statepoint (1):

(6)............ $$\dot{W}_m V_x + \dot{W}_s V_o = (\dot{W}_m + \dot{W}_s) V_1$$

where: $\dot{W}_m$ = motive flow rate (lb/hr)
$\dot{W}_s$ = suction flow rate (lb/hr)

Letting $\lambda = \dot{W}_s / \dot{W}_m$ and since $V_o = 0$

(7a)............ $$V_x = (1 + \lambda) V_1$$

or

(7b)............ $$V_1 = \frac{V_x}{(1+\lambda)}$$

Solving Equation (3) for $V_x$ ,

(8)............ $$V_x = \sqrt{2gJ \quad \Delta h_{i-x}}$$

Solving Equation (5b) for $V_1$ ,

(9)............. $$V_1 = \sqrt{2gJ \quad \Delta h_{4-1}}$$

Substituting Equation (8) into Equation (7b),

(10)............ $$V_1 = \frac{\sqrt{2gJ \quad \Delta h_{i-x}}}{(1+\lambda)}$$

Equating Equation (9) and Equation (10),

(11)........... $$\sqrt{2gJ \ \Delta h_{4-1}} = \frac{\sqrt{2gJ \ \Delta h_{i-x}}}{(1+\lambda)}$$

Simplifying,

(12)........... $$\Delta h_{4-1} = \frac{\Delta h_{i-x}}{(1+\lambda)^2}$$

Therefore:

(13a).......... $$h_1 = h_4 - \frac{\Delta h_{i-x}}{(1+\lambda)^2}$$

or

(13b)........... $h_1 = h_4 - \frac{(h_i - h_x)}{(1+\lambda)^2}$

Since no external work is done,

(14)............ $h_i + \lambda h_o = (1+\lambda) h_4$

or

(15)............ $(1+\lambda) h_4 = h_i + \lambda h_o - h_o + h_o$

Therefore:

(16)............ $h_4 = h_o + \frac{h_i - h_o}{(1+\lambda)}$

Method of Solution:

Known or assumed parameters:

$P_i$ = motive pressure
$h_i$ = motive enthalpy
$P_o$ = suction pressure
$h_o$ = suction enthalpy
$\lambda$ = suction-to-motive flow ratio

1. Solve Equation (16) for $h_4$.
2. Determine $h_x$ from tables knowing $s_x = s_i$ and $P_x = P_o$.

3. Solve Equation (13b) for $h_1$.

4. Determine $s_1$ from tables knowing $P_1 = P_0$ and $h_1$.

5. Determine $P_4$ from tables knowing $s_4 = s_1$ and $h_4$.

The ejector analysis presented in this Appendix is based in part on information contained in Reference 17.

## APPENDIX "C"

## HEAT EXCHANGER DESIGN

The heat exchanger designs used in this report are based upon the procedures set forth in Compact Heat Exchangers, by William Kays and A. L. London (Reference 10). The following is an example of this design procedure as applied to the design of the condenser for system "A".

NOMENCLATURE:

| | |
|---|---|
| $A$ | Heat transfer area $(ft^2)$ |
| $A_c$ | Cross sectional area $(ft^2)$ |
| $A_f$ | Total fin area on one side $(ft^2)$ |
| $A_{fr}$ | Frontal area $(ft^2)$ |
| $b$ | Heat exchanger separator plate spacing (in) |
| $c_p$ | Specific heat (BTU/lbm °F) |
| $C_{min}$ | Minimum flow-stream capacity rate (BTU/hr °F) |
| $C_{max}$ | Maximum flow-stream capacity rate (BTU/hr °F) |
| $f$ | Mean friction factor |
| $h$ | Convective heat transfer coefficient (BTU/hr $ft^2$ °F) |
| $h_{fg}$ | Heat of vaporization (BTU/lbm) |
| $k$ | Thermal conductivity (BTU/hr ft °F) |
| $K_c$ | Contraction loss coefficient for flow at heat exchanger entrance |
| $K_e$ | Expansion loss coefficient for flow at heat exchanger exit |
| $\ell$ | Fin length from root to center (in.) |
| $m$ | Fin effectiveness parameter $[2h/k]$ |
| $\dot{m}$ | Flow rate (lbm/min) |
| $N_{Re}$ | Reynolds number $[4r_H\, \dot{m}/\mu\, A_c]$ |
| $N_{St}$ | Stanton number $[h\, A_c/\dot{m}\, c_p]$ |
| $P$ | Pressure (psia) |
| $Pr$ | Prandtl number $[\mu c_p/k]$ |
| $Q$ | Heat load (BTU/hr) |

| | |
|---|---|
| $r_H$ | Hydraulic, radius (in.) |
| t | Temperature ($^oF$) |
| U | Overall heat transfer coefficient (BTU/hr $ft^2$ $^oF$) |
| v | Specific volume ($ft^3$/lbm) |
| V | Volume ($ft^3$) |
| W | Weight (lbs) |
| $\alpha$ | Ratio of total transfer area on one side of the exchanger to total volume of the exchanger ($ft^2/ft^3$) |
| $\beta$ | Ratio of total heat transfer area on one side of a plate-fin heat exchanger to the volume between the plates on that side ($ft^2/ft^3$) |
| $\delta$ | Fin thickness (in.) |
| $\epsilon$ | Heat exchanger effectiveness |
| $\eta_f$ | Fin temperature effectiveness |
| $\eta_o$ | Total surface temperature effectiveness |
| $\sigma$ | Ratio of free-flow area to frontal area $[A_c/A_f]$ |
| $\mu$ | Viscosity (lbm/ft sec) |
| $\rho$ | Density (lbm/$ft^3$) |

Subscripts:

| | |
|---|---|
| a | Air side of heat exchanger |
| r | Refrigerant side of heat exchanger |
| s | Heat exchanger metal surface |

## Condenser Design

("C" -- is for Condenser)

Design Conditions: $Q = 125{,}800$ BTU/hr $\epsilon = .907$

**Air side:**

$\dot{m} = 179$ lbm/min

$t_{in} = 86^oF$

$t_{out} = 135^oF$

$P_{in} = 2.86$ psia

**Refrigerant side:**

$\dot{m}$ = 26.0 lbm/min

$t_{in}$ = $200^oF$

$t_{out}$ = $140^oF$

$P_{in}$ = 45.12 psia

From Table A-3, Reference 18:

$c_{pa}$ = .240 BTU/lbm $^oF$

$\mu_a$ = $1.301 \times 10^{-5}$ lbm/ft sec

$k_a$ = .0156 BTU/hr ft $^oF$

Pr = .72

$Pr^{2/3}$ = .803

From Reference 19:

$c_{pr}$ = .220 BTU/lbm $^oF$

$\mu_r$ = $2.17 \times 10^{-4}$ lbm/ft sec

$\rho_v$ = 1.050 lbm/ft$^3$

$k_r$ = .0425 BTU/hr ft $^oF$

$h_{fg}$ = 71.4 BTU/lbm

$\rho_\ell$ = 86.7 lbm/ft$^3$

$C_{min} = \dot{m}c_p$ = (179)(.24)(60) = 2580 BTU/hr$^oF$ $\quad C_{min}/C_{max} \approx 0$

From Figure 2-14 (Reference 10 ), NTU = 2.40

UA = $(NTU)(C_{min})$ = (2.40)(2580) = 6190 BTU/hr$^oF$

Use a safety factor of 1.228 (based on comparison of analytical and experimental data on an existing compact heat exchanger)

UA = (6190)(1.228) = 7600 BTU/hr$^oF$

**Air side:** Wavy-fin, plate-fin surface 11.5 - 3/8 W (Figure 10-66, Ref. 10 )

b = .375 in.

$\delta$ = .010 in.

$A_f/A$ = .822

$4r_H$ = .1192 in.

$\beta$ = .345 ft$^2$/ft$^3$

Refrigerant side: Plain plate-fin surface 46.45 T (Fig. 10-37, Ref. 10 ).

$b = .100$ in.

$\delta = .002$ in.

$A_f/A = .837$

$4r_H = .0317$ in.

$\beta = 1332\ ft^2/ft^3$

Try 30 in. x 13 in. x 4 in.

