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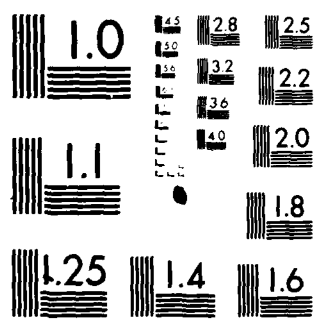

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COMBAT VEHICLE COOLING/HEATING DESIGN INVESTIGATION

FINAL REPORT

Contract Number DAAK70-80-C-0162

Prepared for: U.S. Army Mobility Equipment
Research and Development Command

By: Hamilton Standard
Division of United Technologies

"The views, opinions, and findings contained in this report are those of Hamilton Standard and should not be construed as an official Department of the Army position, policy, or decision, unless so designated by other documentation."

September, 1981

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SUMMARY

This study reviewed various combat vehicle crew compartment thermal conditioning concepts. A preliminary selection of candidate systems, based on performance, power, size, weight, logistics, cost and development status, elimated all concepts from further consideration except for air cycle and vapor cycle. Both the air and vapor cycle systems underwent a preliminary design evaluation and a detailed trade-off. Conclusions drawn from the preliminary design and trade-off of these two concepts are:

1. On a comparative point basis, the air and vapor cycles are equivalent.
2. The vapor cycle has distinct advantages in power and cost.
3. The air cycle has distinct advantages in performance and weight for outdoor conditions other than the main design condition of 120°F dry bulb temperature and 3% relative humidity.
4. The air cycle has a slight advantage in size and logistics.
5. The trade-off categories of reliability and noise, safety, simplicity, maintainability and other are comparable for the two concepts.

Either an air or vapor cycle can provide combat vehicle crew compartment conditioning. The implementation of either concept must therefore be based on a more detailed evaluation of the relative importance of the various trade-off categories, as determined by the Army. For example, if power is the item of primary importance, then the selection of the vapor cycle would be indicated. However, if a vehicle has power available, performance at climates other than hot-dry and conditioning system size and weight may be of primary importance and the installation of an air cycle system would be recommended.

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PREFACE

This study was performed under the direction of Mr. Oddvar Svaeri of the U.S. Army Mobility Equipment Research and Development Command, Ft. Belvoir, Virginia. Many sources were consulted during the vehicle requirements data gathering phase of the study. An attempt was made at each contributing facility to coordinate Hamilton Standard's request for data and the Army's response through a single individual. These facilities and the men who provided this coordinating function are as follows:

Mr. Milton DuBay	Tank Automotive Command (TACOM)	Warren, MI
Mr. Vincent Iacono	Natick Research and Development Laboratories (NLABS)	Natick, MA
Mr. Leonard Beeson	Chemical Systems Laboratory (CSL)	Aberdeen, MD
Capt. Hugh Bracey	Human Engineering Laboratory (HEL)	Aberdeen, MD
Major James Foster	Training and Doctrine Command (TRADOC)	Ft. Monroe, VA

Hamilton Standard personnel performing this program were:

Mr. Jeffrey M. Wehner	Principal Engineer, and author of Final Report
Mr. Donald S. Stein	Study Manager
Mr. George C. Rannenberg	Engineering Manager
Mr. Philip F. Heimlich	Senior Design Engineer
Mr. David L. Faye	Design Engineer
Mr. Richard P. Diefenderfer	Senior Analytical Engineer
Ms. Diane G. Drew	Analytical Engineer

HAMILTON STANDARD Division of UNITED TECHNOLOGIES

(Continued)

Mr.David Kuchinsky	Support Systems Engineering
Mr. Brad Dahlquist	Support Systems Engineering

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SECTION 1.0
INTRODUCTION

Since World War II, the United States, particularly the Department of Defense, has recognized the threats posed to personnel and equipment by the use of Nuclear, Biological or Chemical (NBC) agents in a future wartime situation. Attempts have been made to limit the development and/or stockpiling of weapons designed to deploy those agents by such agreements as the Biological Weapons Convention which came into force in 1975 and the Salt I agreement which was signed in 1976. These arms control measures have only been partially effective in achieving disarmament, and they contain major flaws which allow countries to continue research and development of new agents. U.S. military planners, noting the increasing emphasis among Warsaw Pact countries on NBC warfare, have launched intensive programs to improve the ability of U.S. military personnel to cope with these growing threats.

Historically, bacteriological war has never been employed, but chemical warfare was already well developed by World War I. With the close of World War II, the use of atomic weapons became another viable threat. Since that time, Warsaw Pact forces have continued to develop NBC warfare techniques, and there is evidence that their training includes the actual use of chemical agents. It is also obvious from intelligence reports and Eastern Bloc writings that they do not consider it beyond the realm of possibility to use Chemical or Biological (CB) warfare in conjunction with either nuclear or conventional warfare. The result of a war waged with these weapons against a military force that is not adequately protected is easy to predict. Thus, requirements have been established to update existing combat vehicles for CB warfare and to incorporate CB protection into new combat vehicles.

1.0 (Continued)

For years, thermal conditioning of combat vehicles has been avoided by opening hatches and doors, loosening and opening clothing, and by installing ventilation fans to pump large amounts of ambient air through the vehicle. In a CB environment, the crewmen are likely to be encumbered with CB protective clothing, which possesses thermal insulation properties. In addition, ambient air cooling without collective protection is unacceptable because of the need to keep CB agents out of the vehicle interior. The reduced vehicle interior ventilation rates necessary when operating in a CB environment result in increased crew member thermal stress. Therefore, CB protection requirements have resulted in a need for thermal conditioning systems for the combat vehicles.

Various system concepts have been considered to provide collective protection within vehicles. These system concepts are:

1. positive pressure, which prevents CB infiltration by overpressurizing the closed compartment with filtered, purified air.
2. ventilated facepiece, which delivers purified, filtered air to hoses that pressurize the masks worn by the combat vehicle crewmen who are also wearing individual protective ensembles
3. hybrid, which combines the positive pressure and the ventilated facepiece methods and provides the selection of various operating modes (open or closed hatch, masked or unmasked crewmen, etc.)
4. total, which combines the hybrid system with environmental control of the crew compartment (Reference 1).

For each of these four systems, crew compartment and electronics cooling can be accomplished in several ways:

1.0 (Continued)

1. Crew compartment and electronics are sufficiently cooled in the buttoned-up vehicle environment without ventilation.
2. Crew compartment and electronics are cooled with ambient ventilation air.
3. The entire crew compartment environment is conditioned with closed loop cooling.

On-going U.S. Army studies have concluded that a high crew compartment heat level exists in non-conditioned combat vehicles, especially when buttoned up in an NBC environment. This study, therefore, considered the situation where the entire crew compartment environment is conditioned with closed loop cooling; i.e., the macroclimate system. To avoid contamination of the crew compartment, this study only considered those systems that have hybrid collective protection equipment to prevent CB agent infiltration into the combat vehicle.

Another cooling concept, called the microclimate system, is also currently being studied by the Army to provide cooling to the individual crewmembers. In this concept, a fluid (either liquid or air) is circulated through tubes in a cooling garment worn by the crewman underneath his other clothing. Only the crewmen are cooled by this approach, while the crew compartment remains uncooled. This concept has not been considered in this study.

SECTION 2.0
VEHICLE REQUIREMENTS

INTRODUCTION

This study is divided into two major parts: vehicle requirements and system selection. The first part, vehicle requirements, involved the acquisition, classification, and tabulation of data regarding:

1. vehicle cooling requirements
2. existing cooling equipment
3. filter/overpressure constraints
4. power availability.

Each of these vehicle requirement classifications will be discussed in detail in this section. The second part of the study, system selection, involves trade-off studies and preliminary design of selected concepts.

2.1 VEHICLE COOLING REQUIREMENTS

A comparison of candidate cooling systems requires a baseline cooling load to size the equipment preparatory to trade-off studies. As Army operations are conducted world-wide, it is important that cooling and heating capacities be reviewed in various climates to ensure that the proper system design points are selected. The various climates considered during this study are summarized in Table 2-I (Reference 2).

Although earlier vehicle study results were reviewed to determine the variations of vehicle thermal loads in the various climatic categories, Army regulations concerning climate conditions were recently modified to those presented in Table 2-I. As a point of comparison, earlier studies were performed to the climatic conditions presented in Table 2-II (Reference 3). Thermal load characteristics are presented in Table 2-III for both the uninsulated and insulated M577/A1 Command Post Carrier (Reference 4). Tables 2-IV

TABLE 2-I

CLIMATES

Category	Dry Bulb Temperature (°F)	Relative Humidity (%)	Absolute Humidity (GR/#DA)	Wet-Bulb Temperature (°F)
Hot-Dry	120	3	15	67
Hot-Humid	105	59	201	91
Basic Constant High Humidity	75	100	131	75
Basic Variable High Humidity	95	74	186	87
Basic Hot	110	15	57	72
Basic Cold	-25	100	-	-25
Cold	-50	100	-	-50
Severe Cold	-60	100	-	-60

Reference: "Research, Development, Test and Evaluation of Materiel for Extreme Climatic Conditions"; Army Regulation AR 70-38; 1 August 1979.

TABLE 2-II

CLIMATES USED IN EARLIER ARMY STUDIES

Category	Description	Dry Bulb Temperature (°F)	Relative Humidity (%)	Absolute Humidity (GR/#DA)	Wet Bulb Temperature (°F)
1	Wet-Warm	75	100	131	75
2	Wet-Hot	95	74	186	87.2
3	Humid-Hot	100	64	188	88.5
4	Hot-Dry	125	5	29	71
5	Intermediate Hot-Dry	110	23	89	77.4
6	Intermediate Cold	-25	100		-25
7	Cold	-50	100		-50
8	Extreme Cold	-70	100		-70

Reference: "Research, Development, Test, and Evaluation of Material for Extreme Climatic Conditions"; Army Regulation AR 70-38; 5 May 1969

TABLE 2-III

M577/A1 COMMAND POST CARRIER THERMAL LOADS

Category	Description	Dry Bulb Temperature (°F)	Relative Humidity (%)	Uninsulated Heat Load (Btu/hr)	Insulated Heat Load (Btu/hr)
1	Wet-Warm	75	100	12120	15650
2	Wet-Hot	95	74	43980	28280
3	Humid-Hot	100	64	48370	30110
4	Hot-Dry	125	5	59470	29820
5	Intermediate Hot-Dry	110	23	48610	25920
6	Intermediate Cold	-25	100		
7	Cold	-50	100		
8	Extreme Cold	-70	100	-94925	-25970

Notes:
1. Air condition to 90°F dry bulb, 76°F wet bulb
2. Heat to 60°F
3. Negative latent load of 3800 Btu/hr in Category 4 is neglected

Reference: DuBay, Milton C.; "Engineering Analysis on Crew Compartment Heat Load and Auxiliary Power Unit Problems for the M577/A1 Command Post Carrier with Collective Protection Equipment"; Report AD-B047851 L; U.S. Army Tank Automotive Research and Development Command; Warren, Michigan; March 1980.

TABLE 2-IV

U. S. ROLAND THERMAL LOADS

Category	Description	Dry Bulb Temperature (°F)	Relative Humidity (%)	Heat Load (Btu/hr)
1	Wet-Warm	75	100	
2	Wet-Hot	95	74	
3	Humid-Hot	100	64	
4	Hot-Dry	125	0	24923
5	Intermediate Hot-Dry	110	23	
6	Intermediate Cold	-25	100	-12382
7	Cold	-50	100	
8	Extreme Cold	-70	100	

Notes:
1. Air condition to 90°F dry bulb, 50% relative humidity.
2. Heat to 50°F dry bulb, 50% relative humidity.
3. The hot-dry day was assumed to have 0% relative humidity.

Reference: "U.S. Roland Fire Unit Environmental Control Subsystem, Roland System Description, Volume IV, Book 6, Part B"; Contract No. DAAK40-75-C-0399, Report No. ROL 1040-1; Hughes Aircraft Company; Canoga Park, California; March 1977.

2.1 (Continued)

and 2-V present the thermal loads for the U.S. Roland and the Landing Vehicle Tracked Experimental (LVTX), respectively (Reference 5, 6). The thermal loads for three sizes of LVTX are also presented in Table 2-V. In addition to these tabulated thermal loads, a study of the M-60 estimated a 60,000 Btu/hr thermal load to maintain a 90°F crew compartment when ambient conditions were 110°F dry bulb and 83.2°F wet bulb (Reference 7).

However, detailed thermal analysis did not exist for many of the vehicles being considered in this study, notably the M-1, M-2 and M-3. Therefore, a representative cooling load of 48,000 Btu/hour (4 tons) for the hot-dry climate for an uninsulated vehicle was selected by the Contracting Officer's Representative to compare the various concepts. This compares with 5 tons for the M-60 and uninsulated M577/A1, 2 tons for the U.S. Roland, and 4 tons for an MBT design discussed in Section 2.2. Other climates, notably the hot-humid category, will also be considered because of the performance and sizing effects of high humidity on the crew compartment conditioning equipment. Therefore, in this report, when the term "off-design" is used, it refers to conditions other than the hot-dry climate.

The final item that affects the equipment sizing is the required crew compartment temperature. Previous Army studies had fixed 85°F effective temperature (ET) as the maximum allowable vehicle temperature to prevent personnel performance degradation. Selected design conditions of 81.3°F ET were typical to provide adequate safety margin in the event of system performance degradation.

An additional consideration is that the amount and type of clothing worn by the crew impacts the crew compartment temperature required to prevent or minimize thermal stress. U.S. Army Natick Research and Development Laboratories

TABLE 2-V

LANDING VEHICLE TRACKED, EXPERIMENTAL, THERMAL LOADS

Category	Description	Dry Bulb Temperature (°F)	Relative Humidity (%)	Heat Load (Btu/Hour) Small	Medium	Large
1	Wet-Warm	75	100	26153	33655	41353
2	Wet-Hot	95	74	37910	46224	55726
3	Humid-Hot	100	64	39117	47597	57199
4	Hot-Dry	125	5	33934	43247	53353
5	Intermediate Hot-Dry	110	23	34571	43383	53187
6	Intermediate Cold	-25	100			
7	Cold	-50	100			
8	Extreme Cold	-60	100	- 3536	- 8041	-13926

Notes:
1. Air condition to 90°F dry bulb, 53% relative humidity.
2. Heat to 45°F dry bulb, 50% relative humidity.

Reference: "A Conceptual Evaluation of the Total Environmental Control System for the Landing Vehicle Tracked, Experimental"; Contract N00167-80-M-480; Advanced Technology, Inc.; McLean, Virginia; 26 November 1979.

2.1 (Continued)

(NLABS) analyzed the temperature requirements to hold thermal casualties in the 10 to 30 percent range (for two hours) for seven types of clothing:

1. Combat Vehicle Crewman (CVC) summer suit
2. CVC suit plus armored vest
3. CVC suit plus armored vest plus ventilated facepiece
4. CVC suit plus open CB protective ensemble
5. CVC suit plus closed CB protective ensemble
6. CVC suit plus armored vest plus open CB protective ensemble
7. CVC suit plus armored vest plus closed CB protective ensemble

These categories of clothing were selected to encompass all possible mission scenarios being considered by the U.S. Army Training and Doctrine Command (TRADOC) for incorporation with hybrid collective protection equipment. Table 2-VI presents the results of this casualty analysis by NLABS for a crewman workload of 250 watts and 20 feet/min of air turbulence. This air turbulence assumption yields a conservative temperature requirement as crew compartment conditioning is likely to result in higher air circulation rates (Reference 8).

Combat vehicle mission scenarios, crewmen uniforms and acceptable casualty rates are still being studied by TRADOC. For this reason, a clear-cut selection of a crew compartment design effective temperature was not possible for this study. Agreement with the Contracting Officer's Representative established a crew compartment design goal of 90°F dry bulb temperature and 20 percent relative humidity for the hot dry climate. These conditions, which correspond to an effective temperature of less than 77°F, were used to perform candidate system selections and preliminary system design. However, crew compartment

TABLE 2-VI

NLABS THERMAL CASUALTY SURVEY

	CASUALTY PROBABILITY AT TWO (2) HOURS							
Effective Temperature (°F)	57	57.5	65.3	68	69	73	80	80
Dry Bulb Temperature (°F)	60	60	70	77	75	80	90	95
Relative Humidity (%)	20	40	40	20	40	40	40	22
<u>Crewman Clothing*</u>								
1. CVC Summer						.216	.317	.338
2. CVC + Vest						.239		
3. CVC + Mask + Vest		.124	.198			.280		
4. CVC + CB (Open)		.188						
5. CVC + CB (Closed)		.201						
6. CVC + Vest + CB (Open)		.185						
7. CVC + Vest + CB (Closed)	.202	.206	.269	.307	.302			

Reference: Iacono, V. D.; Memorandum dated 4 December 1980.

Note: Values in table provided by reference memorandum. Other data not available.

*Multiply values by 100 to get percent values.

2.1 (Continued)

relative humidities at off-design high humidity climates exceed the 20% target value. This is caused by several factors:

- for the vapor compression cycle, evaporator performance limits the amount of moisture that can be removed from the cooled air on humid days
- for the air cycle, various amounts of fresh air are bypassed either to satisfy equipment temperature requirements or to operate in non-NBC environments with higher make-up flow rates. This bypassed air at high ambient humidities raises the crew compartment relative humidity.

Detailed performance characteristics will be presented in Section 5.0 of this report.

Judicious placement of the conditioned air distribution ducting would make it possible to provide cool air to the crewman while allowing the return temperature to exceed 90°F by exhausting the air over the electronics equipment. This is tolerable because the electronics equipment is generally rated to 140°F. The effects of this distribution scheme would be to provide sufficient cooling of the crew with smaller equipment and less power.

However, as space within existing combat vehicles is very limited, it was assumed that placement of the distribution ducting needed to blow air on the men before exhausting over the electronics and returning to the conditioning equipment was not possible. Therefore, for purposes of comparing conditioning concepts and performing preliminary design, it was assumed that the crew compartment return temperature was at the specified 90°F crew compartment temperature.

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2.1 (Continued)

It should be noted that crew compartment thermal conditioning will not be required in all climates at all times. Operation in a temperate climate in a non-NBC environment will not require crew compartment conditioning. Thus, while the power requirements of candidate systems are extremely important and will be considered, the fuel consumption impact of crew compartment conditioning cannot be estimated.

2.2 EXISTING COOLING EQUIPMENT

Existing combat vehicle cooling and heating equipment was reviewed as to design, installation, and performance. This review was used to provide insight into existing equipment capabilities and problems associated with crew compartment cooling and emphasized vapor compression refrigeration units for the Main Battle Tank, U.S. Roland, and M-60. Air cycle refrigeration units being developed for European applications were also reviewed. Table 2-VII summarizes the characteristics of the equipment reviewed.

TABLE 2-VII
EXISTING COOLING EQUIPMENT

Vehicle	Type of System	Design Cooling Capacity (Btu/Hr)	System Weight (Pounds)	Required Power (kw)	Overall COP
MBT	Vapor	48,000	667	14	1.0
U.S. Roland	Vapor	30,000	547	9	1.0
M-60	Vapor	60,000	500	26	0.7

MBT - The Main Battle Tank environmental control unit is a vapor compression system using R-12 as the refrigerant. The unit consists of dual compressor condenser modules, a recirculating air module, a control panel, and a central equipment enclosure which houses contactors and relays to control the recirculatory air fan motors (References 9, 10). The compressor condenser modules are

2.2 (Continued)

mounted on the exterior of the vehicle via a cantilever frame from the sidewall of the tank. Each module weighs 264 pounds. The compressor is a six cylinder axial automotive air conditioning unit, and a special motor design was employed to minimize power consumption. The condenser is a plate and fin design, with a vane axial fan used to circulate 93 lb/min of air to each condenser to dissipate the heat.

The recirculating air module interfaces with the collective protection equipment. The unit consists of dual evaporators, dual fans, and liquid and electric heaters. Gross cooling capacity of the unit is approximately 59,000 Btu/hr under maximum design load conditions with a design requirement of 48,000 Btu/hr. The evaporators are basic automotive-type plate and fin units. The air circulating fans are high speed vane-axial designs with a 6.25 inch impeller diameter designed for a flow of 500 cfm and an air static rise of 6 inches water gage. The plate and fin liquid heater is designed for a 55,000 Btu/hr heating requirement with 180°F liquid. The tube and fin electric heater supplements the liquid heater at low temperatures. The overall weight of the recirculating air module is 120 pounds, and the normal air flow rate is 950 cfm.

The Main Battle Tank environmental control unit weighs 667 pounds, provides a net capacity of approximately 4 tons of cooling, and has a coefficient of performance of 3.48 Btu/Hr/Watt.

<u>U.S. Roland</u> - The U.S. Roland is a missile fire unit mounted on an M109 chassis (Reference 5). The environmental control unit is a closed vapor cycle system using R-22 as the refrigerant mounted externally on the left rear of the unit. Power for the environmental control unit is supplied by alternating current

2.2 (Continued)

power from the auxiliary power unit (PPU) which is also mounted on the rear of the vehicle. The mounting of the environmental control unit and auxiliary power unit are presented schematically in Figure 2-1. A hybrid collective protection unit has been added to the U.S. Roland and is mounted on the rear directly underneath the environmental control unit.

The refrigerant compressor is a welded hermetically sealed unit. Power for the unit is supplied by an induction motor with a motor input of 5900 watts. The condenser is an aluminum plate fin unit with a capacity of 55,100 Btu/hr at a condensing temperature of 150°F.

Both recirculating cabin air and outside make-up air are circulated over the evaporator. The evaporator is a tubular fin design with a capacity of 33,375 Btu/hr. This provides an exit air temperature of 58°F for a cabin return temperature of 90°F. A centrifugal fan circulates the air to the cabin at a rate of 800 cfm. Two banks of three tubular finned electric heating elements provide a total heating capability of 15,000 Btu/hr.

The complete environmental control system weighs 547 pounds. The system provides 30,000 Btu/hr of cooling while consuming approximately 9 kw of power.

M-60 - In the M-60 study, a vapor compression system was selected to provide 5 tons of air conditioning capacity (Reference 7). Dual evaporators of 30,000 Btu/hr capacity each are recommended with a separate compressor for each evaporator. The selected system uses dual condensers and thermal expansion valves. A single condenser fan and a single evaporator blower circulate the required air.

Power to the compressors and blowers is supplied by a hydraulic power

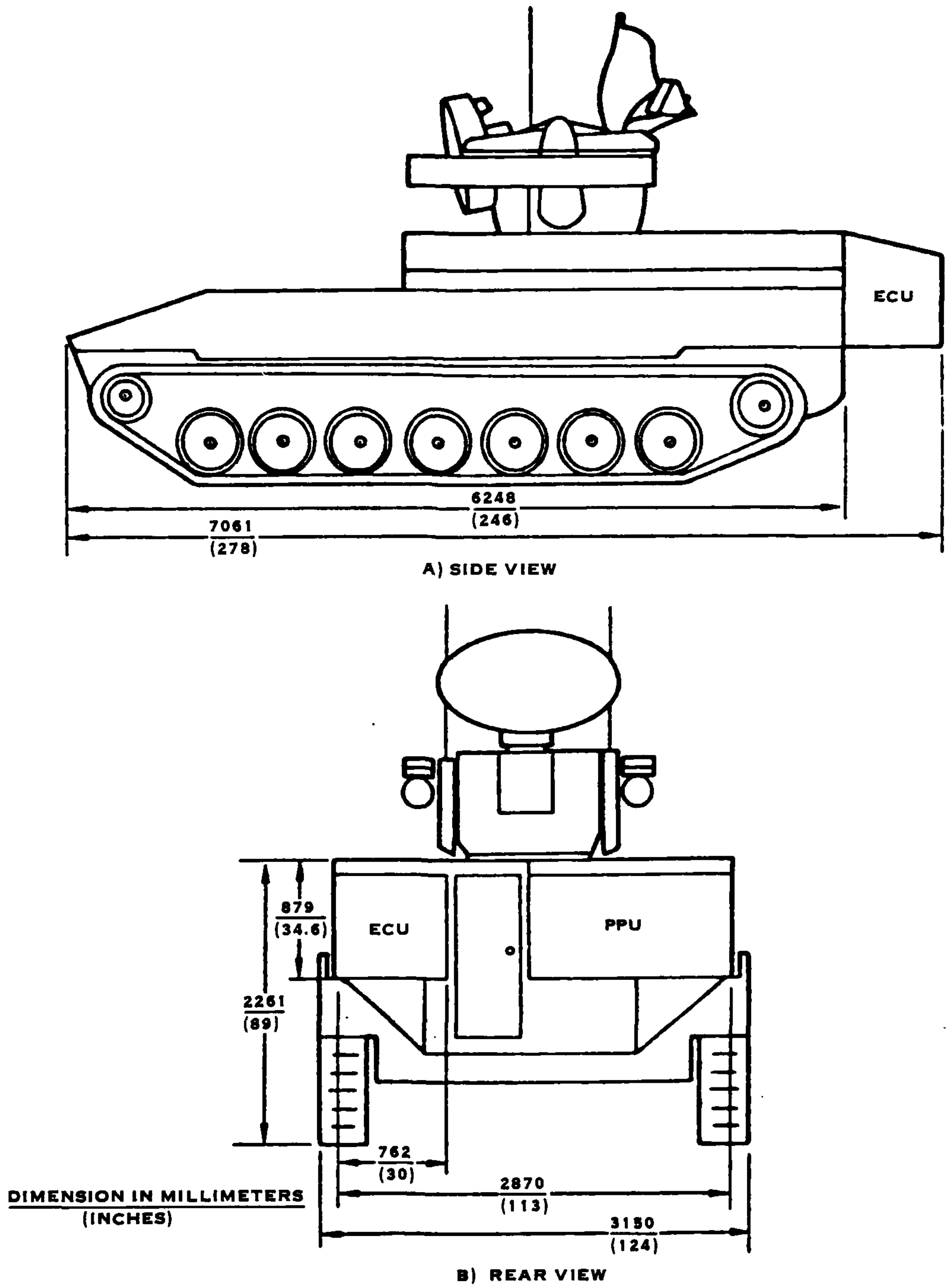

FIGURE 2-1. U.S. ROLAND ENVIRONMENTAL CONTROL UNIT MOUNTING

REFERENCE 5

2.2 (Continued)

system using existing components from other vehicles. This power system consists of dual hydraulic motors and a hydraulic pump. The hydraulic pump is driven through a right angle drive and a magnetic clutch by the tank engine at the transmission power take-off. However, a new gearbox design is needed to transmit power from the dual hydraulic motors to the blowers and compressors.

The complete system is estimated to weigh 500 pounds and to consume 35 hp. The major components are located in an enclosure mounted on the outside of the turret bustle to allow installation of an air conditioning system without requiring mission critical equipment removal. This mounting arrangement is presented schematically in Figure 2-2. This mounting arrangement could cause degradation of turret performance due to unbalance. However, this could only be checked by testing. An additional configuration concept where the condenser module is mounted on the outside of the turret bustle and the evaporator module is mounted on the inside of the turret bustle was also presented. This concept would have required the removal of 4 rounds of ammunition and the revision of interior turret stowage.

European Applications - An air cycle system is currently being developed by British Aerospace, Dynamics Group, a licensee of Hamilton Standard, for a European combat vehicle application. This system uses a turbocompressor cooling unit and a power compressor that is driven by a hydraulic motor. The system is designed for a 5°C (41°F) cooling air supply temperature and an overall cooling capacity of approximately 2.7 tons. System sizes, weights, and vehicle packaging were unavailable for inclusion in this report.

2.3 FILTER/OVERPRESSURE CONSTRAINTS

For the combat vehicles being studied, collective protection is required

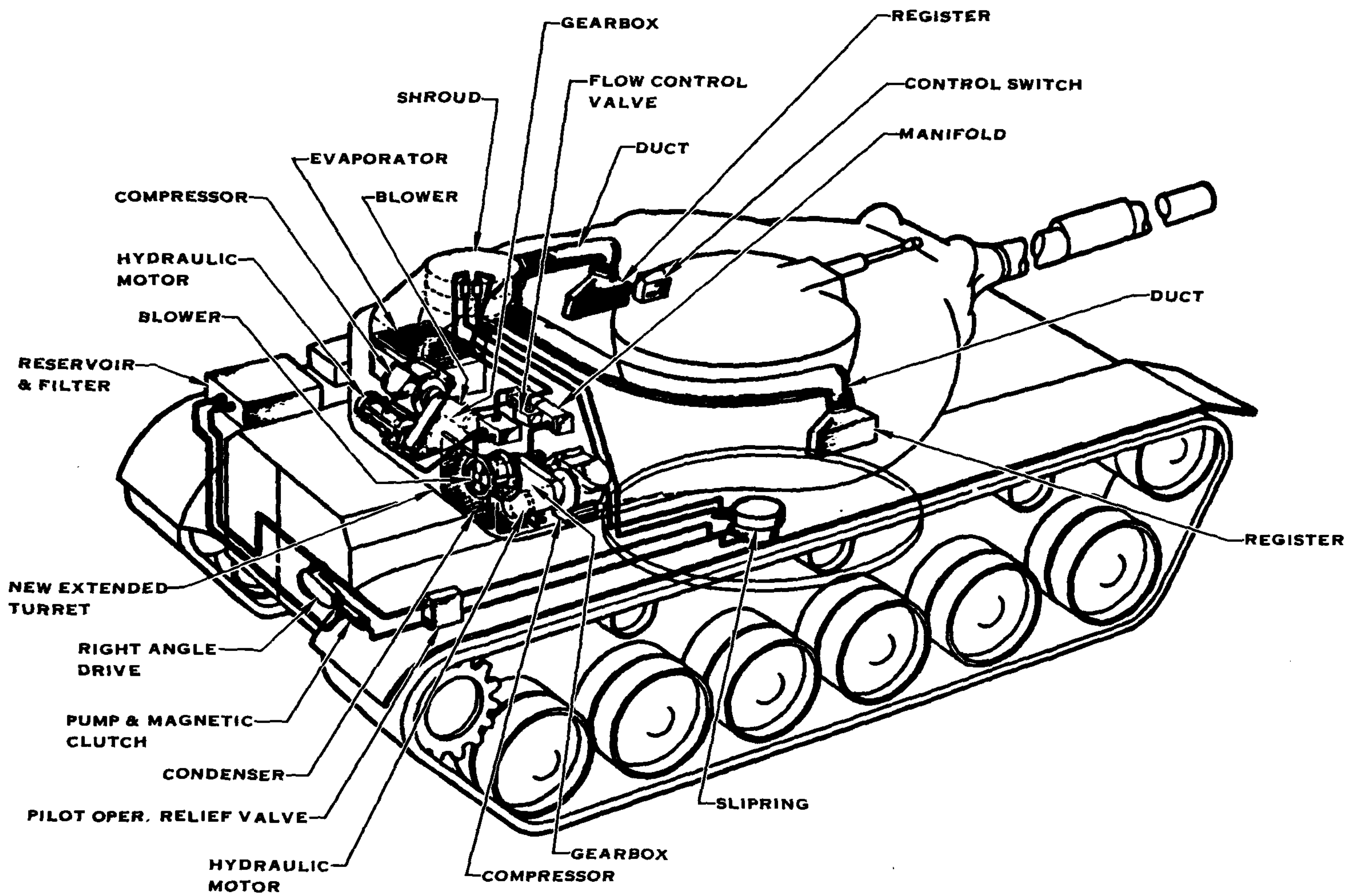

FIGURE 2-2. M-60 ENVIRONMENTAL CONTROL UNIT MOUNTING

REFERENCE 7

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2.3 (Continued)

to provide the crewmen with protection against chemical and biological agents. Therefore, it was assumed that a hybrid collective protection system integrating positive pressure and ventilated facepiece hardware will be part of the combat vehicle equipment.

As the hybrid collective protection equipment (HCPE) affects the air refrigerant make-up rates and crew compartment pressure levels, it was necessary to determine the operating characteristics of the HCPE to properly integrate it with the crew compartment thermal conditioning equipment. For this purpose, a coordination meeting was held with personnel from Chemical Systems Laboratory and Honeywell, the HCPE contractor. The HCPE, presented schematically in Figure 2-3 and pictorially in Figure 2-4, combines filters and blowers in a pressurized plenum. Operating characteristics were defined for three conditions:

a. NBC mode with air conditioning on
b. Non-NBC mode with air conditioning on
c. Full bypass mode with air conditioning off.

As the full bypass mode requires no air conditioning, its operating characteristics were not considered further in this study. However, ducting was provided so that this bypass air could reach the crew compartment without flowing through the cooling equipment of the conditioning systems. The operating characteristics of the HCPE for the first two modes with the air conditioning systems operating are summarized in Table 2-VIII.

