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CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS

TECHNICAL DOCUMENTARY REPORT No. MRL-TDR-62-7

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LIFE SUPPORT SYSTEMS LABORATORY
6570th AEROSPACE MEDICAL RESEARCH LABORATORIES
AEROSPACE MEDICAL DIVISION
AIR FORCE SYSTEMS COMMAND
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

PROJECT No. 6373, TASK No. 637302

(Prepared under Contract AF 33(616)-7768 by
C. C. Wright of AiResearch Manufacturing Co.
A Division of The Garrett Corporation, Los Angeles 45, Calif.)

<p>Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio Rpt. No. NRL-TDR-62-7. CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS. Final report, Feb 1962. 118p. incl illus., tables, 9 refs.</p> <p>Unclassified report</p> <p>An analytical feasibility study was conducted on contaminant freeze-out (water vapor and CO₂) systems employing the endothermic conversion of liquid to gaseous oxygen as a heat sink. Simple freeze-out systems, combination freeze-out and adsorption systems, and systems with auxiliary cooling were studied and compared. For minimum</p> <p style="text-align: right;">(over)</p>	<p style="text-align: center;">UNCLASSIFIED</p> <p>I. Space Biophysics, Closed-Cycle Ecological Systems</p> <p>I. AFSC Project 6373, Task 637302</p> <p>II. Contract AF 33(616)-7768</p> <p>III. AIR Research Manufacturing Co., Los Angeles 45, Calif.</p> <p>IV. Wright, C. C.</p> <p>V. Secondary Rpt No. SS-596-R, Rev. I</p> <p style="text-align: center;">UNCLASSIFIED</p>	<p style="text-align: center;">UNCLASSIFIED</p> <p>Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio Rpt. No. NRL-TDR-62-7. CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS. Final report, Feb. 1962. 118p. incl illus., tables, 9 refs.</p> <p>Unclassified report</p> <p>An analytical feasibility study was conducted on contaminant freeze-out (water vapor and CO₂) systems employing the endothermic conversion of liquid to gaseous oxygen as a heat sink. Simple freeze-out systems, combination freeze-out and adsorption systems, and systems with auxiliary cooling were studied and compared. For minimum</p> <p style="text-align: right;">(over)</p>	<p style="text-align: center;">UNCLASSIFIED</p> <p>I. Space Biophysics, Closed-Cycle Ecological Systems</p> <p>I. AFSC Project 6373, Task 637302</p> <p>II. Contract AF 33(616)-7768</p> <p>III. AIR Research Manufacturing Co., Los Angeles 45, Calif.</p> <p>IV. Wright, C. C.</p> <p>V. Secondary Rpt No. SS-596-R, Rev. I</p> <p style="text-align: center;">UNCLASSIFIED</p>
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FOREWORD

This report is a summary of all effort which was carried out by the AiResearch Manufacturing Company, a Division of The Garrett Corporation, Los Angeles 45, California, on a Contaminant Freeze-Out Study for Closed Respiratory Systems. The work was performed under Air Force Contract AF 33(616)-7768, Project No. 6373, "Equipment for Life Support in Aerospace," Task No. 637302, "Respiratory Support Equipment." Mr. Clyde G. Roach, Chief of the Respiratory Equipment Section, Sustenance Branch, Life Support Systems Laboratory, 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio, was the project monitor.

The report was prepared by Mr. C. C. Wright of the AiResearch Manufacturing Company. The study was entirely analytical in nature and was performed during the period from January 1961 to August 1961. The AiResearch report number is SS-596-R, Rev. 1.

ABSTRACT

The purpose of this program was to conduct an analytical feasibility study on contaminant freeze-out (water vapor and carbon dioxide) systems, which employ the endothermic conversion of liquid oxygen to gaseous oxygen as a heat sink. Simple freeze-out systems, combination freeze-out and adsorption systems, and systems with auxiliary cooling were studied and compared.

From the point of view of minimum weight and minimum number of components, the simple freeze-out system with water removal (dumped overboard as a vapor to a vacuum) appeared to be the best system. A desirable feature of this system is that the liquid oxygen requirements are low (about equal to the normal metabolic oxygen consumption rate). A possible undesirable feature is that the frozen out water is not recovered. The system uses a four-channel regenerative switching heat exchanger in which cooling with freeze-out, regeneration of the effluent air, and sublimation of the frozen out contaminants occur simultaneously.

The final portion of the study consists of the preliminary design of a model version of the simple freeze-out system with water removal. Recommendations as to the amount and type of additional effort required to prove feasibility and to reduce the design approach to hardware practice are also given.

PUBLICATION REVIEW

This technical documentary report has been reviewed and is approved.