$A_{fr_a} = 390\ in^2$

$A_{fr_r} = 52\ in^2$

$V = 1560\ in^3$

$$\alpha_a = \frac{b_a\ a}{b_a + b_r + 2_a} = 261 \qquad \alpha_r = 269$$

$$A_a = \alpha_a V = \frac{(261)(1560)}{(1728)} = 236 \qquad A_r = \frac{(269)(1560)}{(1728)} = 243$$

$$\sigma_a = \alpha_a r_{H_a} = .648 \qquad \sigma_r = .1777$$

$$A_{c_a} = \sigma A_{fr_a} = (.648)(390) = 253 \qquad A_{c_r} = (.1777)(52) = 9.24$$

$$N_{Re_a} = \frac{(.1192)(179)}{(1.301 \times 10^{-5})(5)(253)} = 1296$$

$$N_{Re_r}\left(\frac{\rho_l}{\rho_v}\right)^{1/2} = \frac{(.0317)(26.0)}{(2.17 \times 10^{-4})(9.24)(5)}\left(\frac{86.7}{1.05}\right)^{.5} = 748$$

From Figure 10-66 (Reference 10 ):

$$N_{St}Pr^{2/3} = .0147 \qquad t_s = 137$$

$$N_{St} = .01831$$

$$h_a = \frac{(.01831)(.24)(60)(179.0)}{(253)} = .1867$$

$$h_r = 13.8\left[\frac{.0425}{(.0317)(12)}\right]\left[\frac{(.22)(2.17\text{x}10^{-4})(3.6\text{x}10^{3})}{(.0425)}\right]^{1/3}\left[\frac{(71.4)}{(.22)(140-137)}\right]^{1/6}$$

$$\text{x } (748)^{.2} = 20.2$$

$$m_a = \sqrt{(24)(.1867)} = 2.12 \qquad m_r = \sqrt{(120)(20.2)} = 49.2$$

$$m_a \ell_a = (2.12)(.1875) = .398 \qquad m_r \ell_r = (49.2)(.05) = 2.46$$

$$\eta_{fa} = \frac{m_a \ell_a}{\tanh (m_a \ell_a)} = \frac{.378}{.398} = .950 \qquad \eta_{fr} = \frac{.985}{2.46} = .400$$

$$\eta_{oa} = 1 - \frac{A_{fr}}{A}(1-\eta_f) = 1 - (.822)(.050) = .959$$

$$\eta_{or} = 1 - (.837)(600) = .498$$

$$1/U_a = \frac{1}{\eta_{oa} h_a} + \frac{1}{(A_r/A_a)\eta_{or} h_r}$$

$$1/U_a = \frac{1}{(.959)(.1867)} + \frac{1}{\frac{243}{236}(.498)(20.2)} = 5.59 + .10 = 5.69$$

$$U_a A_a = \frac{(236)(144)}{(5.69)} = 5970 \text{ vs. } 7600$$

Try 30 in. x 13 in. x 5 in.

$$A_{fr_a} = 390 \text{ in}^2$$

$$A_{fr_r} = 65 \text{ in.}$$

$$V = 1950 \text{ in}^3$$

$$A_a = \frac{(261)(1950)}{(1728)} = 294$$

$$A_r = \frac{(269)(1950)}{(1728)} = 303$$

$$A_{c_r} = (.1777)(65) = 11.55$$

$$N_{Re_r}\left(\rho_l/\rho_v\right)^{1/2} = 6908/11.55 = 598$$

$$t_s = 138$$

$$h_r = 2.46\left(\frac{325}{2}\right)^{1/6}(598)^{.2} = 20.7$$

$$m_r = \sqrt{(120)(20.7)} = 49.8$$

$$m_r \ell_r = (49.8)(.05) = 2.49$$

$$\eta_{f_r} = \frac{.986}{2.49} = .396$$

$$\eta_{o_r} = 1 - (.837)(.604) = .495$$

$$1/U_a = \frac{1}{(.959)(.1867)} + \frac{1}{\left(\frac{303}{294}\right)(.495)(20.7)} = 5.59 + .09 = 5.68$$

$$U_a A_a = \frac{(294)(144)}{(5.68)} = 7450 \text{ vs. } 7600$$

Try 30 in. x 13 in. x 5.1 in.

$$A_{fr_a} = 390 \text{ in}^2$$

$$A_{fr} = 66.3 \text{ in.}$$

$$V = 1989 \text{ in}^3$$

$$A_a = \frac{(261)(1989)}{(1728)} = 300$$

$$A_r = \frac{(269)(1989)}{(1728)} = 310$$

$$A_{c_r} = (.1777)(66.3) = 11.78$$

$$N_{Re_r} = \left(\rho_\ell/\rho_v\right)^{1/2} = 6908/11.78 = 586$$

$$t_s = 138$$

$$h_r = 2.46\left(\frac{325}{2}\right)^{1/6}(586)^{.2} = 20.6$$

$$m_r = \sqrt{(120)(20.6)} = 49.7$$

$$m_r \ell_r = (49.7)(.05) = 2.48$$

$$\eta f_r = \frac{.986}{2.48} = .398$$

$$\eta o_r = 1 - (.837)(.602) = .496$$

$$1/U_a = \frac{1}{(.959)(.1867)} + \frac{1}{\left(\frac{310}{300}\right)(.496)(20.6)} = 5.59 + .09 = 5.68$$

$$U_a A_a = \frac{(300)(144)}{(5.68)} = 7610 \text{ vs } 7600$$

From Figure 5-4, (Reference 10 ):

$$K_c = 1.02 \qquad K_c = .02$$

$$v_1 = \frac{(53.35)(546)}{(144)\ (2.35)} = 86.2 \qquad v_2 = \frac{(53.35)(595)}{(144)\ (2.25)} = 96.1$$

$$v_m = 91.1 \qquad f = .0147$$

$$\Delta P/p = \frac{(179)^2\ (86.2)(144)}{(253)^2\ (64.4)(2.35)(3600)} \left[ (1.02 + 1.0 - (.648)^2 + 2\left(\frac{96.1}{86.2} - 1\right) \right.$$

$$\left. + (.0147)\left(\frac{236}{253}\right)(144)\left(\frac{91.1}{86.2}\right) - \left(1 - (.648)^2 - .02\right)\left(\frac{96.1}{86.2}\right) \right]$$

$$\Delta P/p = .01141\ (.600 + .230 + 2.087 - .624) = .0262$$

$$\Delta P = (.0262)(2.86) = .0749 \text{ psi} = 2.07 \text{ in } H_2O$$

Actual Dimensions:

29.423 in. x 13.255 in. x 5.100 in.

27 air passages

26 refrigerant passages

54 separator plates

$$V_{SP} = \frac{(29.423)(5.1)(.010)(54)}{(1728)} = .0468 \text{ ft}^3$$

$$V_{AP} = \frac{(29.423)(11.5)(.462)(1.083)(5.1)(.010)(27)}{1728} = .1349$$

$$V_{RP} = \frac{(5.1)(46.45)(.1023)(29.423)(.002)(26)}{1728} = .0215$$

$$V_{Total} = .0468 + .1349 + .0215 = .203$$

$$W = (.203)(169) = 24.3 \text{ lbs + headers}$$

Using 25% manifold allowance,

$$W = (1.25)(24.3) = \underline{30.4 \text{ lbs}}$$

# APPENDIX "D"

## SYSTEM PERFORMANCE ANALYSIS

The following is a typical system performance analysis. The design condition for the analysis is the maximum heat load condition.

The purpose of this analysis is to determine the air expander performance characteristics that are required in order to meet the system design conditions. For this purpose, it is necessary to assume the system operating pressures and state-of-the-art performance for the remaining components.

The results are presented in tabular form as a function of condenser capacity. The term $\dot{m}_E(135-T_F)$ required is a measure of the required condenser capacity and is a function of the bleed air expander efficiency. The term $\dot{m}_E(135-T_F)$ available is a measure of the available condenser cooling capability and is a function of the ram air expander efficiency. If the required condenser capacity is less than the available condenser capability, the system will meet the system design conditions.

Included is a system schematic with the statepoints identified. The nomenclature is defined in the text of the analysis.

### SYSTEM "A"

**Design Condition:** Maximum cabin cooling load

From Table 4-1: Condition No. 14,
ICAO Standard Day.