As the fresh make-up air passing through the HCPE can affect the overall system cooling requirements, the heating effects of the HCPE must be considered. There are two sources of heat gain in the air passing through the HCPE. These are (1) heat transferred from the hot armor surrounding the inlet ducting

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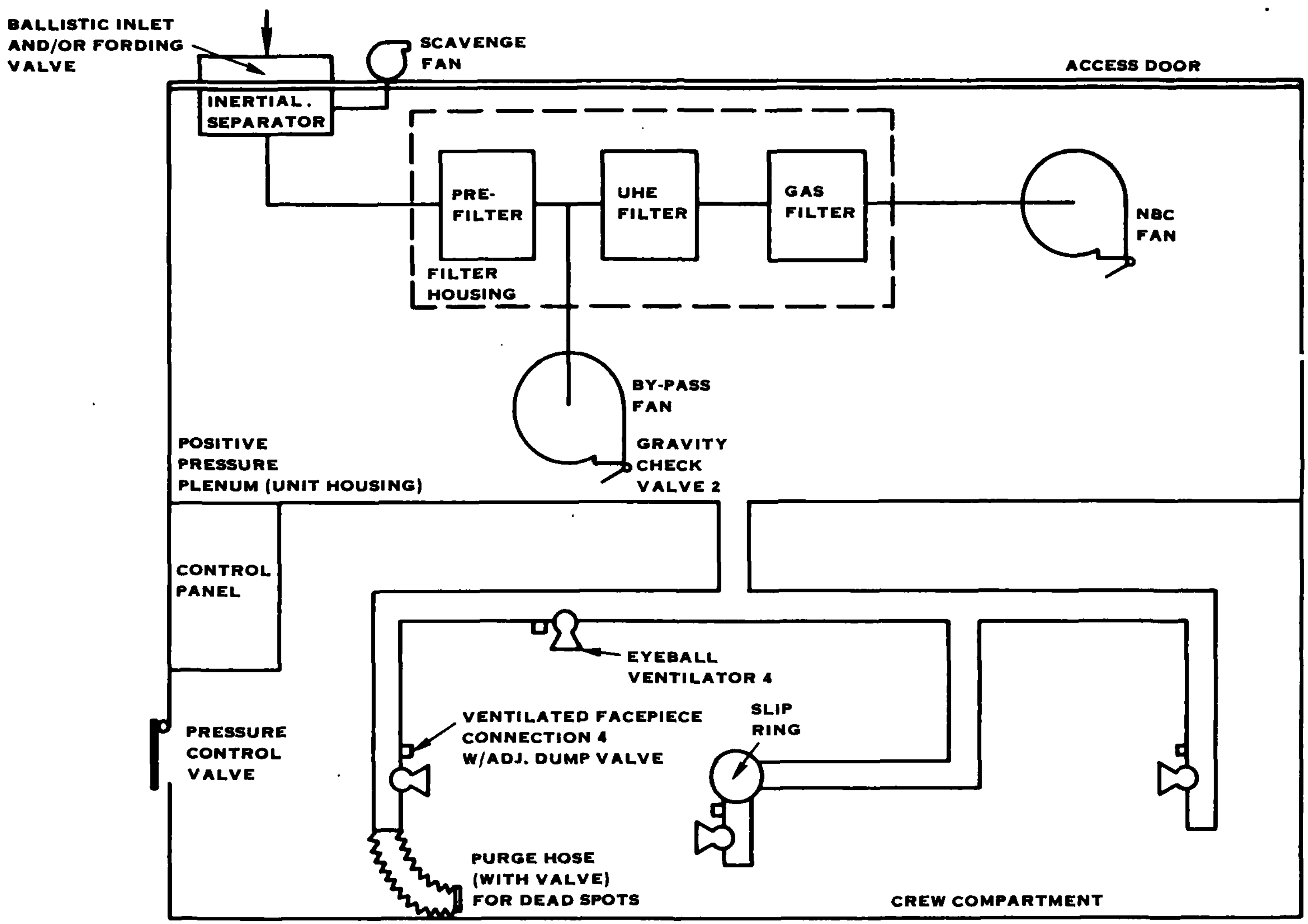

FIGURE 2-3. HCPE SYSTEM

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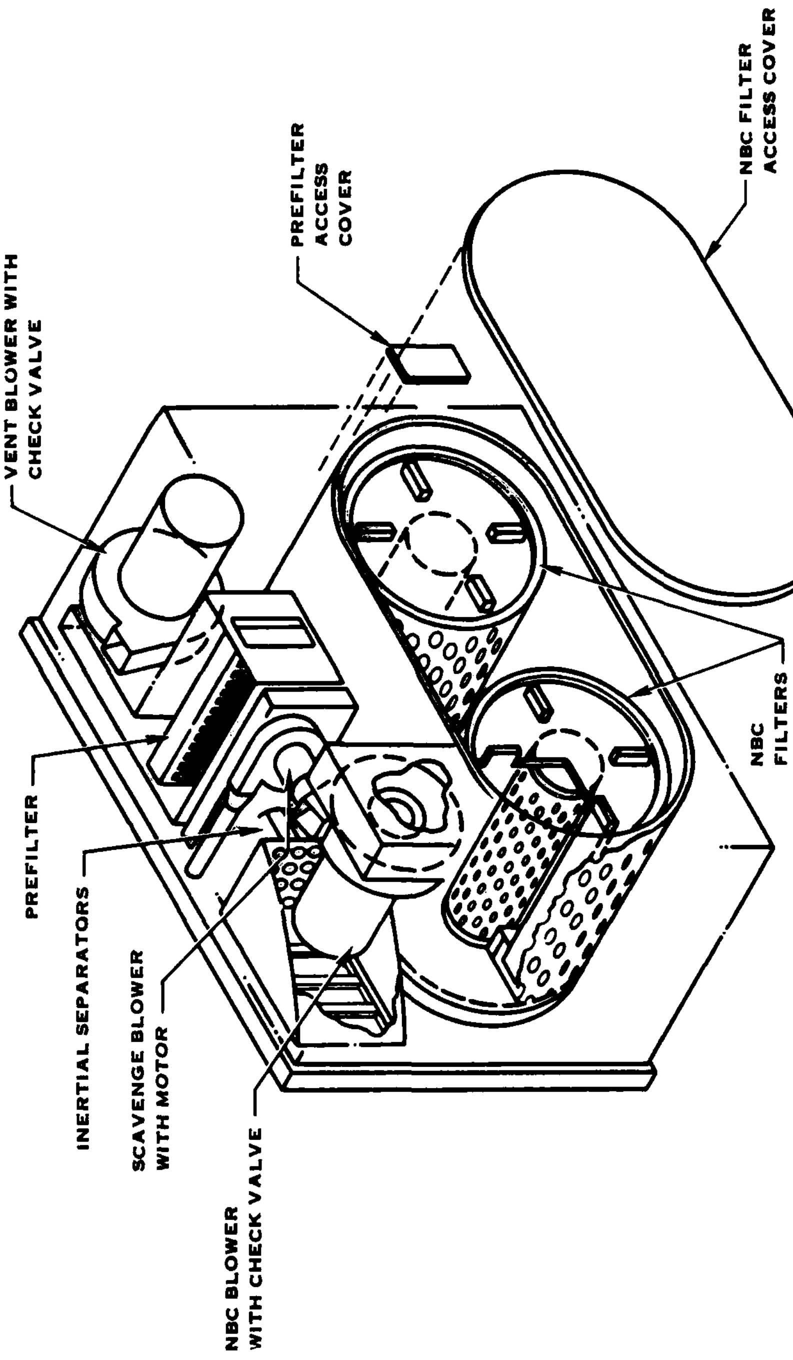

FIGURE 2-4. BASELINE FILTER - BLOWER PACK

TABLE 2-VIII

HYBRID COLLECTIVE PROTECTION EQUIPMENT OPERATING CHARACTERISTICS

A. NBC Mode, Air Conditioning On

Pressure at HCPE interface - 7" H_2O

Pressure in crew compartment - 1.5" H_2O

Filter flow - 300 cfm (with 4-5 cfm per man available for ventilated facepiece)

Blower power - 2 KW

B. Non-NBC Mode, Air Conditioning On

Pressure at HCPE interface - 7" H_2O

Filter flow - 600 cfm

Blower power - 2 KW

2.3 (Continued)

on a hot day and (2) heat from the blower motor. It was assumed that there would be a 5°F temperature rise with air flow of 300 cfm resulting from the hot armor surrounding the inlet ducting. This was proportioned to a 2.5°F rise for an air flow of 600 cfm during non-NBC operation. It was also conservatively assumed that all blower power (2 KW) was converted to heat and transferred to the HCPE air flow in the unit plenum. This is a worst case assumption for the effects of the blower motor on the crew compartment conditioning requirement.

2.4 POWER AVAILABILITY

Typically, existing combat vehicles have no excess power available. Only the XM-1 presently has available power of 12 KW when the vehicle is running. Furthermore, without an APU or other power source, these vehicles have no available power when in an engine-off mode, such as silent watch, which can be a high thermal stress mode if the vehicle is buttoned-up in an NBC protective posture. Therefore, one of the critical issues in crew compartment environmental conditioning is the power required to operate the air conditioning equipment.

During the program kick-off meeting, the Contracting Officer's Representative agreed that Hamilton Standard should minimize the power consumption of any selected environmental control concept undergoing preliminary design, but that no competitive concept should be eliminated strictly on the basis of power. Rather, the issues of power availability will be reviewed in detail by the Army after the selected crew compartment conditioning concepts are designed and traded-off.

However, applicable sources of power for crew compartment conditioning were reviewed during this study. Earlier studies of the M577/A1 (Reference 4)

2.3 (Continued)

considered several methods of powering crew compartment conditioning equipment. These methods included:

1. compressor driven by main engine hydraulic drive; electric condenser and evaporator fan
2. compressor belt driven from main engine through constant speed belt drive; electric condenser and evaporator fans
3. compressor and condenser fan driven from APU engines; electric evaporator fan
4. compressor and condenser fan driven from separate engine; electric evaporator fan
5. AC electric driven air cycle
6. AC electric driven vapor cycle.

During this earlier study, it was determined that the hydraulic drive lacked adequate power at engine idle and both the hydraulic drive and constant speed belt drive required continuous engine operation to provide cooling. The use of a non-standard APU required policy decisions from the Army and was also considered noisy. The use of a separate engine was eliminated because of suspected increased maintenance and size and weight penalties in addition to the standard APU. The electrically driven systems were both considered to be good, having long MTBF and reduced maintenance.

For this combat vehicle study, electrically driven components are favored for the vapor cycle application. When DC motors with brushes are used on vehicles, open or automotive type compressors are used because the brushes cannot be sealed hermetically. When an AC electric driven unit is employed, a hermetically sealed vapor cycle compressor-motor is used to eliminate refrigerant leakage around the compressor shaft. A motor driven system provides

2.4 (Continued)

more flexibility in the vehicle than engine mounted direct drives because they can be installed anywhere space is available. Additionally, belt drives are not recommended because of questionable reliability.

For the air cycle system, either an electric or hydraulic power source is favored for the flexibility of packaging the system in the vehicle. The hydraulic power source more closely matches the air cycle system power requirement and therefore may be more favorable for this application. If space is available, a direct gear drive of the power compressor may also be feasible.

SECTION 3.0
PRELIMINARY SELECTION OF CANDIDATE SYSTEMS

INTRODUCTION

Various refrigeration concepts were considered for macroclimate conditioning of the crew compartment after the vehicle requirements presented in the preceding section were determined. These concepts included air cycle, vapor compression cycle, positive displacement air cycle, thermoelectric, vortex tube, expendable heat sink, absorption cooling, and evaporative cooling. The advantages and disadvantages of each concept in the area of performance, power, size/weight, vulnerability, and logistics were considered. Other factors such as noise and development status were considered as secondary elements.

3.1 AIR CYCLE

The air cycle is a fully developed concept that has been used in aircraft compartment cooling for many years, particularly when a high pressure air engine bleed source is available. Air cycle systems are also used in applications requiring shaft driven compressors. This cycle is depicted schematically in Figure 3-1. The air cycle package is generally smaller and lighter than the vapor cycle, and, for this reason, is presently being applied to European combat vehicle cooling applications. This concept has the added advantage of possessing a combined cooling/heating capability.

The air cycle requires less logistical support than the vapor compression cycle in that leak detectors, make-up refrigerant, and vacuum pumps are not required. The system can also continue operation with minor pinhole leaks. However, if leakage areas in the high pressure ducting are of the same order of magnitude as the nozzle area, such as would result from a bullet hole, system performance will degrade.

3.1 (Continued)

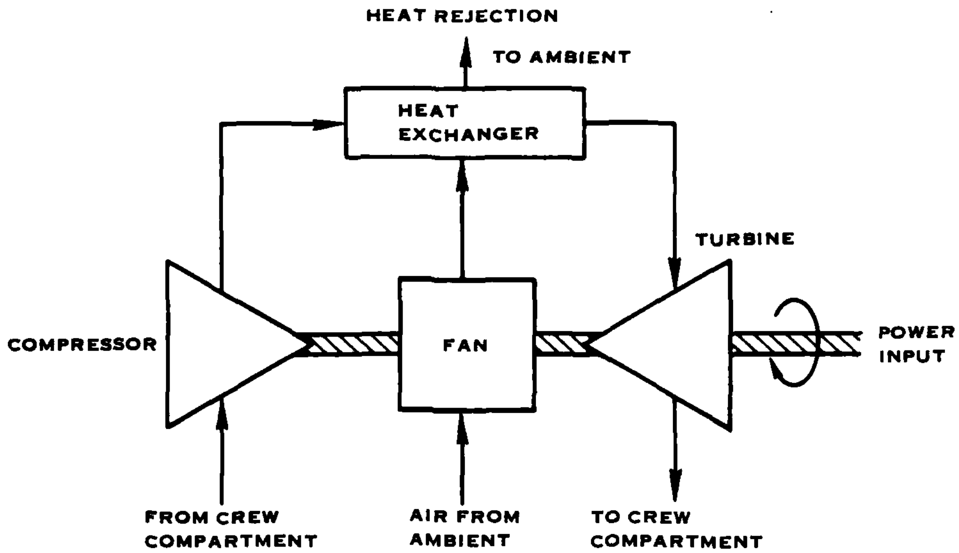

FIGURE 3-1. AIR CYCLE SYSTEM SCHEMATIC

The major disadvantage of the air cycle is that it requires more power than the vapor cycle. However, recent advances in the concept, such as "chilled recirc", can significantly reduce power as compared to the conventional air cycle in certain applications (Reference 11).

For the reasons listed above, the air cycle will be considered in more detail in the following sections. A description of the system operation will also be presented.

3.2 VAPOR COMPRESSION CYCLE

Like the air cycle, the vapor compression cycle (Figure 3-2) is commonly used for many refrigeration applications, most notably home and automobile air conditioning. The major advantage of this concept is that it requires less power than the air cycle.

3.2 (Continued)

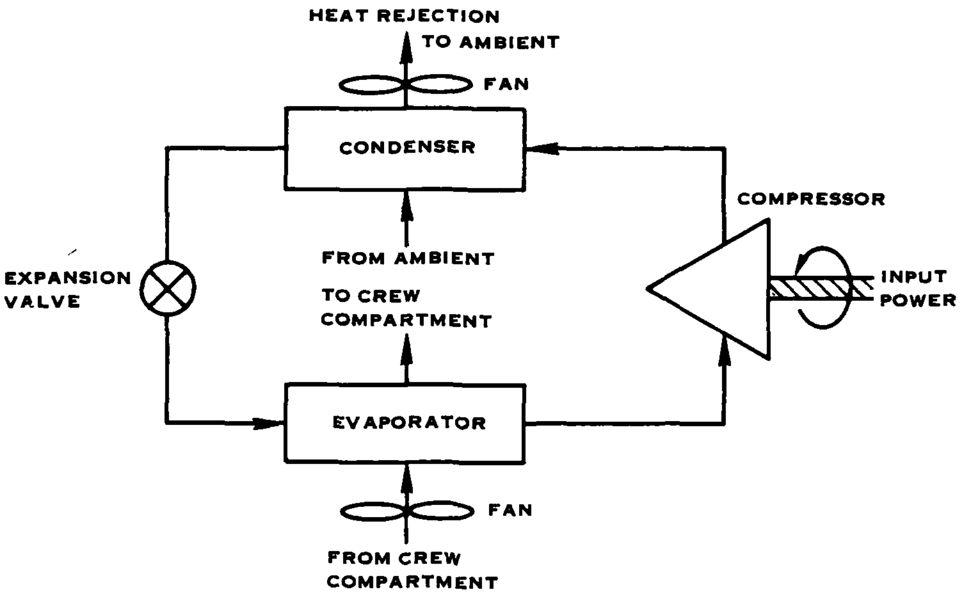

FIGURE 3-2. VAPOR COMPRESSION CYCLE SYSTEM SCHEMATIC

The major disadvantages of this concept when compared to the air cycle are:

- No combined cooling/heating capability.
- More stringent logistical requirements (refrigerant, leak detector, vacuum pump).
- Larger and heavier than the air cycle equipment.
- Minor pinhole leaks will shut down the system.

However, because of this concept's low power, it will also be considered in more detail in the following sections. A description of the system operation will also be presented.

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3.3 POSITIVE DISPLACEMENT AIR CYCLE

This concept currently under development for the Army uses proprietary vaned rotary machines to circulate air in a closed loop coolant system. This closed loop system, depicted schematically in Figure 3-3, interfaces with circulating air or liquid systems that provide the actual cooling. Several applications of this concept were studied. One involves a common housing for the compressor-expander, while another employs separate housings.

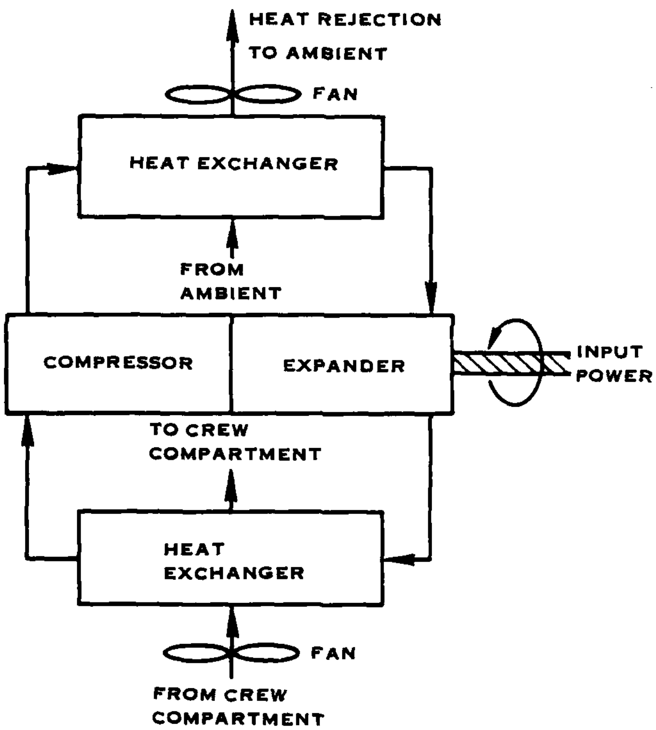

FIGURE 3-3. POSITIVE DISPLACEMENT AIR CYCLE SYSTEM SCHEMATIC

The common housing compressor-expander concept has been tested for low thermal load applications such as personnel portable conditioning schemes. This unit required a 2 1/2 horsepower motor and supplied 0.3 tons of cooling (COP = 0.57). (Reference 12)

3.3 (Continued)

More recent work on this concept utilizing separate housings for compression and expansion to eliminate leakage and thermal effects between the compressor and expander involves a 1 1/2 ton unit that requires 8-10 hp for operation (COP = .71). This system is currently under development and only the expander has been built and tested. However, this expansion device had an isentropic efficiency comparable to the air cycle discussed above.

This cycle is also more complex than either the air or vapor compression cycles described above if oil mist is used for lubrication of the compressor-expander. This oil becomes entrained in the circulating air loops and requires oil separators and coolers to treat the oil. The presence of oil in the air loop, as well as rubbings and shavings of vaned rotary equipment or housings, requires the circulating air to be in a closed loop rather than being discharged to the crew compartment as in the conventional air cycle. Thus, it requires an additional interface heat exchanger and air circulation fan that is not needed in the conventional air cycle. This results in additional system weight. For example, the weight goal for the 0.3 ton system described above is 50 pounds (167 pounds/ton) as compared to an estimated weight of 125 pounds for a 4 ton air cycle system (31 pounds/ton).

Recent studies of this concept indicate that dry lubrication may be feasible from the standpoint of machine life. This would allow the system to be operated open loop, eliminating the heat exchanger that is required when oil lubrication is used. However, the toxicity of the discharged air may still be a problem. This concept is still being studied as the life of the machine using dry lubrication and the physiological effects of breathing the dry lubrication are unknown. Development work also continues on much larger cooling load applications.

3.3 (Continued)

Because of the current development of this equipment, and the on-going studies and design activity, positive displacement vane air cycles were not considered further.

3.4 THERMOELECTRIC

A typical thermoelectric system schematic is presented in Figure 3-4. For the high temperature differentials required at the design condition (such as 40°F evaporator and 140°F condenser temperatures), the performance of thermoelectric modules is less efficient than the performance of an air cycle system. Coefficients of performance are in the range of 0.35 maximum for a two-stage unit (Reference 13), with practical designs including fans having a COP as low as 0.26.

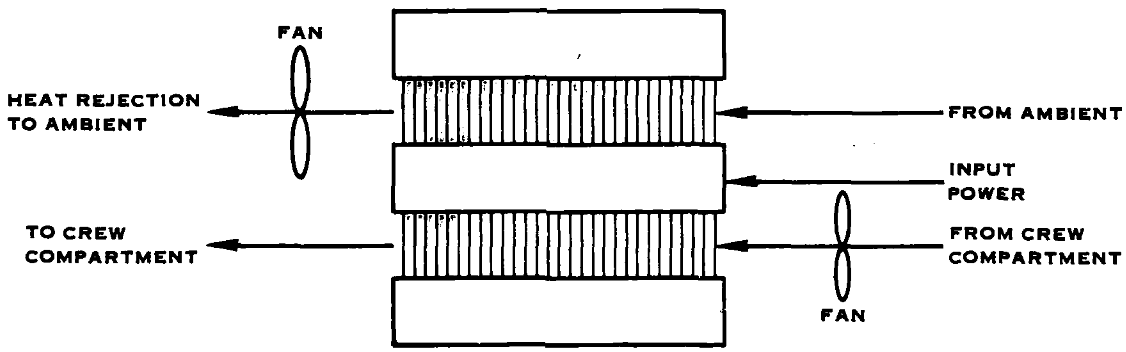

FIGURE 3-4. THERMOELECTRIC SYSTEM SCHEMATIC

While the size of each thermoelectric module (chip) is small, an extremely large number of them would be required to provide 4 tons of cooling. The number of chips required is estimated to be as high as 5600 for a two-stage unit, with a total module weight of approximately 380 pounds.

3.4 (Continued)

A major development effort would also be required to effectively package this concept. With a 28 VDC power source, 12 chips would be wired in series and the unit would then require 470 parallel circuits of the 12 chips in series to provide the required cooling. These would then have to be packaged in such a manner as to provide proper heat transfer areas and flow passages.

Another disadvantage of this concept is the cost of each thermoelectric module. Each two-stage chip is estimated to cost $52. This amounts to a complete thermoelectric unit cost of approximately $300,000 for a 4 ton cooling capacity with the existing ambient and air supply temperature constraints. This cost does not include any packaging effort or the cost of additional required equipment such as fans.

In addition, earlier Army experience with thermoelectric air conditioners indicated a problem of catastrophic failure of the solder joints if the cooling system fails on hot junctions. This failure occurs so rapidly that no known protective device can prevent it.

Because of these problem areas, this concept was not considered further.

3.5 VORTEX TUBES

The vortex tube is a device with no moving parts that is capable of producing a significant refrigeration effect. Cold air resides at the center of the vortex due to kinetic energy effects while the outer vortex is at elevated temperatures due to the energy exchange and friction. The main advantages of this concept are its reliability and maintainability resulting from no moving parts.

The major disadvantage of this system, depicted schematically in Figure 3-5, is its overall poor efficiency and high power requirements. Based

3.5 (Continued)

on a thermal expansion efficiency of 50% (Reference 14) and an inlet temperature of 140°F, an expansion ratio of approximately 4 to 1 is needed to provide the desired supply temperature. Because of the combined hot and cold flow in the unit, only 30-40% of the total unit flow is available for crew compartment conditioning. This results in an overall system COP of less than 0.10.

If a source of compressed air is not available, then a compressor and heat exchanger are needed to provide air at the proper temperature and pressure to the vortex tube. This system also requires a fan to pull ambient air through the heat exchanger. Thus, system simplicity is not much improved over the air cycle.

For these reasons, vortex tubes were not considered further.

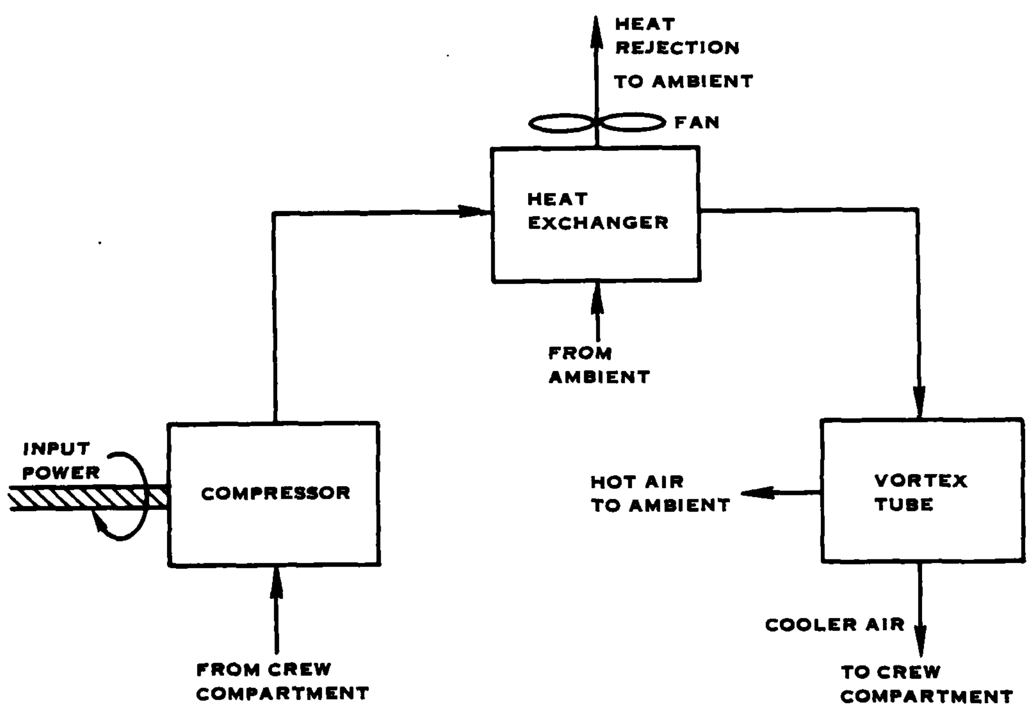

FIGURE 3-5. VORTEX TUBE SYSTEM SCHEMATIC

3.6 EXPENDABLE HEAT SINK

Expendable heat sink cooling can be accomplished with ice, liquid air, or liquid oxygen. A typical system is presented in Figure 3-6. All of these alternatives were reviewed for applicability to a 4 ton cooling requirement with an 85°F crew compartment dry bulb temperature.

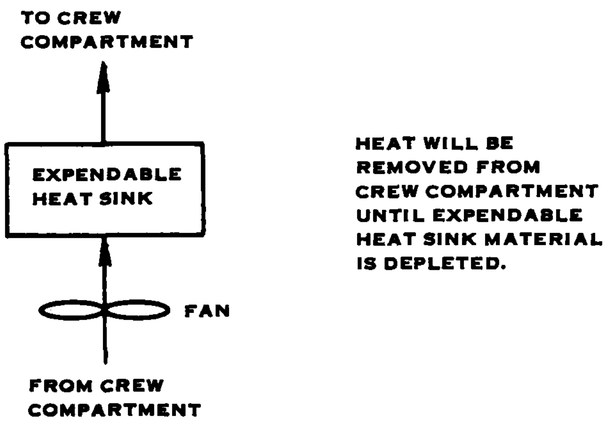

FIGURE 3-6. EXPENDABLE HEAT SINK SYSTEM SCHEMATIC

The major disadvantage of this concept is the weight and volume of the stored ice, liquid air, or liquid oxygen. For the baseline condition, the calculated weights of necessary refrigerant per mission are:

ice	-	243.7 lb/hr
liquid air	-	263.7 lb/hr
liquid oxygen	-	277.5 lb/hr

For missions of several hours duration, the size and weight penalty for this type of crew compartment conditioning is prohibitive.

Another disadvantage of this cooling concept is logistics. Supply areas would be required to either have the refrigeration capability to make large amounts of ice or to transport insulated containers of liquid air or oxygen to replenish depleted supplies.

For these reasons, this concept was not considered further.

3.7 ABSORPTION COOLING

This concept has the advantage of using heat as the power source rather than shaft power. Although absorption cooling is a developed concept, it requires more equipment than the vapor compression cycle, as shown in the system schematic presented in Figure 3-7.

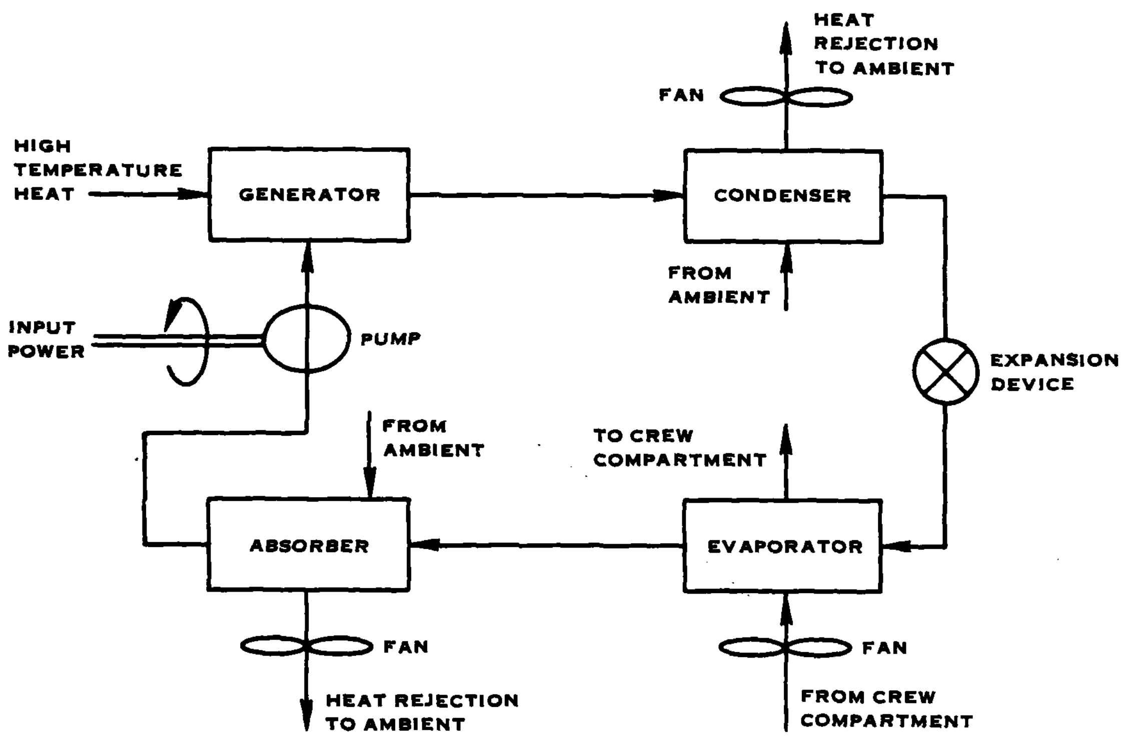

FIGURE 3-7. ABSORPTION COOLING SYSTEM SCHEMATIC

The major disadvantage of this concept is that it is not applicable to existing combat vehicles unless the vehicle undergoes major modifications. A high temperature source of heat (200°F - 400°F) is needed to drive vapor from the solution in the generator. It may be possible to supply this high temperature source by means of the engine exhaust, but this would require major vehicle configuration changes and does not appear practical. Another method of desorbing vapor would be to use a vacuum pump. The addition of this pump

3.7 (Continued)

would further complicate the system, while also impacting the critical power requirements.

Since present absorption cooling systems are also extremely gravity sensitive, development work would be needed to overcome this problem in a mobile unit.

For these reasons, the use of absorption chilling is not recommended and the concept was not considered further.

3.8 EVAPORATIVE COOLING

In this concept, water is applied to the surfaces of the vehicle and cooling produced by its evaporation is used for crew compartment conditioning. In one approach, water is contained between the vehicle walls and a porous material exposed to ambient, through which water vapor can pass. In a second approach, a porous wicking material protected by metal plates is combined with a water distribution system to apply water to the surfaces of the vehicle (Reference 4).

The evaporative cooling rate of both of these nproaches is affected by the ambient environment. On days of high relative humidity, sufficient crew compartment cooling cannot be attained.

Evaporative cooling also causes installation difficulties. When mounted on the outside of a combat vehicle, the system is vulnerable to damage in battle or in transit. The constant wetness against the tank hull is also likely to result in unfavorable effects from fungi or bacteria formation or from corrosion.

The system is also heavy and must make allowances for water loss due to damage or spills in addition to having sufficient on-board water to provide

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3.8 (Continued)

cooling over a several hour mission. Replenishment may also be difficult if a water source is not available, such as is likely to occur in a desert environment. There is also no heating capability in this approach.

Because of these disadvantages, evaporative cooling was not considered further.

3.9 SUMMARY

The items reviewed for each concept are presented in summary form in Table 3-I. The advantages and disadvantages of each concept leading to its selection or rejection for further consideration are presented. Based on the discussions presented above, air cycle and vapor compression crew compartment conditioning concepts are selected for preliminary design and trade-offs.

SYSTEM	PERFORMANCE	POWER	SIZE/WEIGHT	VULNERABILITY	LOGISTICS	ADVANTAGES	DISADVANTAGES	SELECT
AIR CYCLE (TURBO-MACHINERY)	1) MEETS DESIRED LOAD	1) NEED MORE POWER THAN VAPOR CYCLE	1) LOW WEIGHT	1) CAN RUN WITH SLIGHT LEAKS OR DAMAGE	1) MINIMUM	1) FULLY DEVELOPED CONCEPT 2) MINIMUM LOGISTICS 3) CONTINUES OPERATION EVEN WITH MINOR DAMAGE 4) HEATING CAPABILITY	1) NEED MORE POWER THAN VAPOR 2) NOISIER	✓
VAPOR CYCLE	1) MEETS DESIRED LOAD	1) LOW POWER REQ'T	1) HEAVIER THAN AIR CYCLE	1) LEAKAGE WILL SHUT DOWN SYSTEM	1) NEED REFRIGERANT, LEAK DETECTOR, AND VACUUM PUMP	1) LESS POWER THAN AIR CYCLE 2) FULLY DEVELOPED CONCEPT	1) NEED ADDITIONAL COMPONENTS FOR HEATING 2) LEAKAGE WILL SHUT DOWN SYSTEM 3) MORE LOGISTICAL NEEDS	✓
AIR CYCLE (POSITIVE DISPLACE-MENT)	1) CURRENTLY UNDER DEVELOPMENT. TO DATE, ONLY LOADS $<$ 2 TONS TESTED. 2) WET LUBE REQUIRES CLOSED LOOP DUE TO POTENTIAL TOXICITY OF ENTRAINED OIL AND METALLIC DEBRIS	1) ESTIMATED TO EVENTUALLY APPROACH TURBO-MACHINERY AIR CYCLE LEVELS	1) CLOSED LOOP NEEDS EXTRA HX 2) DEVELOPMENT UNITS ARE STILL HEAVY	1) SIMILAR TO AIR CYCLE	1) NEED OIL MIST FOR LUBRICATION 2) ROTOR MUST BE PRECISELY REFERENCED IN COMMON HOUSING APPROACH		1) CLOSED LOOP REQUIRES BLOWERS TO TRANSFER AIR TO THE HX 2) SIMPLISTIC ARRANGEMENT IS LOST WITH CLOSED LOOP 3) CONCEPT IS STILL UNDER DEVELOPMENT	
THERMO-ELECTRIC	1) 10–35% COP EVEN IN MODULES	1) CONSUMES MORE POWER THAN AIR CYCLE	1) STACKING CAN PRODUCE SMALL SIZE UNITS, BUT FAIRLY HEAVY		1) NEED REPLACEMENT MODULES		1) NEEDS PACKAGING DEVELOPMENT FOR LARGE COOLING LOAD. 2) EXPENSIVE	
VORTEX TUBE	1) CAN MEET DESIRED SUPPLY TEMP	1) NEEDS MORE POWER THAN AIR OR VAPOR CYCLE (COP $<$ 0.10)	1) WHILE UNIT ITSELF IS SMALL, NEED COMPRESSED AIR SOURCE AND HEAT EXCHANGER	1) NO MOVING PARTS	1) MINIMUM	1) CAN GET BOTH HOT AND COLD AIR 2) ONLY MOVING PARTS ARE IN POWER COMPRESSOR AND FANS	1) NOISY 2) NEEDS MORE POWER THAN AIR OR VAPOR CYCLE 3) NOT MUCH SIMPLER THAN AIR CYCLE WITH AIR SOURCE AND HEAT EXCHANGER	
EXPENDABLE HEAT SINK	1) CAN PROVIDE COOLING	1) MINIMUM POWER	1) NEEDS LARGE AMOUNTS OF REFRIGERANT	1) LEAKS WILL SHUT DOWN SYSTEM 2) O_2 LEAK IS HAZARD	1) COMPLEX WITH REFRIGERATION EQUIPMENT FOR HEAT SINK REPLACEMENT		1) WEIGHT AND VOLUME 2) COMPLEX LOGISTICS	
ABSORPTION	1) NEEDS HIGH TEMP SOURCE TO OPERATE	1) LOWER POWER BECAUSE LIQUID PHASE	1) MORE EQUIPMENT THAN VAPOR CYCLE	1) LEAKAGE WILL SHUT DOWN SYSTEM	1) NEED MAKE-UP REFRIGERANT, LEAK DETECTOR	1) FULLY DEVELOPED CONCEPT	1) HIGH TEMP HEAT SOURCE IS NOT AVAILABLE - SYSTEM WILL NOT PERFORM UNLESS HIGH TEMP (EG. EXHAUST) IS PLUMBED IN	
EVAPORATIVE COOLING	1) MINIMUM OR NO PERFORMANCE DURING HIGH RH	1) NO POWER REQUIREMENT	1) COVERS MAJOR EXTERIOR AREAS OF VEHICLE 2) HEAVY	1) SUSCEPTIBLE TO DAMAGE DURING TRANSPORT OR BATTLE	1) MUST HAVE LARGE WATER SUPPLY FOR REPLENISHMENT		1) DIFFICULT TO APPLY 2) INSUFFICIENT COOLING 3) SUSCEPTIBLE TO OUTSIDE CORROSION, FUNGI, BACTERIA, AND DAMAGE	

TABLE 3-I. PRELIMINARY SELECTION OF CANDIDATE SYSTEMS

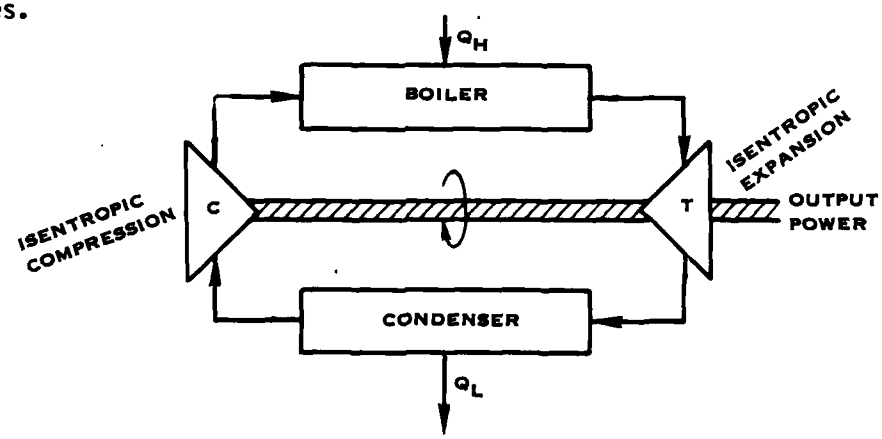
SECTION 4.0
COMPARATIVE SYSTEM PERFORMANCE

INTRODUCTION

This section presents a general description of both the air and vapor cycle operation as well as a description of the baseline Carnot concept. Measures of system performance will be defined and an explanation and comparison of typical system differences from Carnot performance will be presented.