Wayne H. McCandless
WAYNE H. McCANDLESS
Chief, Life Support Systems Laboratory

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FOR CLOSED RESPIRATORY SYSTEMS

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LIST OF SYMBOLS

English Letter Symbols

- A** = Heat transfer area, ft^2
- A_c** = Free flow area, ft^2
- A_f** = Face area, ft^2
- B** = Concentration of adsorbate in an adsorber bed at saturation, lb adsorbate per lb adsorber
- b** = Fin height, in.
- C** = Constant in the Equation for pressure drop in adsorber beds (See Eqn IV-22)
- c_p** = Specific heat at constant pressure, Btu per lb °R
- D** = Diameter, in.
- E** = $\Delta T_h / \Delta T_o$ = Temperature effectiveness of a heat exchanger
- f** = Fanning friction factor, dimensionless. This factor vs Reynolds number defines the flow friction characteristics of a heat transfer surface (See Figure 24)
- G** = w/A_c = Mass velocity, lb per sec ft^2 unless otherwise noted
- g** = Inertia proportionality factor, $\text{lb}_m \text{ft}$ per $\text{lb}_f \text{ft}^2$
- H** = Humidity, lb water per lb dry air
- h** = Heat transfer coefficient, Btu per sec ft^2 °R; or enthalpy, Btu per lb_m
- j** = $\text{St Pr}^{2/3}$ = Colburn modulus, dimensionless. This modulus vs Reynolds number defines the heat transfer characteristics of a heat transfer surface. (See Figure 24)
- K** = Ratio of free flow area to face area, dimensionless
- k** = Thermal conductivity, Btu per sec ft °R
- L** = Length, ft
- m** = Molecular weight, lb per lb mol
- n** = Exponent in the equation for pressure drop in adsorber beds (See Eqn IV-22)
- N** = Number of adjacent fins in a plate fin flow passage

LIST OF SYMBOLS (CONTINUED)

- $NTU = UA/wc_p$ = Number of heat transfer units, a dimensionless heat exchanger parameter
- P = Pressure, lb_f per ft^2 unless otherwise noted
- $Pr = \mu c_p/k$ = Prandtl number, dimensionless
- Q = Heat transfer rate, Btu per day
- R = The gas constant in $Pv = RT$, $ft\ lb_f$ per $lb_m\ ^\circ R$
- $Re = 4rG/\mu$ = Reynolds number, a flow modulus, dimensionless
- $St = h/G c_p$ = Stanton number, dimensionless
- T = Temperature, $^\circ R$
- t = Thickness, in.; or contact time, sec
- U = Overall coefficient of heat transfer, Btu per sec $ft^2\ ^\circ R$
- V = Volume, ft^3 ; or velocity, ft per sec
- v = Specific volume, ft^3 per lb
- W = Weight, lb_m
- w = Flow rate, lb per sec unless otherwise noted
- X = Contaminant concentration in the air, lb contaminant per lb air
- y = Carbon dioxide concentration in the air, lb carbon dioxide per lb dry air
- $Z = (wc_p)_1/(wc_p)_2$ = Capacity rate ratio, a dimensionless heat exchanger parameter

Greek Letter Symbols

- $\alpha = \frac{Pr^{2/3}}{c_p}$ = A fluid property grouping, $lb\ ^\circ R$ per Btu
- $\gamma = c_p/c_v$ = Ratio of specific heats, dimensionless
- Δ = Denotes a difference
- θ = Time, sec unless otherwise noted
- η_o = Overall fin effectiveness of an extended surface, dimensionless
- μ = Viscosity, lb_m per sec ft

LIST OF SYMBOLS (CONTINUED)

- ρ = Density, lb_m per ft^3
 σ = Ratio of density to standard density
 λ = Latent heat of vaporization, Btu per lb_m

Subscripts

- a = Dry air
c = Carbon dioxide or cold side
f = Face
h = Hot side
i = Ice
m = Metabolic
o = Oxygen or overall
w = Water
1, 2, 3, etc. = Refer to locations in a system or component

Note: Some of the symbols carry a subscript as part of the basic symbol.
They are as follows:

- A_c , A_f , c_p , and η_o
 lb_f = Pounds force
 lb_m = Pounds mass

LIST OF ABBREVIATIONS

Aux	=	Auxiliary
Atm	=	Atmosphere of pressure
Btu	=	British thermal unit
ft	=	Feet
hr	=	Hour
in.	=	Inch
min	=	Minute
mmHg	=	Millimeter of mercury
psia	=	Pounds force per square inch
sat	=	Saturated
$^{\circ}\text{R}$	=	Degree Rankine

CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS

SECTION I

INTRODUCTION

1. Background - This report describes a design study of various methods of removing water and carbon dioxide contaminants from closed respiratory systems. All methods employ the endothermic conversion of liquid oxygen to gaseous oxygen as a heat sink.

The support of human life in the closed respiratory systems of space vehicles requires careful environmental control. The breathing gas atmosphere must be controlled within certain limits with respect to temperature, pressure, and composition. One of the major problems concerns the control of contaminants, which are represented by gaseous constituents other than oxygen and nitrogen (or other diluent). The metabolism of food by the occupants results in the addition of water vapor and carbon dioxide to the cabin atmosphere by respiration. Additional water vapor is obtained by perspiration. These substances must not only be removed, but their concentrations must be held to within very definite limits. A number of other contaminants from various sources are possible. For example, hydrogen sulphide, hydrogen, methane, and ammonia are contained in flatus. The electronic equipment, power generating and energy conversion systems, and surface finishes can produce a wide variety of contaminating gases. Because of the relative volatilities and allowable contaminant partial pressures, only carbon dioxide and water vapor (of the more probable contaminants) can be removed by freeze-out methods using the cryogenic atmospheric makeup fluids as the heat sink. Other methods must then be used to assure maintenance of contaminant concentrations to levels below toxic or explosive limits.