Compressor bleed air temperature, $(T_A)$ = $952^oF$

Compressor bleed air pressure, $(P_A)$ = 162.0 psia

Ram air temperature, $(T_D)$ = $325^oF$

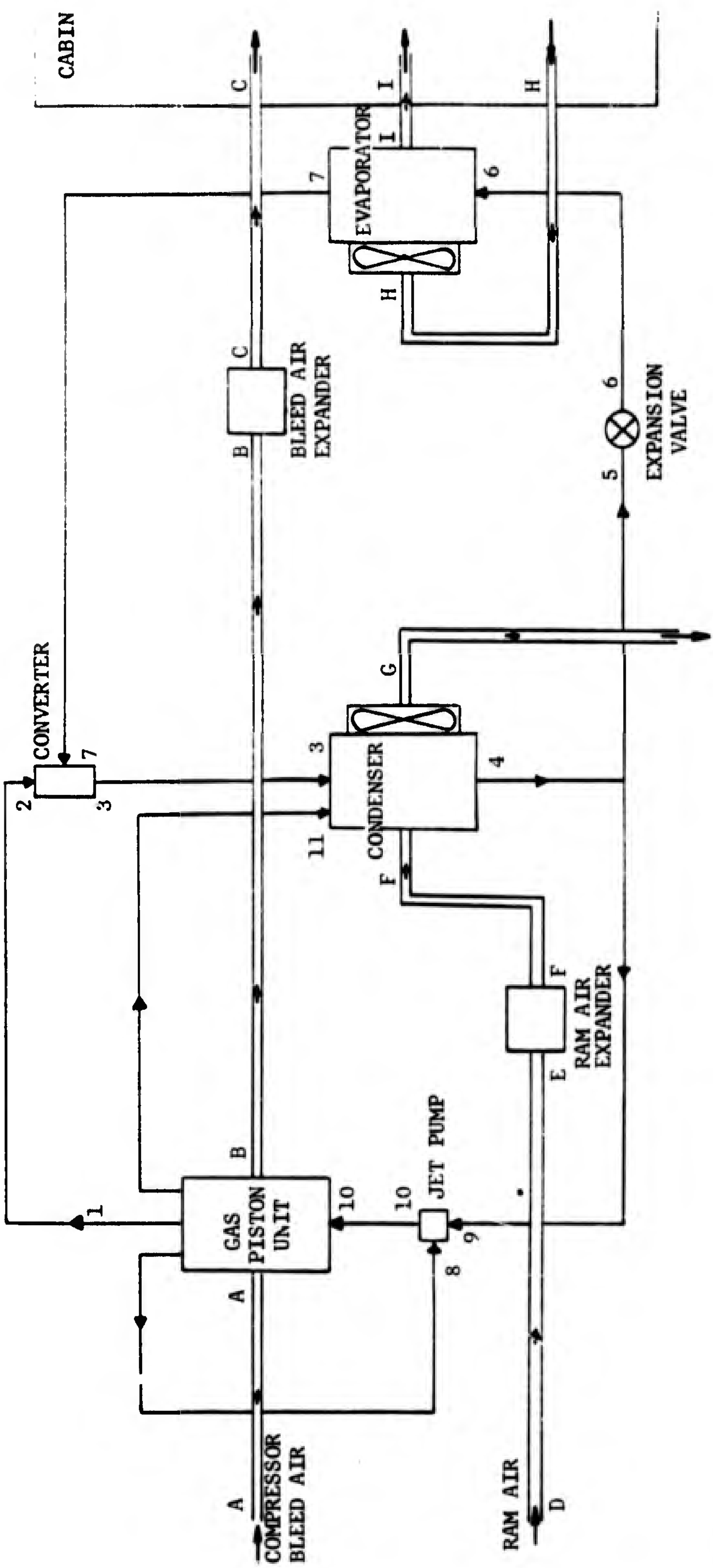
Ram air pressure, $(P_D)$ = 31.50 psia

Ambient pressure, $(P_\infty)$ = 2.72 psia

Cabin temperature, $(T_H)$ = $100^oF$

Cabin pressure, $(P_C)$, $(P_H)$ = 7.72 psia

Cabin heat load, $(Q_{CAB})$ = 38,500 BTU/hr




SYSTEM "A" SCHEMATIC

FIGURE D.1

Assume:

| | |
|---|---|
| Evaporator pressure, $(P_6)$, $(P_7)$ = | 6.25 psia |
| Evaporator refrigerant inlet temperature, $(T_6)$ = | $35^oF$ |
| Evaporator refrigerant outlet temperature, $(T_7)$ = | $40^oF$ |
| Evaporator air inlet temperature, $(T_H)$ = | $100^oF$ |
| Evaporator air outlet temperature, $(T_I)$ = | $45^oF$ |
| Condenser inlet pressure, $(P_3)$ = | 46.00 psia |
| Condenser outlet pressure, $(P_4)$, $(P_5)$, $(P_9)$ = | 45.12 psia |
| Condenser refrigerant outlet temperature, $(T_4)$ = | $140^oF$ |
| Condenser air outlet temperature, $(T_G)$ = | $135^oF$ |
| Depressurization temperature, $(T_{11})$ = | $310^oF$ |
| Boiler pressure, $(P_1)$, $(P_2)$, $(P_8)$, $(P_{11})$ = | 300.21 psia |
| Boiler refrigerant outlet temperature, $(T_1)$, $(T_2)$, $(T_8)$, $(T_{11})$ = | $310^oF$ |
| Converter efficiency, $(\eta_C)$ = | .35 |

Depressurization flow equal to 10% of boiler flow,
$\dot{m}_{11} = .1\ \dot{m}_1$

Jet pump motive flow equal to 8% of jet pump suction flow,
$\dot{m}_8 = .08\ \dot{m}_9$

From Table T-11, Reference 19:

$$h_1 = h_2 = h_8 = h_{11} = 125.59 \text{ BTU/lbm}$$

$$h_4 = h_5 = h_6 = h_9 = 37.15 \text{ BTU/lbm}$$

$$h_7 = 96.86 \text{ BTU/lbm}$$

$$s_2 = s_{11} = .19468 \text{ BTU/lbm } ^oR$$

$$s_7 = .19839 \text{ BTU/lbm } ^oR$$

Heat balance on the jet pump,

Assuming an adiabatic efficiency of .98

$$.98\ \dot{m}_8(h_8-h_{10}) = \dot{m}_9\ (h_{10}-h_9)$$

Since: $\dot{m}_8 = .08\ \dot{m}_9$

$$h_{10} = \frac{h_9 + .0784\, h_8}{1.0784} = \frac{(37.15) + (.0784)(125.59)}{1.0784}$$

$$h_{10} = 43.58 \text{ BTU/lbm}$$

Heat balance on the converter,

Adiabatic efficiency is defined as:

$$\eta_C = \frac{\lambda \text{ actual}}{\lambda \text{ ideal}}$$

Where:

$$\lambda = \frac{\dot{m}_7}{\dot{m}_2} = \frac{h_2 - h_{3i}}{h_{3i} - h_7}$$

With $h_{3i}$ evaluated at:

$$s_{3i} = \frac{s_2 + s_7}{2} = \frac{.19468 + .19839}{2} = .19654 \text{ BTU/lbm } {}^{\circ}\text{R}$$

Therefore:

$$\dot{m}_2 = \dot{m}_7/\eta_C \left(\frac{h_2 - h_{3i}}{h_{3i} - h_7}\right)$$

From Table T-11:

$$h_{3i} = 111.20 \text{ BTU/lbm}$$

$$\dot{m}_2 = \dot{m}_7/(.35)\left(\frac{125.59 - 111.20}{111.20 - 96.86}\right) = 2.85\, \dot{m}_7$$

$$\dot{m}_2 (h_2 - h_3) = \dot{m}_7 (h_3 - h_7)$$

Substituting and simplifying -

$$h_3 = \frac{2.85\, h_2 + h_7}{3.85} = \frac{(2.85)(125.59) + 96.86}{3.85}$$

$$h_3 = 118.13 \text{ BTU/lbm}$$

Mass balances:

$$\dot{m}_A = \dot{m}_B = \dot{m}_C$$

$$\dot{m}_D = \dot{m}_E = \dot{m}_F = \dot{m}_G$$

$$\dot{m}_H = \dot{m}_I$$

$$\dot{m}_1 = \dot{m}_{10} = 3.46\ \dot{m}_7$$

$$\dot{m}_2 = 2.85\ \dot{m}_7$$

$$\dot{m}_3 = 3.85\ \dot{m}_7$$

$$\dot{m}_4 = 4.20\ \dot{m}_7$$

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_7$$

$$\dot{m}_8 = .26\ \dot{m}_7$$

$$\dot{m}_9 = 3.20\ \dot{m}_7$$

$$\dot{m}_{11} = .35\ \dot{m}_7$$

Heat balance on the cabin,

$$Q_{CAB} = \dot{m}_A C_P\ (T_H - T_C) + \dot{m}_I\ C_P\ (T_H - T_I)$$

$$C_P = (.240\ \text{BTU/lbm}\ ^oF)\ (60\ \text{min/hr}) = 14.4\ \frac{\text{BTU min}}{\text{lbm hr}\ ^oF}$$

This value will be assumed constant throughout the analysis.