4.1 CARNOT CYCLE

The Carnot cycle is the most efficient cycle that can operate between two constant temperature reservoirs. In this cycle every process is reversible, and thus the complete cycle is also reversible. Heat transfer with both the high and low temperature reservoirs is reversible isothermal, and the changes in fluid temperature in both the turbine and compressor are reversible adiabatic processes.



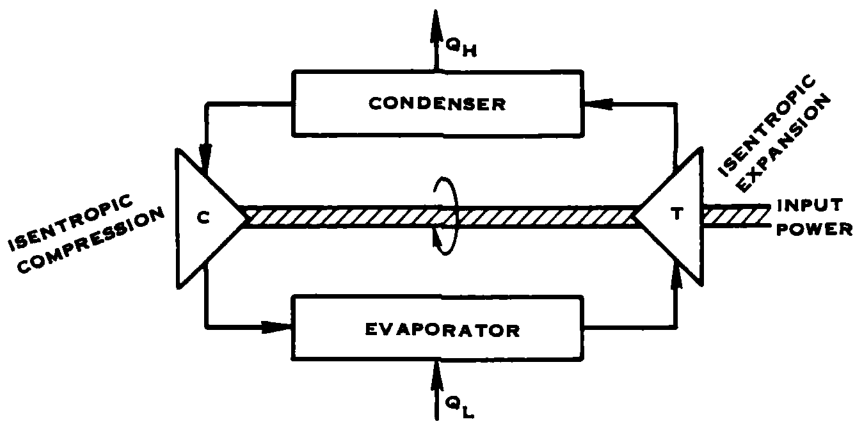
For the ideal heat engine depicted above, the thermal efficiency of the cycle is defined as the output energy (work) divided by the input energy. In this case

$$\eta = \frac{\text{output}}{\text{input}} = \frac{Q_H - Q_L}{Q_H} = 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H}$$

To increase the efficiency of this cycle, it is necessary to increase the

4.1 (Continued)

temperature at which heat is rejected to the boiler (T_H) or decrease the temperature at which heat is rejected from the condenser (T_L).



For the refrigerator depicted above, the heat engine cycle is reversed. This is possible because all Carnot processes are reversible. For this refrigeration process, the coefficient of performance (COP) is defined as energy removed by the refrigerant (Q_L) divided by the input power (Q_H - Q_L)

$$\text{or COP} = \frac{\text{output}}{\text{input}} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{\frac{Q_H}{Q_1} - 1} = \frac{1}{\frac{T_H}{T_L} - 1}$$

For this refrigeration process, the COP can be improved either by decreasing the temperature at which heat is rejected from the condenser (T_H) or increasing the temperature at which heat is rejected to the evaporator (T_L). The effect of either of these two actions is to reduce the temperature differential, or temperature "lift", between the evaporator and condenser.

For this study, the evaporator and condenser temperature values are essentially fixed by ambient conditions and crew compartment supply temperature requirements. Specifically

4.1 (Continued)

$T_H = T_{ambient} = 120°F = 579.7°R$

$T_L = T_{crew\ compartment\ supply} = 50°F$, typical supply $= 509.7°R$

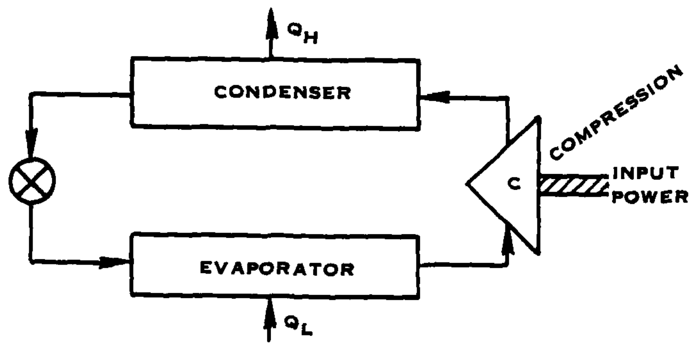
Therefore, the maximum coefficient of performance (Carnot) that could be expected for this application would be:

$$COP = \frac{1}{\frac{T_H}{T_L} - 1} = \frac{1}{\frac{579.7}{509.7} - 1} = 7.28$$

In actual practice, all system losses and inefficiencies must be considered to determine actual system performance. These losses and inefficiencies include:

- fluid characteristics
- compression losses
- expansion losses
- ducting pressure losses
- heat transfer inefficiencies
- motor/power losses.

4.2 VAPOR COMPRESSION REFRIGERATION CYCLE



This cycle, depicted schematically above, consists of a compressor, condenser, and evaporator. In this cycle, the working fluid is compressed as

4.2 (Continued)

vapor, using work produced by an outside source. This outside source can be a power loop driven by steam heat in a boiler or an electric motor. The vapor is condensed at high pressure, rejecting heat and leaving the condenser as a liquid. The refrigerant is then expanded to a lower pressure through a valve and enters the evaporator. The fluid is vaporized in the evaporator which produces cooling.

For this application, losses occur in the fluid, compressor and ducting. The fluid losses result from the fact that no refrigerant can be expanded isentropically through the expansion valve. Rather, an isenthalpic expansion occurs which is equivalent to an isentropic efficiency of approximately 93% for Freon 12 using the calculation procedures of Reference 16.

Compressor efficiency is defined as the input shaft power of an isentropic compressor divided by the input power of the actual compressor required to produce the same pressure ratio with the same inlet conditions. A typical value to use for compressor efficiency for this application is 65%.

Ducting pressure losses are modeled by the $\Delta P_{loss}/P$ of the cycle. This variable is defined as the sum of the individual ratios of local pressure loss divided by the local pressure. A typical value for this application for the ducting pressure losses is 5%.

The coefficient of performance of the vapor cycle can be calculated by

$$\frac{COP}{\eta_{Motor}} = \frac{\text{Refrigeration}}{\text{Input Work}}$$

For the vapor cycle hot-dry system performance presented in a subsequent section, the system COP/ η_{motor} is approximately 1.41. For power generation with no losses (i.e., $\eta_{motor} = 1$), the COP for this vapor cycle would be 1.41, compared to a Carnot cycle COP of 7.28.

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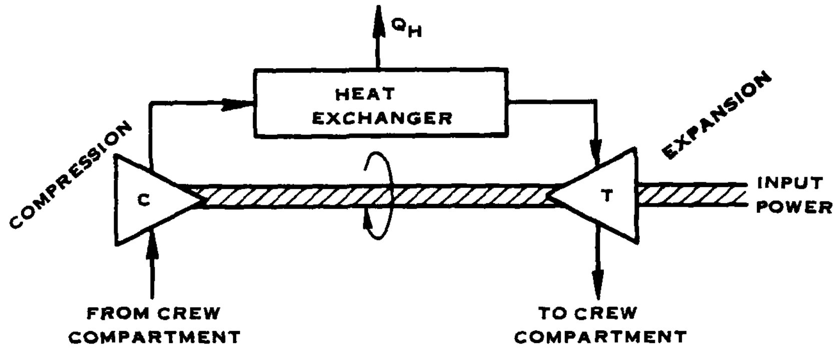
4.3 AIR CYCLE REFRIGERATION



In the air cycle depicted above, warm air from the crew compartment and make-up ambient air are compressed to high pressure and temperature. Heat is rejected to the atmosphere in the heat exchanger, and the air then enters the turbine while still at high pressure. Expansion through the turbine reduces the pressure and temperature, and the exiting cold air is then used to cool the crew compartment. Power generated by the turbine is used to drive the compressor. An additional compressor independently powered by an electric motor, auxiliary power unit, belt drive, etc., is used to overcome the cycle losses and inefficiencies.

One of the advantages of an open loop air cycle with recirculation is that, system temperatures permitting, it requires one less heat exchanger than the vapor cycle, thus reducing system components and weights. In the air cycle, the air refrigerant leaving the expansion turbine can be used directly to cool the crew compartment by running the system open loop. If a cooling system is operated closed loop, an interface heat exchanger is needed to cool the crew compartment cooling fluid. This is analogous to circulating air from the crew compartment over the evaporator in the vapor compression cycle.

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4.3 (Continued)

A general advantage of an air cycle is that it can be regenerated. For this cooling application, a regenerative heat exchanger that uses turbine outlet air to cool the turbine inlet air is feasible and allows the use of the turbine refrigeration potential while eliminating the formation of ice. This will be discussed in more detail in subsequent sections of this report.

For the air cycle depicted in the sketch, losses occur in expansion, compression, ducting, and heat transfer. Air requires the use of an expansion turbine instead of the expansion valve used in the vapor cycle. Turbine efficiency, defined as the actual turbine output shaft power divided by the output power of an isentropic turbine operating at the same inlet conditions and pressure ratio, will be in the range of 80 to 88%. This represents a larger loss than the isenthalpic expansion value of the vapor cycle which had an isentropic efficiency of approximately 0.93. This larger expansion loss is one of the general disadvantages of the air cycle when compared to the vapor compression cycle.

The coefficient of performance of the air cycle can be calculated by

$$\frac{\text{COP}}{\eta_{\text{motor}}} = \frac{\text{Refrigeration}}{\text{Input Work}}$$

where refrigeration is the crew compartment cooling load and input work is the power required by the drive compressor and fans. To determine the input work requirements, the air cycle system performance must be analyzed. This was done for two versions of HCPE interface and the results are presented in Figures 4-1 and 4-2. COP/ η_{motor} for the two modes of operation depicted in Figures 4-1 and 4-2 are 0.480 and 0.651, respectively. These values of COP/ η_{motor} clearly indicate the larger power demands of the air cycle when compared to the vapor

4.3 (Continued)

cycle. Both of these cases are preliminary system schematics performed prior to preliminary design activities, but the significance of the placement of the hybrid collective protection equipment interface is clear. Less power is required when this interface is upstream of the shaft driven compressor (or other appropriate power source) than when the interface is downstream of the turbine. This is a result of the higher crew compartment supply temperature when the HCPE interfaces downstream of the turbine. This higher supply temperature requires a higher coolant flow rate to maintain a constant crew compartment temperature. This results in more power being required from the shaft driven compressor.

Despite this decreased performance in the air cycle, the air cycle has several advantages over the vapor cycle which warrant selection for a detailed trade-off. These include:

- combined heating and cooling capability
- reduced size and weight
- minor leakage will not force the system to shut down

Detailed design and trade-offs of the air cycle and vapor cycle will be presented in subsequent sections of this report.

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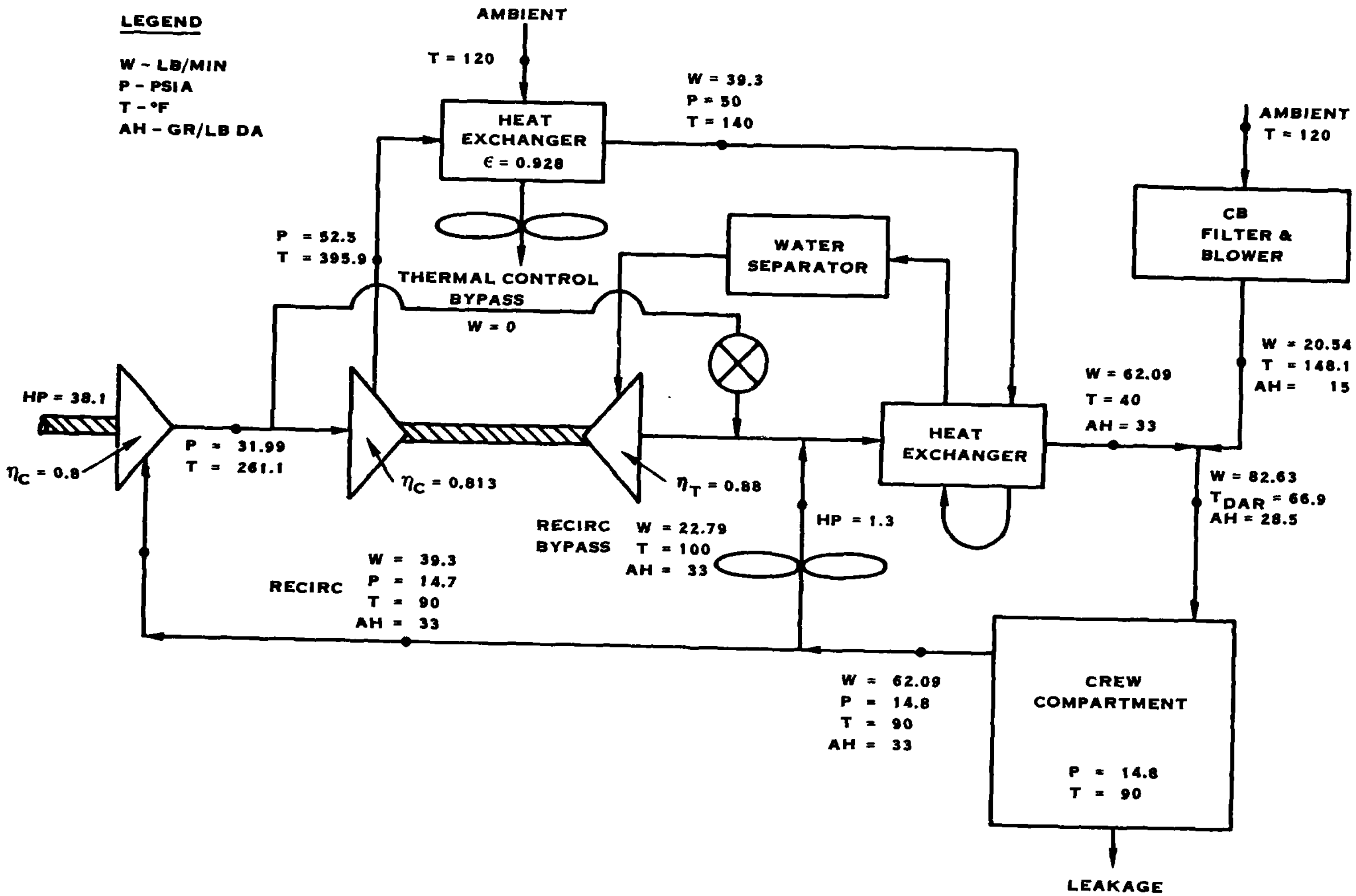

FIGURE 4.-1. HOT DRY DAY; 4 TON COOLING DOWNSTREAM HCPE INTERFACE

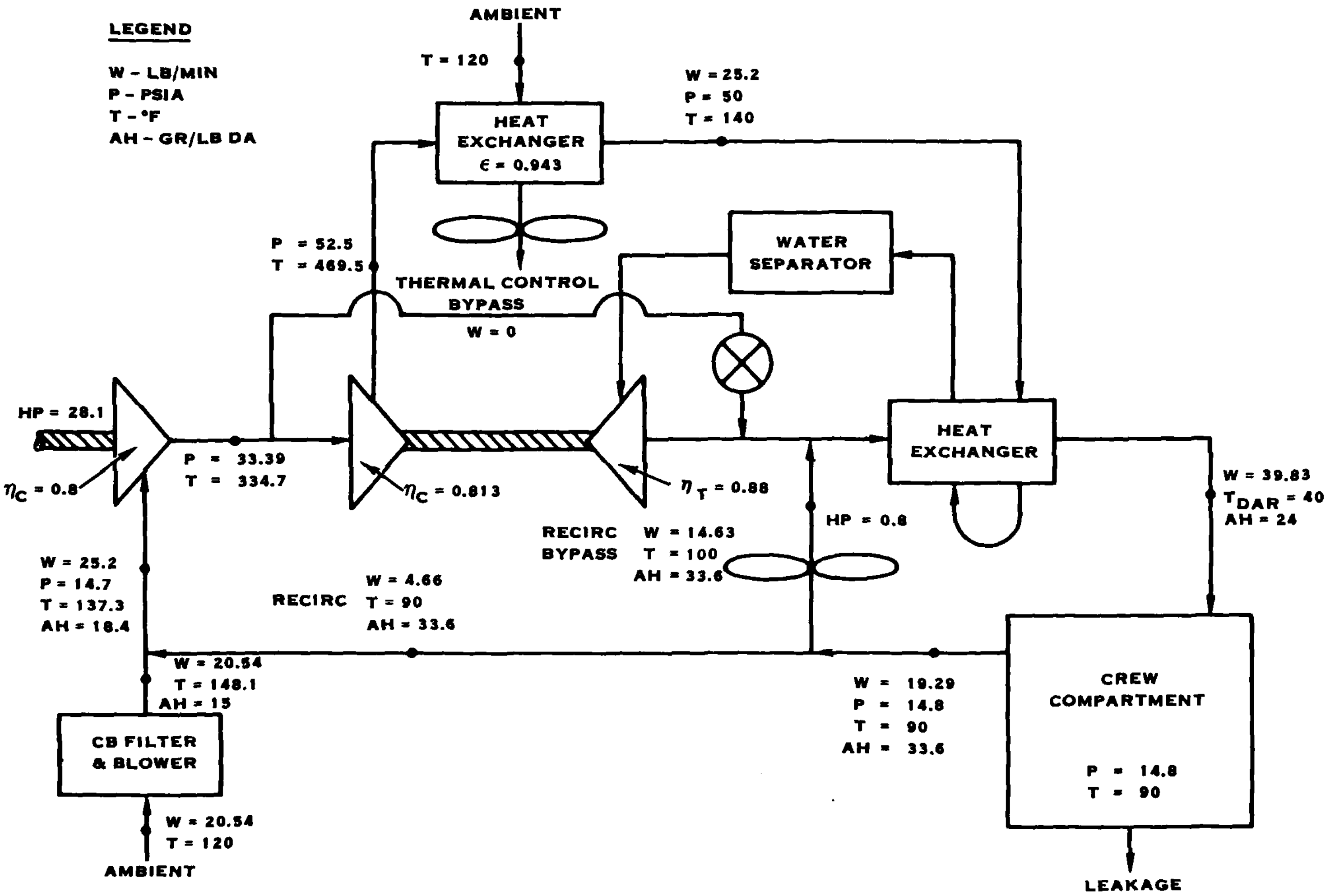

FIGURE 4-2. HOT-DRY DAY; 4 TON COOLING UPSTREAM HCPE INTERFACE

SECTION 5.0
CANDIDATE SYSTEM PERFORMANCE

INTRODUCTION

This section presents the results of the system performance of the preliminary design of the air cycle and vapor compression refrigeration concepts selected as candidate systems for crew compartment cooling. The design specifications will be defined, the derivation of thermal loads will be presented, and system thermal performance will be summarized. Sizing conditions for the various components and control schemes will be explained.

5.1 DESIGN SPECIFICATION

Both the air and vapor cycles must meet the following specification:

1. 48,000 Btu/hr (4 tons) of equivalent evaporator cooling for the hot-dry climate
2. Crew compartment design goal conditions of 90°F dry bulb temperature, 20% relative humidity
3. Crew compartment pressurized to 1.5" H_2O
4. Systems to interface with the hybrid collective protection equipment.

Both concepts are to minimize power since available power on combat vehicles is marginal. For the purpose of estimating power needs on a comparable level for both systems, 100% motor efficiency was assumed for both cycles.

5.2 THERMAL LOAD DERIVATION

As the conditioning systems will be operated in climates other than the hot-dry design point and only the hot-dry cooling requirement was defined, it was necessary to estimate the various elements of the crew compartment cooling

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5.2 (Continued)

load so that off-design cooling requirements (i.e., other than the hot-dry climate) could be approximated. Because there is no evaporator in an air cycle system, a system performance comparison between air and vapor cycles is difficult. Therefore, the first step of this estimation procedure was to define the actual system cooling provided by the candidate systems so that they could be compared. This cooling consists of four elements:

1. the cooling provided to system recirculating air
2. the cooling provided to the fresh make-up air passing through the HCPE
3. the equivalent cooling of water removed from the air
4. the cooling necessary to remove heat generated by recirculating air fans

Detailed equations for each of these elements are presented in Table 5-I.

It was then necessary to estimate the various components of the 4 ton cooling requirement (such as electronics, solar, personnel, etc.) so that a crew compartment heat transfer coefficient (UA) could be defined for estimating cooling requirements at other than the hot-dry design climate. First, equivalent evaporator performance for the hot-dry day with 300 cfm HCPE fresh air make-up and a 4 ton cooling requirement was modeled. This performance was based on the cooling elements of Table 5-I and is presented schematically in Figure 5-1. Then, based on thermal load proportions from earlier Army studies of the M577/A1 (Reference 4) scaled to a 4 ton cooling load, a UA of 355.2 Btu/hr-°F was estimated for the crew compartment heat transfer coefficient. This heat transfer coefficient is estimated by subtracting ventilation and fixed cooling requirements from the total crew compartment sensible load. Table 5-II presents the detailed derivation of this value.

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TABLE 5-I

COOLING ELEMENTS

1. System Recirculation Air

$$Q = (W_{crew\ compartment\ supply} - W_{fresh\ air}) \times 0.24 \times (T_{crew\ compartment\ db} - T_{crew\ compartment\ supply\ db})$$

2. Fresh Make-Up Air

$$Q = W_{fresh\ air} \times 0.24 \times (T_{HCPE\ out} - T_{crew\ compartment\ supply\ db})$$

3. Water Removed

$$Q = W_{H_2O\ removed} \times h_{fg}$$

4. Recirculating Air Fans

$$Q = W_{fan} \times 0.24 \times \Delta T_{fan}$$

Notes:

Q	defined as Btu/hr
W	defined as lb/hr
T	defined as °F
h_{fg}	defined as Btu/lb

5.2 (Continued)

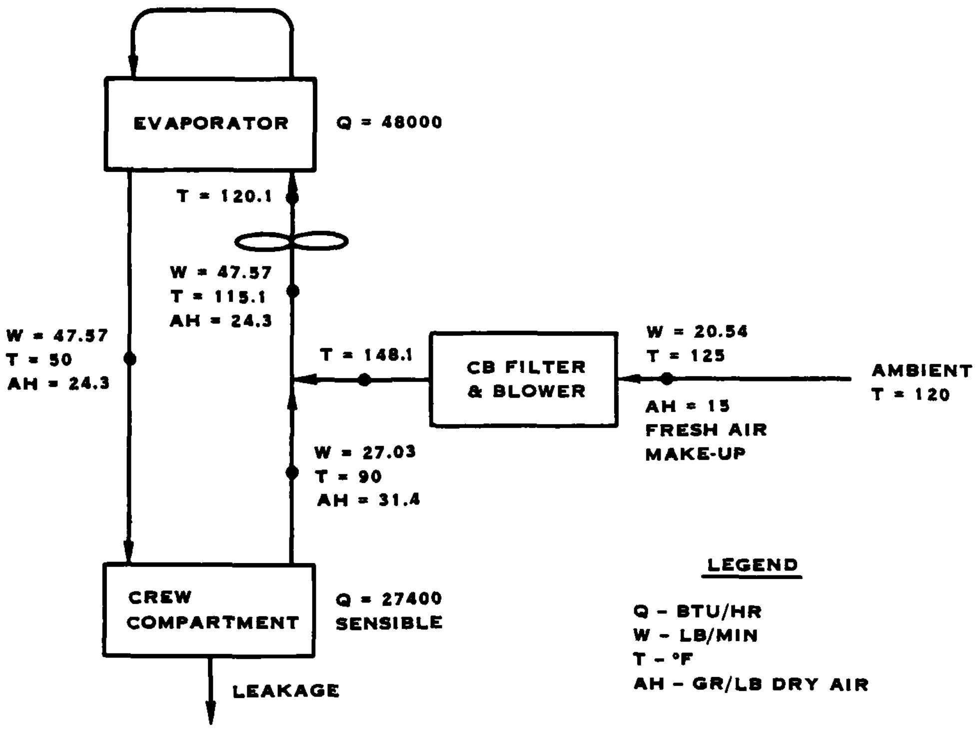

FIGURE 5-1. EQUIVALENT EVAPORATOR SCHEMATIC

This cooling load element estimation allows cooling loads and system performance to be determined for the hot-humid, basic constant high humidity, basic variable high humidity, and basic hot climates in addition to the hot-dry design point. The severe cold climate was also reviewed for system heating capability. Ambient conditions for these climates were summarized in Table 2-I.

Both the air and vapor cycles were designed to interface with the HCPE. The cooling systems were designed for 300 cfm fresh air make-up through the NBC filter to account for crew compartment leakage. In addition, the 600 cfm fresh air flow case was also analyzed for each of the climates to determine performance when the NBC filter is bypassed and the leakage rate is higher. The HCPE operating modes were summarized in Table 2-VIII.

TABLE 5-II
DERIVATION OF CREW COMPARTMENT HEAT TRANSFER COEFFICIENT

A. ESTIMATED-FIXED LOADS BASED ON EARLIER STUDIES

1. INTERIOR AND MISCELLANEOUS = 15,780 BTU/HR
2. SOLAR = 10,815 BTU/HR

B. CREW COMPARTMENT ESTIMATED FIXED LOADS FOR THIS INVESTIGATION

1. INTERIOR AND MISCELLANEOUS IN THE CREW COMPARTMENT IS EQUAL TO THE TOTAL INTERIOR AND MISCELLANEOUS MINUS THE HCPE BLOWER (2 KW) AND THE RECIRC FAN IN FIGURE 5-1.

   CREW COMPARTMENT INTERIOR AND MISCELLANEOUS=15780-6826-3425 = 5529 BTU/HR

2. SOLAR = 10815 BTU/HR

3. PERSONNEL:

   (4 MEN EACH AT 250 WATTS TOTAL METABOLIC LOAD.)
   FOR 90°F, TOTAL SENSIBLE LOAD = 400 BTU/HR

C. CREW COMPARTMENT UA

THE CREW COMPARTMENT LOAD AFFECTED BY UA IS THE TOTAL CREW COMPARTMENT SENSIBLE LOAD FROM FIGURE 5-1 MINUS THE FIXED INTERIOR AND MISCELLANEOUS, SOLAR, AND PERSONNEL LOADS:

$$Q_{UA\Delta T} = Q_{CREW\ COMPARTMENT}\ SENSIBLE - (Q_{INTERIOR\ \&\ MISCELLANEOUS} + Q_{SOLAR} + Q_{PERSONNEL}\ SENSIBLE) = 27400-(5529 + 10815 + 400) = 10656\ BTU/HR$$

FOR ΔT OF 30°F (120°F AMBIENT – 90°F CREW COMPARTMENT), UA = 355.2 BTU/HR-°F

5.2 (Continued)

Another item affecting system sizing was ambient cooling air used in heat exchangers, condensers, etc. It was assumed that this air would be 5°F higher than ambient because of heat transfer from the surrounding armor on days when cooling is required. For severe cold day operation, air to the heat exchangers was assumed to be at ambient conditions.

The increase in cabin humidity due to the evaporation of sweat from the vehicle crew also affects equipment sizing and moisture removal requirements. Crewman casualty work performed by the Army to determine vehicle requirements for this study assumed a metabolic load of 250 watts per crewman (Reference 8). Therefore, this was the personnel cooling requirement used during this design effort. For these preliminary design activities, it was assumed that the metabolic load was all latent at crew compartment temperatures of 96°F or higher. For temperatures below 96°F, the sensible load was equivalent to 68 Btu/hr-°F and the latent load was adjusted so that the total personnel load was 1000 watts. This modeling approach was based on earlier aircraft environmental control system designs.

With the design specification and cooling load derivation completed, preliminary design of both the air and vapor compression systems was initiated. A detailed explanation of these efforts is presented below.

5.3 AIR CYCLE SYSTEM DESCRIPTION

The air cycle system that was studied is presented schematically in Figure 5-2. This system uses the 3-wheel air cycle machine previously developed by Hamilton Standard. In this system a small portion of the turbine power goes to the cooling air fan, with the major portion going to the bootstrap compressor. Fresh air from the HCPE is compressed in the system drive compressor before

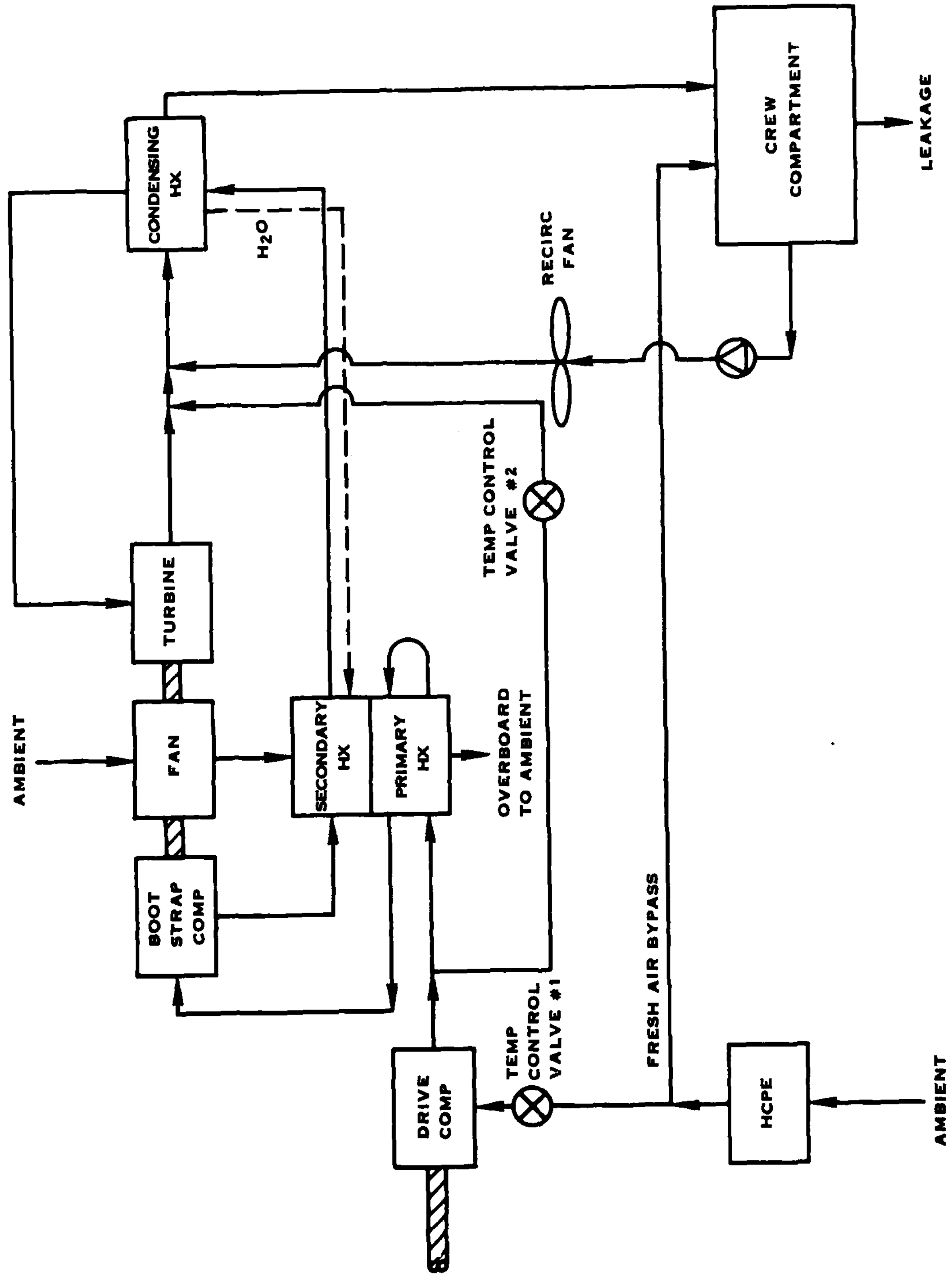

FIGURE 5-2. AIR CYCLE SYSTEM SCHEMATIC

5.3 (Continued)

entering the 3-wheel air cycle machine. While a single compressor could be designed to combine the drive and bootstrap functions, separate bootstrap and drive compressors were selected for preliminary design because of packaging flexibility and the current development status for aircraft applications. The system also uses both a primary and secondary heat exchanger to cool the air entering both the bootstrap compressor and the high pressure side of the condensing heat exchanger.

This system also utilizes an upstream water separator and a unique "recirc chiller" concept (Recirc Air ™) developed by Hamilton Standard. Air cycle refrigeration concepts in the past typically used a coalescing water separator downstream of the turbine to eliminate moisture in the supply air. This coalescing water separator has several negative aspects:

1. the coalescer results in a large pressure drop which essentially backpressures the turbine, increasing system losses
2. temperatures at the water separator inlet must be maintained above 32°F to prevent ice formation in the water separator. Thus, there is no advantage in increasing turbine efficiency or heat exchanger sizes if these changes result in a water separator inlet temperature that is below freezing. Furthermore, the subfreezing cooling potential of the turbine cannot be effectively utilized when high moisture levels are present.

The upstream condensing heat exchanger was the first major breakthrough in eliminating the need for a coalescer. However, even in earlier system applications with an upstream condenser, crew compartment recirculation air was not incorporated into the cycle. Therefore, these earlier systems did not have

5.3 (Continued)

the capability of using all of the turbine cooling potential because of ice build-up in the condenser. Rather, the recirculation air was mixed downstream of the condenser where it could not be used to melt ice.

With the incorporation of the "recirc chiller" concept (i.e., mix crew compartment recirculation air so as to melt ice with the recirculation air), the problems of the earlier systems are corrected. The coalescer is eliminated by condensing moisture in the high pressure side of the condenser upstream of the turbine. Condensed moisture is withdrawn by means of a "scupper" and injected into the secondary heat exchanger where it aids in cooling bootstrap compressor outlet air. This water removal reduces the amount of ice present at the turbine outlet. The crew compartment recirculation air is mixed with the turbine discharge prior to entering the low pressure side of the condenser. This mix temperature is designed to be high enough to eliminate condenser icing problems. The flow to the crew compartment is the sum of air through the air cycle machine plus the recirculated air. This reduces the amount of air that must flow through the drive compressor and air cycle machine which results in lower power requirements and smaller equipment.

The system is designed to provide 4 tons of cooling in the hot-dry climate with an HCPE flow rate of 300 cfm. For cases where 600 cfm flows through the HCPE, the flow that cannot be handled by the drive compressor and air cycle machine is bypassed to the distribution ducting downstream of the condenser. This is depicted by the "fresh air bypass" line in Figure 5-2. There are several reasons for this bypass ducting selection:

1. for HCPE operation with the crew compartment conditioning system off, proper air flow distribution will be obtained

5.3 (Continued)

2. mixing at the turbine discharge (along with the recirculated air) upstream of the low pressure condenser will raise the temperatures in the condenser which will result in less water removal and undesirably higher moisture levels in the crew compartment.

As 600 cfm is a condition where the NBC filter is being bypassed, it was assumed that the increased ventilation rate would result from more openings (such as hatches) in the vehicle and that the thermal load problem is not as severe as in the NBC protective posture with 300 cfm fresh air flow.

Both a primary heat exchanger upstream of the bootstrap compressor and a secondary heat exchanger downstream of the bootstrap compressor are used. The secondary heat exchanger is used to cool the air entering the high pressure side of the condensing heat exchanger to obtain proper water removal and turbine inlet conditions. The primary heat exchanger is used to limit bootstrap compressor outlet temperature. This is a proven concept in aircraft environmental control system applications and is part of an off-the-shelf system.

Water removed from the condensing heat exchanger is injected into the conditioned air outlet-fan air inlet portion of the secondary heat exchanger to take advantage of the evaporative cooling potential of the condensed moisture. Approximately 80% of the evaporative cooling potential is available for this use. The heat exchangers use a previously developed concept where the primary and secondary sections are brazed together as a single unit to minimize the size of the two heat exchangers by eliminating headers and connecting ductwork.

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5.3 (Continued)

Thermal control for off-design cooling and heating operation is also provided in this system. This thermal control is provided by a combination of a drive compressor throttling valve, labeled "temperature control valve #1" on Figure 5-2, and a hot thermal bypass controlled by "temperature control valve #2." For design operation at the hot-dry climate, the drive compressor throttling valve is completely open and the hot thermal bypass is completely closed. The crew compartment recirculation flow is then selected so that 35°F air is supplied at the condenser outlet at the hot-dry design point.