2. Purpose of the Study - Several liquid oxygen heat sink freeze-out methods of removing contaminants in closed respiratory systems are possible, but not all of them are feasible for use in space vehicles where minimum overall system weight and size are necessary. Only a limited supply of liquid oxygen is available - the ideal use rate being just equal to or only slightly higher than the metabolic consumption rate (normally, about 2.0 lb per man day). The use of such a small amount of liquid oxygen for freeze-out places a severe restriction on the design of any freeze-out system, and extremely careful and ingenious studies are required to arrive at an optimum freeze-out system. The purpose of this study was to find out if freeze-out of contaminants by use of a liquid oxygen heat sink is thermodynamically feasible for space vehicle closed respiratory systems. This report is, therefore, a feasibility study which consists of schematic diagrams and cycle analysis calculations for various freeze-out systems, quantitative (weight, size, and power requirement) and qualitative (advantage and disadvantage) comparisons of various freeze-out systems, a preliminary design of a model contaminant removal system which is considered to be most worthy of future development, and finally, a recommendation as to the amount and type of additional effort required to prove feasibility and to reduce the design approach to hardware practice.

3. Possible Freeze-Out Systems - The removal of contaminants from respiratory system air may be accomplished by various combinations of heat exchangers, adsorbers, absorbers, expanders, refrigeration devices and auxiliary heat sinks. In this study only the following systems, all of which involve some form of freeze-out, were considered: simple freeze-out of water and carbon dioxide, freeze-out of water plus adsorption of carbon dioxide, adsorption of water plus freeze-out of carbon dioxide, freeze-out of water plus adsorption of carbon dioxide in combination with auxiliary cooling devices. The use of absorbers was not considered.

The two major problems associated with the freeze-out systems were the regeneration of the frozen-out contaminants and the recovery of water. Regeneration is necessary to prevent excessive conversion of liquid oxygen to gaseous oxygen, and water recovery may be necessary to maintain an over-all water balance for the space vehicle.

SECTION II

GENERAL CONSIDERATIONS

1. Integration of Contaminant Freeze-out and Cabin Environmental System - A contaminant freeze-out system is a part of the space vehicle cabin environmental system. The integration of the contaminant freeze-out system with the cabin environmental system is shown schematically in Figure 1. Notice that the cabin air is cooled by the cabin air cooler to a dew point of 35°F. Some of the air is then directed to the freeze-out system where the carbon dioxide and remaining water vapor are removed. The remaining air by-passes the freeze-out system and mixes with the purified-oxygen enriched air from the freeze-out system. The cool air mixture then returns to the cabin. Odors, carbon monoxide, and other trace contaminants are adsorbed on the way.

The cabin environmental system maintains the air at a temperature of about 70°F and the relative humidity at about 50 per cent. The freeze-out system maintains the carbon dioxide concentration at a level which is tolerable by the occupants for long durations. This level is not precisely known at the present time, but the range of concentrations is somewhere between 0.5 and 1.0 per cent carbon dioxide by volume. The range of cabin total pressure will probably be somewhere between 0.5 and 1.0 atmosphere.

The primary function of the contaminant freeze-out system is to remove carbon dioxide from the air. However, because of thermodynamic reasons, water freezes-out or is adsorbed before carbon dioxide. Therefore, the freeze-out system must also remove water from the air. The water may be returned after removal of the carbon dioxide.