$$38{,}500 = \dot{m}_A\ (14.4)\ (100 - T_C) + \dot{m}_I\ (100 - 45)$$

$$2674 = \dot{m}_A\ (100 - T_C) + 55\ \dot{m}_I$$

Heat balance on the evaporator,

$$Q_E = \dot{m}_I\ C_P\ (T_H - T_I) = \dot{m}_7\ (h_7 - h_6)$$

$$14.4\ \dot{m}_I\ (100 - 45) = 60\ \dot{m}_7\ (96.86 - 37.15)$$

$$\dot{m}_7 = .221\ \dot{m}_I$$

Heat balance on the condenser,

$$Q_C = \dot{m}_D C_P (T_G - T_F) = \dot{m}_3 (h_3 - h_4) + \dot{m}_{11} (h_{11} - h_4)$$

$$14.4\, \dot{m}_E (135 - T_E) = (60)(3.85)\, \dot{m}_7 (118.14 - 37.15) + + (60)(.35)\, \dot{m}_7 (125.59 - 37.15)$$

$$\dot{m}_E (135 - T_E) = 1428\, \dot{m}_7$$

Heat balance on the boiler,

$$Q_B = \dot{m}_A C_P (T_A - T_B) = \dot{m}_1 (h_1 - h_{10})$$

$$14.4\, \dot{m}_A (952 - T_B) = (3.46)(60)\, \dot{m}_7 (125.59 - 43.58)$$

$$\dot{m}_A (952 - T_B) = 1182\, \dot{m}_7$$

Temperature out of air expanders,

Bleed air expander:

$$T_C = T_B \left\{ 1 - \eta_{ExB} \left[ 1 - \frac{1}{\left(\frac{P_B}{P_C}\right)^{\frac{\gamma-1}{\gamma}}} \right] \right\}$$

For air: $\gamma = 1.395$

$$T_C = T_B \left\{ 1 - \eta_{ExB} \left[ 1 - \frac{1}{\left(\frac{162.0}{7.72}\right)^{.283}} \right] \right\}$$

$$T_C = T_B (1 - .578\, \eta_{ExB})$$

In $F^o$:

$$T_C = (T_B + 460)(1 - .578\, \eta_{ExB}) - 460$$

Ram air expander:

$$T_F = T_E \left\{ 1 - \eta_{ExR} \left[ 1 - \frac{1}{\left(\frac{P_E}{P_F}\right)^{\frac{\gamma-1}{\gamma}}} \right] \right\}$$

$$T_F = 785 \left\{ 1 - \eta_{ExR} \left[ 1 - \frac{1}{\left(\frac{P_E}{2.25}\right)^{.283}} \right] \right\}$$

In $F^o$:

$$T_F = 785 \left\{ 1 - \eta_{ExR} \left[ 1 - {}^{1}/\left(\frac{P_E}{2.25}\right)^{.283} \right] \right\} - 460$$

Ram air pressure recovery,

Theoretical ram air flow rate:

$$\dot{m}_{TH} = \sqrt{\gamma g R} \left(\frac{60}{144}\right) \sqrt{T} \quad A \, \rho \, M$$

For Air: $\quad R = 53.35 \dfrac{\text{ft lbs}}{\text{lbm } ^oR}$

From McDonnell Specification:

Mach number, (M) = 2.25

Ram air duct area, (A) = 16.96 $in^2$

Ambient air temperature, (T) = $390^oR$

$$\rho = P/_{RT} = \frac{(144)\ (2.25)}{(53.35)(390)} = .01557 \text{ lbm/ft}^3$$

$$\dot{m}_{TH} = \sqrt{(1.4)\ (32.2)\ (53.35)} \left(\frac{60}{144}\right) \sqrt{390}\ (.01557)\ 2.72 = 289 \text{ lbm/min}$$

The pressure recovery factor $(P_{actual}/P_{total})$ is determined from Figure D.2 as a function of the flow rate ratio $(\dot{m}_{actual}/\dot{m}_{theoretical})$.

$$\dot{m}_E = \dot{m}_{actual}$$

$$P_E = P_D \ (P_{actual}/P_{total})$$

$$T_E = T_D$$

The following is a parametric solution of the above equations:

Assume: $\quad T_B = 350^oF$ and $\eta_{ExB} = .85$

$$T_C = 810 \left[ 1 - (.578)(.85) \right] - 460 = -48^oF$$

RAM AIR PRESSURE RECOVERY FOR
McDONNELL F-4 RAM AIR SCOOP (M = 2.2)

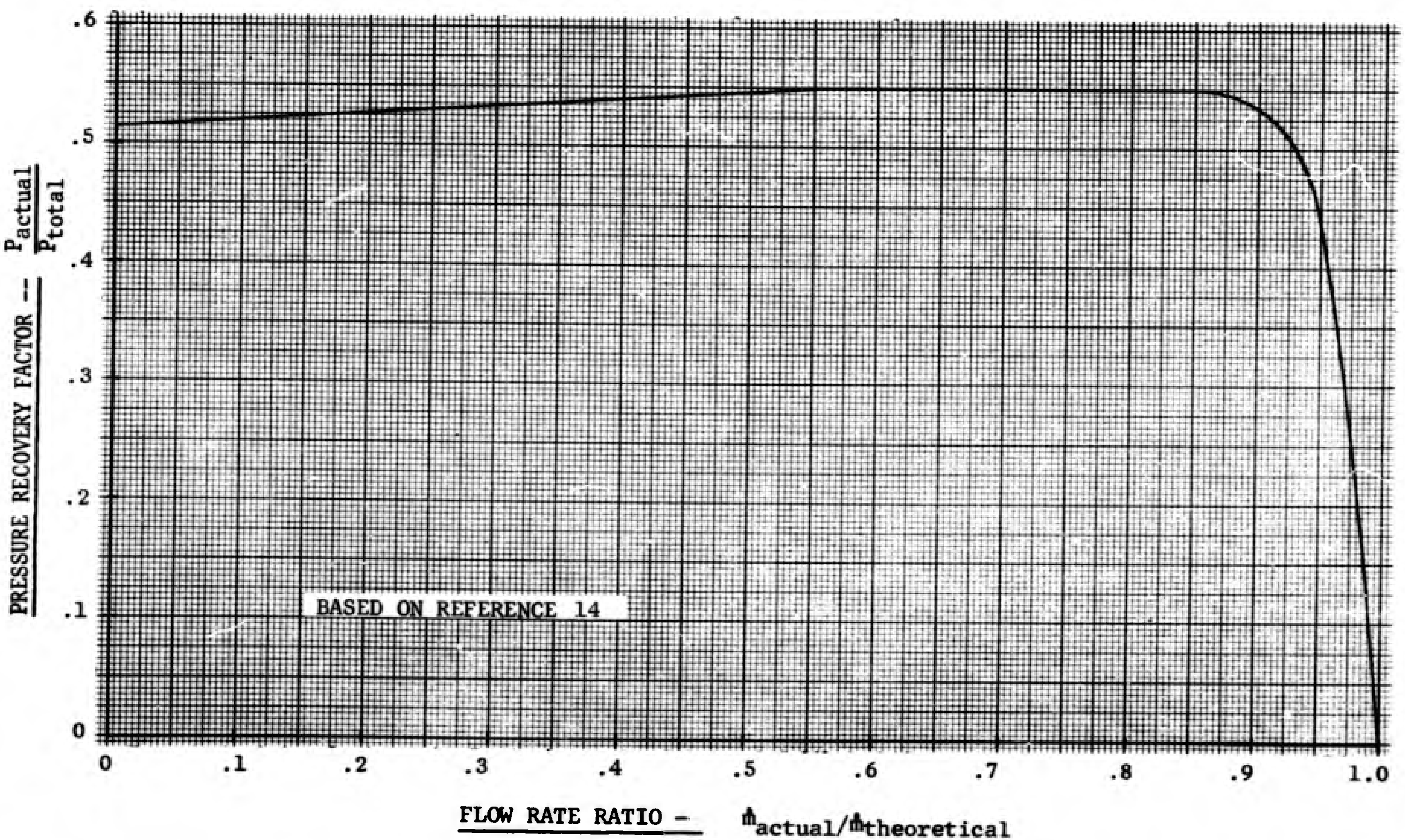





FIGURE D.2

$$2674 = \frac{1182\ \dot{m}_7\ (100 - T_C)}{(952 - T_B)} + \frac{55\ \dot{m}_7}{.221}$$

$$\dot{m}_7 = 2674/\left[\ 1.98\ (100 - T_C) + 249\right]$$

$$\dot{m}_7 = 2674/\ (1.98\left[100 - (-48)\right] + 249) = 4.93\ \text{lbm/min}$$

$$\dot{m}_A = 1.98\ \dot{m}_7 = (1.98)\ (4.93) = 9.76\ \text{lbm/min}$$

$$\dot{m}_I = \dot{m}_7/.221 = (4.53)\ (4.93) = 22.4\ \text{lbm/min}$$

$$\dot{m}_E\ (135 - T_F) = 1428\ \dot{m}_7 = (1428)\ (4.93) = 7040\ \frac{\text{lbm}\ {}^{\circ}\text{F}}{\text{min}}$$