For off-design cooling conditions, condenser outlet temperatures will fall below 35°F for certain climates if the conditioning system is operated at full capacity. This low temperature at the condenser outlet can cause icing problems. It is possible to increase the condenser outlet temperature to 35°F by opening the hot thermal bypass. However, this would not reduce the flow through the drive compressor, and there would be no reduction in required system power. Rather than open the hot thermal bypass, a throttling valve on the drive compressor inlet is used instead. As the condenser outlet temperature falls below 35°F, this throttling valve closes, reducing the flow through the conditioning system and diverting the remainder of the HCPE flow to the fresh air bypass. The throttling valve adjusts to a setting where the condenser outlet temperature is 35°F. This approach reduces the flow through the drive compressor and minimizes the power required by the system at off-design cooling cases.

The hot thermal bypass ("temperature control valve #2") can then be opened to adjust to a warmer crew compartment temperature, if so desired. For the purposes of this study, the off-design cases are shown as "full cold" operation to demonstrate the cooling capacity with no hot thermal bypass.

5.3 (Continued)

For cases where heating of the crew compartment is desired on a severe cold day, the first step in the control sequence is to throttle the drive compressor to its minimum setting in an attempt to meet the condenser outlet temperature requirement of 35°F. Because of the severe cold ambient conditions and the low temperatures out of the secondary heat exchanger, the condenser outlet temperature will be below 35°F even with the throttling valve at its minimum setting. Then, the hot thermal bypass will open in an attempt to satisfy a 60°F crew compartment temperature. If the crew compartment temperature remains below the 60°F setting with the hot thermal bypass wide open, the drive compressor throttling valve will begin to open. This puts more flow through the drive compressor, reducing the amount of cold fresh air bypass flow and increasing the amount of flow available for the hot thermal bypass.

The complete control sequence is depicted schematically in Figure 5-3.

5.4 AIR CYCLE SYSTEM PERFORMANCE

The air cycle system was designed for a 4 ton cooling capacity at the hot-dry climate with a 300 cfm HCPE fresh air flow rate. The system performance at this design condition is presented schematically in Figure 5-4. As discussed previously, the drive compressor throttling valve is wide open with no fresh air bypass and the hot thermal bypass control valve is fully closed. Crew compartment air at a rate of 13.9 lb/min is recirculated to maintain a condenser outlet temperature of 35°F. The total power requirement for system operation at this condition is 37.51 horsepower for the drive compressor and recirculation fan, ignoring any power source inefficiencies.

For air cycle system performance, the dry air rated supply temperature rather than the dry bulb supply temperature is used to determine cooling

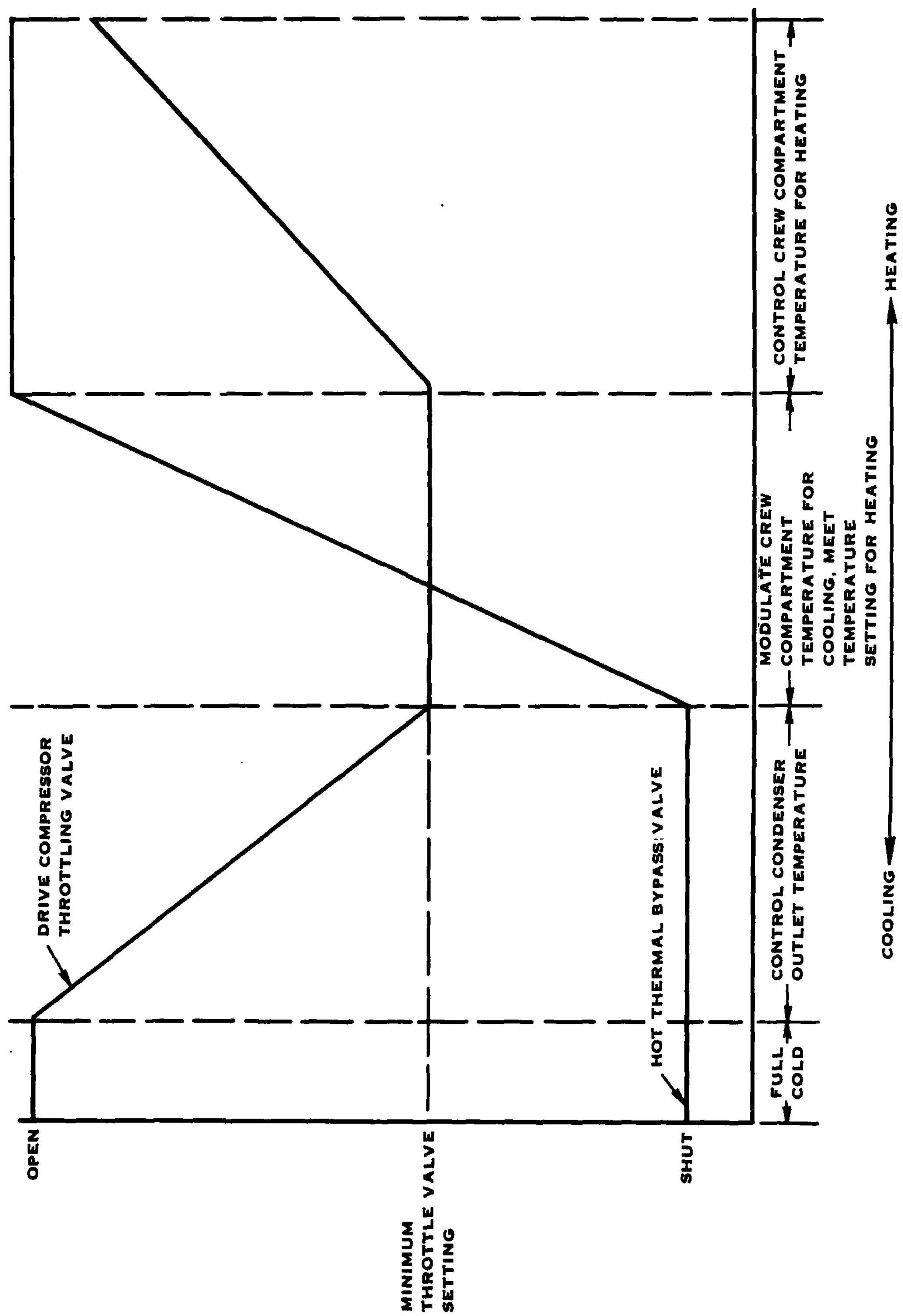


FIGURE 5-3. AIR CYCLE CONTROL SEQUENCE

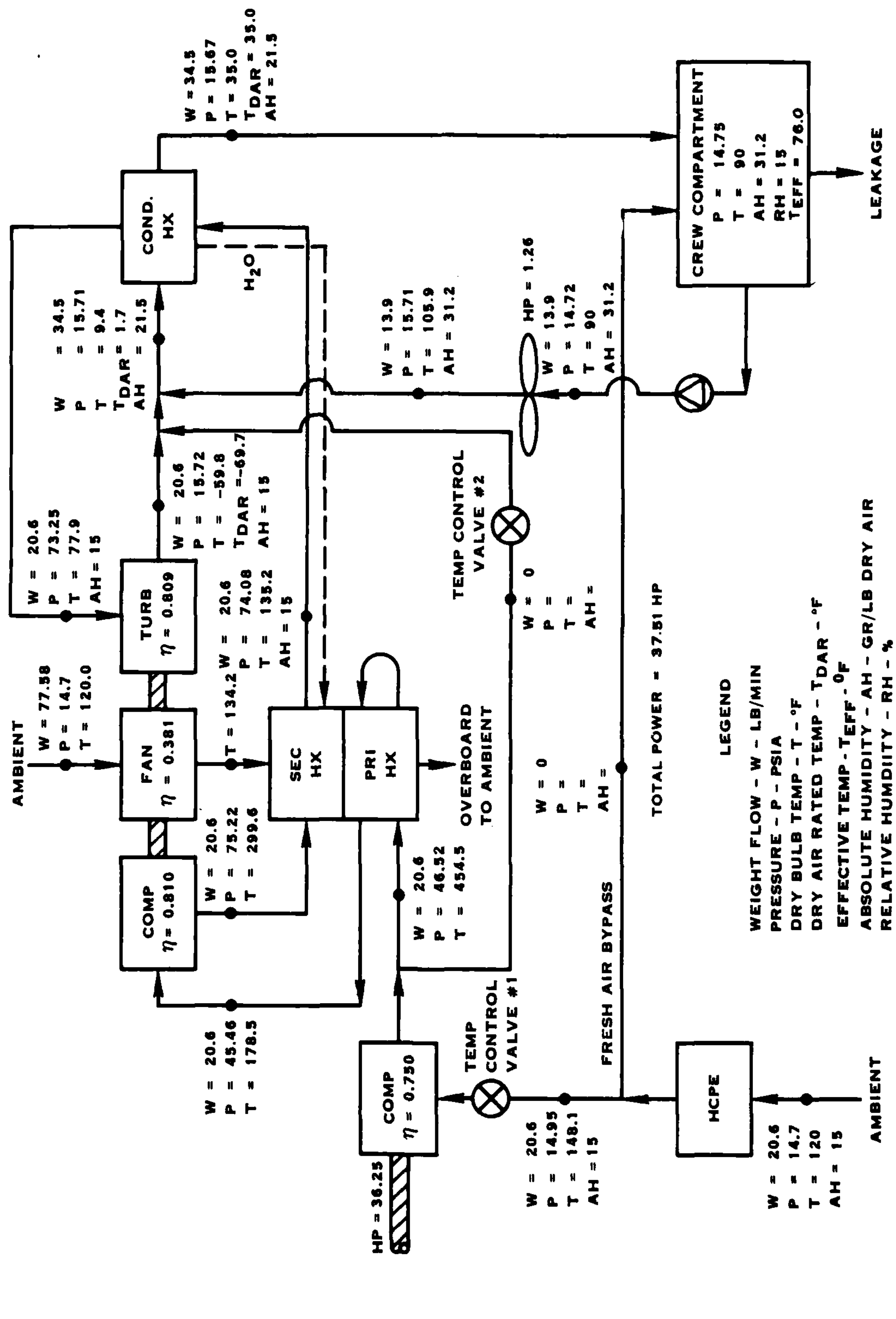


FIGURE 5-4. AIR CYCLE HOT-DRY CLIMATE

5.4 (Continued)

capacity when free moisture levels are 15 gr/pound of dry air or less. Because of high speed air flow through the turbine, the entrained moisture exists as a fine, dispersed mist rather than as water droplets. It has been determined in current aircraft environmental control system applications that the fine dispersed mist can aid in crew compartment cooling, provided free moisture in the cooling air is 15 gr/lb or less. Therefore, the dry air rated supply temperature, (T_{DAR}) which is an adjusted temperature to account for the evaporative cooling capability of the entrained mist, is used to determine system cooling capacity. This approach does not apply to the vapor compression system where entrained moisture is in the form of droplets off of the evaporator coils rather than as a fine mist, and dry bulb supply temperature is used to determine cooling capacity for the vapor compression system.

The air cycle system performance for the hot-humid, basic constant high humidity, basic variable high humidity, and basic hot climates are presented schematically in Figures 5-5 through 5-8 inclusive. The thermal loads were estimated by the procedures presented earlier in Table 5-II, and the system performance is determined for "full cold" operation (no hot thermal bypass). For the basic constant high humidity (Figure 5-6) and basic variable high humidity (Figure 5-7) cases, the drive compressor throttling valve is used to maintain a condenser outlet temperature of 35°F. For the hot-humid and basic hot cases (Figures 5-5 and 5-8, respectively), the drive compressor throttling valve is wide open and the condenser outlet temperature is greater than 35°F. Crew compartment temperatures range from 61.7°F to 82.6°F, with relative humidities ranging from 28 to 80 percent. However, the highest crew compartment effective temperature is 74.2°F at the hot-humid climate, compared to

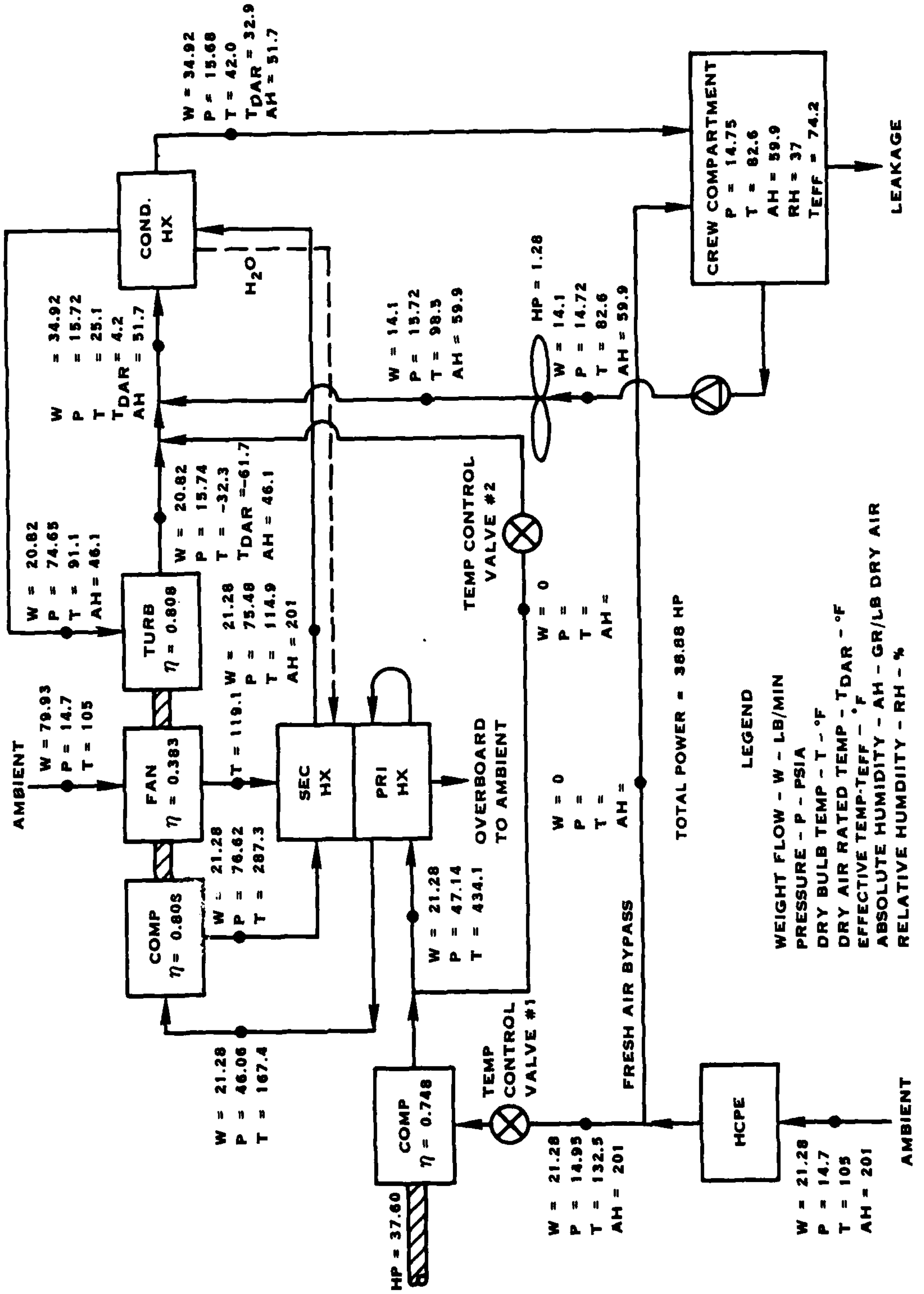


FIGURE 5-5. AIR CYCLE HOT-HUMID CLIMATE

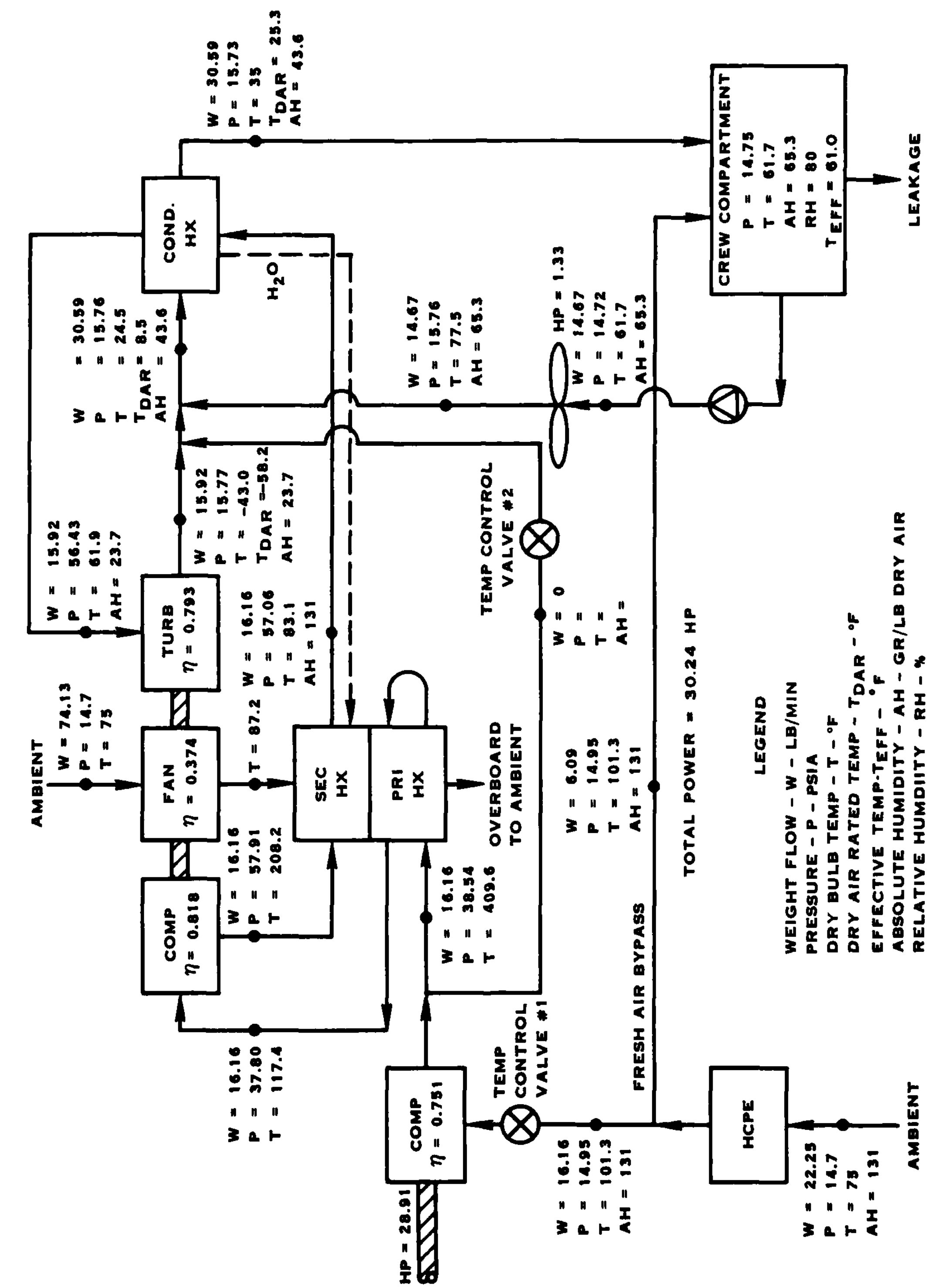


FIGURE 5-6. AIR CYCLE BASIC CONSTANT HIGH HUMIDITY CLIMATE

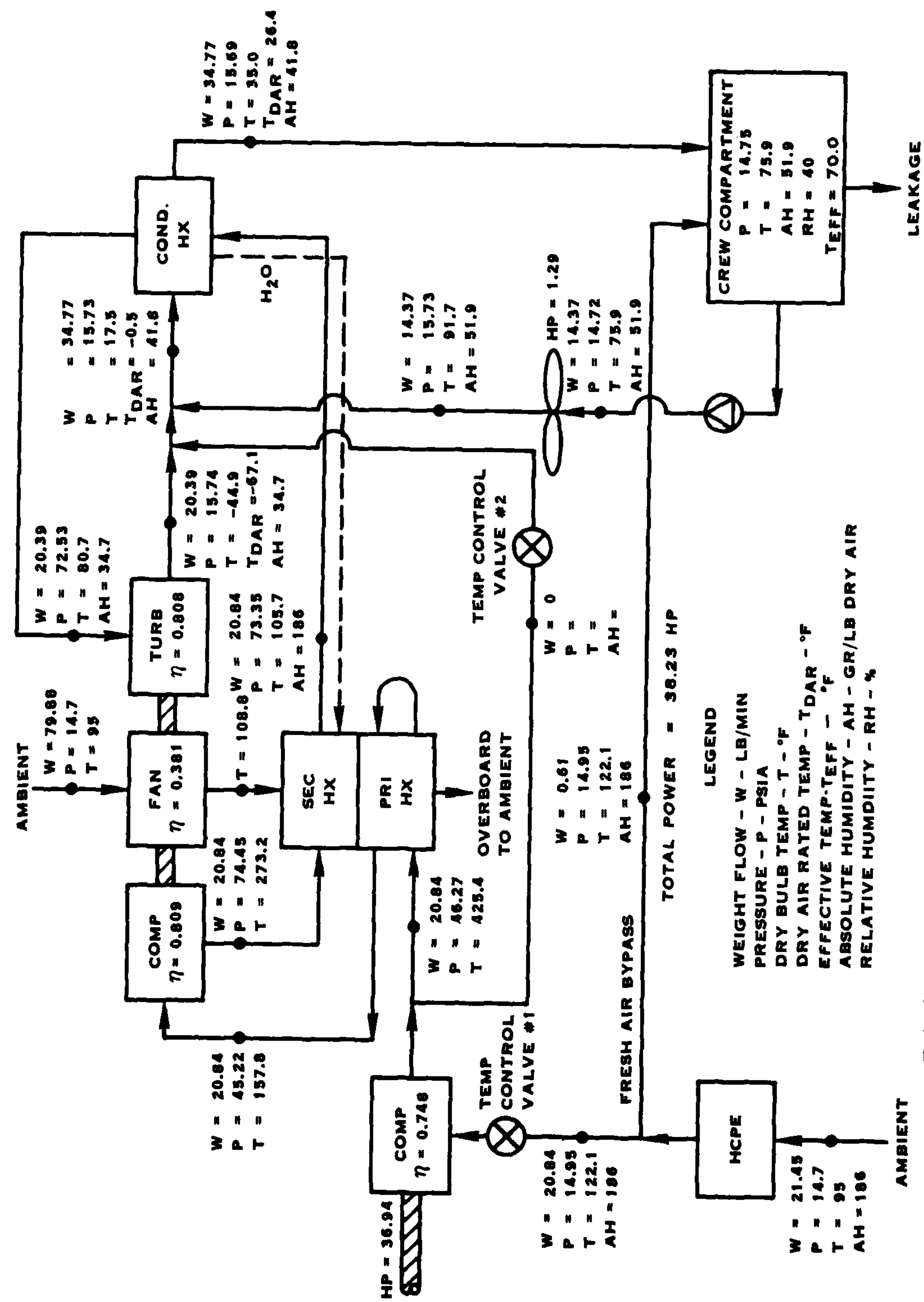


FIGURE 5-7. AIR CYCLE BASIC VARIABLE HIGH HUMIDITY CLIMATE

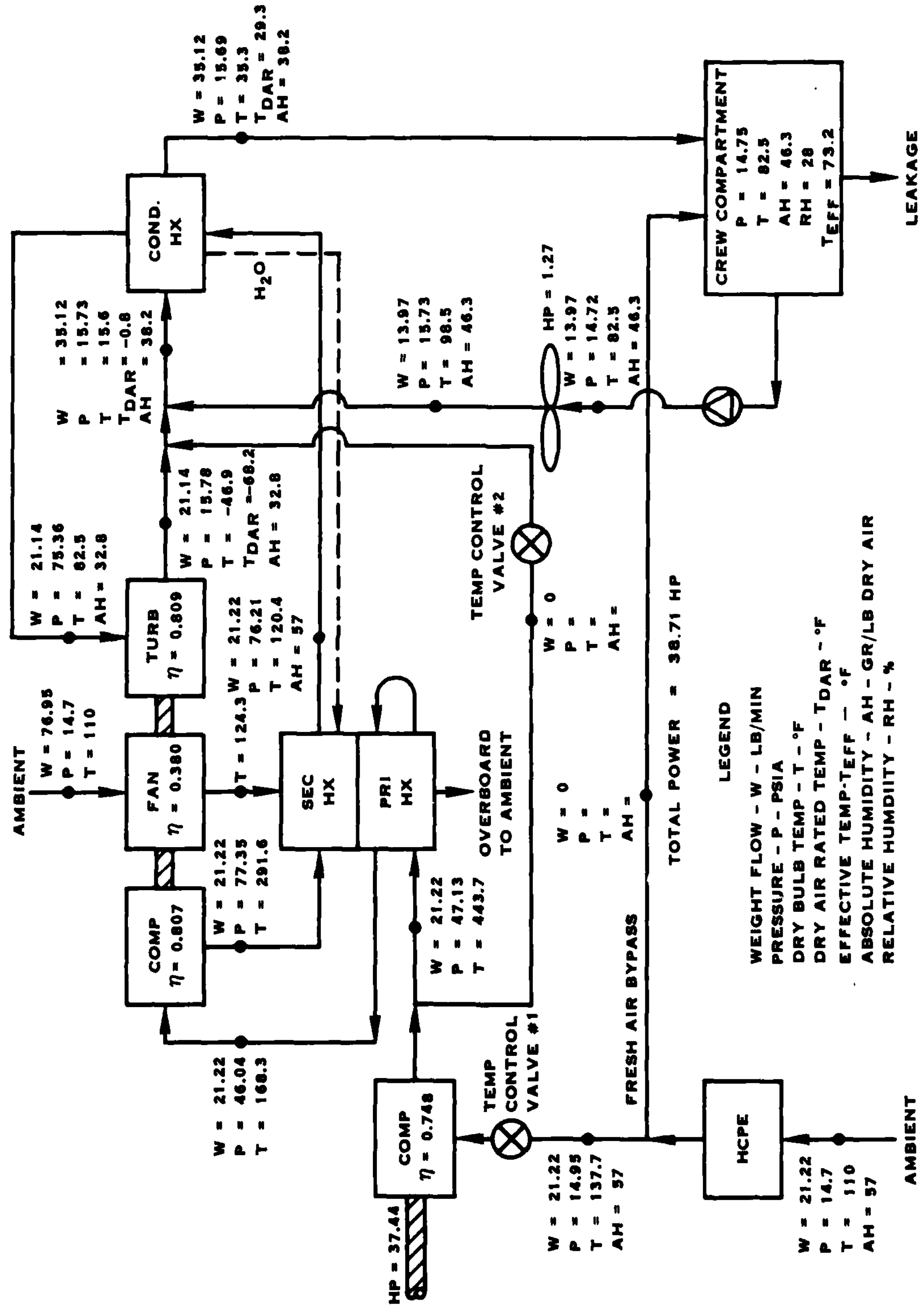


FIGURE 5-8. AIR CYCLE BASIC HOT CLIMATE

5.4 (Continued)

a target value of 76.8°F based on 90°F dry bulb temperature at 20% relative humidity. A complete summary and comparison of all system performance is presented in Section 7.1.

Figures 5-9 through 5-13 present air cycle system operation for the hot-dry, hot-humid, basic constant high humidity, basic variable high humidity, and basic hot climates with 600 cfm fresh make-up air in a non-NBC environment. As the drive compressor was sized for a 300 cfm capacity with a wide open throttling valve, a large proportion of the fresh make-up air is bypassed directly to the crew compartment for non-NBC operation. This infusion of warm make-up air directly into the crew compartment results in higher crew compartment temperatures for all of these climates than exists for the NBC environment presented in Figures 5-4 through 5-8. For non-NBC operation with the air cycle system, a 2 speed blower in the HCPE may be needed to reduce the ventilation rate to limit the infusion of unconditioned warm ambient air on hot days or cold ambient air on severe cold days.

As in the previous cases, the drive compressor throttling valve is set to obtain a 35°F dry bulb condenser outlet temperature with no hot thermal bypass. This valve will only actuate for operation in the basic constant high humidity climate (Reference Figure 5-11). At all other climates requiring cooling in a non-NBC environment, the throttling valve is wide open and the condenser outlet dry bulb temperature is greater than 35°F.

One of the advantages of an air cycle conditioning scheme is the inherent heating capability of the cycle. This heating capability is demonstrated for the severe cold climate in an NBC environment (300 cfm make-up) in Figure 5-14. For this case, the hot thermal bypass is wide open and the drive compressor