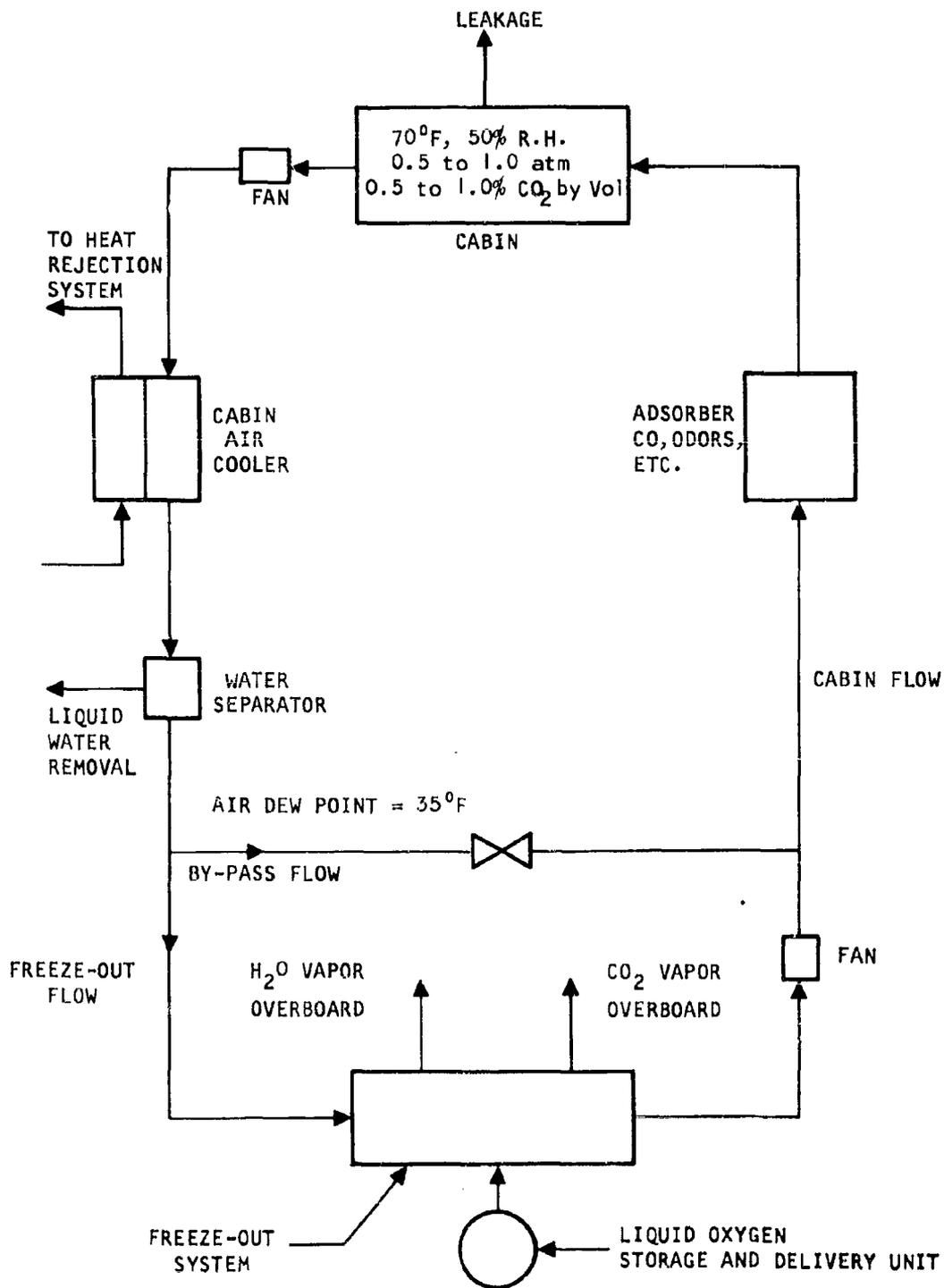


Figure 1. Schematic Flow Diagram of the Cabin Environmental and Contaminant Freeze-out Systems

2. Air Handling Requirements - The amount of cabin air flow may be determined from an energy balance for the cabin. Neglecting air leakage and heat leak, the metabolic heat generation rate must be equal to the change in enthalpy of the air times the air flow rate. This energy balance may be expressed as

$$Q_m = w_{\text{cabin air}} \left[(h_{a1} - h_{a0}) + (y_1 h_{c1} - y_0 h_{c0}) + (H_1 h_{w1} - H_0 h_{w0}) \right] \quad \text{II-1}$$

Assuming perfect gas behavior, Equation II-1 may be written

$$Q_m = w_{\text{cabin air}} \left\{ \left[c_{pa} + (y_1 - y_0) c_{pc} + (H_1 - H_0) c_{pw} \right] (T_1 - T_0) + (H_1 - H_0) \lambda_w \right\} \quad \text{II-2}$$

where the symbols are defined in the nomenclature, and the subscripts 0 and 1 refer to cabin inlet and outlet conditions respectively. If the cabin total pressure is 14.7 psia, the inlet absolute humidity should be about $H_0 = 30$ grains water per lb of air (0.00429 lb water per lb air),

and the outlet absolute humidity (70 per cent relative humidity) would be 77 grains water per lb air (0.0110 lb water per lb air). The change in carbon dioxide enthalpy is small and may be neglected. The latent heat of vaporization of the water is about $\lambda_w = 1060$ Btu per lb, the water vapor specific heat is 0.45 Btu per lb $^{\circ}\text{F}$, the air specific heat is 0.24 Btu per lb $^{\circ}\text{F}$, the air temperature rise is about $T_1 - T_0 = 35^{\circ}\text{F}$,

and the normal metabolic heat production rate is about 540 Btu per man-hr. If these numbers are substituted into Equation II-2, the cabin air flow rate at one atmosphere of pressure is $w_{\text{cabin air}} = 34.6$ lb air per man hr.

If the cabin air pressure is reduced to one half an atmosphere, the relative humidity should still be maintained at 50 per cent for the same degree of comfort. The absolute humidities would be twice as great, and the oxygen enriched air specific heat would be about 0.235 Btu per lb $^{\circ}\text{F}$. Therefore, the cabin air flow rate at one half an atmosphere would be reduced to $w_{\text{cabin air}} = 24.4$ lb air per hr.