**Assume:** $T_B = 350^{\circ}\text{F}$ and $\eta_{ExB} = .80$

$$T_C = 810\left[1 - (.578)\ (.80)\right] - 460 = -24^{\circ}\text{F}$$

$$\dot{m}_7 = 2674/\left[1.98\ (100 - (-24)) + 249\right] = 5.40\ \text{lbm/min}$$

$$\dot{m}_A = (1.98)\ (5.40) = 10.69\ \text{lbm/min}$$

$$\dot{m}_I = (4.53)\ (5.40) = 24.5\ \text{lbm/min}$$

$$\dot{m}_E\ (135 - T_F) = (1427)\ (5.40) = 7710\ \frac{\text{lbm}\ {}^{\circ}\text{F}}{\text{min}}$$

**Assume:** $T_B = 350^{\circ}\text{F}$ and $\eta_{ExB} = .75$

$$T_C = 810\left[1 - (.578)\ (.75)\right] - 460 = -2^{\circ}\text{F}$$

$$\dot{m}_7 = 2674/\left[\ 1.98(100 - (-2)) + 249\right] = 5.93\ \text{lbm/min}$$

$$\dot{m}_A = (1.98)\ (5.93) = 11.74\ \text{lbm/min}$$

$$\dot{m}_I = (4.53)\ (5.93) = 26.9\ \text{lbm/min}$$

$$\dot{m}_E\ (135 - T_F) = (1427)\ (5.93) = 8460\ \frac{\text{lbm}\ {}^{\circ}\text{F}}{\text{min}}$$

Assume: $T_B = 350^oF$ and $\eta_{ExB} = .70$

$$T_C = 810\left[1 - (.578)(.70)\right] - 460 = 22^oF$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 22) + 249\right] = 6.64\ lbm/min$$

$$\dot{m}_A = (1.98)(6.64) = 13.15\ lbm/min$$

$$\dot{m}_I = (4.53)(6.64) = 30.1\ lbm/min$$

$$\dot{m}_E\,(135 - T_F) = (1427)(6.64) = 9480\ \frac{lbm\ ^oF}{min}$$

Assume: $T_B = 350^oF$ and $\eta_{ExB} = .65$

$$T_C = 810\left[1 - (.578)(.65)\right] - 460 = 45^oF$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 45) + 249\right] = 7.47\ lbm/min$$

$$\dot{m}_A = (1.98)(7.47) = 14.79\ lbm/min$$

$$\dot{m}_I = (4.53)(7.47) = 33.8\ lbm/min$$

$$\dot{m}_E\,(135 - T_F) = (1427)(7.47) = 10{,}660\ \frac{lbm\ ^oF}{min}$$

Assume: $T_B = 350^oF$ and $\eta_{ExB} = .60$

$$T_C = 810\left[1 - (.578)(.60)\right] - 460 = 69^oF$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 69) + 249\right] = 8.63\ lbm/min$$

$$\dot{m}_A = (1.98)(8.63) = 17.09\ lbm/min$$

$$\dot{m}_I = (4.53)(8.63) = 39.1\ lbm/min$$

$$\dot{m}_E\,(135 - T_F) = (1427)(8.63) = 12{,}320\ \frac{lbm\ ^oF}{min}$$

Assume: $T_B = 400^oF$ and $\eta_{ExB} = .85$

$$T_C = 860\left[1 - (.578)(.85)\right] - 460 = -22^oF$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - (-22)) + 249\right] = 5.46 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)\,(5.46) = 10.81 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)\,(5.46) = 24.7 \text{ lbm/min}$$

$$\dot{m}_E\,(135 - T_F) = (1427)\,(5.46) = 7790 \ \frac{\text{lbm } {}^{\circ}\text{F}}{\text{min}}$$

**Assume:** $T_B = 400^{\circ}F$ and $\eta_{ExB} = .80$

$$T_C = 860\left[1 - (.578)\,(.80)\right] - 460 = 3^{\circ}F$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 3) + 249\right] = 5.90 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)\,(5.90) = 11.68 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)\,(5.90) = 26.7 \text{ lbm/min}$$

$$\dot{m}_E\,(135 - T_F) = (1427)\,(5.90) = 8420 \ \frac{\text{lbm } {}^{\circ}\text{F}}{\text{min}}$$

**Assume:** $T_B = 400^{\circ}F$ and $\eta_{ExB} = .75$

$$T_C = 860\left[1 - (.578)\,(.75)\right] - 460 = 27^{\circ}F$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 27) + 249\right] = 6.79 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)\,(6.79) = 13.44 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)\,(6.79) = 30.8 \text{ lbm/min}$$

$$\dot{m}_E\,(135 - T_F) = (1427)\,(6.79) = 9690 \ \frac{\text{lbm } {}^{\circ}\text{F}}{\text{min}}$$

**Assume:** $T_B = 400^{\circ}F$ and $\eta_{ExB} = .70$

$$T_C = 860\left[1 - (.578)\,(.70)\right] - 460 = 52^{\circ}F$$

$$\dot{m}_7 = 2674/\left[1.98\,(100 - 52) + 249\right] = 7.77 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)\,(7.77) = 15.38 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)(7.77) = 35.2 \text{ lbm/min}$$

$$\dot{m}_E (135 - T_F) = (1427)(7.77) = 11{,}090 \frac{\text{lbm } ^\circ\text{F}}{\text{min}}$$

Assume: $T_B = 400^\circ F$ and $\eta_{ExB} = .65$

$$T_C = 860 \left[ 1 - (.578)(.65) \right] - 460 = 77^\circ F$$

$$\dot{m}_7 = 2674 / \left[ 1.98 (100 - 77) + 249 \right] = 9.06 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)(9.06) = 17.94 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)(9.06) = 41.0 \text{ lbm/min}$$

$$\dot{m}_E (135 - T_F) = (1427)(9.06) = 12{,}930 \frac{\text{lbm } ^\circ\text{F}}{\text{min}}$$

Assume: $T_B = 400^\circ F$ and $\eta_{ExB} = .60$

$$T_C = 860 \left[ 1 - (.578)(.60) \right] - 460 = 102^\circ F$$

$$\dot{m}_7 = 2674 / \left[ 1.98 (100 - 102) + 249 \right] = 10.91 \text{ lbm/min}$$

$$\dot{m}_A = (1.98)(10.91) = 21.6 \text{ lbm/min}$$

$$\dot{m}_I = (4.53)(10.91) = 49.4 \text{ lbm/min}$$

$$\dot{m}_E (135 - T_F) = (1427)(10.91) = 15{,}570 \frac{\text{lbm } ^\circ\text{F}}{\text{min}}$$

**Ram Air:**

It was determined by trial-and-error solution that the maximum ram air cooling capacity is achieved at a flow rate ratio:

$$\dot{m}_{actual}/\dot{m}_{theoretical} = .89$$

From Figure D-2:

$$P_{actual}/P_{total} = .540$$

$$\dot{m}_E = (.89)(289) = 258 \text{ lbm/min}$$

$$P_E = (.54)(31.5) = 17.01 \text{ psia}$$

Assume: $\eta_{ExR} = .6$

$$T_F = 785 \left\{ 1 - \eta_{ExR} \left[ 1 - 1/\left(\frac{17.01}{2.25}\right)^{.283} \right] \right\} - 460$$