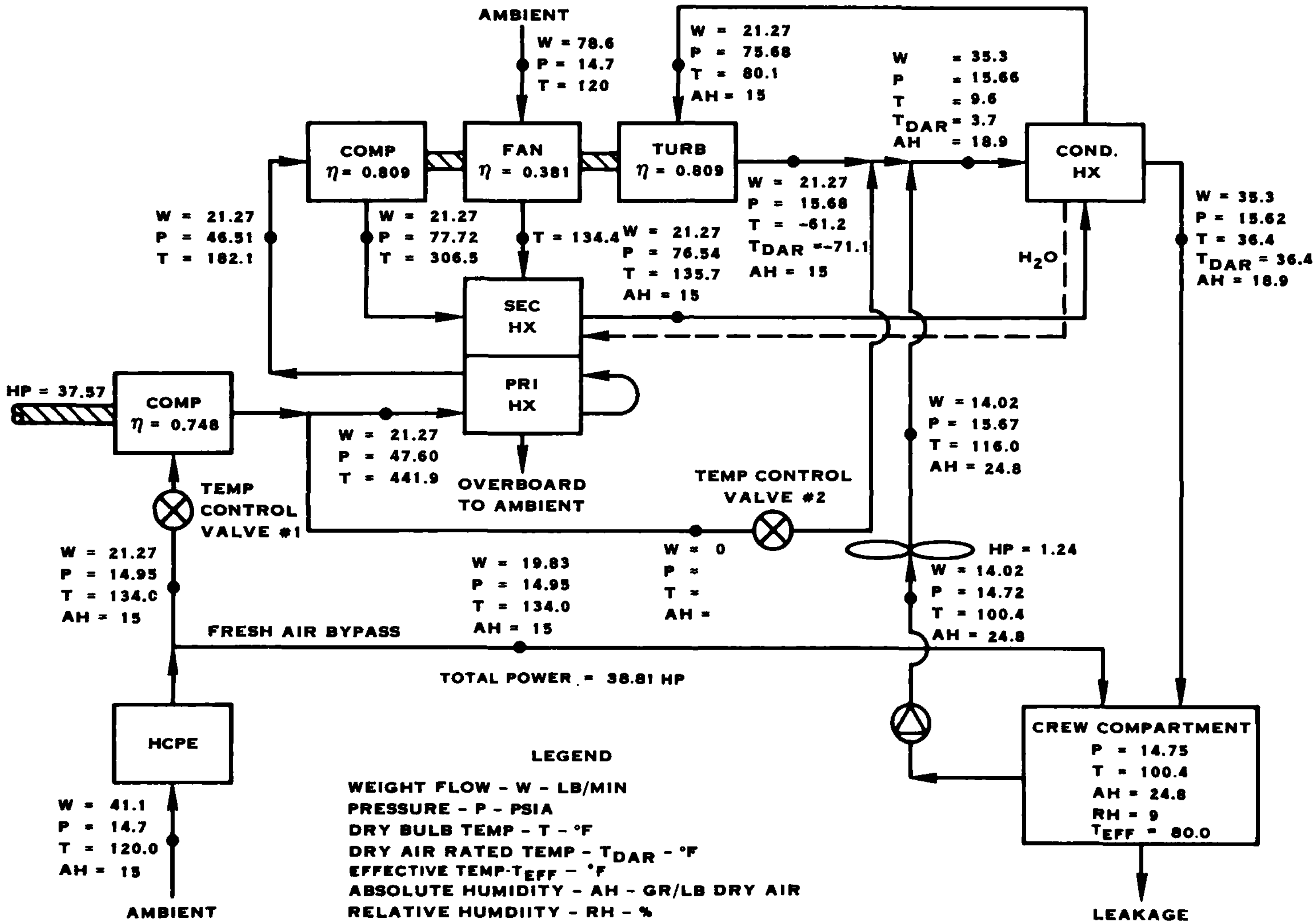


FIGURE 5-9. AIR CYCLE HOT DRY CLIMATE, NON-NBC

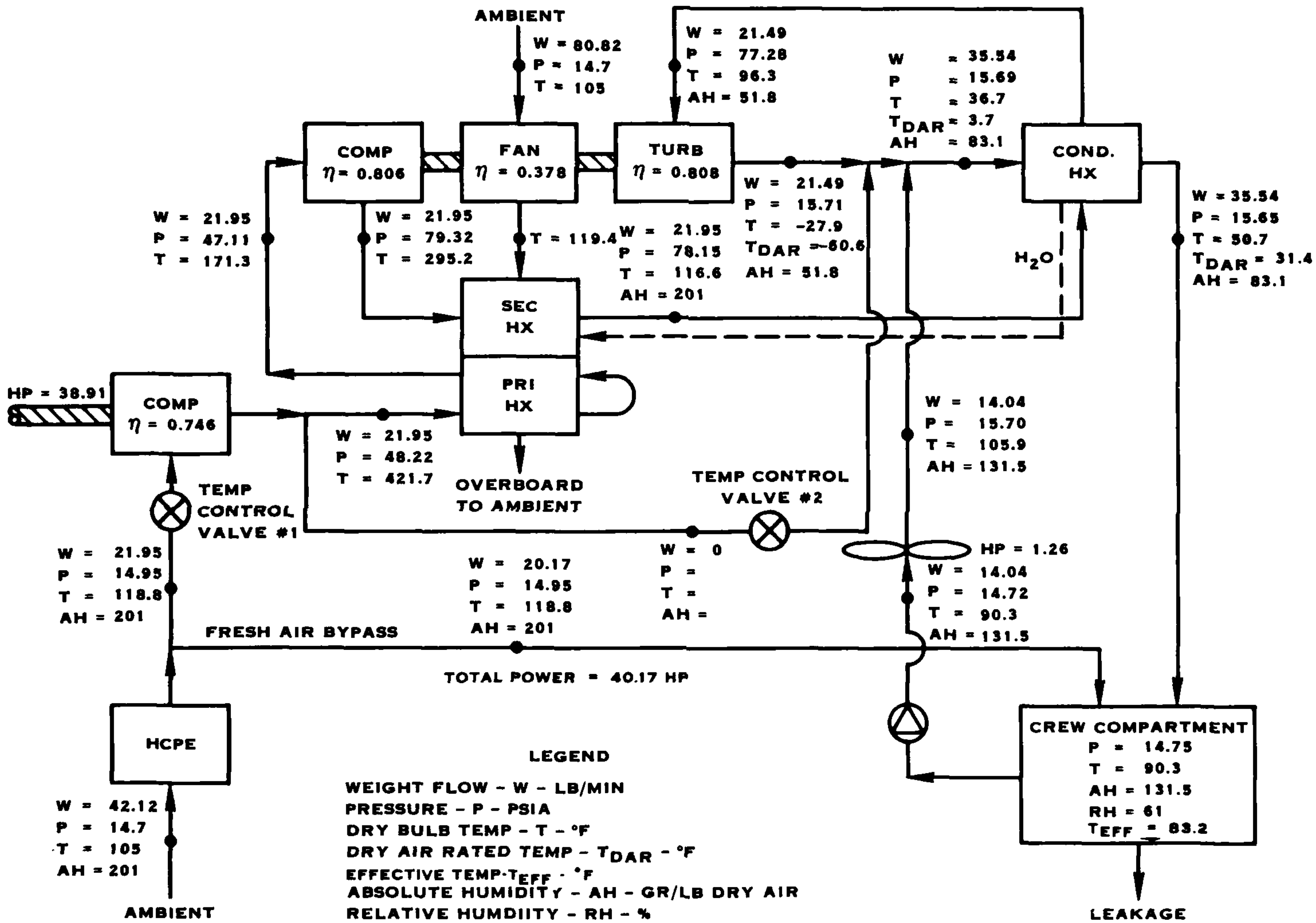


FIGURE 5-10. AIR CYCLE HOT HUMID CLIMATE, NON-NBC

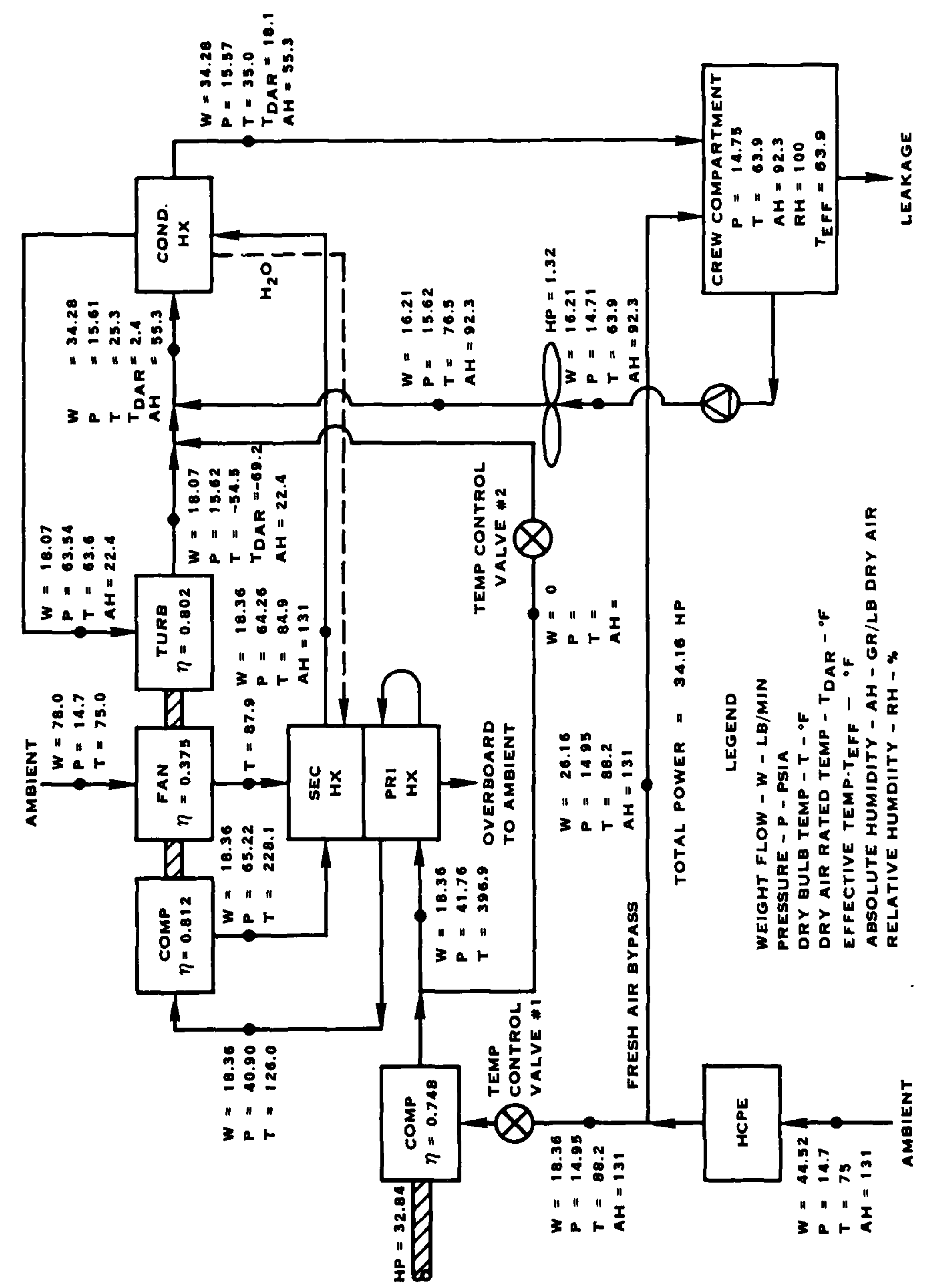


FIGURE 5-11. AIR CYCLE BASIC CONSTANT HIGH HUMIDITY CLIMATE, NON-NBC

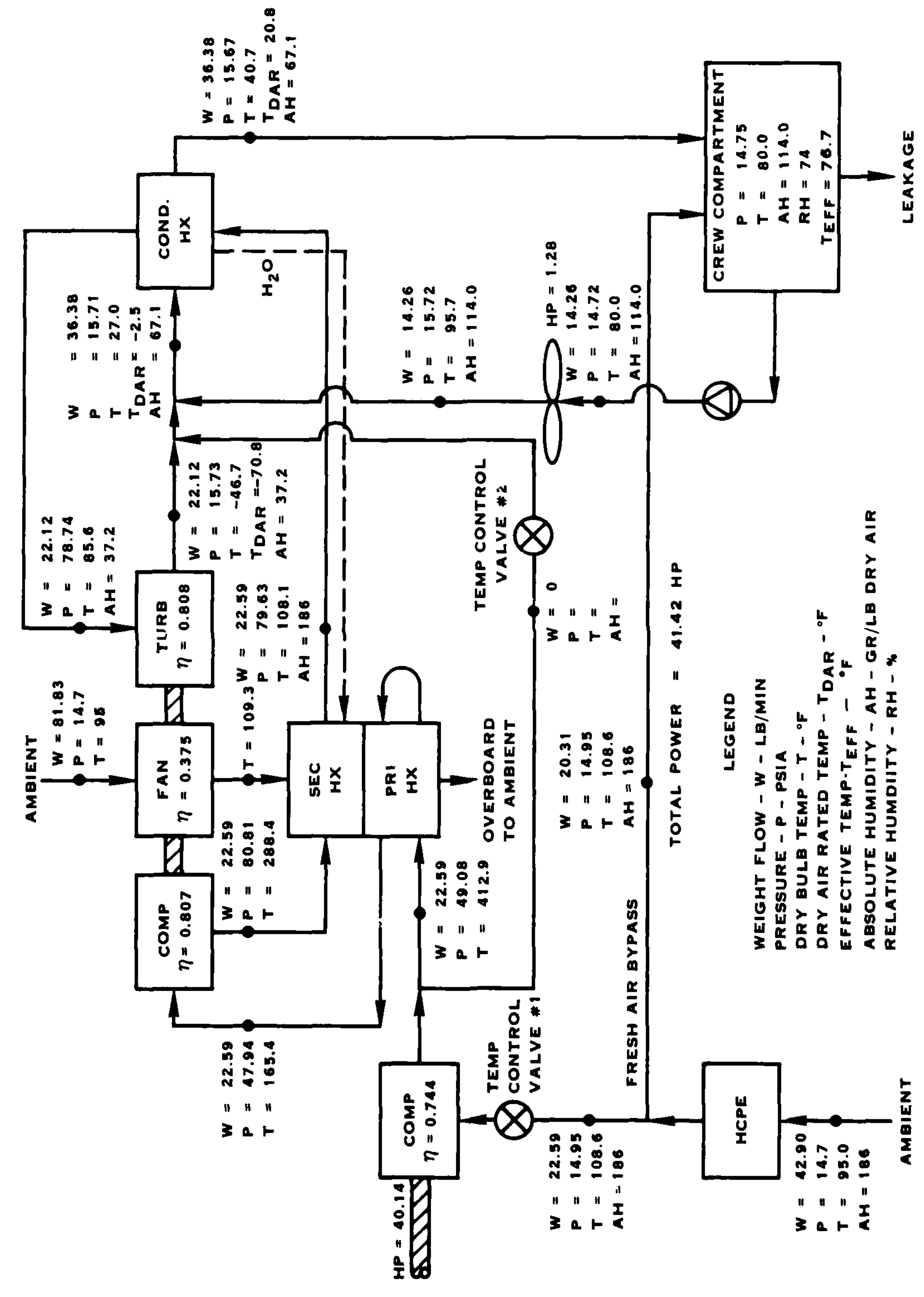
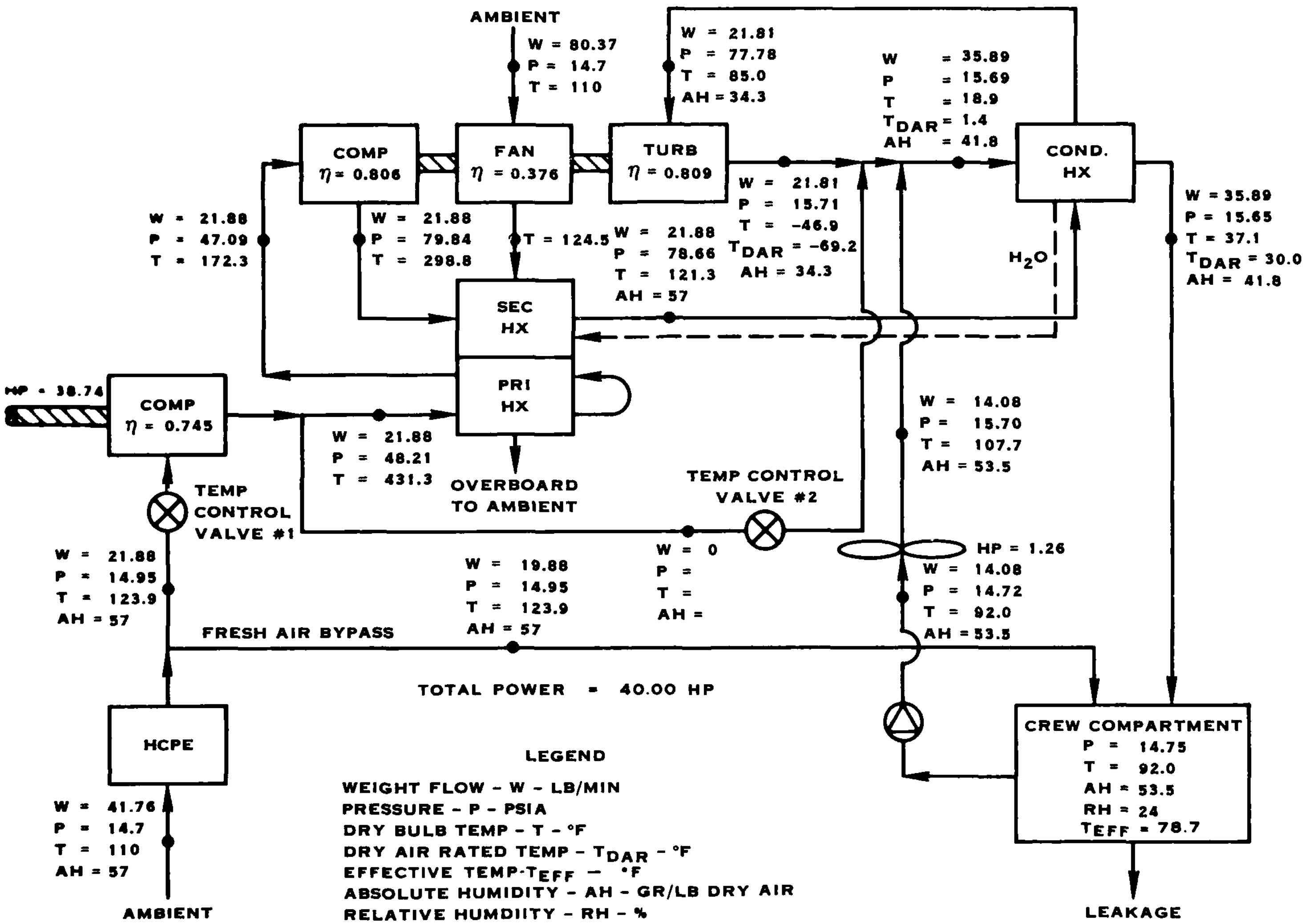


FIGURE 5-12. AIR CYCLE BASIC VARIABLE HIGH HUMIDITY CLIMATE, NON-NBC



LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMP - T - °F
DRY AIR RATED TEMP - T_{DAR} - °F
EFFECTIVE TEMP-T_{EFF} — °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMDIITY - RH - %

FIGURE 5-13. AIR CYCLE BASIC HOT CLIMATE, NON-NBC

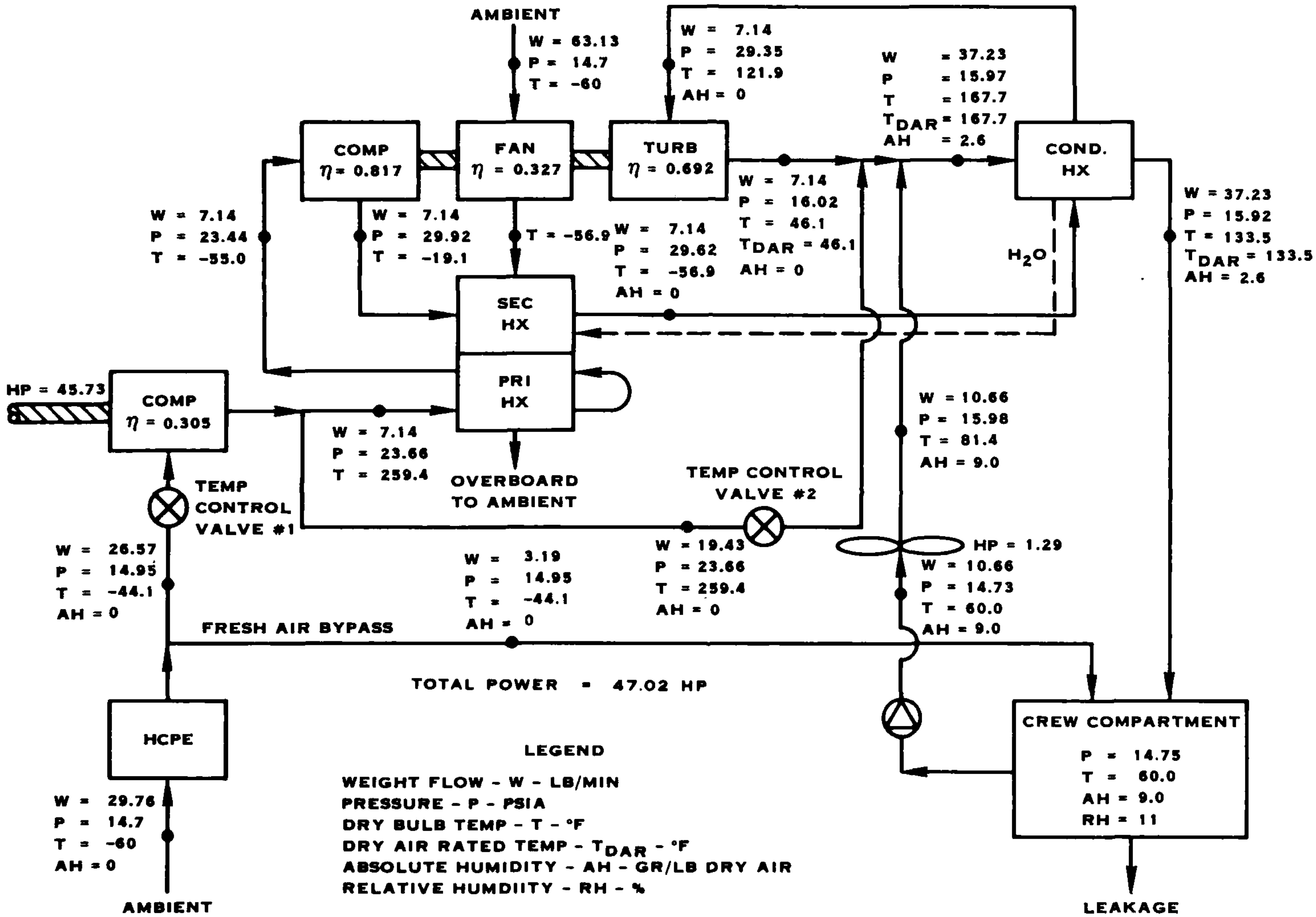


FIGURE 5-14. AIR CYCLE SEVERE COLD CLIMATE

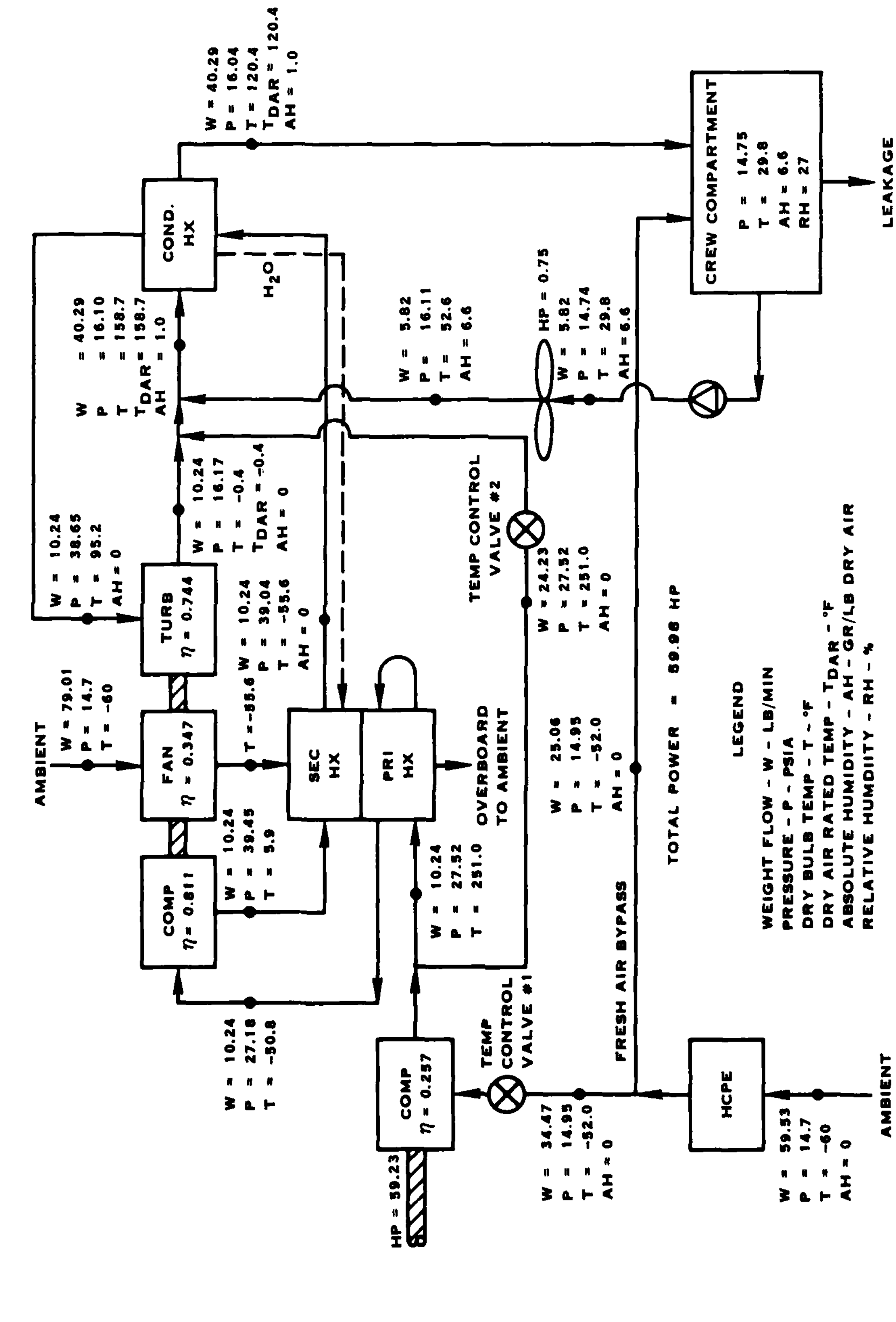


FIGURE 5-15. AIR CYCLE SEVERE COLD CLIMATE, NON-NBC

5.4 (Continued)

throttling valve has partially reopened to hold a 60°F crew compartment temperature.

Figure 5-15 depicts the system heating capability for the severe cold climate in a non-NBC environment. In this case, both the hot thermal bypass and the drive compressor throttling valve are wide open but due to the amount of cold fresh air being bypassed, the crew compartment temperature is only 29.8°F. However, this is still a large amount of heating capability (approximately 90°F temperature increase) and the amount of ventilation through the HCPE can be reduced if higher crew compartment temperatures are desired, such as with the use of a 2 speed non-NBC blower as discussed above.

Figures 5-14 and 5-15 demonstrate the heating capability of the air cycle. However, to obtain this capability, large amounts of power are required by the drive compressor (45.7 and 59.2 hp, respectively). In actuality, the power available for the drive compressor may be limited, and actual heating capacity will be less than that presented in the figures.

The last air cycle performance figures presented, Figures 5-16 and 5-17, indicate system performance when the ventilated facepiece is operational with 300 cfm HCPE flow. Figure 5-16 presents operation in the hot-dry climate and Figure 5-17 presents operation on a hot-humid day. For these cases, twenty cfm (5 cfm per man) is tapped off to the ventilated facepieces upstream of the drive compressor. This provides filtered but uncooled air to the ventilated facepieces and can be used whether the air cycle is being operated or not.

Another alternative would be to tap cooled air for the facepieces from a location downstream of the condensing heat exchanger and upstream of the crew compartment. To prevent the possibility of breathing contaminated air, this

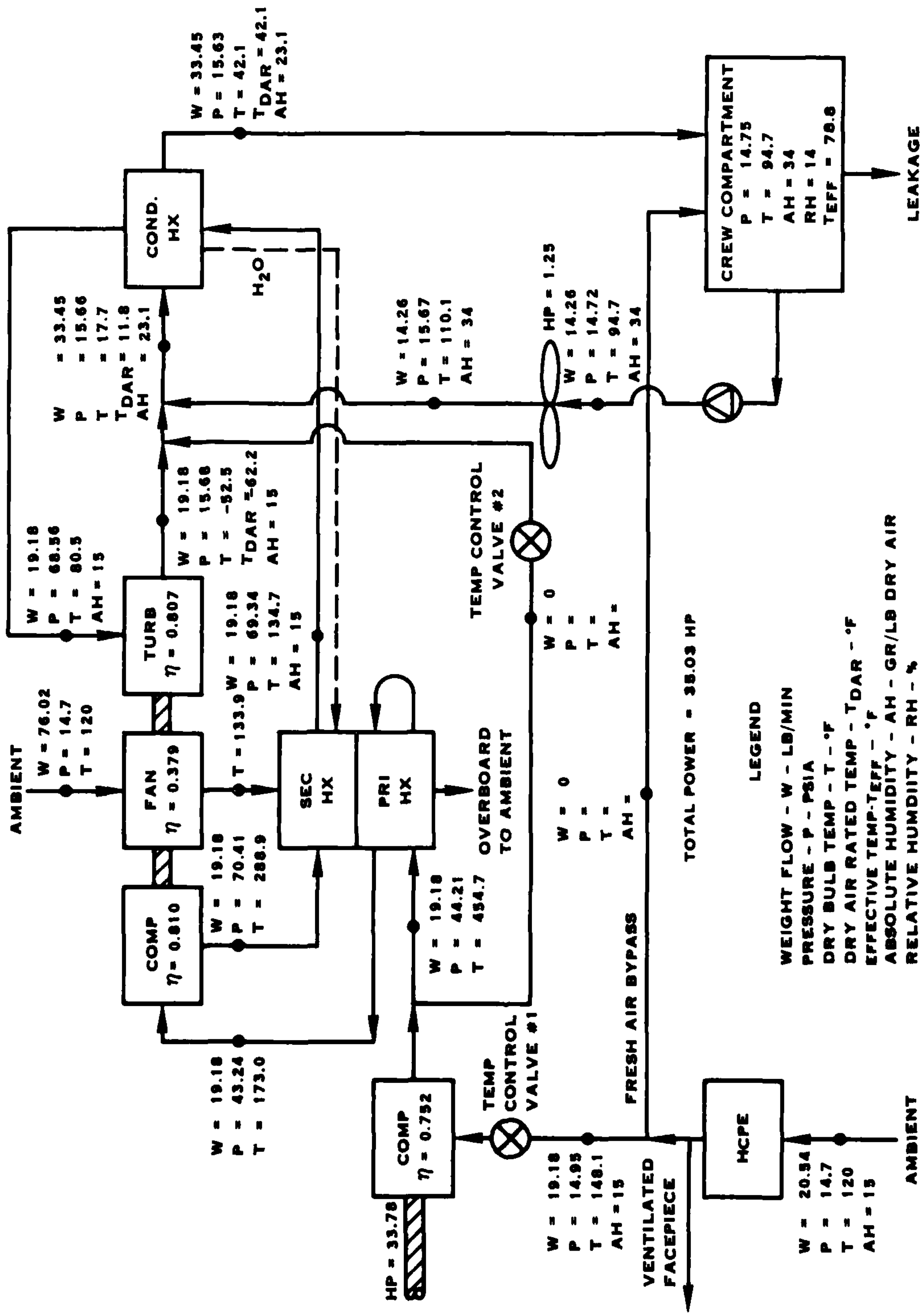


FIGURE 5-16. AIR CYCLE HOT-DRY CLIMATE WITH VENTILATED FACEPIECE

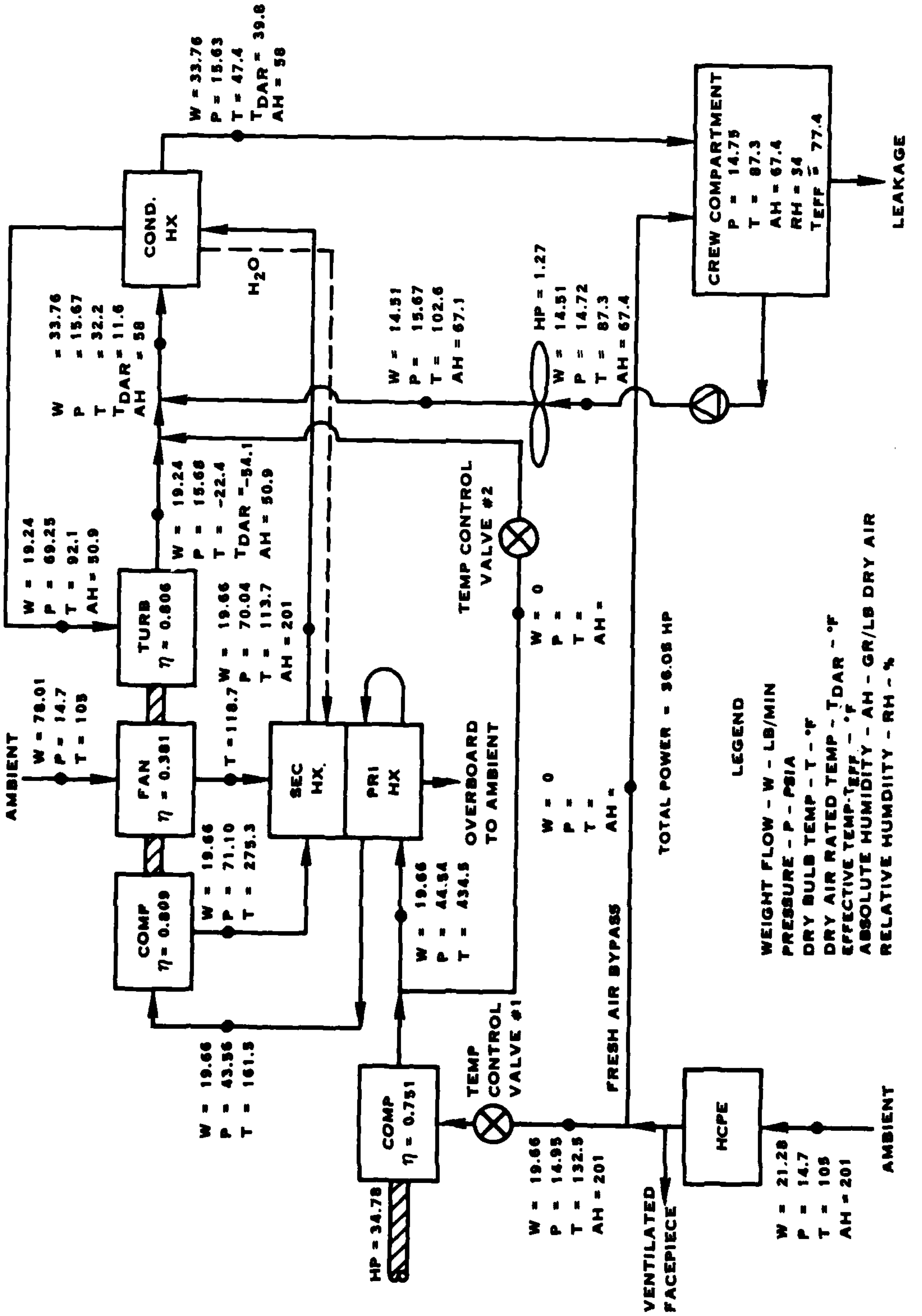


FIGURE 5-17. AIR CYCLE HOT-HUMID CLIMATE WITH VENTILATED FACEPIECE

5.4 (Continued)

tap off could only be allowed to mate with individual filter fittings. Operation in this mode is identical to system operation with an HCPE flow rate of 300 cfm as presented in Figures 5-4 and 5-5. This alternative will not be functional when the air conditioning system is turned off. The ventilated facepiece tap off location of Figures 5-16 and 5-17 must be used when cooling is not provided.

5.5 VAPOR COMPRESSION CYCLE SYSTEM DESCRIPTION

The vapor compression system that was studied is depicted schematically in Figure 5-18. This system employs the existing technology that has been extensively used for home and automobile air conditioning.

Air recirculated from the crew compartment is mixed with fresh make-up air from the HCPE. Heat is rejected from this air mixture by evaporating refrigerant in the evaporator. The cooled air is then distributed in the crew compartment.

Superheated vapor at low pressure from the evaporator is compressed before entering the condenser. Heat is rejected from the refrigerant to circulating ambient air in the condenser, and the refrigerant leaves the condenser as liquid at a high pressure. The refrigerant then passes through an expansion valve where the pressure is lowered prior to entering the evaporator. Freon-12 is used as the refrigerant because of its stability, commercial availability, and vapor pressure characteristics.

As in the air cycle preliminary design, the vapor cycle is designed to provide 4 tons of cooling in the hot-dry climate with an HCPE flow rate of 300 cfm. Make-up flow rates of 600 cfm when the NBC filter is bypassed were considered as off-design operation. The recirculating air fan was assumed

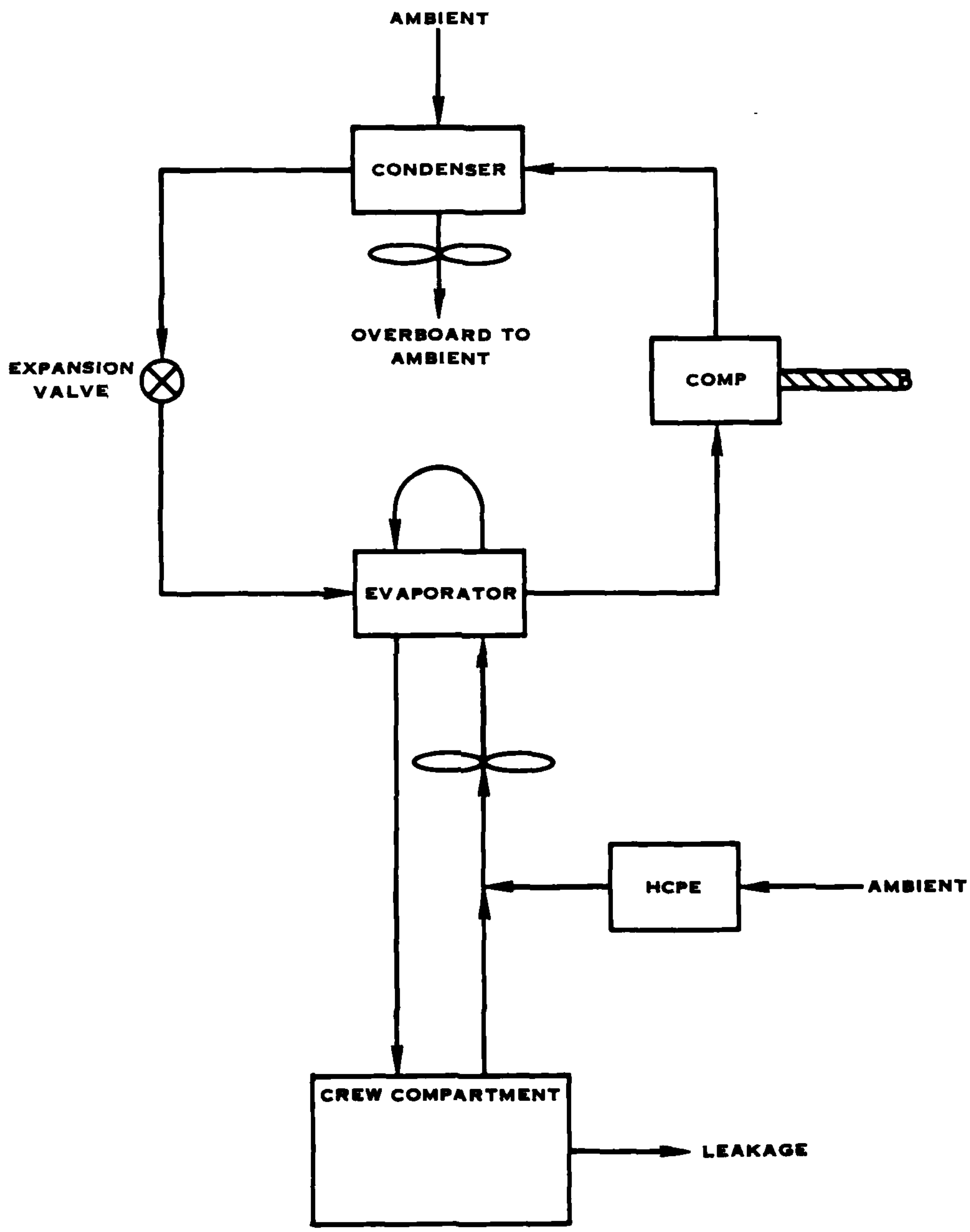


FIGURE 5-18. VAPOR COMPRESSION CYCLE SYSTEM SCHEMATIC

5.5 (Continued)

to pass the same flow rate regardless of the fresh air make-up rate because of equivalent interface pressures.

The placement of the HCPE interface with recirculated air upstream of the evaporator was selected for several reasons. Placing the HCPE interface upstream of the evaporator requires less air flow to satisfy the cooling requirements than placing the interface in the air return line between the evaporator and the crew compartment. This is a result of the evaporator air outlet temperature, which is constant regardless of the HCPE interface location. Thus, placing the interface downstream of the evaporator causes a higher crew compartment supply temperature, which in turn requires a higher air flow to maintain a constant crew compartment temperature. The lower air flow of an upstream HCPE interface is desirable because of lower pressure drops in the evaporator and lower fan power requirements. The upstream HCPE interface also results in lower crew compartment supply humidities because moisture is removed in the evaporator from all of the air flowing to the crew compartment. If the fresh air is mixed in downstream of the evaporator or is blown directly into the crew compartment, no moisture removal is accomplished on the make-up air and the relative humidity of the crew compartment will be higher than it is with the HCPE interface upstream of the evaporator. Finally, with the recirculating air fan placed between the upstream HCPE interface and the evaporator, some fresh make-up air will be drawn into the system even if the HCPE blower fails. The recirculating air fan could also be placed downstream of the evaporator. However, with the constraint on evaporator air outlet temperature, this would result in a higher crew compartment supply air temperature, requiring higher air flow rates and more recirculating fan power. For this reason, the

5.5 (Continued)

recirculating air fan is placed upstream of the evaporator.