The air flow rate to the freeze-out system depends upon the amount of carbon dioxide that must be removed per man-hr, the carbon dioxide removal efficiency, and the concentration of carbon dioxide in the air coming out of the cabin. The carbon dioxide removal rate may be expressed as

$$w_c = \eta y_1 w_{\text{freeze-out air}} \quad \text{II-3}$$

where, according to Dalton's law of partial pressures for perfect gases,

$$y_1 = \frac{m_c P_{c1}}{m_a P_a} = 1.52 \frac{P_{c1}}{P_a} \quad \text{II-4}$$

Assuming a 90 per cent removal efficiency and a concentration of 1.0 per cent by volume ($P_{c1}/P_a = .01$), the air flow to the freeze-out system for a removal rate equal to the normal metabolic production rate of 2.25 lb CO₂ per man day (0.0937 lb CO₂ per man-hr) is $w_{\text{freeze-out air}} = \frac{w_c}{\eta y_1}$
 $= \frac{.0937}{.9 \times .0152} = 6.85$ lb air per man hr. If the carbon dioxide concentration is reduced to 0.50 per cent by volume, the air flow rate to the freeze-out system is doubled.

3. Freeze-Out Temperatures - The temperatures required for freeze-out of contaminants may be determined by referring to the vapor pressure vs temperature curves for carbon monoxide, carbon dioxide and water in Figure 2. The contaminants must be cooled to their saturation temperatures before any freeze-out takes place. The air enters the freeze-out system saturated with water vapor at a temperature of 495°R. Freeze-out starts here and is essentially complete (99 per cent) at about 400°R, and only trace amounts remain at 360°R.

Carbon dioxide freeze-out starts at a much lower temperature. For long duration flights of a space vehicle, the maximum allowable carbon dioxide partial pressure for human safety is about 7.6 mm Hg (0.147 psia). Therefore, the air must be cooled to a temperature of about 273°R before any solid carbon dioxide is formed. Essentially complete freeze-out (99 percent) may be obtained by cooling to a temperature of about 220°R. Both of these temperatures are substantially above the boiling point of liquid oxygen at low pressures (151°R at 0.5 atm and 162°R at 1.0 atm) and, therefore, freeze-out of carbon dioxide is thermodynamically possible.

For safety reasons, the carbon monoxide partial pressure in the respiratory system should be kept below about 0.1 mm Hg (0.0019 psia). Figure 2 indicates that the air would have to be cooled to about 80°R before any carbon monoxide freeze-out would occur. Removal by freeze-out with a liquid oxygen heat sink is thermodynamically impossible. Carbon monoxide removal must be accomplished outside the freeze-out system.