$$T_F = 785 \left[ 1 - (.6)(.436) \right] - 460 = 134^oF$$

$$\dot{m}_E (135 - T_F) = 258 (135 - 134) = 258 \frac{lbm\ ^oF}{min}$$

Assume: $\eta_{ExR} = .65$

$$T_F = 785 \left[ 1 - (.436)(.65) \right] - 460 = 119^oF$$

$$\dot{m}_E (135 - T_F) = 258 (135 - 119) = 4{,}128 \frac{lbm\ ^oF}{min}$$

Assume: $\eta_{ExR} = .70$

$$T_F = 785 \left[ 1 - (.436)(.70) \right] - 460 = 103^oF$$

$$\dot{m}_E (135 - T_F) = 258 (135 - 103) = 8{,}256 \frac{lbm\ ^oF}{min}$$

Assume: $\eta_{ExR} = .75$

$$T_F = 785 \left[ 1 - (.436)(.75) \right] - 460 = 86^oF$$

$$\dot{m}_E (135 - T_F) = 258 (135 - 86) = 12{,}640 \frac{lbm\ ^oF}{min}$$

Assume: $\eta_{ExR} = .80$

$$T_F = 785 \left[ 1 - (.436)(.80) \right] - 460 = 71^oF$$

$$\dot{m}_E (135 - T_F) = 258 (135 - 71) = 16{,}510 \frac{lbm\ ^oF}{min}$$

TABLE D-1

SUMMARY OF RESULTS

| $\eta_{ExB}$ (Bleed) | $T_B$ ($^oF$) | $\eta_{ExR}$ (Ram) | $T_C$ ($^oF$) | $T_F$ ($^oF$) | $\dot{m}_7$ (lbm/min) | $\dot{m}_A$ (lbm/min) | $\dot{m}_I$ (lbm/min) | $\dot{m}_E$ (lbm/min) | $\dot{m}_E(135-T_F)$ (Required) | $\dot{m}_E(135-T_F)$ (Available) |
|---|---|---|---|---|---|---|---|---|---|---|
| .85 | 400 | .70 | -22 | 103 | 5.46 | 10.81 | 24.7 | 213 | 7,790 | 8,260 |
| .85 | 350 | .70 | -48 | 103 | 4.93 | 9.76 | 22.4 | 213 | 7,040 | 8,260 |
| .80 | 400 | .75 | 3 | 86 | 5.90 | 11.68 | 26.7 | 213 | 8,420 | 12,640 |
| .80 | 350 | .70 | -24 | 103 | 5.40 | 10.69 | 24.5 | 213 | 7,710 | 8,260 |
| .75 | 400 | .75 | 27 | 86 | 6.79 | 13.44 | 30.8 | 213 | 9,690 | 12,640 |
| .75 | 350 | .75 | -2 | 86 | 5.93 | 11.74 | 26.9 | 213 | 8,460 | 12,640 |
| .70 | 400 | .75 | 52 | 86 | 7.77 | 15.38 | 35.2 | 213 | 11,090 | 12,640 |
| .70 | 350 | .75 | 22 | 86 | 6.64 | 13.15 | 30.1 | 213 | 9,480 | 12,640 |
| .65 | 400 | .80 | 77 | 71 | 9.06 | 17.94 | 41.0 | 213 | 12,930 | 16,510 |
| .65 | 350 | .75 | 45 | 86 | 7.47 | 14.79 | 33.8 | 213 | 10,660 | 12,640 |
| .60 | 400 | .80 | 102 | 71 | 10.91 | 21.60 | 49.4 | 213 | 15,570 | 16,510 |
| .60 | 350 | .75 | 69 | 86 | 8.63 | 17.09 | 39.1 | 213 | 12,320 | 12,640 |

## APPENDIX "E"

## MAXIMUM SYSTEM CAPABILITY ANALYSIS

The following is a typical system capability analysis. The equations, nomenclature, and system schematic are identical to those used in the system performance analysis described in Appendix "D" Figure E.1 (Reference 9 ) was used to predict the ram air pressure recovery factor as data on the F-4 ram air pressure recovery, as a function of Mach number, was not available. This curve was found to be slightly conservative for the F-4 pressure recovery factor at Mach number 2.2.

The purpose of this analysis was to determine the maximum cooling capacity of the system as a function of altitude. For this purpose, state-of-the-art performance was assumed for all components. Since the cooling load at a specific altitude is a function of Mach number, the results are stated in terms of the maximum Mach number at which the system can meet the incident cabin heat loads. The capability analysis shown here applies to atmospheric conditions on an ICAO Standard Day.

SYSTEM "A"

Bleed air expander efficiency, $\eta_{ExB} = .8$

Ram air expander efficiency, $\eta_{ExR} = .7$

Sea Level

From Figure 3.5, Figure 4.1, and Reference 2:

$Q_{CAB} = 30{,}700$ $T_D = 231$ $M = 1.3$ $P_D = 295.0$

$P = 14.69$ $P_A = 35.65$

From Figure E.1: $P_E/P_D = .877$ $P_E = 31.3$

$$T_F = 691\left\{1 - .7\left[1 - 1\Big/\left(\frac{31.3}{14.69}\right)^{.283}\right]\right\} - 460 = 138^{\circ}F > \underline{135^{\circ} \text{ too high}}$$

$Q_{CAB} = 25{,}500$ $T_D = 207$ $M = 1.2$

From Figure E.1: $P_E/P_D = .897$ $P_E = 32.0$

$$T_F = 667\left\{1 - .7\left[1 - 1\Big/\left(\frac{32.0}{14.69}\right)^{.283}\right]\right\} - 460 = 115^{\circ}F$$

RAM AIR SCOOP PRESSURE RECOVERY

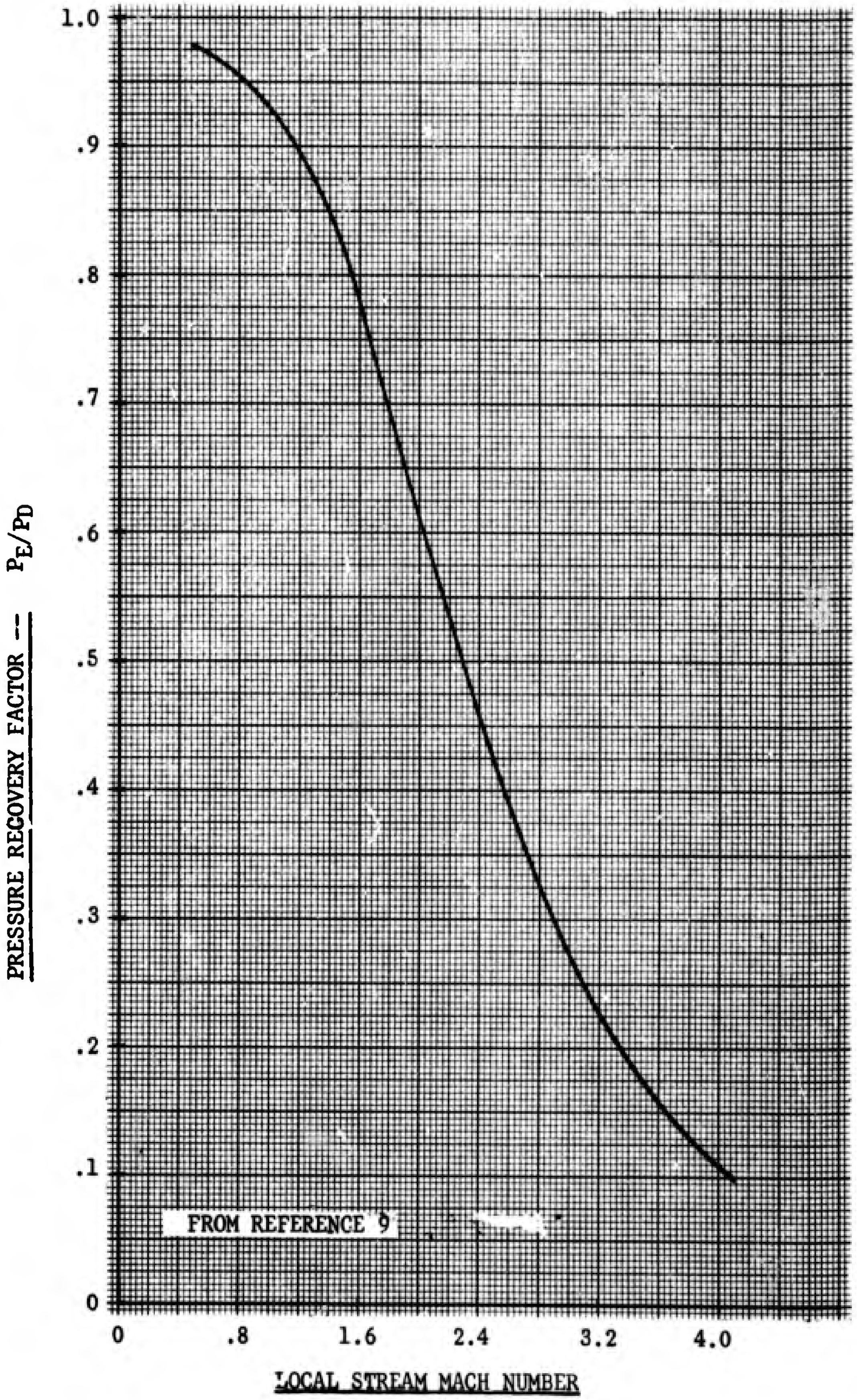





FIGURE E.1

$$\dot{m}_E = (831)(1.3)(14.69)/\sqrt{519} = 1141 \text{ lb/min}$$

$$\dot{m}_E (135 - T_F) = 1141 (135 - 115) = \underline{22,820 \text{ lb}^oF\text{/min}}$$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - 1/\left(\frac{295.0}{14.67}\right)^{.283} \right] \right\} - 460 = -22^oF$$

$$\dot{m}_7 = \frac{25,500}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182 \ (-35-45)}{(860 - 350)} \right. = 3.15 \text{ lb/min}$$

$$\dot{m}_E (135 - T_G) = (1428)(2.04) = \underline{4,496 \text{ lb}^oF\text{/min}}$$