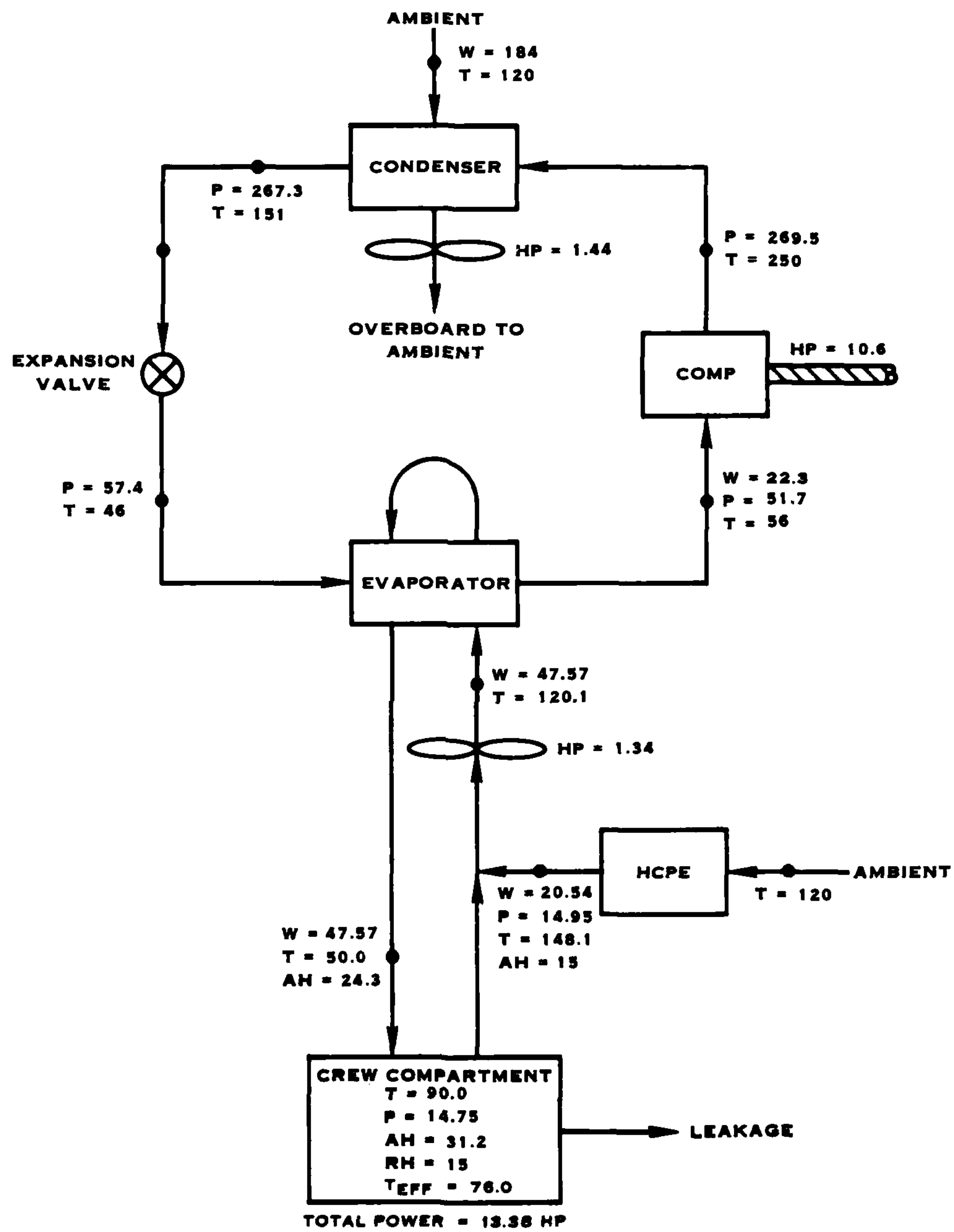
The refrigerant loop is a duplex design with dual evaporators, condensers, condenser air fans and compressors. This permits system modulation in climates where reduced cooling is needed, thus saving on power consumption. The system capacity is controlled by the cycling of the refrigerant compressor, as determined by the thermostat on the recirculating crew compartment air. When the air temperature rises a preset amount (e.g., 5°F) above the temperature setting, the second Freon system will then cycle to hold this new condition.

The vapor cycle also uses a thermostatic expansion valve to regulate the flow of refrigerant into the evaporator under changing conditions. This valve is controlled by the amount of superheat in the compressor suction gas. Under low loads as the amount of superheat decreases, the valve closes to protect the unit from flooding. The valve also limits the maximum suction pressure at high loads as all the valve fluid evaporates and limits the valve opening to a design maximum.

5.6 VAPOR COMPRESSION CYCLE SYSTEM PERFORMANCE

The design case for this system is presented in Figure 5-19. This is the hot-dry climate with a 4 ton cooling capacity and 300 cfm fresh air make-up through the HCPE. The total power requirement for system operation at this condition is 13.38 horsepower at 100% motor efficiencies for the refrigerant compressors and condenser and evaporator air circulation fans. This power requirement is approximately 36% of the power required by the air cycle for the same condition.

Off-design performance for the hot-humid, basic constant high humidity, basic variable high humidity, and basic hot climates with 300 cfm HCPE flow



LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE-T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-19. VAPOR COMPRESSION CYCLE HOT-DRY CLIMATE

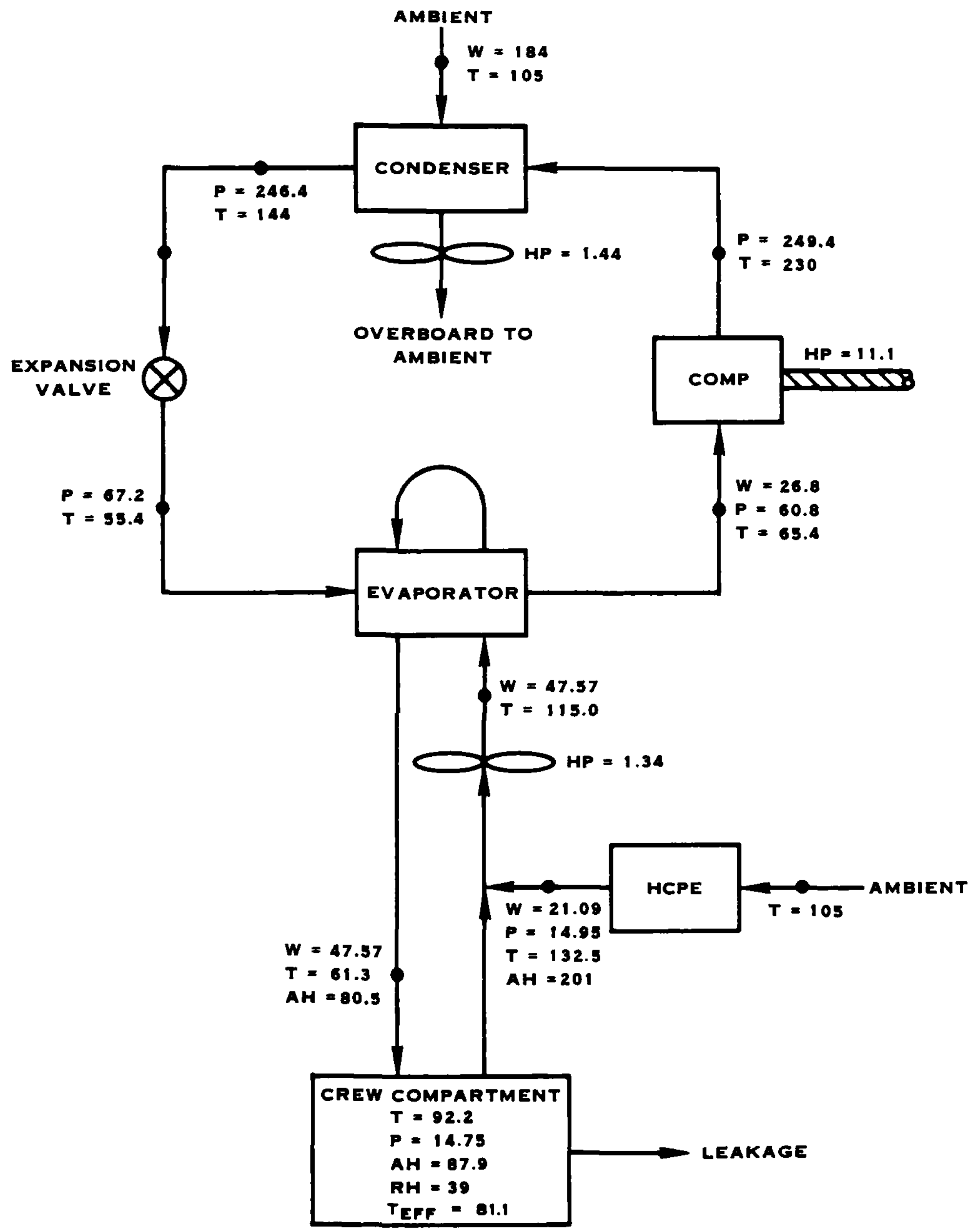
5.6 (Continued)

is presented in Figures 5-20 through 5-23, respectively. For operation in the basic constant high humidity climate (Figure 5-21), only half of the refrigerant cycle need be operated to hold a crew compartment dry bulb temperature of 72.4°F. The 90°F crew compartment dry bulb temperature requirement is exceeded only for the hot-humid climate (Figure 5-20), which also has the highest crew compartment effective temperature of 81.1°F.

The vapor compression system performance for the non-NBC environment with 600 cfm fresh air make-up flow is presented in Figures 5-24 through 5-28 for the hot-dry, hot-humid, basic constant high humidity, basic variable high humidity, and basic hot climates, respectively. The crew compartment temperature rises as high as 102°F for the hot-humid climate, where the maximum suction pressure is limited so as to limit the refrigerant system temperatures. This condition also results in a high crew compartment effective temperature of 89.0°F.

As the vapor compression system has no inherent heating capability, system operation at the severe cold climate is not applicable. Either electric or liquid heaters would have to be added to the recirculating crew compartment air loop to obtain heating in any new vehicle application. It has been assumed that the necessary heaters already exist in combat vehicles being considered for a crew compartment conditioning retrofit, and they have not been duplicated in this study.

If the ventilated facepieces are used, the required 20 cfm (5 cfm per crewman) can be tapped off either just downstream of the HCPE or just downstream of the evaporator. If the tap-off is just downstream of the HCPE, the air will not be cooled, but it will be filtered and purified and can be used

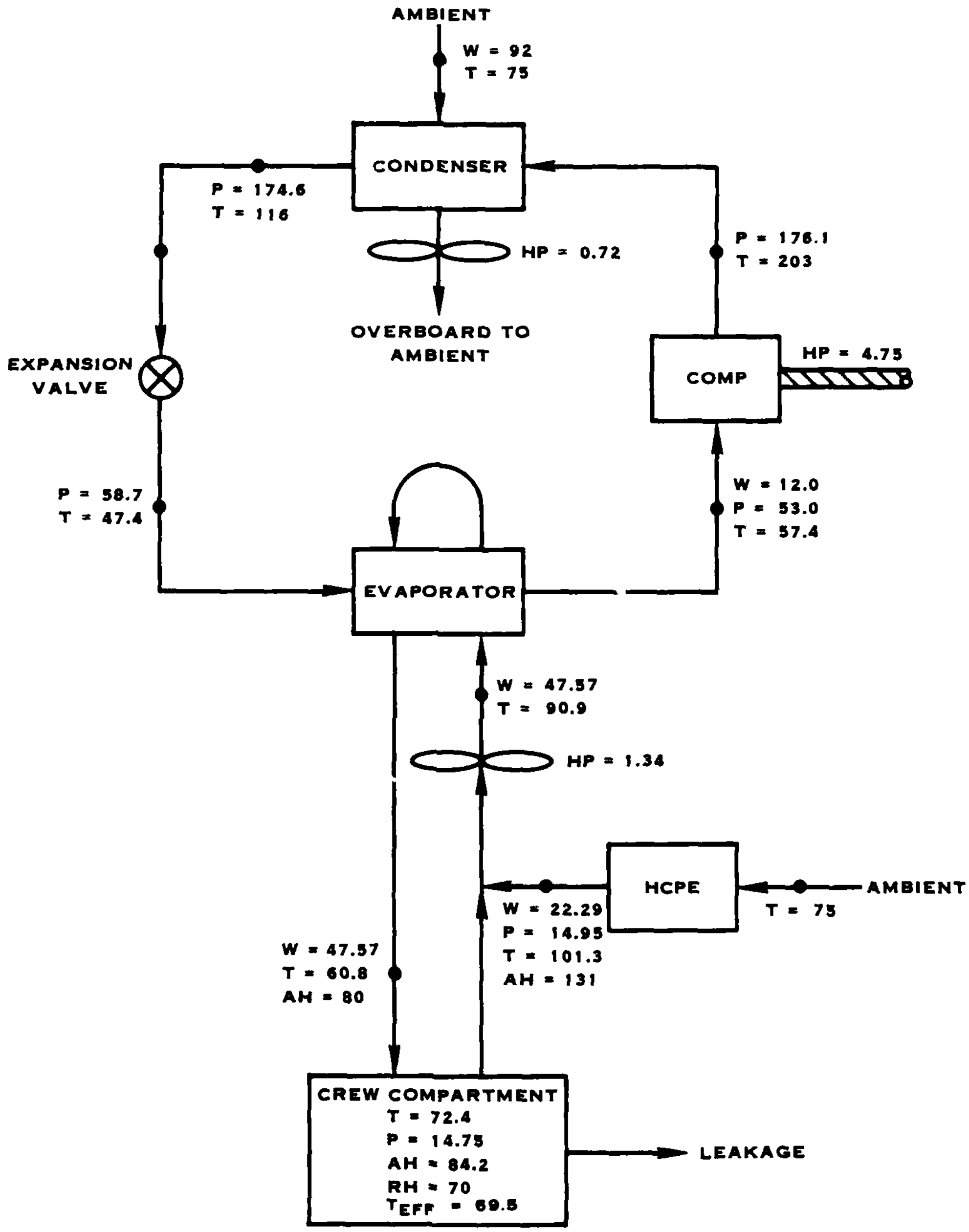


TOTAL POWER = 13.88 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE-T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-20. VAPOR COMPRESSION CYCLE HOT HUMID CLIMATE



TOTAL POWER = 6.81 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE-T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-21. VAPOR COMPRESSION CYCLE BASIC CONSTANT HIGH HUMIDITY CLIMATE

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COMBAT VEHICLE COOLING/HEATING DESIGN INVESTIGATION.(U)
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UNCLASSIFIED NL

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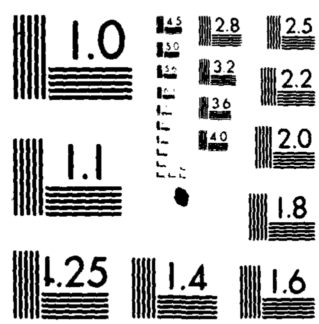

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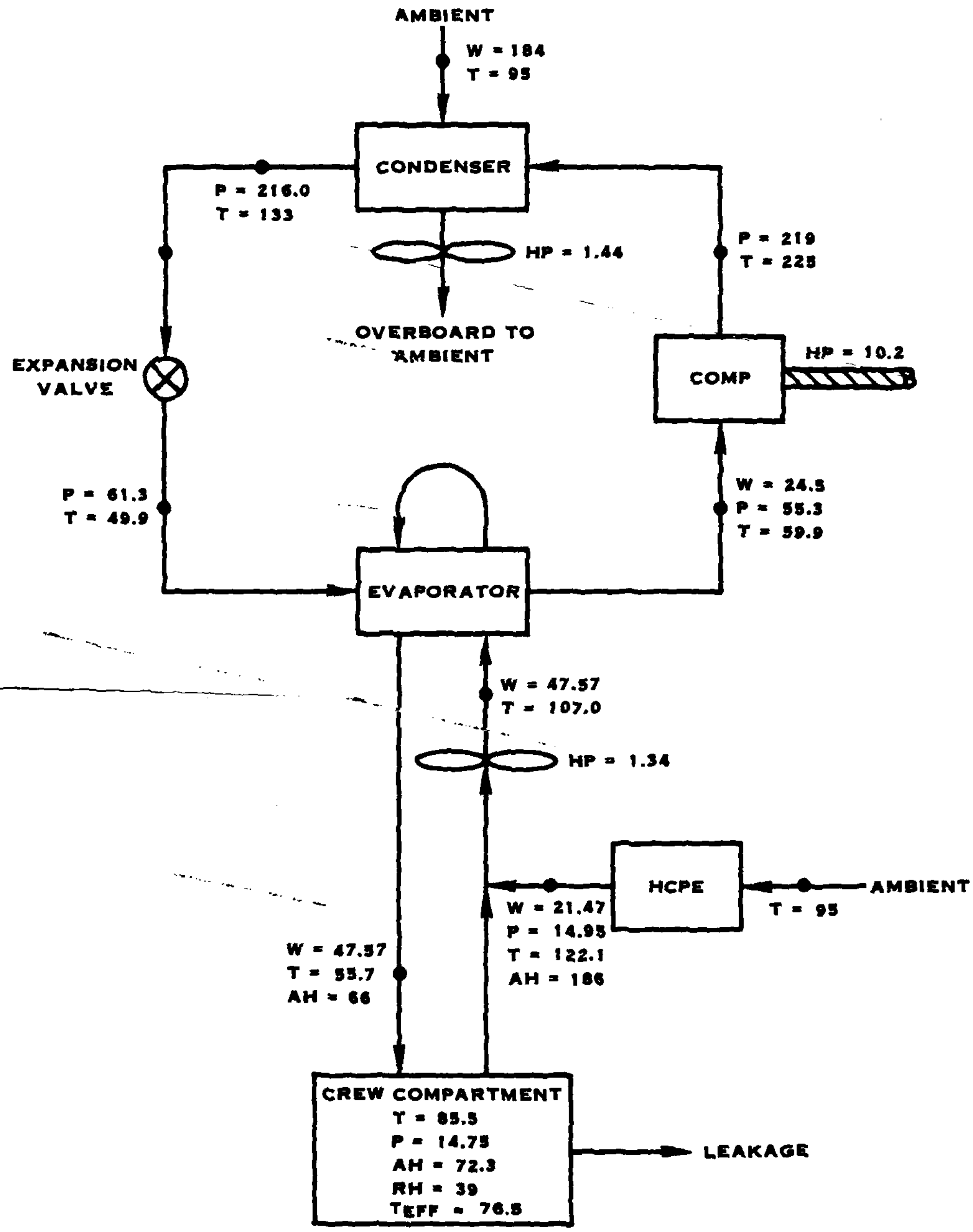
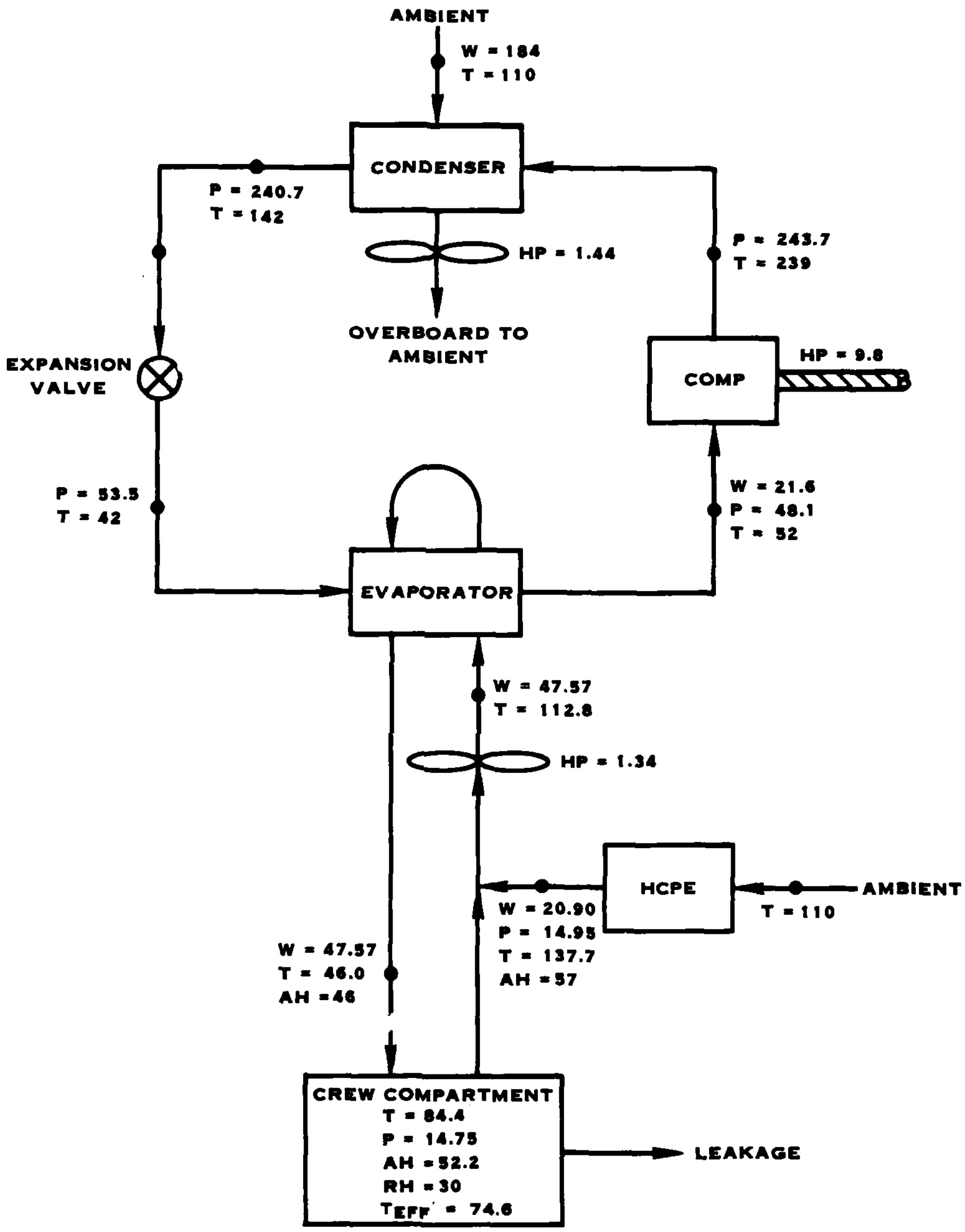


TOTAL POWER = 12.98 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE-T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-22. VAPOR COMPRESSION CYCLE BASIC VARIABLE HIGH HUMIDITY CLIMATE

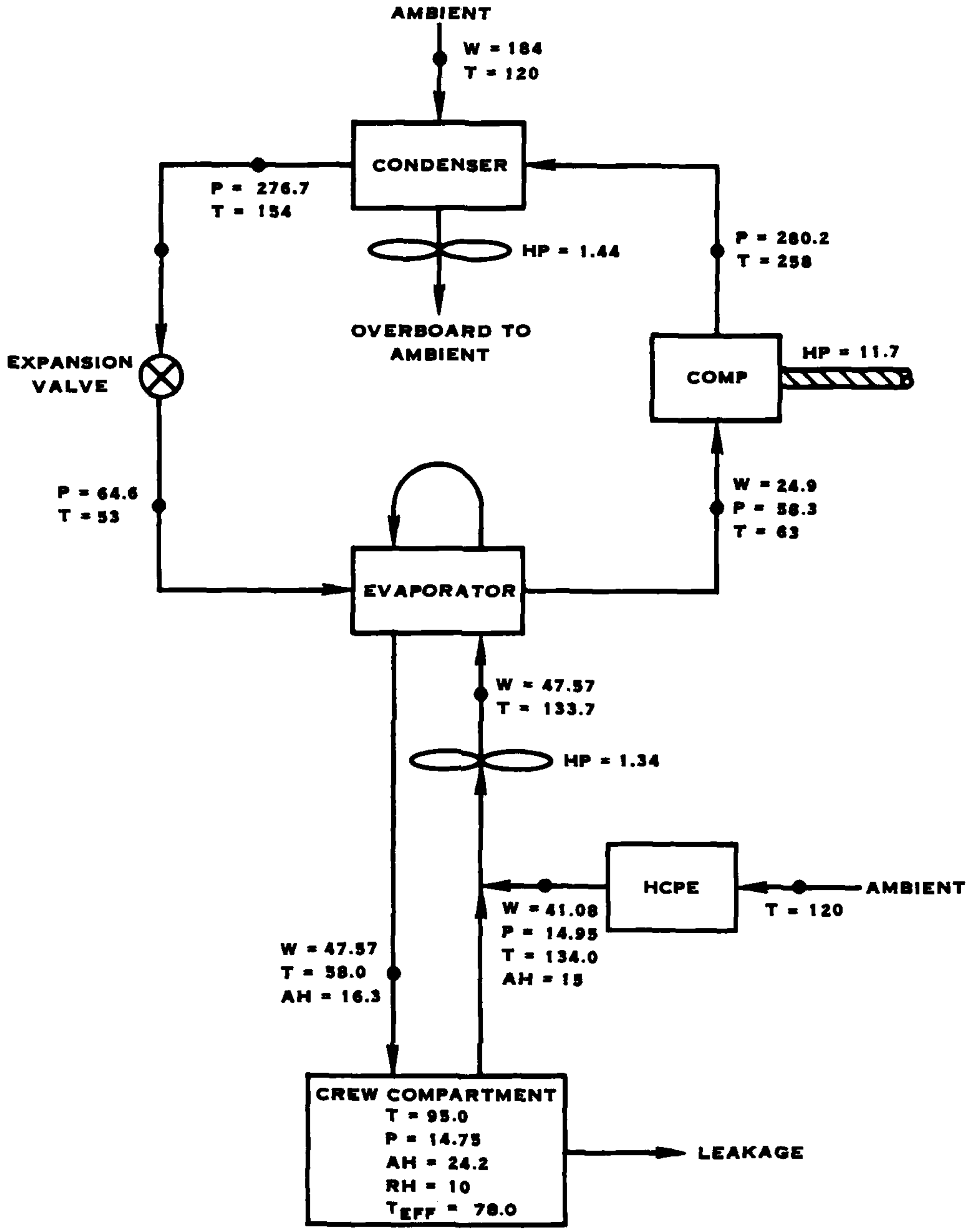


TOTAL POWER = 12.58 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE - T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-23. VAPOR COMPRESSION CYCLE BASIC HOT CLIMATE

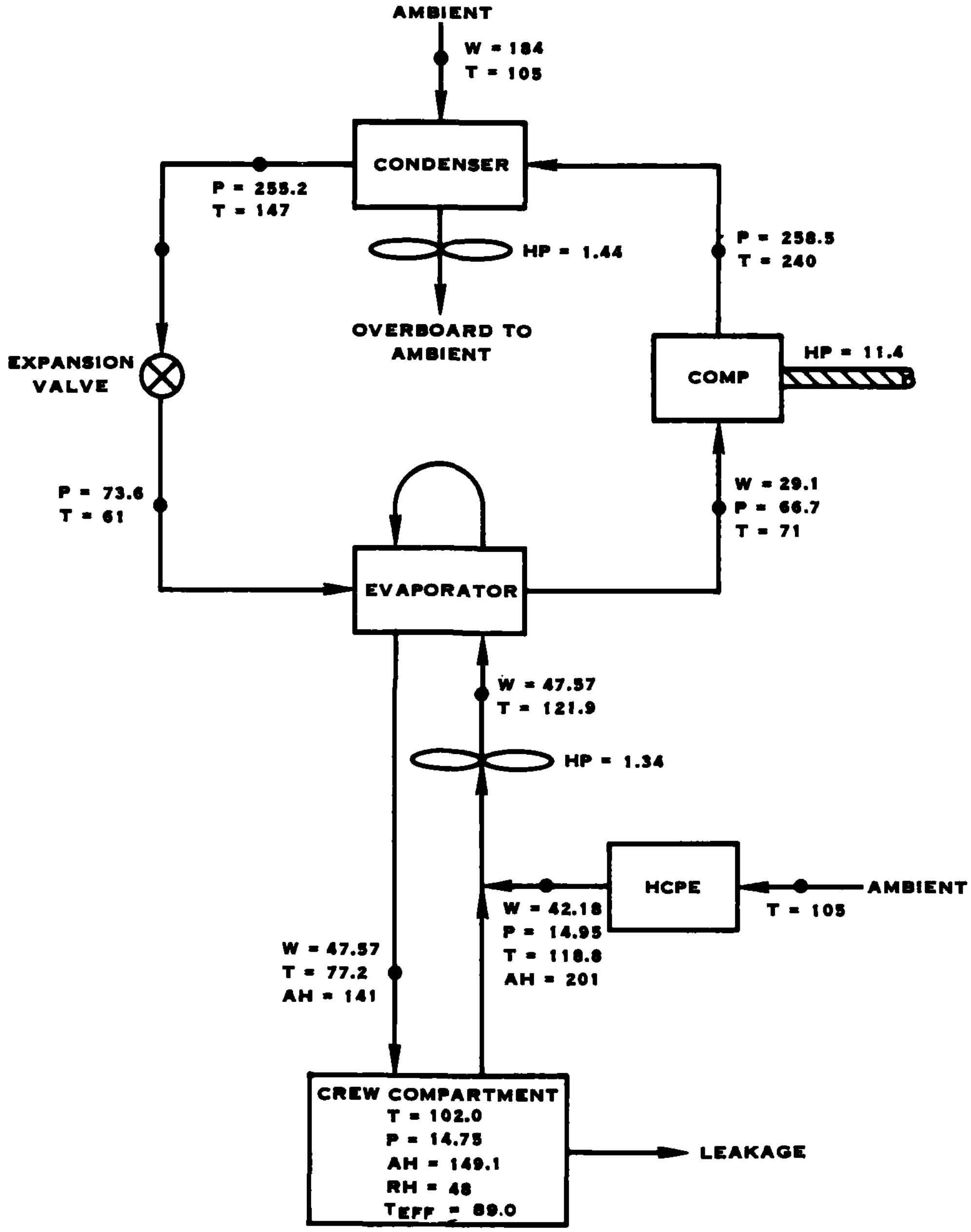


TOTAL POWER = 14.48 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE - T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-24. VAPOR COMPRESSION CYCLE HOT-DRY CLIMATE, NON-NBC

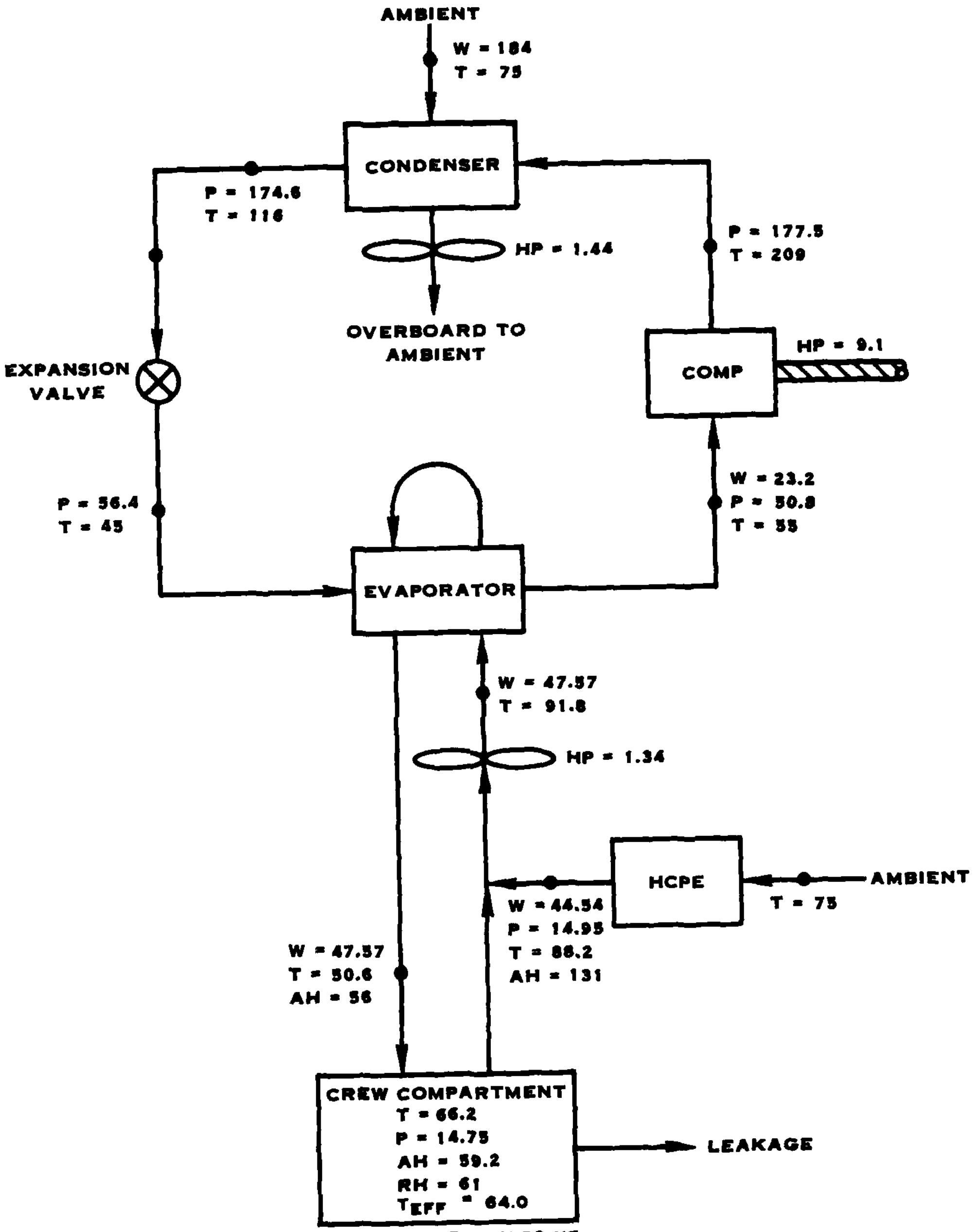


TOTAL POWER = 14.18 HP

LEGEND

WEIGHT FLOW – W – LB/MIN
PRESSURE – P – PSIA
DRY BULB TEMPERATURE – T – °F
EFFECTIVE TEMPERATURE-T_{EFF} – °F
ABSOLUTE HUMIDITY – AH – GR/LB DRY AIR
RELATIVE HUMIDITY – RH – %

FIGURE 5-25. VAPOR COMPRESSION CYCLE HOT-HUMID CLIMATE, NON-NBC

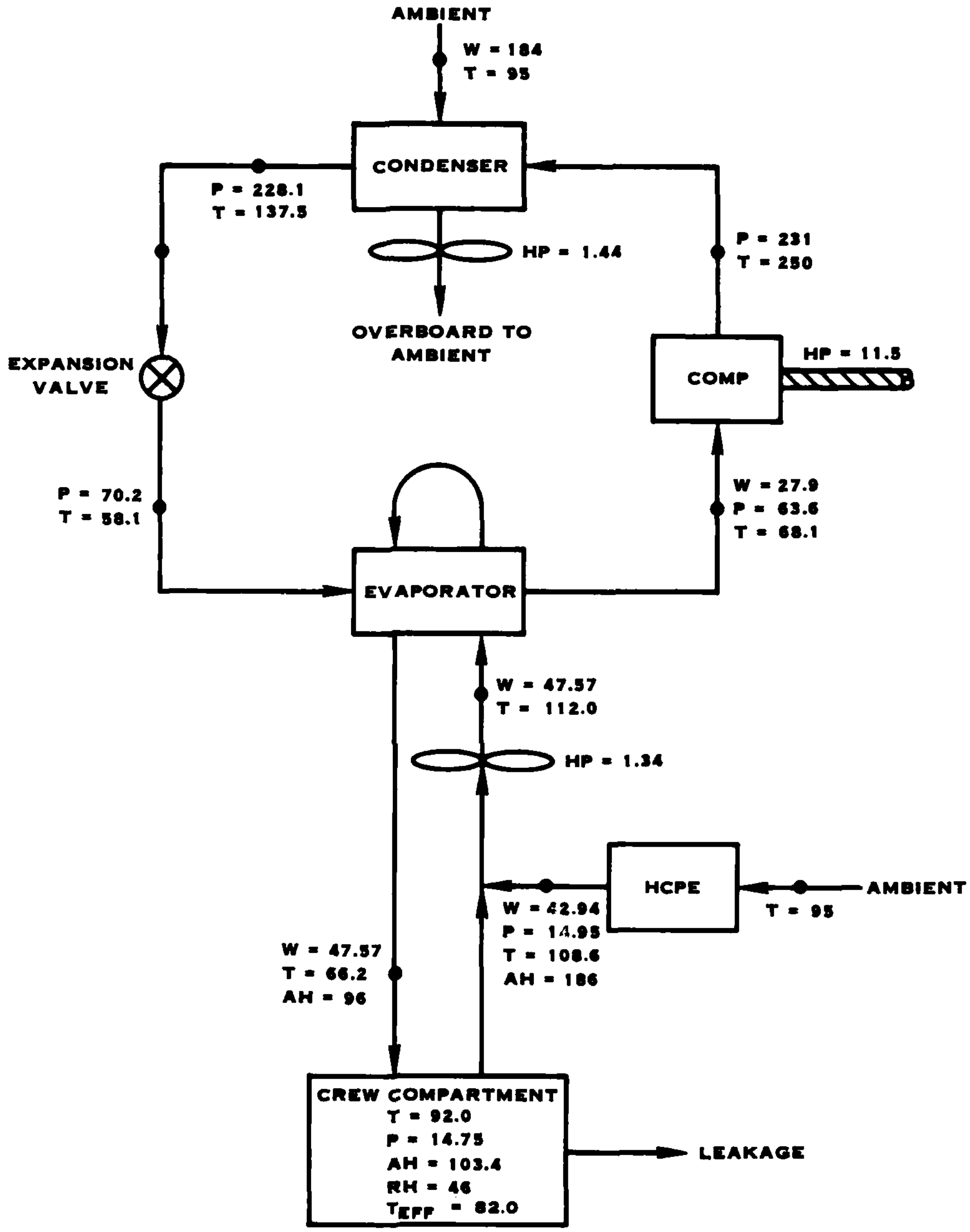


TOTAL POWER = 11.88 HP

LEGEND

WEIGHT FLOW – W – LB/MIN
PRESSURE – P – PSIA
DRY BULB TEMPERATURE – T – °F
EFFECTIVE TEMPERATURE – T_{EFF} – °F
ABSOLUTE HUMIDITY – AH – GR/LB DRY AIR
RELATIVE HUMIDITY – RH – %