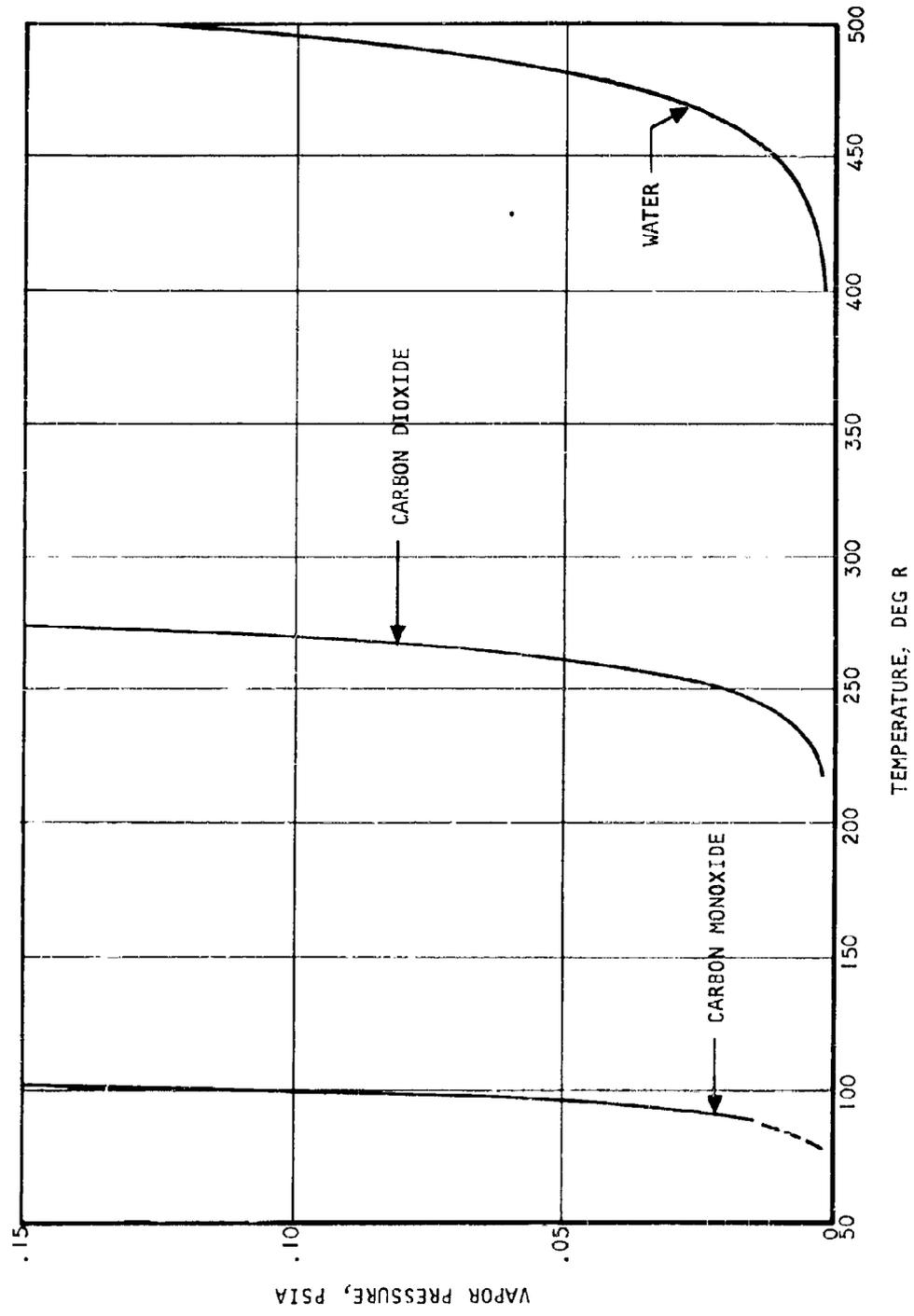


Figure 2. Vapor Pressure of Carbon Monoxide, Carbon Dioxide and Water

4. Freeze-Out Processes - A total of five freeze-out processes are possible. They are as follows:
- (a) Non-regenerative freeze-out. Water and carbon dioxide are recovered.
 - (b) Freeze-out with regeneration of the processed air. Water and carbon dioxide are recovered.
 - (c) Freeze-out with regeneration of the processed air and carbon dioxide. Water is recovered.
 - (d) Freeze-out with regeneration of the processed air and water. Carbon dioxide is recovered.
 - (e) Regenerative freeze-out. Water and carbon dioxide are dumped overboard.

Process (a) may be rejected because the liquid oxygen requirements are far in excess of the metabolic rate. Process (b) is better than (a), but the liquid oxygen requirement is still excessive. Process (c) is desirable because water is recovered. Although, the liquid oxygen requirement is less than process (b), the oxygen usage is high. Process (d) is about the same as (c), but is not as desirable because water is not recovered. Process (e) requires the least amount of liquid oxygen. The two processes which appear to be most worthy of analysis are (c) and (e).