$Q_{CAB}$ = 27,500 $T_B$ = 216 M = 1.25

From Figure E.1: $P_E/P_D$ = .886 $P_E$ = 31.6

$$T_F = 676 \left\{ 1 - .7 \left[ 1 - 1/\left(\frac{31.6}{14.69}\right)^{.283} \right] \right\} - 460 = 124^oF$$

$\dot{m}_E$ = 1192 lb/min $\dot{m}_E (135 - T_F)$ = 1192 (135 - 124) = $\underline{13,110 \text{ lb}^oF\text{/min}}$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - 1/\left(\frac{295.0}{14.67}\right)^{.283} \right] \right\} - 460 = -22^oF$$

$$\dot{m}_7 = \frac{27,500}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182 \ (-35-45)}{(860 - 350)} \right] = 3.40 \text{ lb/min}$$

$$\dot{m}_E (135 - T_G) = (1428)(3.40) = \underline{4,850 \text{ lb}^oF\text{/min}}$$

| | | | |
|---|---|---|---|
| By interpolation: | ALTITUDE | = | Sea Level |
| | MACH NO. | = | 1.27 |
| | HEAT LOAD | = | 28,800 BTU/hr |

---

**15,000 Feet**

From Figure 3.5, Figure 4.1, Reference 2:

| | | | |
|---|---|---|---|
| $T_A$ = 908 | $P_A$ = 270.0 | $T_D$ = 271 | $P_D$ = 35.20 |
| $Q_{CAB}$ = 35,700 | $P_C$ = 10.90 | M = 1.7 | P = 8.2 |

From Figure E.1: $P_E/P_D = .736$ $P_E = 25.91$

$$T_F = 731 \left\{ 1 - .7 \left[ 1 - 1/\left(\frac{25.91}{8.2}\right)^{.283} \right] \right\} - 460 = 128^oF$$

$\dot{m}_E = 538$ lb/min $\dot{m}_E (135 - T_F) = 538 (135 - 128) = \underline{3,770 \text{ lb}^oF/\text{min}}$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - 1/\left(\frac{270.0}{10.90}\right)^{.283} \right] \right\} - 460 = 11^oF$$

$$\dot{m}_7 = \frac{35,700}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182 (11-45)}{(908 - 350)} \right] = 5.66 \text{ lb/min}$$

$\dot{m}_E (135 - T_F) = (1428)(5.66) = \underline{8,080 \text{ lb}^oF/\text{min}}$

$Q_{CAB} = 30,500$ $T_D = 243$ $M = 1.6$

From Figure E.1: $P_E/P_D = .779$ $P_E = 27.4$

$$T_F = 703 \left\{ 1 - .7 \left[ 1 - 1/\left(\frac{27.4}{8.2}\right)^{.283} \right] \right\} - 460 = 100^oF$$

$\dot{m}_E = 506$ lb/min $\dot{m}_E (135 - T_F) = 506 (135 - 100) = \underline{17,710 \text{ lb}^oF/\text{min}}$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - 1/\left(\frac{270.0}{10.9}\right)^{.283} \right] \right\} - 460 = 11^oF$$

$$\dot{m}_7 = \frac{30,500}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182 (11-45)}{(908 - 350)} \right] = 4.83 \text{ lb/min}$$

$\dot{m}_E (135 - T_F) = (1428)(4.83) = \underline{6,900 \text{ lb}^oF/\text{min}}$

By interpolation:

| | | |
|---|---|---|
| ALTITUDE | = | 15,000 feet |
| MACH NO. | = | 1.67 |
| HEAT LOAD | = | 34,100 BTU/hr |

---

25,000 Feet

From Figure 3.5, Figure 4.1, and Reference 2:

$T_A = 911$ $P_A = 231.0$ $T_D = 249$ $P_D = 31.50$

$Q_{CAB} = 30{,}500$ $P_C = 10.4$ $M = 1.8$ $P = 5.4$

From Figure E.1: $P_E/P_D = .692$ $P_E = (31.5)(.692) = 21.8$

$$T_F = 709\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{21.8}{5.4}\right)^{.283}\right]\right\} - 460 = 87^{\circ}F$$

$\dot{m}_E = 389$ lb/min $\dot{m}_E\,(135 - T_F) = 389\,(135 - 87) = \underline{18{,}670 \text{ lb}^{\circ}\text{F/min}}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{231.0}{10.4}\right)^{.283}\right]\right\} - 460 = 18^{\circ}F$$

$$\dot{m}_7 = \frac{30{,}500}{14.4}\Big/\left[249 + \frac{65000 - 1182\,(18-45)}{(911 - 350)}\right] = 5.02 \text{ lb/min}$$

$\dot{m}_E\,(135 - T_F) = (1428)\,(5.02) = \underline{7{,}170 \text{ lb}^{\circ}\text{F/min}}$

$Q_{CAB} = 35{,}400$ $T_D = 282^{\circ}F$ $M = 1.9$

From Figure E.1: $P_E/P_D = .652$ $P_E = 20.5$

$$T_F = 742\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{20.5}{5.4}\right)^{.283}\right]\right\} - 460 = 123^{\circ}F$$

$\dot{m}_E = 411$ lb/min $\dot{m}_E\,(135 - T_F) = 411\,(135 - 123) = \underline{4{,}930 \text{ lb}^{\circ}\text{F/min}}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{231.0}{10.4}\right)^{.283}\right]\right\} - 460 = 18^{\circ}F$$

$$\dot{m}_7 = \frac{35{,}400}{14.4} \quad 249 + \frac{65000 - 1182\,(18-45)}{(911 - 350)} = 5.83 \text{ lb/min}$$

$\dot{m}_E\,(135 - T_F) = (1428)\,(5.83) = \underline{8{,}330 \text{ lb}^{\circ}\text{F/min}}$

By interpolation:

| | | |
|---|---|---|
| ALTITUDE | = | 25,000 feet |
| MACH NO. | = | 1.88 |
| HEAT LOAD | = | 34,900 BTU/hr |

36,000 Feet

From Figure 3.5, Figure 4.1, and Reference 2:

$T_A = 926$ $P_A = 192.0$ $T_D = 291$ $P_D = 30.10$

$Q_{CAB} = 34,900$ $P_C = 8.25$ $M = 2.15$ $P = 3.25$

From Figure E.1: $P_E/P_D = .555$ $P_E = 16.70$

$$T_F = 751 \left\{ 1 - .7 \left[ 1 - {}^{1}/\left(\frac{16.7}{3.25}\right)^{.283} \right] \right\} - 460 = 96^{o}F$$

$\dot{m}_E = 294$ lb/min $\quad \dot{m}_E\,(135 - T_F) = 294\,(135 - 96) = \underline{11,470 \text{ lb}^{o}\text{F/min}}$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - {}^{1}/\left(\frac{192.0}{8.25}\right)^{.283} \right] \right\} - 460 = -32^{o}F$$

$$\dot{m}_7 = \frac{34,900}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182\,(-32-45)}{(926 - 350)} \right] = 4.65 \text{ lb/min}$$

$$\dot{m}_E\,(135 - T_F) = (1428)\,(4.65) = \underline{6,640 \text{ lb}^{o}\text{F/min}}$$

$Q_{CAB} = 37,300$ $T_D = 309$ $M = 2.2$

From Figure E.1: $P_E/P_D = .533$ $P_E = 16.04$

$$T_F = 769 \left\{ 1 - .7 \left[ 1 - {}^{1}/\left(\frac{16.04}{3.25}\right)^{.283} \right] \right\} - 460 = 114^{o}F$$

$\dot{m}_E = 301$ lb/min $\quad \dot{m}_E\,(135 - T_F) = 301\,(135 - 114) = \underline{6,320 \text{ lb}^{o}\text{F/min}}$

$$T_C = 810 \left\{ 1 - .8 \left[ 1 - {}^{1}/\left(\frac{192.0}{8.25}\right)^{.283} \right] \right\} - 460 = -32^{o}F$$

$$\dot{m}_7 = \frac{37,300}{14.4} \Big/ \left[ 249 + \frac{65000 - 1182\,(-32-45)}{(926 - 350)} \right] = 4.97 \text{ lb/min}$$

$$\dot{m}_E\,(135 - T_F) = (1428)\,(4.97) = \underline{7,100 \text{ lb}^{o}\text{F/min}}$$