FIGURE 5-26. VAPOR COMPRESSION CYCLE BASIC CONSTANT HIGH HUMIDITY CLIMATE, NON-NBC



TOTAL POWER = 14.28 HP

LEGEND

WEIGHT FLOW - W - LB/MIN
PRESSURE - P - PSIA
DRY BULB TEMPERATURE - T - °F
EFFECTIVE TEMPERATURE-T_{EFF} - °F
ABSOLUTE HUMIDITY - AH - GR/LB DRY AIR
RELATIVE HUMIDITY - RH - %

FIGURE 5-27. VAPOR COMPRESSION CYCLE BASIC VARIABLE HIGH HUMIDITY CLIMATE, NON-NBC

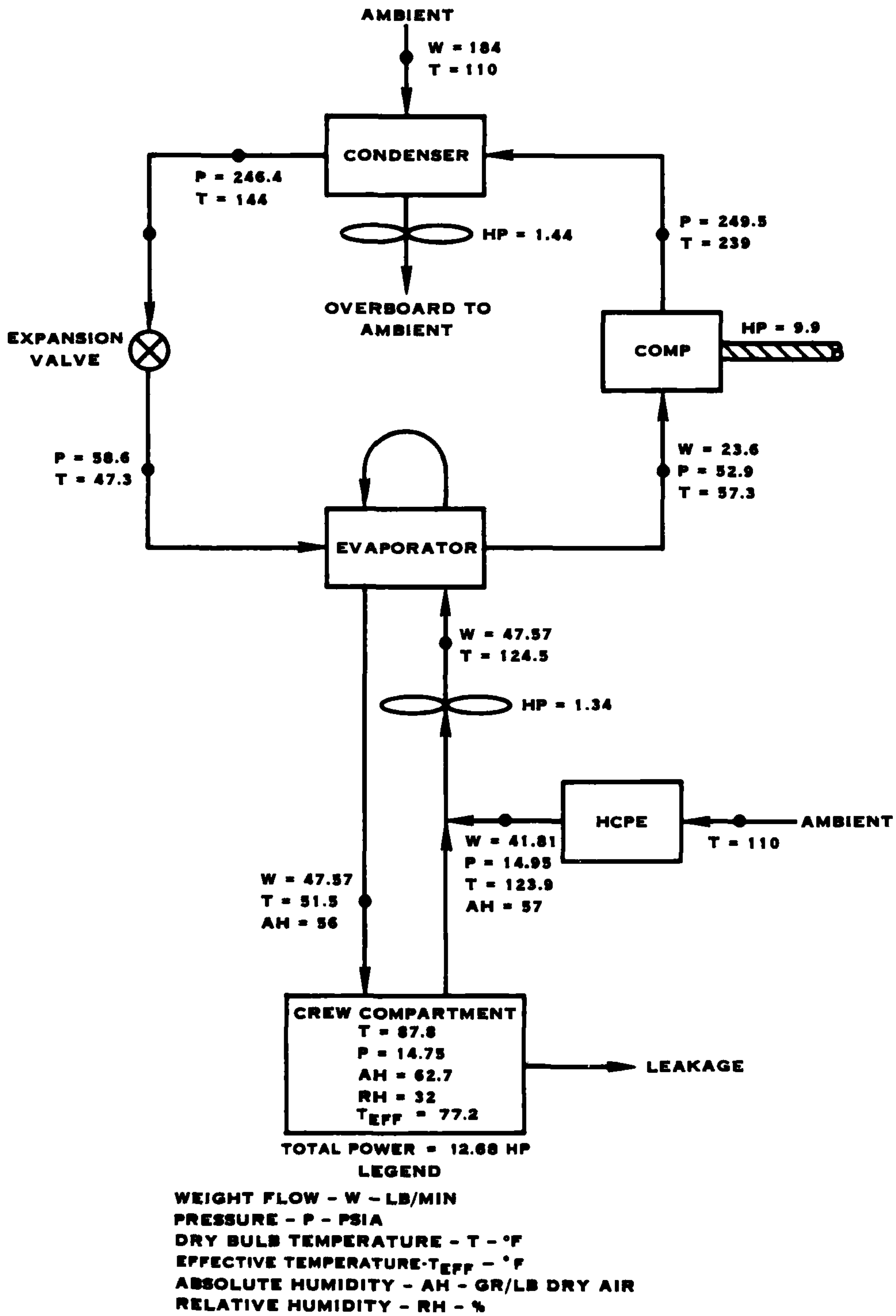


FIGURE 5-28. VAPOR COMPRESSION CYCLE BASIC HOT CLIMATE, NON-NBC

5.6 (Continued)

whether the system is operating or not. If the ventilated facepiece tap-off is downstream of the evaporator, the air will be cooled and system performance will be as depicted in Figures 5-19 through 5-23. However, this tap-off must be mated to a personal filter inlet to prevent possible CB contamination from recirculated crew compartment air.

SECTION 6.0
CANDIDATE SYSTEM EQUIPMENT

INTRODUCTION

This section will present approximate component and package envelopes for both the air and vapor cycle systems. Since the vapor cycle is commonly used in vehicle applications and the vapor cycle system for this application is essentially the MBT system (References 9, 10), an extensive equipment description will not be presented. However, the air cycle equipment description will be covered in greater depth because it is more commonly associated with aircraft rather than vehicle conditioning. The technology associated with the air cycle system is also newer and less commonly known than for the vapor cycle.

The approximate weight of the complete system without armor shielding will also be presented. Armor shielding will be required if the systems are mounted externally. However, it was assumed that the weight of the armor shielding would be equivalent for either concept.

6.1 AIR CYCLE

The air cycle system depicted schematically in Figure 5-2 is a slightly modified version of the DHC-8 aircraft environmental control system which is currently under development at Hamilton Standard. The major components of the air cycle system are:

- air cycle machine
- primary/secondary heat exchanger
- condenser/mixer
- recirculating air fan
- drive compressor/gearbox

Air Cycle Machine - A typical installation drawing for the 3-wheel air cycle machine is presented in Figure 6-1. The cast magnesium turbine housing provides

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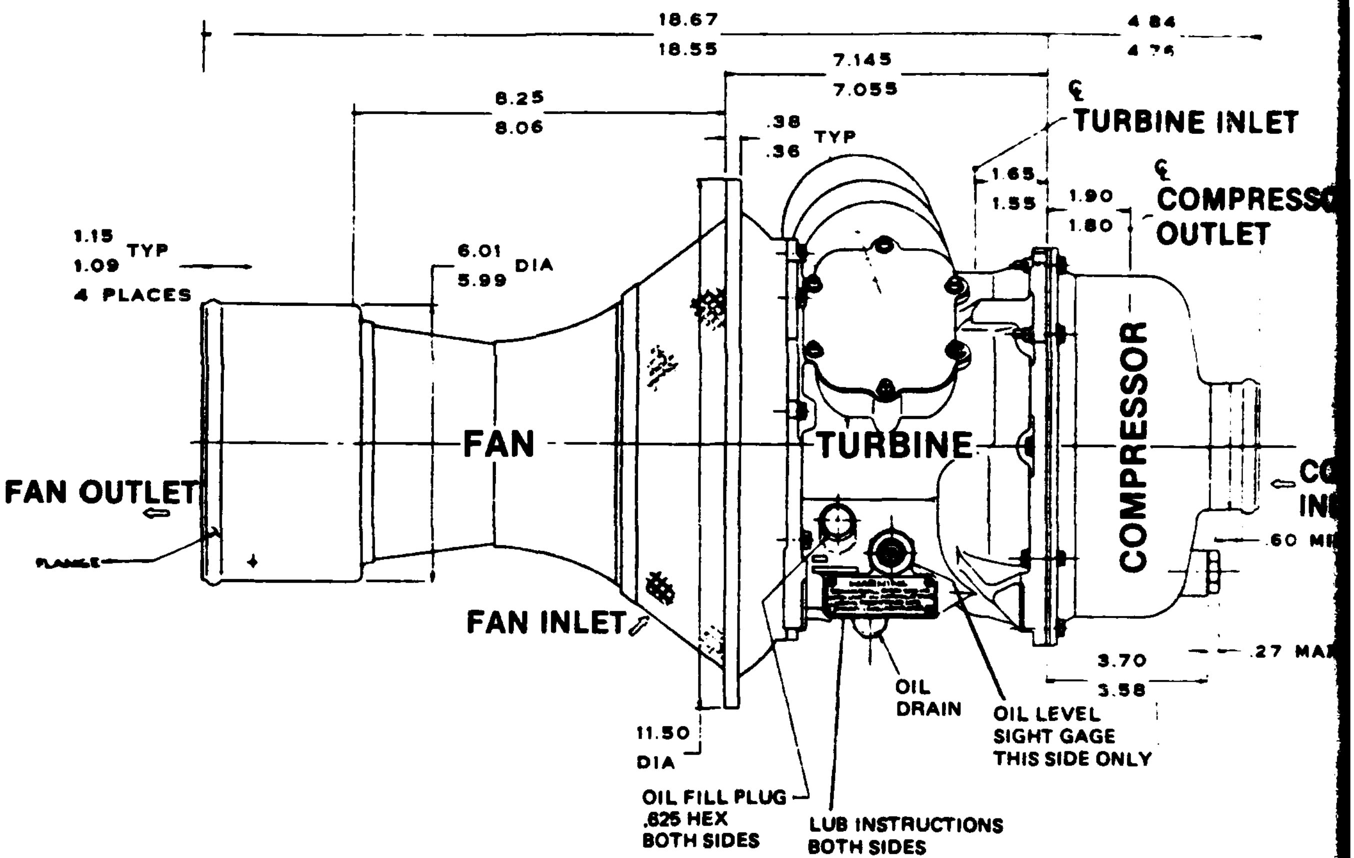

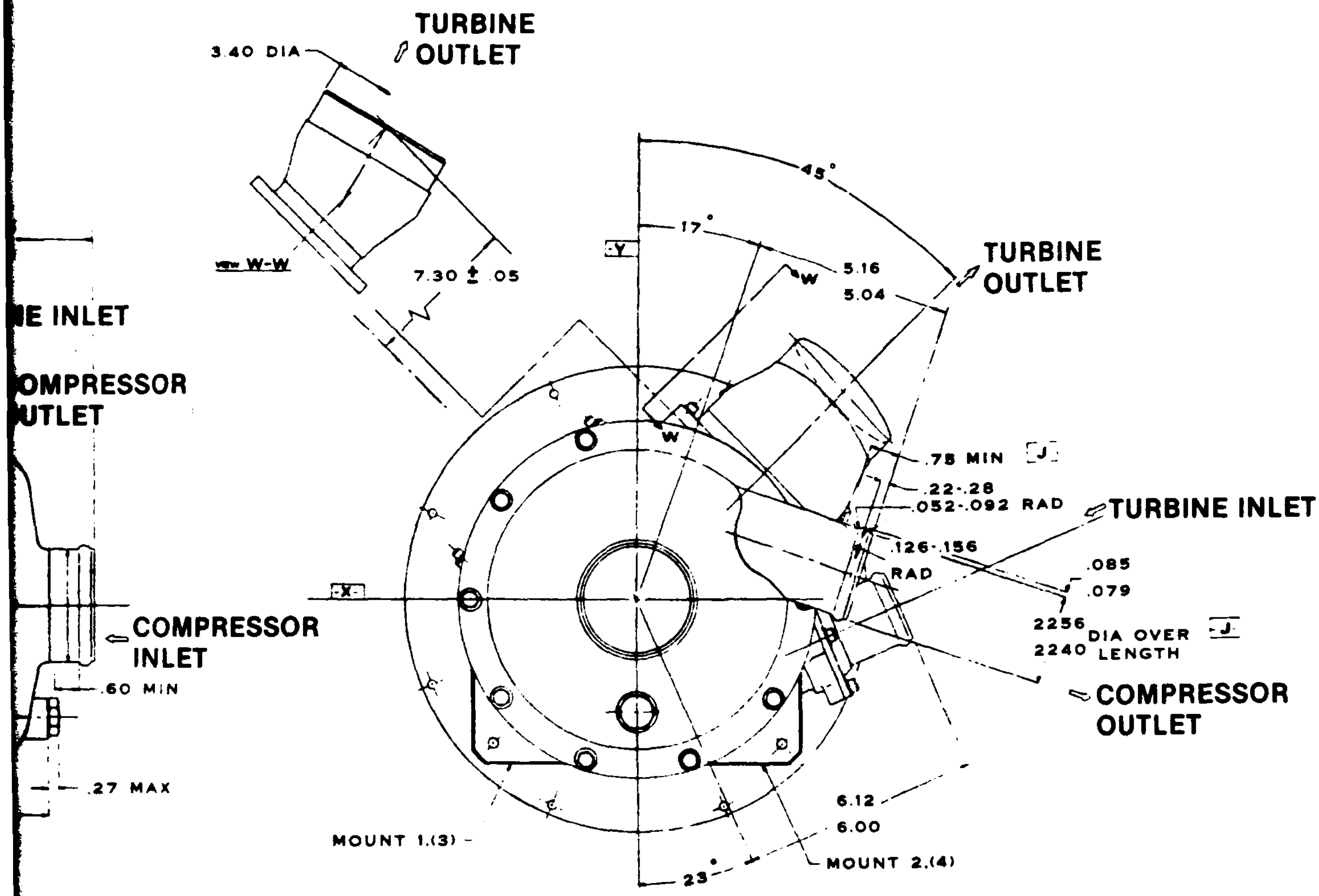

FIGURE 6-1. AIR CYCLE MACHINE INSTALLATION

6.1 (Continued)

support for the bearing cartridge, compressor and fan housings, and can also serve as the main structural member for mounting the unit. The turbine inlet torus and discharge plenum are part of the turbine housing.

A simple dome shaped compressor housing is constructed of high strength aluminum and provides an inlet segment as well as an outlet collector plenum for compressor flow. The fan inlet housing is cast aluminum and incorporates an internal steel section for fan rotor containment and a screen at the inlet to prevent foreign object damage to the fan rotor. The aluminum sheet metal fan diffuser is supported directly from the fan inlet housing. Radial turbine and compressor rotors are machined from high strength wrought aluminum. The axial flow fan rotor is a precision forged wrought aluminum component.

All rotors are fully contained to provide safe operation for normal and possible failure modes. The air cycle machine incorporates a speed limiting device in the form of a machined groove in the turbine rotor, i.e., a turbine "fuse". This fuse section causes the rim to separate from the hub within a defined speed range which is well above the maximum normal operating speed. Since the turbine is the driving member, neither the fan nor compressor rotor speeds can ever exceed the "fuse" burst speed of the turbine. The housings are designed to contain each of the three rotors at the maximum "fuse" speed.

Primary/Secondary Heat Exchanger - The primary/secondary heat exchanger presented in Figure 6-2 is of aluminum plate-and-fin type construction. Its core is an aluminum, crossflow, plate and fin design consisting of a primary and secondary section brazed together as a single unit utilizing Hamilton Standard's fluxless, furnace brazing process. Both the primary and secondary sections are identical in fin configuration. Air from the drive compressor makes a

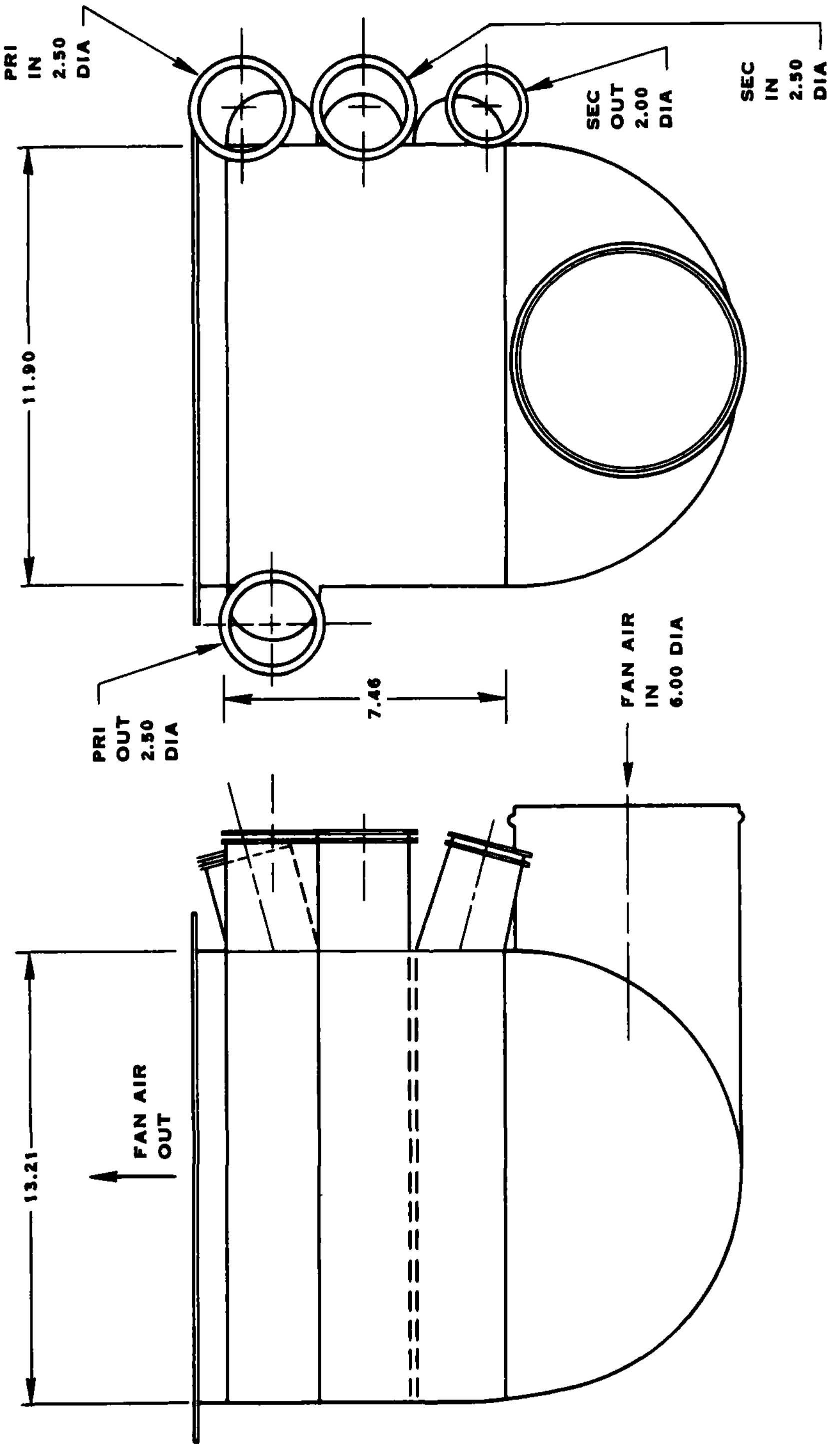

FIGURE 6-2. PRIMARY/SECONDARY HEAT EXCHANGER INSTALLATION

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6.1 (Continued)

single pass through the primary section and two passes through the secondary section. Fan air makes a single pass through the core flowing in series first through the secondary and then through the primary section. The core flow pattern and fin configuration is the same as that successfully employed in the DHC-7 refrigeration package heat exchanger; however, the core size has been modified for this specific application and will be slightly larger. The core sizes are:

Hot Flow Length, inches	11.9
Cold Flow Length, inches	7.46
No Flow Length, inches	13.21

A boss is provided in the fan air inlet header for installation of the spray nozzle, which sprays water drained from the water collector section of the condenser/mixer onto the fan air inlet face of the secondary heat exchanger. Evaporation of this water in the core provides additional cooling of the crew compartment air. For maximum effect, the water is sprayed on the conditioned air outlet side of the fan air inlet to increase the temperature differential between conditioned and fan air in that region.

Condenser/Mixer - The condenser and mixer are designed for ice-free operation and minimum maintenance. The condenser/mixer consists of three sections: a condenser, a mixer and a water collector fabricated as a single unit in order to minimize volume and weight, and to improve operating efficiency. Installation of the condenser/mixer and water collector is shown in Figure 6-3.

The condenser is an aluminum, crossflow, plate-and-fin design utilizing a brazed core, manufactured via Hamilton Standard's fluxless, furnace, braze process. Hot air from the secondary heat exchanger and cold turbine discharge

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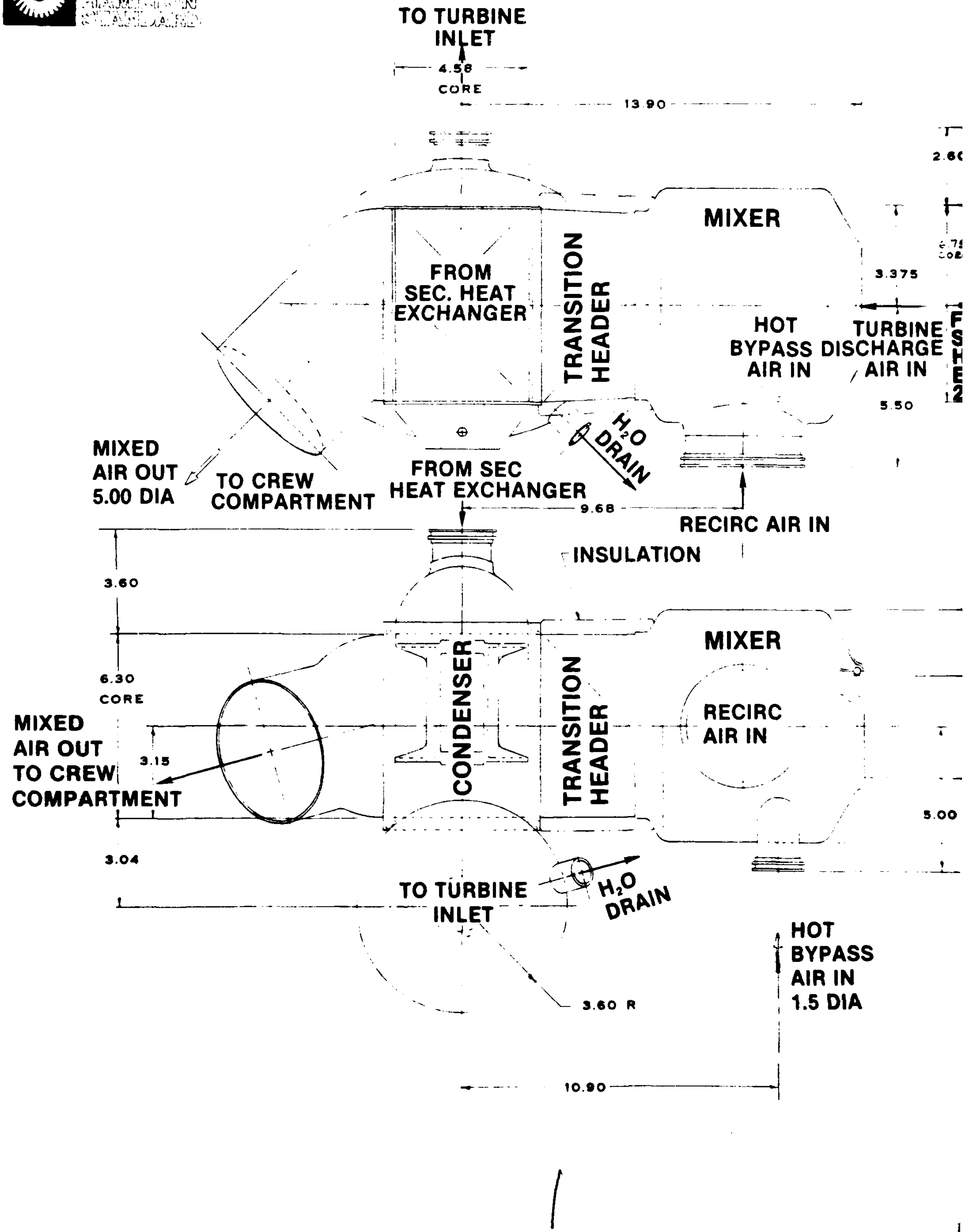

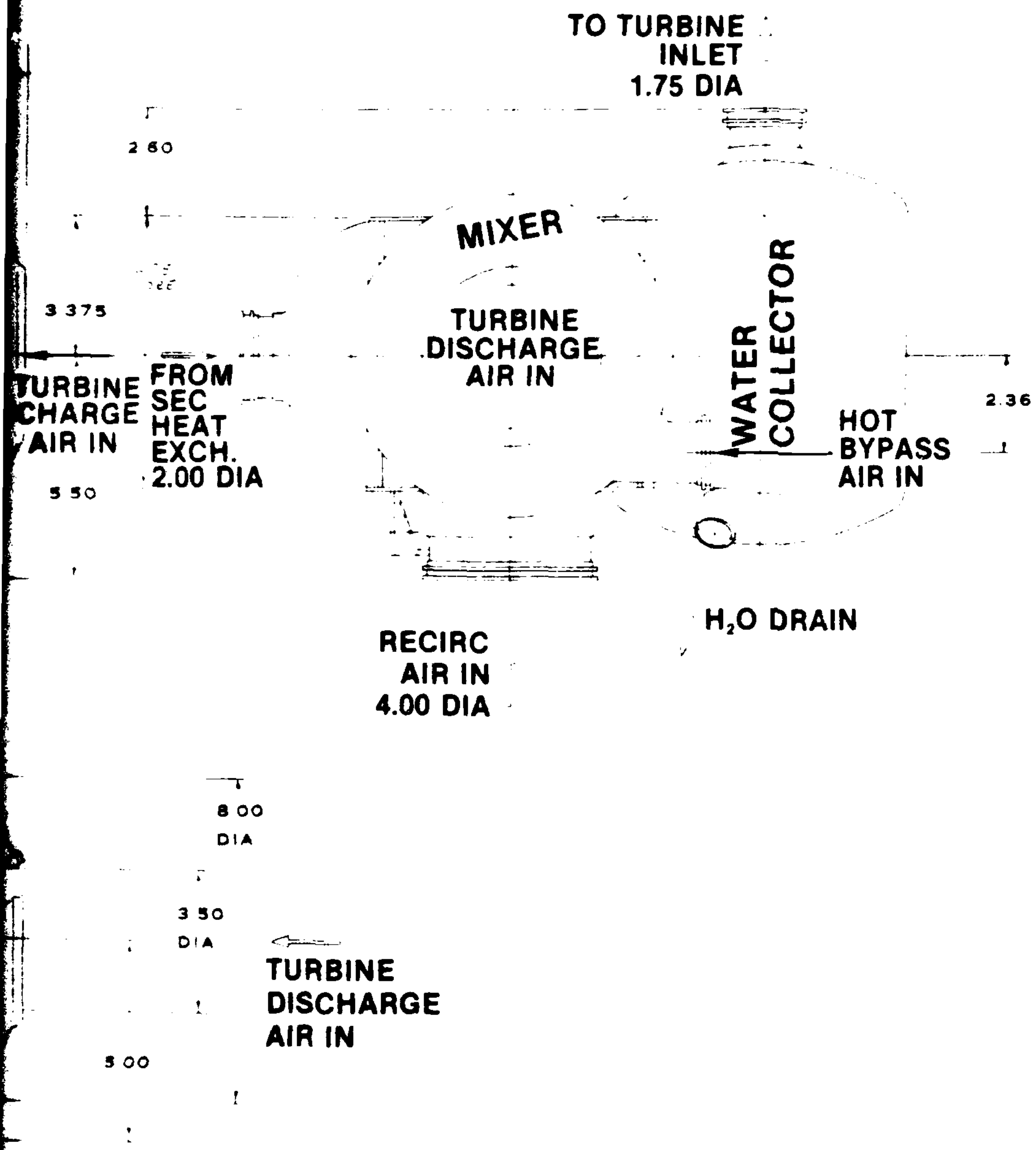

FIGURE 6-3. CONDENSER/MIXER INSTALLATION

6.1 (Continued)

air each make a single pass through the core. The cold air flows between straight fins with a low fins per inch density. This configuration is used to minimize the cold flow pressure drop and, thereby, reduce the recirculation fan horsepower requirement, and to minimize the frontal area on which snow generated by the air conditioning system may accumulate.

The mixer consists of two concentric, aluminum, cylindrical shells welded to the condenser cold side inlet transition header. Recirculated air, turbine discharge air and hot thermal bypass air are mixed to provide a more uniform temperature entering the condenser core and to minimize the formation of snow. The outer shell contains the recirculation air and bypass air inlet ducts. These inlets supply air to an annular area created by the inner and outer shells where the inner shell forms the mixing chamber. Mixing of warm and cold air is initiated close to the turbine discharge inlet. A series of distribution holes around the periphery of the conical inner shell allows penetration of warm recirculation air through the cold turbine discharge plume along the length of the mixer. As the mixed air is diffused in the mixer conical section, mass heat transport takes place between the mixed air molecules at lower velocities. This results in an evenly mixed bulk air temperature prior to its entering the condenser core.

The coaxial mixer is welded to the condenser transition header to integrate both functions of mixing and condensation into a single lightweight component. The external surface of the mixing chamber is covered with one-half inch thick insulating material to preclude the possibility of condensing water forming on the cold walls during normal operation.

6.1 (Continued)

In this system, water is condensed out in some instances in both the secondary heat exchanger and the condenser. For convenience, it is desirable to collect this water at a single point rather than at two. Collecting the water directly in the condenser high pressure outlet header before it can become re-entrained in the exiting air will provide optimum collection efficiency. This is accomplished by an in-line scupper and drain arrangement in the high pressure outlet header.

Recirculation Fan - The recirculation fan installation is presented in Figure 6-4. The crew compartment air recirculating fan is a highly efficient centrifugal fan, utilizing backward swept blades. Its rotor is backward swept to minimize blade loading and maximize performance, and is of low speed design to minimize noise. Pressure recovery is achieved by a short radial diffuser and a constant velocity scroll diffuser. The fan is powered by a basic AC motor with an inverter to utilize a 28 vdc power supply, while improving the life that could be obtained with a DC motor which would utilize brushes.

Drive Compressor and Gearbox - The installation of the drive compressor and gearbox is presented in Figure 6-5, depicting an approximate envelope for shaft drive, belt drive or hydraulic motor power sources.

The drive compressor wheel size is based on an earlier Hamilton Standard design for an aircraft supercharger. Because of the drive compressor pressure and temperature levels, the compressor wheel is titanium and the housing is stainless steel. The gearbox would be of a new design with an aluminum housing and steel gears.

System - Typical package installation drawings for earlier aircraft ECS applications without the drive compressor are presented in Figures 6-6 and 6-7. The