5. Adsorber Characteristics - Two types of adsorbent materials are considered in this study. They are silica gel for the adsorption of water and molecular sieves for the adsorption of carbon dioxide. Silica gel is a granulated form of silicon dioxide, and is manufactured in various mesh sizes. (The term gel refers to a state of the material during the manufacturing process). Molecular sieves are synthetic zeolites which are manufactured in the form of small pellets. These materials are placed inside cylindrical canisters which fit into the air duct to form a porous bed. The air flows through the bed and is purified.

The overall operating characteristics of a typical adsorber bed during adsorption are shown in Figure 3. The concentration of the contaminant in the outlet air is essentially zero until a break through point is reached. After this the concentration of the contaminant in the outlet air increases rapidly, and the air must be switched to another bed to continue the adsorption process. The loaded bed must then be desorbed by heating with hot, pure air or by heating and venting to a vacuum. During adsorption the air temperature rises because of the release of the latent heat of vaporization and the heat of wetting. The sum of these two items is called the heat of adsorption and is equal to about 1400 Btu per lb for silica gel adsorption of water and about 300 Btu per lb for molecular sieve adsorption of carbon dioxide. Figure 3 also shows how the concentration of contaminants in the bed material varies with distance along the bed.

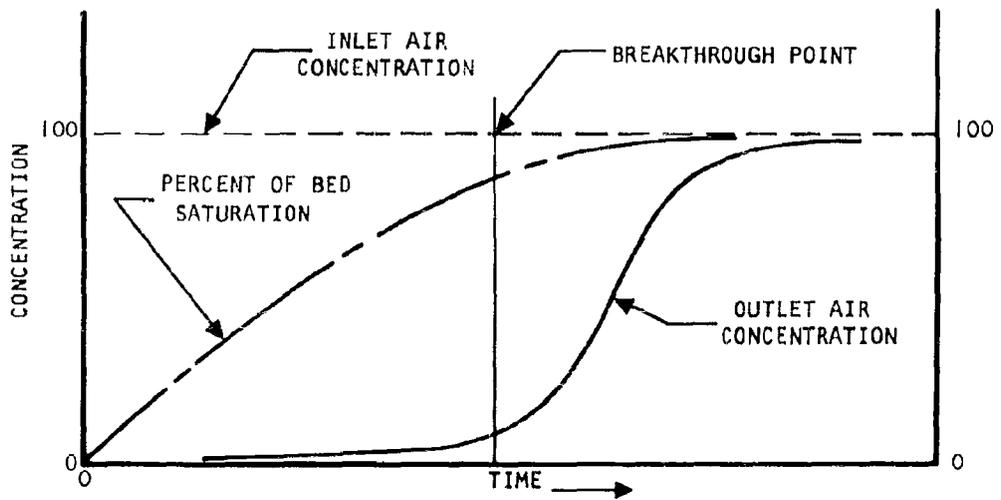
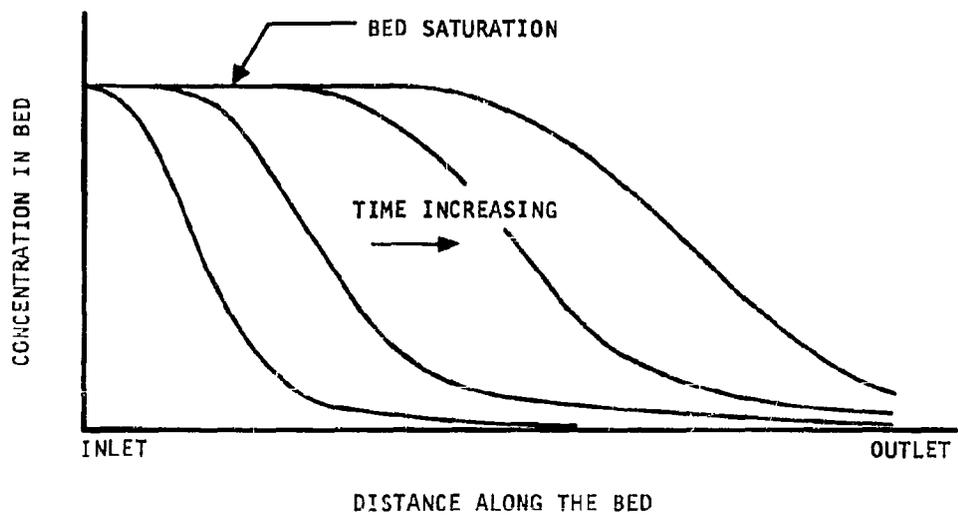