By interpolation:

| | | |
|---|---|---|
| ALTITUDE | = | 36,000 feet |
| MACH NO. | = | 2.19 |
| HEAT LOAD | = | 37,100 BTU/hr |

## 40,000 Feet

From Figure 3.5, Figure 4.1, and Reference 2:

$T_A = 952$ $P_A = 162.0$ $T_D = 325$ $P_D = 31.50$

$Q_{CAB} = 38{,}500$ $P_C = 7.72$ $M = 2.25$ $P = 2.72$

From Figure E.1: $P_E/P_D = .513$ $P_E = 16.16$

$$T_F = 785\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{16.16}{2.72}\right)^{.283}\right]\right\} - 460 = 108^{\circ}F$$

$\dot{m}_E = 258$ lb/min $\dot{m}_E\,(135 - T_F) = 258\,(135 - 108) = \underline{6{,}970\ lb^{\circ}F/min}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{162.0}{7.72}\right)^{.283}\right]\right\} - 460 = -24^{\circ}F$$

$$\dot{m}_7 = \frac{38{,}500}{14.4}\Big/\left[249 + \frac{65000 - 1182\,(-24-45)}{(952 - 350)}\right] = 5.42\ lb/min$$

$\dot{m}_E\,(135 - T_F) = (1428)\,(5.42) = \underline{7{,}740\ lb^{\circ}F/min}$

$Q_{CAB} = 36{,}100$ $T_D = 309$ $M = 2.2$

$T_A = 937$ $P_A = 158.1$ $P_D = 30.6$

From Figure E.1: $P_E/P_D = .533$ $P_E = 16.31$

$$T_F = 769\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{16.31}{2.72}\right)^{.283}\right]\right\} - 460 = 94^{\circ}F$$

$\dot{m}_E = 252$ lb/min $\dot{m}_E\,(135 - T_F) = 252\,(135 - 94) = \underline{10{,}330\ lb^{\circ}F/min}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{162.0}{7.72}\right)^{.283}\right]\right\} - 460 = -24^{\circ}F$$

$$\dot{m}_7 = \frac{36{,}100}{14.4}\Big/\left[249 + \frac{65000 - 1182\,(-24-45)}{(937 - 350)}\right] = 4.91\ lb/min$$

$\dot{m}_E\,(135 - T_F) = (1428)\,(4.91) = \underline{7{,}010\ lb^{\circ}F/min}$

By interpolation:

ALTITUDE = 40,000 feet
MACH NO. = 2.24
HEAT LOAD = 38,000 BTU/hr

55,000 Feet

From Figure 3.5, Figure 4.1, and Reference 2:

$T_A = 910$ $P_A = 74.6$ $T_D = 242$ $P_D = 10.32$

$Q_{CAB} = 25{,}000$ $P_C = 6.35$ $M = 2.0$ $P = 1.35$

From Figure E.1: $P_E/P_D = .610$ $P_E = 6.30$

$$T_F = 702\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{6.30}{1.35}\right)^{.283}\right]\right\} - 460 = 69^{\circ}F$$

$\dot{m}_E = 113.6$ lb/min $\dot{m}_E\ (135 - T_F) = 113.6\ (135 - 69) = \underline{7{,}500\ lb^{\circ}F/min}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{74.6}{6.35}\right)^{.283}\right]\right\} - 460 = 24^{\circ}F$$

$$\dot{m}_7 = \frac{25{,}000}{14.4}\Big/\left[249 + \frac{65000 - 1182\ (24-45)}{(910 - 350)}\right] = 4.24\ lb/min$$

$\dot{m}_E\ (135 - T_F) = (1428)\ (4.24) = \underline{6{,}050\ lb^{\circ}F/min}$

$Q_{CAB} = 26{,}700$ $T_D = 258$ $M = 2.05$

From Figure E.1: $P_E/P_D = .591$ $P_E = 6.10$

$$T_F = 718\left\{1 - .7\left[1 - {}^{1}\!/\!\left(\frac{6.10}{1.35}\right)^{.283}\right]\right\} - 460 = 84^{\circ}F$$

$\dot{m}_E = 116.4$ lb/min $\dot{m}_E\ (135 - T_F) = 116.4\ (135 - 84) = \underline{5{,}940\ lb^{\circ}F/min}$

$$T_C = 810\left\{1 - .8\left[1 - {}^{1}\!/\!\left(\frac{74.6}{6.35}\right)^{.283}\right]\right\} - 460 = 24^{\circ}F$$

$$\dot{m}_7 = \frac{26{,}700}{14.4}\Big/\left[249 + \frac{65000 - 1182\ (24-45)}{(910 - 350)}\right] = 4.53\ lb/min$$

$\dot{m}_E\ (135 - T_F) = (1428)\ (4.53) = \underline{6{,}470\ lb^{\circ}F/min}$

By interpolation:

| | | |
|---|---|---|
| ALTITUDE | = | 55,000 feet |
| MACH NO. | = | 2.04 |
| HEAT LOAD | = | 26,400 BTU/hr |

57,500 Feet

From Figure 3.5, Figure 4.1, and Reference 2:

$T_A = 877$ $\quad P_A = 59.0$ $\quad T_D = 212$ $\quad P_D = 6.73$

$Q_{CAB} = 20,600$ $\quad P_C = 6.20$ $\quad M = 1.9$ $\quad P = 1.20$

From Figure E.1: $\quad P_E/P_D = .651$ $\quad P_E = 4.38$

$$T_F = 672 \left\{ 1 - .7 \left[ 1 - 1/\left(\frac{4.38}{1.20}\right)^{.283} \right] \right\} - 460 = 68^oF$$

$\dot{m}_E = 96.0$ lb/min $\quad \dot{m}_E\,(135 - T_F) = 96.0\,(135 - 68) = \underline{6,430 \text{ lb}^oF/\text{min}}$

$$\dot{m}_7 = \left(\frac{20,600}{16,500}\right) 3.06 = 3.82 \text{ lb/min}$$

$\dot{m}_E\,(135 - T_F) = (1428)(3.82) = \underline{5,450 \text{ lb}^oF/\text{min}}$

$Q_{CAB} = 22,600$ $\quad T_F = 227$ $\quad M = 1.95$

From Figure E.1: $\quad P_E/P_D = .627$ $\quad P_E = 4.22$

$$T_F = 687 \left\{ 1 - .7 \left[ 1 - 1/\left(\frac{4.22}{1.20}\right)^{.283} \right] \right\} - 460 = 83^oF$$

$\dot{m}_E = 98.4$ lb/min $\quad \dot{m}_E\,(135 - T_F) = 98.4\,(135 - 83) = \underline{5,120 \text{ lb}^oF/\text{min}}$

$$\dot{m}_7 = \left(\frac{22,600}{16,500}\right) 3.06 = 4.19 \text{ lb/min}$$

$\dot{m}_E\,(135 - T_F) = (1428)(4.19) = \underline{5,980 \text{ lb}^oF/\text{min}}$

By interpolation:

| | | |
|---|---|---|
| ALTITUDE | = | 57,500 feet |
| MACH NO. | = | 1.93 |
| HEAT LOAD | = | 21,800 BTU/hr |

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13. ABSTRACT

The results of an analytical study investigating the feasibility of a new and unique environmental control concept for the cabin air conditioning requirements of the Navy F-4 fighter aircraft are reported. The heat-actuated Conductron environmental control concept is basically a vapor compression system which utilizes two thermodynamic loops and a unique device which replaces the conventional compressor. It is shown that an environmental control system based on the Conductron concept can be adapted to the Navy F-4 fighter aircraft with the existing constraints on weight, size, and power. The Conductron system can provide both cooling and heating under all aircraft operating conditions, including ground-static. Significant reductions in the compressor bleed air requirements, as compared to the present Navy F-4 air-cycle air conditioning system, can be realized with the Conductron environmental control system.

The Conductron ECS can be used for cabin and equipment environmental control of all high-performance and subsonic aircraft. The specific results of this study, however, are applicable to the Navy F-4 fighter aircraft only.

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| 14. KEY WORDS | LINK A | | LINK B | | LINK C | |
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| Supersonic fighters<br>Cabin air conditioning<br>Heat-actuated environmental control system<br>Dual-vapor compression cycle<br><br>Navy F-4 (airplane) | | | | | | |

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