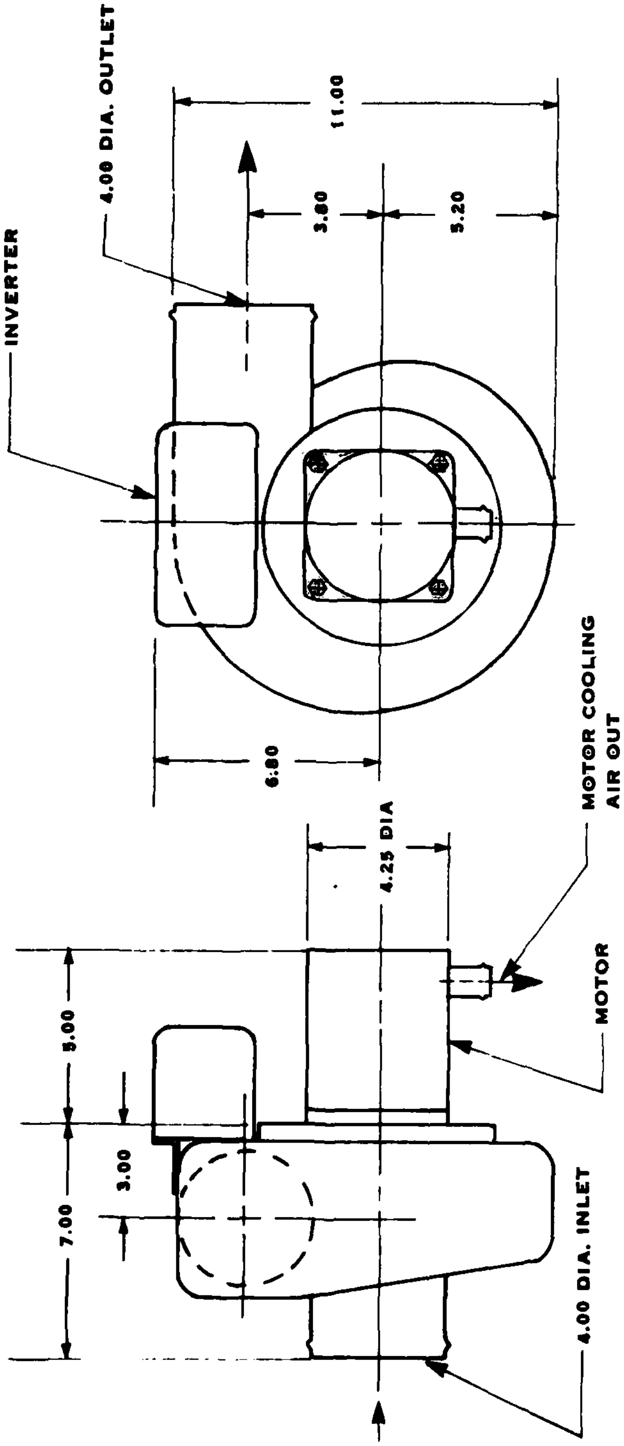

FIGURE 6-4. RECIRCULATION FAN INSTALLATION

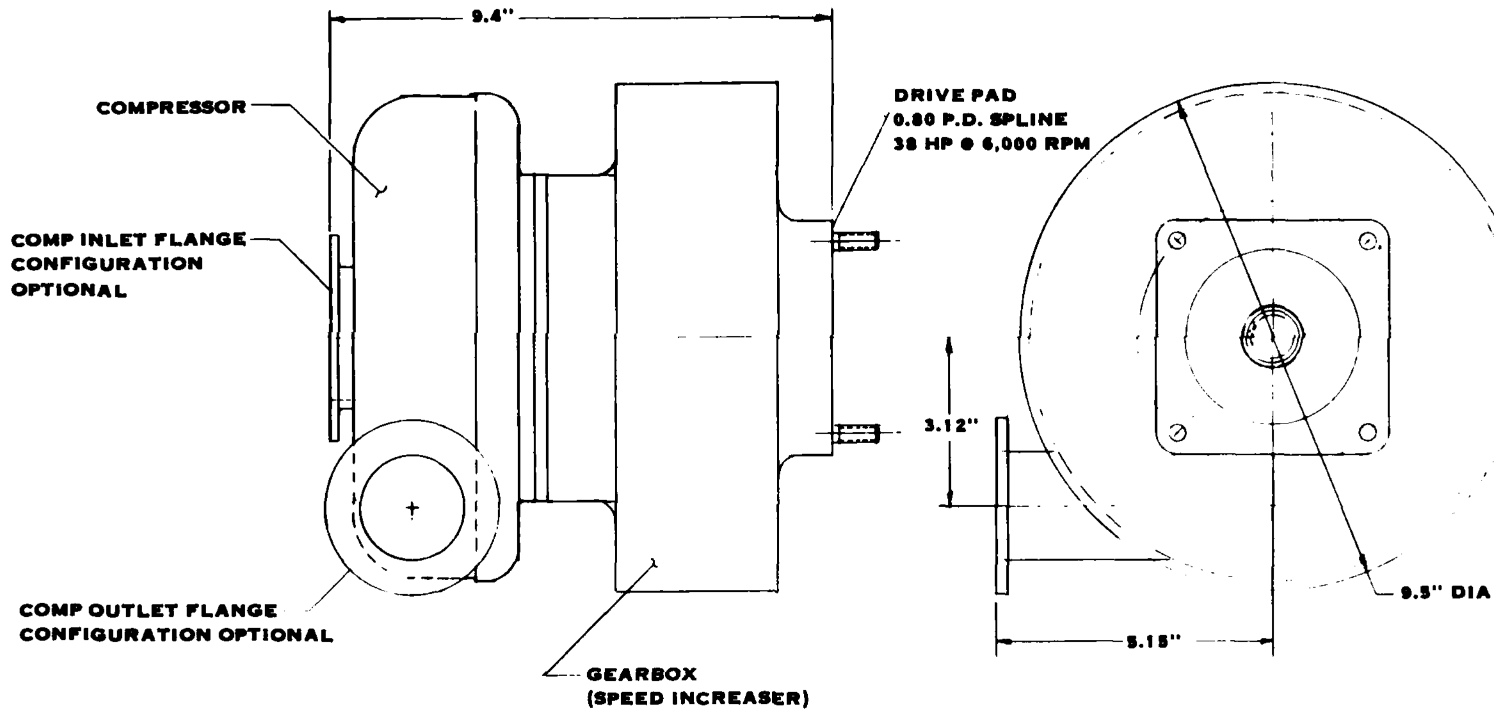

FIGURE 6-5. DRIVE COMPRESSOR AND GEARBOX INSTALLATION

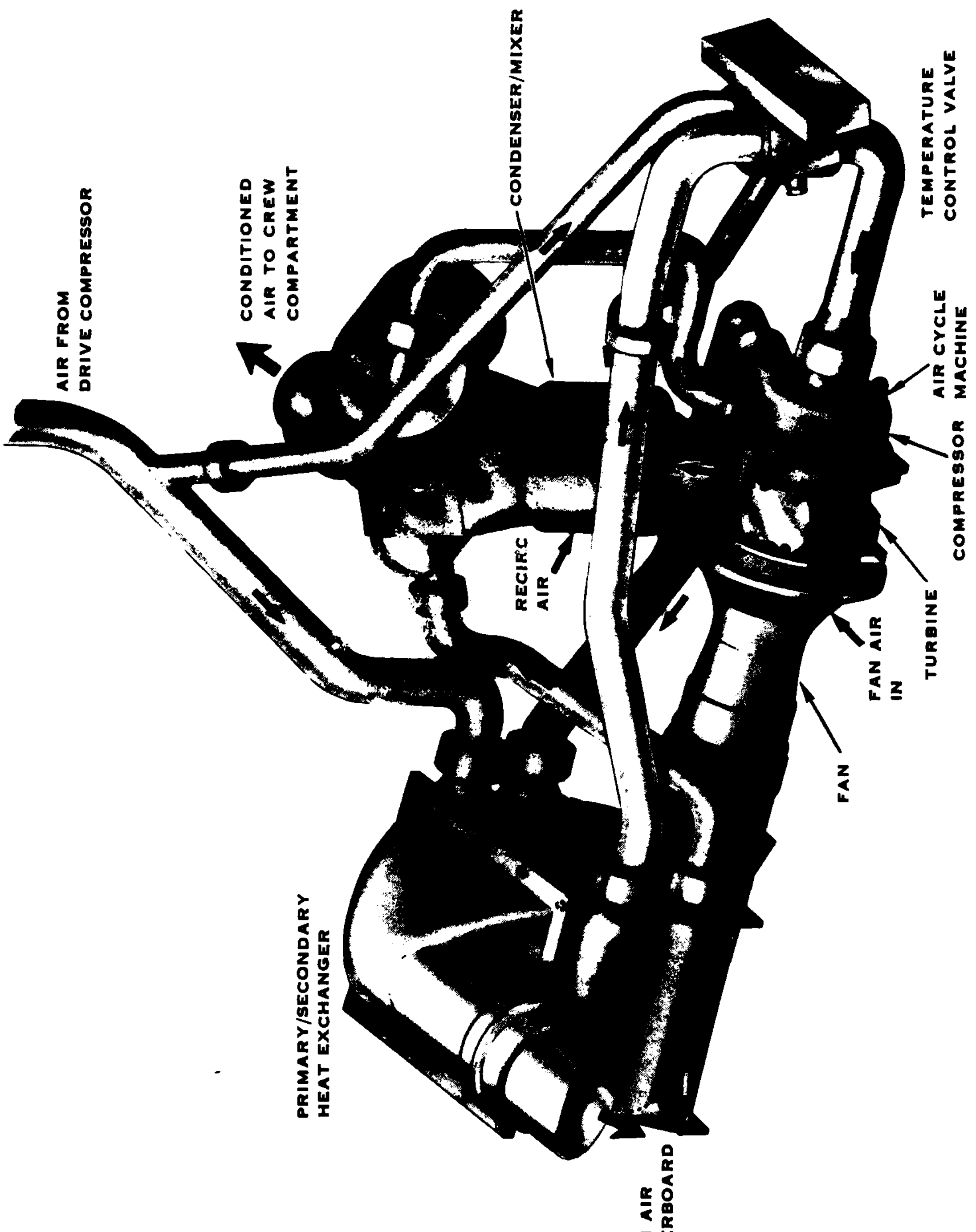

FIGURE 6-6. TYPICAL PACKAGE INSTALLATION

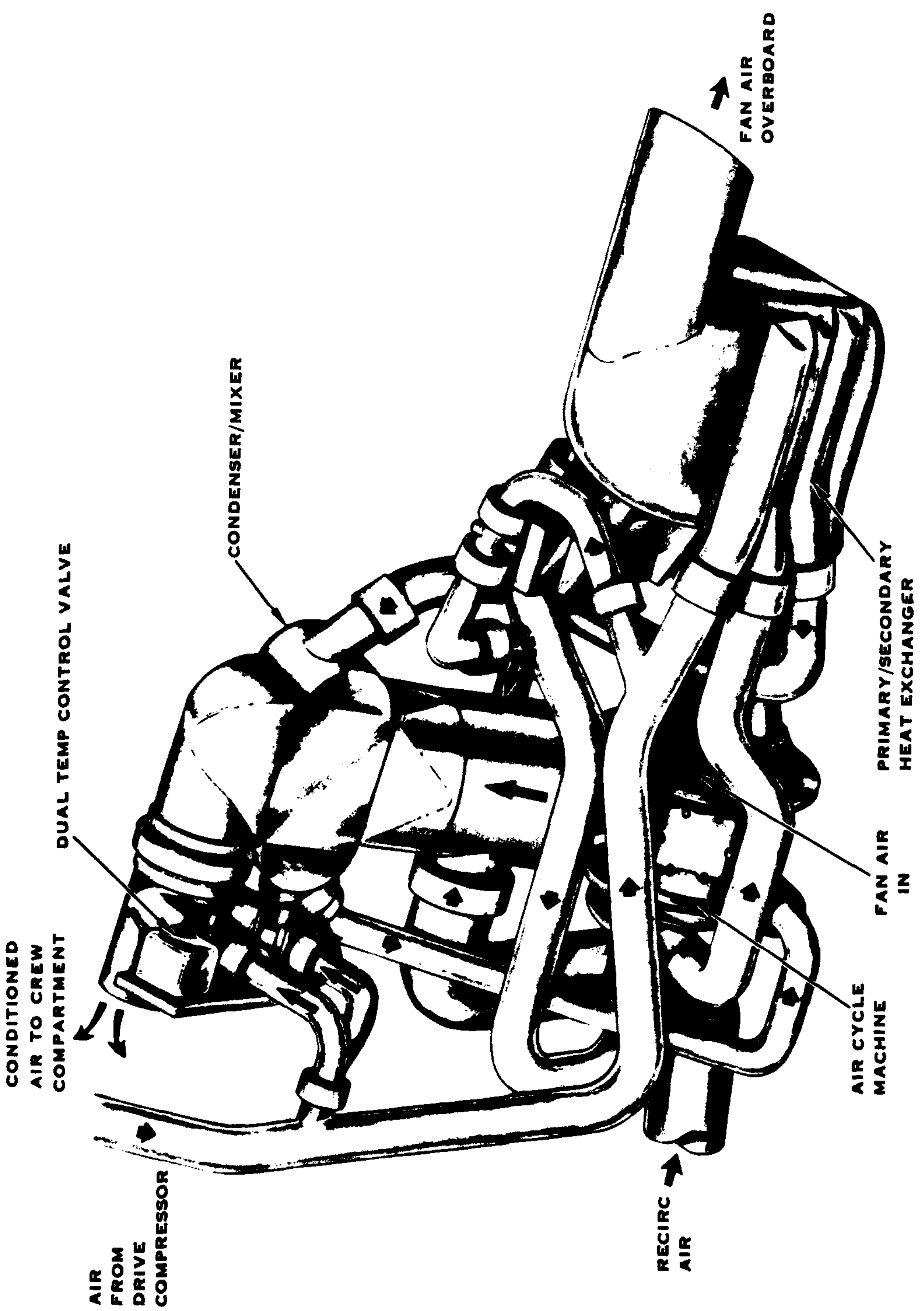

FIGURE 6-7. TYPICAL PACKAGE INSTALLATION

6.1 (Continued)

approximate total system envelope is 16 ft^3 (4 feet long by 2 feet wide by 2 feet high) for these aircraft applications. These envelopes have been developed for minimum weight rather than minimum volume. If connecting duct work were to consist of box-shaped plenums rather than smooth curving, small diameter ducting, the system envelope could be reduced with an increase in system weight. With a detailed system packaging development study, it is anticipated that this volume could be significantly reduced from 16 ft^3.

The overall system weight is estimated to be 125 pounds, based on the following component weights:

air cycle machine	21.4	pounds
primary/secondary heat exchanger	33.0	pounds
condenser/mixer	9.0	pounds
recirculation fan	12.0	pounds
drive compressor and gearbox	28.0	pounds
frame, ducting, and controls	21.6	pounds
Total	125.0	pounds

6.2 VAPOR CYCLE

The vapor compression system depicted schematically in Figure 5-18 uses R12 as the refrigerant and is a slightly modified version of the MBT conditioning system developed by Harrison Radiator (References 9, 10). The system employs dual refrigerant loops consisting of dual compressors, condensers, condenser fans and evaporators to provide some system redundancy in the event of damage. The major components of the system when packaged modularly are a single recirculating air module and dual compressor/condenser modules.

6.2 (Continued)

Recirculating Air Module - The recirculating air module composed of dual evaporators, expansion valves, and crew compartment recirculating air fans is presented in Figure 6-8. The module provides for mixing of the recirculated crew compartment air with fresh make-up air from the HCPE prior to being cooled by the evaporators.

The two evaporators were sized by Hamilton Standard to provide 2 tons of cooling each in the hot dry climate. Each evaporator is a tubular fin design with horizontal copper Freon tubes with aluminum fins. There are 12 fins per inch and each evaporator has an air flow face area of 1 ft^2.

Thermostatic expansion valves constructed of brass with a 2.5 ton capacity are used to regulate the flow of refrigerant into the evaporator under changing conditions. This valve is controlled by the amount of superheat in the suction gas. It protects the unit from flooding under periods of low loads by closing the valve as the amount of superheat decreases. It also limits the maximum suction pressure at high loads as all the valve fluid evaporates and limits the valve opening to a design maximum. For this application, 66 psia is the desired design maximum compressor suction pressure.

Dual vane axial air circulating fans are mounted between the HCPE interface and the evaporator inlet. The envelope has been lengthened from the MBT configuration to aid the air flow distribution over the dual evaporators with the recirculating fans mounted upstream of the evaporators. Both recirculating fans are operated even if only one evaporator is being used. Actual flow characteristics would have to be determined by system testing of the module. However, the evaporator design has a desired low air flow pressure drop of 0.2" H_2O. As an alternative, system tuning of the recirculating air and HCPE interface pressure would be possible with a manually adjustable damper plate where

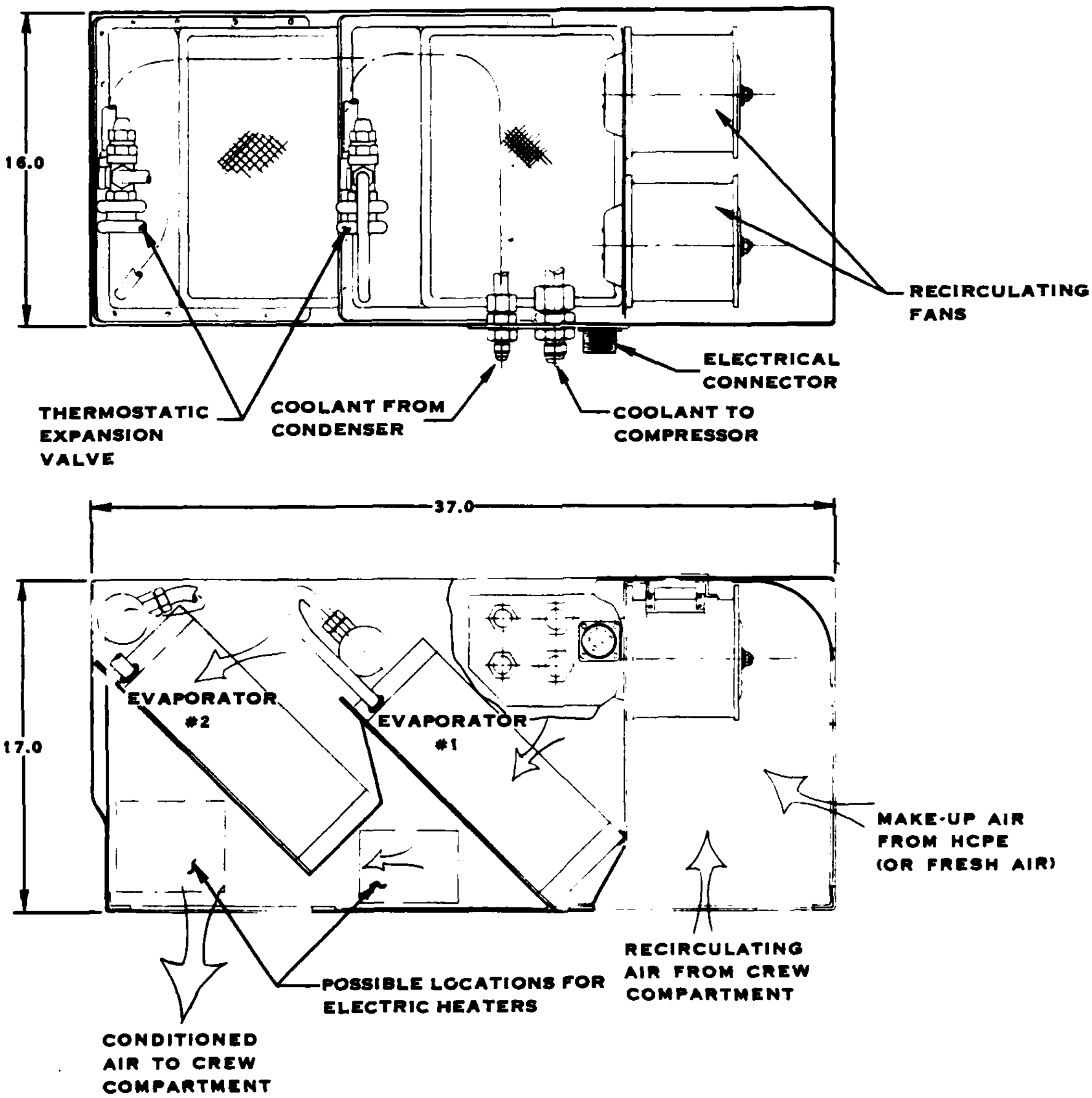

FIGURE 6-8. RECIRCULATING AIR MODULE

6.2 (Continued)

the HCPE make-up air flow enters the recirculating air module. The module also provides some space to add heaters if desired. These areas are so designated on Figure 6-8.

Compressor-Condenser Modules - The dual compressor-condenser modules from the Main Battle Tank development were not modified for this study. The envelope for each module is presented in Figure 6-9.

The compressor is a six-cylinder axial compressor used for automotive applications. Modifications to the automotive design were made to ensure lubrication while the vehicle is on slopes and to modify the drive coupling. The motor was of a special design and is a compound wound totally enclosed unit with a direct drive. An aluminum adapter and a cast iron and neoprene coupling are used to align the compressor and motor shafts and to transfer the power. Another alternative would be to use a hermetically sealed motor-compressor unit so as to eliminate all possible compressor shaft leakage. However, this alternative should not have a major impact in the approximate envelope presented in Figure 6-9. The system capacity is controlled by cycling the compressor, as determined by the thermostat on the recirculating crew compartment air.

The condenser is an aluminum plate and fin design with a 227 sq. in. face area. Cooling air is drawn through the unit with an aluminum vane-axial fan with a 12.75 inch impeller diameter and a DC motor. Proper air flow paths are maintained with an .060 inch thick aluminum shroud between the condenser and the fan.

The compressor-condenser module also contains a modified automotive-type receiver-dehydrator. This aluminum component provides some refrigerant storage, a refrigerant sightglass, a moisture indicator, a desiccant to absorb any residual moisture in the refrigerant, and a screen to filter the liquid.

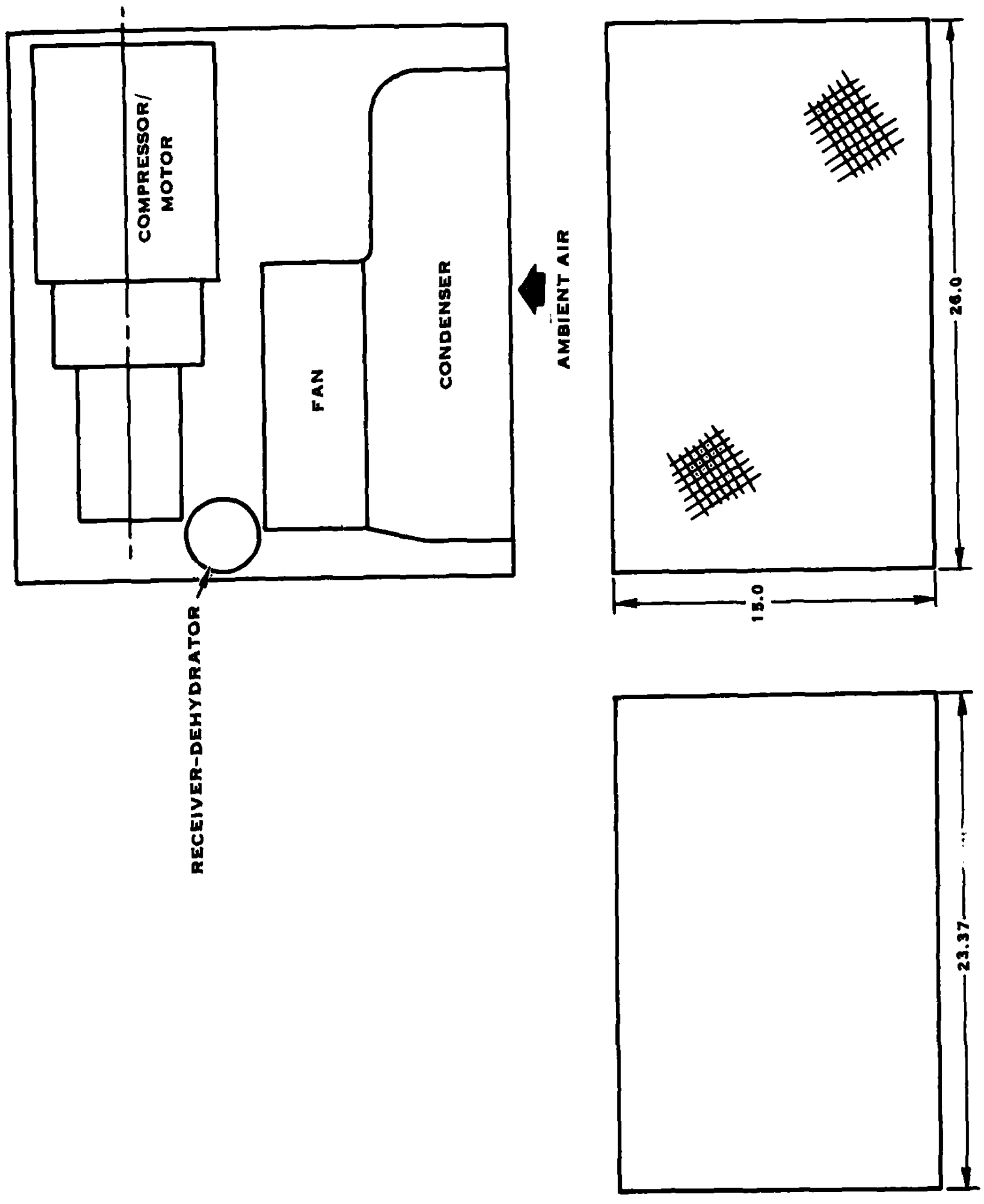

FIGURE 6-9. COMPRESSOR/CONDENSER MODULE

6.2 (Continued)

System - The complete system is estimated to require a volume of 16.6 ft^3. This allows 5.4 ft^3 for each condenser module and 5.8 ft^3 for the recirculating air module.

The total system weight is approximately 683 pounds. Each compressor-condenser module weighs 264 pounds, with 110 pounds of this being the compressor motor. The recirculating air module weighs 130 pounds without heaters and 150 pounds with heaters. An additional 25 pounds is allotted for controls, ducting, etc.

SECTION 7.0
TRADE-OFFS

INTRODUCTION

This section presents a detailed comparison of both the air and vapor cycle refrigeration system designs. The trade-off criteria and their relative weighting are summarized in Table 7-I. As performance/power and size are the

TABLE 7-I
TRADE-OFF CRITERIA

ITEM	ASSIGNED POINTS
Performance/Power	25
Size	25
Weight	10
Reliability	10
Logistics	10
Cost	10
Noise, Safety, Simplicity, Maintainability, Other	10
Maximum Total	100

most critical items, they have been assigned the highest importance. Weight, reliability, logistics, and cost have been assigned a secondary level of importance. Noise, safety, simplicity, maintainability, and other items have been given the lowest relative importance and are grouped together into a single category. Each of the candidate concepts is reviewed independently and can be awarded from 0 to the maximum number of points in each category. Where there is a significant advantage for one concept, it has been awarded the maximum number of points and the other concept has been assigned 0 points.

7.0 (Continued)

Where there is no distinct advantage, an attempt has been made to award proportionate points. Each of these trade-off issues and their scoring will be discussed in detail below.

7.1 PERFORMANCE/POWER

In this category worth 25 points, 15 points were assigned to power and 1 point each was assigned for system performance in each of the 10 cooling environments studied. As discussed previously in the "Comparative System Performance" section, the vapor compression system has fewer sources of process inefficiencies than an equivalent air cycle and, thus, has lower power requirements. A comparison of the total system power requirements for the studied cases based on 100% motor efficiency is presented in Table 7-II for make-up flow rates of both 300 and 600 cfm. A motor efficiency of 100% was used to compare the system power demands because the actual methods of powering the system are unknown. Furthermore, motor efficiency does not have an impact on the relative power needs of the two concepts. Disregarding the severe cold climate, the power requirements of the vapor compression cycle are approximately 36% of the power requirements of the air cycle. This lower power requirement is a major advantage of the vapor compression system.

The cooling capacity of the candidate systems at off-design conditions must also be considered in any performance comparison. Table 7-III presents a summary of crew compartment dry bulb temperature, effective temperature, and relative humidity for both systems with 300 cfm HCPE flow in an NBC environment. The air cycle always has the cooling capacity to meet a 90°F crew compartment dry bulb temperature, whereas the vapor cycle system exceeds this target in a hot-humid climate. The air cycle has also good moisture

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TABLE 7-II

POWER REQUIREMENTS (HP) [1]

CLIMATE	AIR CYCLE		VAPOR COMPRESSION CYCLE	
	300 CFM	600 CFM	300 CFM	600 CFM
Hot-Dry	37.51	38.81	13.38	14.48
Hot-Humid	38.88	40.17	13.88	14.18
Basic Constant High Humidity	30.24	34.16	6.81 [2]	11.88
Basic Variable High Humidity	38.23	41.42	12.98	14.28
Basic Hot	38.71	40.00	12.58	12.68
Severe Cold	47.02	59.98	N/A	N/A

NOTE: (1) Power requirements are based on motor efficiencies of 100%.

(2) Only half of system is operated at this condition.

TABLE 7-III

NBC PERFORMANCE COMPARISON

CLIMATE	AIR CYCLE			VAPOR COMPRESSION CYCLE		
	$T_{crew\ comp}$		RH	$T_{crew\ comp}$		RH
	Dry Bulb	Effective		Dry Bulb	Effective	
Hot-Dry	90.0	76.0	15	90.0	76.0	15
Hot-Humid	82.6	74.2	37	92.2	81.1	39
Basic Constant High Humidity	61.7	61.0	80	72.4 [1]	69.5 [1]	70 [1]
Basic Variable High Humidity	75.9	70.0	40	85.5	76.5	39
Basic Hot	82.5	73.2	28	84.4	74.6	30
Severe Cold	60	56.6	11	N/A	N/A	N/A

NOTE: (1) Only half of the system is operated at this condition.

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7.1 (Continued)

removal characteristics. This combination of cooling capacity and water removal results in equivalent or lower crew compartment effective temperatures with the air cycle system in climates requiring cooling in an NBC environment.

Table 7-IV presents a similar summary for operation in a non-NBC environment with 600 cfm HCPE flow. The air cycle has lower crew compartment dry

TABLE 7-IV
NON-NBC PERFORMANCE COMPARISON

CLIMATE	AIR CYCLE			VAPOR COMPRESSION CYCLE		
	$T_{crew\ comp}$		RH	$T_{crew\ comp}$		RH
	Dry Bulb	Effective		Dry Bulb	Effective	
Hot-dry	100.4	80.0	9	95.0	78.0	10
Hot-Humid	90.3	83.2	61	102.0	89.0	48
Basic Constant High Humidity	63.9	63.9	100	66.2	64.0	61
Basic Variable High Humidity	80.0	76.7	74	92.0	82.0	46
Basic Hot	92.0	78.7	24	87.8	77.2	32
Severe Cold	29.8	-	27	N/A	N/A	N/A

bulb and effective temperatures where moisture removal is more critical; i.e., the hot-humid, basic constant high humidity, and basic variable high humidity climates. Where entrained moisture is not as prevalent and the ambient temperature is high (hot-dry and basic hot climates), the vapor cycle has lower dry bulb and effective temperatures. This is a result of the flow capacity of the air cycle drive compressor necessitating the direct bypass of a large proportion of the hot fresh make-up air. In the vapor compression system, the

7.1 (Continued)

recirculating fan handles more than 600 cfm, so all the HCPE flow will pass through the evaporator. While the fresh air bypass in the air cycle system always passes a high proportion of the HCPE flow in a non-NBC environment, the effect is more pronounced on the hot-dry and basic hot days because of (1) the high temperatures out of the HCPE when compared to the other climates and (2) the low ambient humidities of these two climates which minimize the water removal advantages of the air cycle.

In summary, the vapor cycle has a clear-cut advantage in power, while the air cycle has superior performance in at least 6 of the 9 off-design conditions studied (the NBC mode, basic constant high humidity climate was considered equal because only one of the duplex Freon systems was operating). As power is considered more critical, 18 points were assigned to the vapor cycle and 7 points to the air cycle.

7.2 SIZE

The estimated package size for the air cycle system in an aircraft application is 16 ft^3 (4 feet long X 2 feet high X 2 feet wide). As discussed previously, this air cycle system size should be significantly reduced after a development packaging program stressing volume rather than weight. The estimated package size for the vapor compression system, based on previously developed combat vehicle units, is 16.6 ft^3.

Both systems are equally suited for modularization. The vapor cycle evaporators, compressors and condensers can be packaged together or subdivided. The air cycle's major components, consisting of air cycle machine, dual heat exchanger, condenser/mixer, recirculation fan, and drive compressor and gearbox, can be mounted in available spaces with interconnecting ducting.

7.2 (Continued)

Because of the anticipated reduction in air cycle volume, the air cycle is awarded 20 points, while the vapor cycle was assigned 15 points for reasonable envelope and modularization.

7.3 WEIGHT

Low weight is a major advantage of the air cycle system. The smaller component sizes result in lower overall system weight. The air cycle system is estimated to weigh 125 pounds, while the vapor cycle system weighs approximately 460 pounds on a comparable basis without heaters or drive compressor motors. The full 10 points assigned to this category were given to the air cycle system.

7.4 RELIABILITY

Both the air and vapor cycle systems are considered reliable and comparable. The vapor cycle is a completely developed concept that has been used for many years in home and automobile applications. The air cycle is also a completely developed concept that has been used extensively in aircraft air conditioning applications, utilizing both engine bleed air and shaft driven compressors. Both of these concepts have demonstrated reliability, and both are assigned the full 10 points in this scoring category.

7.5 LOGISTICS

The logistics requirements for the air cycle system would be somewhat simpler than for the vapor cycle. The air cycle only requires conditioning components to replace any damaged or failed equipment. In addition to replacement components, the vapor cycle system also has a requirement to maintain a refrigerant charge in the system. This requires a leak detector to determine if refrigerant is escaping from the system and a vacuum pump and make-up

7.5 (Continued)

refrigerant to replace any refrigerant that has leaked out.

For this comparison item of 10 possible points, 8 points were given to the air cycle and 5 points were given to the vapor cycle.

7.6 COST

The cost of the conditioning equipment is an advantage for the vapor cycle system. The commonality of vapor cycle components produced by widespread use of vapor cycle equipment results in a fairly low estimated cost for a commercially-available 4 ton vapor compression refrigeration system.

The comparable air cycle system is more expensive due to amortization of development costs and a lower commonality base of air cycle components. The air cycle system for this application is a modified DHC-8 aircraft air conditioning package that is estimated to have a recurring cost of approximately 6 times that of the vapor cycle. In the cost category, the vapor cycle system is awarded a full 10 points.

7.7 NOISE, SAFETY, SIMPLICITY, MAINTAINABILITY, OTHER

This single category is of the lowest relative importance and is intended to address all other items not discussed previously.

The high speed operating characteristics of the air cycle machine tend to make it noisier than the vapor cycle. However, during vehicle operating maneuvers, this noise should not be annoying to the crew, particularly if the unit is mounted on the exterior of the vehicle. Both the air and vapor cycle systems are sufficiently noisy to require system shutdown during silent watch operations.

Both of these systems are considered safe. The major hazard associated with vapor cycle systems is refrigerant leakage. In this study, R-12 was

7.7 (Continued)

selected because of its commonality, commercial availability, stability, and non-toxicity (R-12 also has desirable vapor pressure characteristics and power requirements). The major hazard associated with the air cycle is the potential breakage of the high speed rotating turbomachinery. However, these problems have been thoroughly studied in our aircraft air conditioning applications and are accounted for in all of our designs by providing housing containment of the "shrapnel". Also, the turbine is "fused" as a speed limiting device to minimize this problem.

In general, the vapor cycle system is simpler than the air cycle system. It has fewer moving parts and does not require the same high degree of accuracy in design and manufacture that high speed, rotating equipment does.

In terms of maintainability, both systems are comparable. If packaged as a single unit, the difficulty of replacing any given part can be the same for either an air or vapor cycle system. If modularized, the maintainability of each system is simplified.

Other general items were also considered. These items include vulnerability and combined heating/cooling capability. The vapor cycle system is vulnerable to minor damage in the form of refrigerant leaks because these leaks will eventually shut the system down. In comparison, minor leakage will not shut the air cycle system down. However, a bullet hole in a high pressure air cycle line, resulting in a hole equivalent in size to the turbine nozzle area, will result in excessive leakage that will impair cooling capability.

The air cycle has a further advantage over the vapor cycle system in that it also has a heating capability. By bypassing hot air from the drive compressor around the heat exchangers and the 3-wheel air cycle machine, warm

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7.7 (Continued)

air can be supplied to the crew compartment on cold days. With the vapor cycle system, electric or liquid heaters must be included to heat the crew compartment.

In this trade-off category, 9 points were given to the air cycle because of its advantages in vulnerability and combined heating/cooling. Six points were given to the vapor cycle because of its advantages in noise and simplicity.

7.8 SUMMMARY

The summary of the trade-off items is presented in Table 7-V. Based on the total points awarded, the vapor cycle and the air cycle are equally applicable to combat vehicle crew compartment conditioning.

TABLE 7-V
TRADE-OFF COMPARISON

ITEM	ASSIGNED POINTS	AIR CYCLE	VAPOR CYCLE
Performance/Power	25	7	18
Size	25	20	15
Weight	10	10	0
Reliability	10	10	10
Logistics	10	8	5
Cost	10	0	10
Noise, Safety, Simplicity, Mantainability, Other	10	9	6
Total Points	100	64	64

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SECTION 8.0
CONCLUSIONS AND RECOMMENDATIONS

INTRODUCTION

This study was intended to review and trade off a variety of candidate combat vehicle conditioning concepts. On the basis of performance, power, size, weight, logistics, cost and development status, the following concepts were eliminated from further consideration: positive displacement air cycle, thermoelectric, vortex tubes, expendable heat sink, absorption cooling and evaporative cooling. The remaining concepts considered, air cycle and vapor compression refrigeration, underwent a preliminary design, system performance analysis and detailed trade study. The conclusions and recommendations resulting from the design, analysis and trade-off are presented below.

8.1 CONCLUSIONS

Both the air and vapor cycles were designed (1) for a 4 ton cooling capacity on a hot-dry day,(2) to provide a target crew compartment dry bulb temperature of 90°F at 20% relative humidity and (3) to interface with the hybrid collective protection equipment. Off-design performance was analyzed in other climates for both the NBC and non-NBC environments. Finally, both concepts were compared on the basis of performance/power, size, weight, reliability, logistics, cost and other factors. The conclusions resulting from these activities are:

1. The vapor cycle has a distinct advantage in both the power and cost categories, requiring approximately 36% of the power needed by the air cycle.
2. The air cycle has a distinct advantage in the categories of performance and weight. In the nine off-design environments requiring cooling, the air cycle will provide a lower crew

8.1 (Continued)

compartment effective temperature in six environments. In addition, the air cycle has a built-in heating capability while the vapor cycle requires liquid or electric heaters. Additionally, the air cycle weighs several hundred pounds less than the vapor cycle.

3. The air cycle has a slight advantage in the categories of size and logistics. An air cycle packaging effort geared to minimizing volume rather than minimizing system weight would result in an air cycle packaging size reduction from the current aircraft applications. The logistics of the air cycle are simpler than the vapor cycle because there is no need for refrigerant charging.
4. On a total comparative point basis, the air and vapor cycles are equivalent. Both are applicable to combat vehicle crew compartment conditioning.

8.2 RECOMMENDATIONS

Because of the comparative equivalency of the trade-offs of the air and vapor cycles, the implementation of either system is dependent on the relative importance of the trade-off categories to the Army. For example, if a specific combat vehicle has sufficient on-board power for operation of an air cycle system, the air cycle would be favored over the vapor cycle because of off-design cooling performance, combined heating capability and size. However, a detailed system packaging study is required to minimize the system size and to effectively package the system within a specific vehicle. This study is required because current aircraft air cycle systems are weight-optimized rather than volume optimized.

8.2 (Continued)

If power availability is of primary importance, a vapor cycle system would be recommended because its power requirements are approximately 36% of the air cycle power requirements. Adequate cooling performance is attained by the vapor cycle when in an NBC protective posture, as the calculated crew compartment effective temperature did not exceed 81.1°F for any of the climates analyzed. (Reference Table 7-III).

The implementation of microclimate (crewman suit) cooling would greatly reduce the power requirements for vehicle cooling systems because of the major reduction in cooling load. For a 4 man crew, microclimate cooling of approximately 4000 Btu/hr (1/3 ton) would be needed compared to the 4-5 tons of cooling needed for a macroclimate (crew compartment) conditioning scheme.

Finally, the retrofit of existing combat vehicles to accommodate hybrid collective protection equipment and either macroclimate or microclimate cooling equipment causes packaging difficulties. To alleviate these packaging difficulties in future vehicle applications, the Army is considering an integrated engine concept called Multi-Purpose, Survivability - Enhancement Life-Support System (MUSELS). With this integrated systems approach, a turbine engine could provide engine prime power, electric power, collective protection, cooling and heating in a space envelope that is probably not much larger than for present prime movers. For an application such as the M1, particulates would be removed by a series of filters. Temperatures resulting from compression, combined with a gas filter, would remove much of the contamination. Cool turbine discharge air could be used directly for crew compartment cooling and leakage air make-up or in an interchanger to cool circulating coolant in a microclimate system. However, one expected negative aspect of this

8.2 (Continued)

turbine engine approach is increased fuel consumption. This approach would be applicable to both vehicle prime power plants and auxiliary power units for shelters, vans and other enclosures.

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