Figure 3. Concentration Profiles in Adsorber Beds

The amount of contaminant required to saturate an adsorber is a function of the vapor pressure of the contaminant (adsorbate) and the temperature of the bed. This equilibrium data may be plotted as lbs of contaminant per lb of adsorbent vs vapor pressure with temperature as a parameter. The water vapor adsorption characteristics of silica gel and molecular sieves at a temperature of 25°C(77°F) are shown in Figure 4. The effect of temperature on the water vapor adsorption of silica gel is shown in Figure 5, and the effect of temperature on the carbon dioxide adsorption of molecular sieves is shown in Figure 6. Notice that the adsorption capability increases with increasing pressure and decreases with increasing temperature. The data in Figures 5 and 6 indicates that high temperatures are required to desorb the beds. The high temperature requirement may be relieved by venting to the vacuum of space. The pressure drop characteristics of two molecular sieve beds and one silica gel bed are shown in Figure 7.

6. Heat Sink Requirements - The amount of liquid oxygen required for freeze-out of water and carbon dioxide may be easily determined from energy balance considerations. These are discussed in Section III.

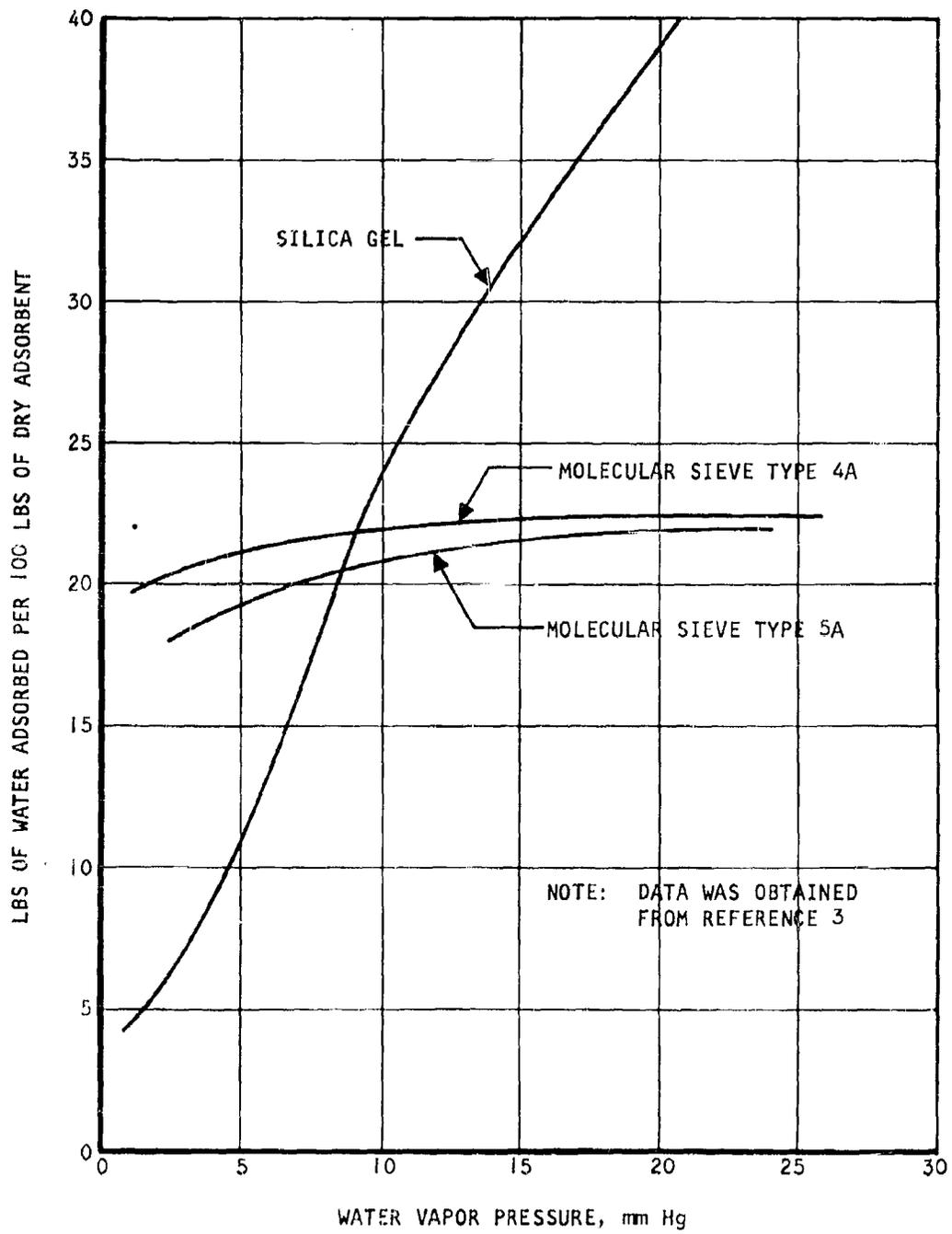


Figure 4. Water Vapor Adsorption of Silica Gel and Molecular Sieves at 77°F

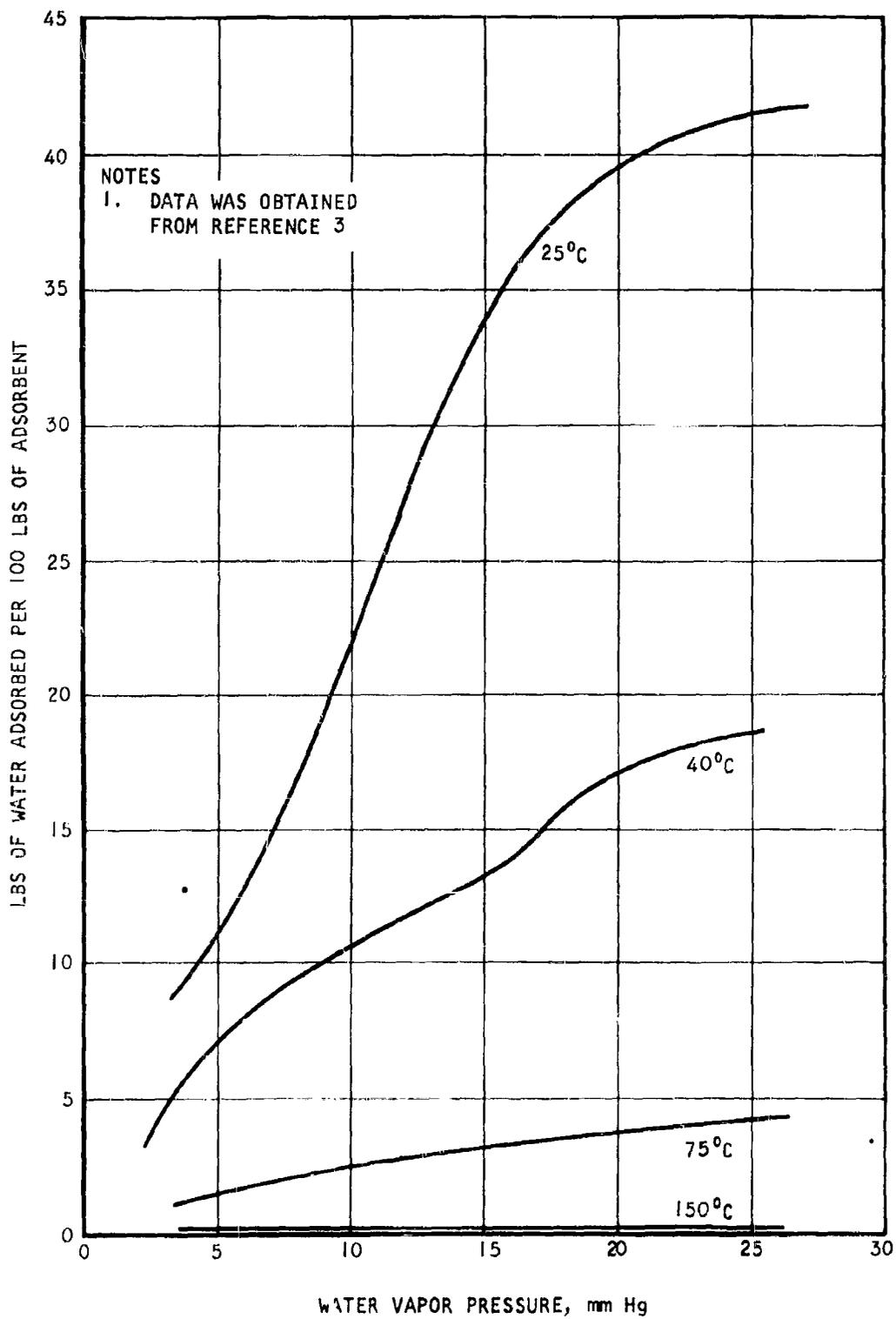


Figure 5. Water Vapor Adsorption Isotherms for Silica Gel

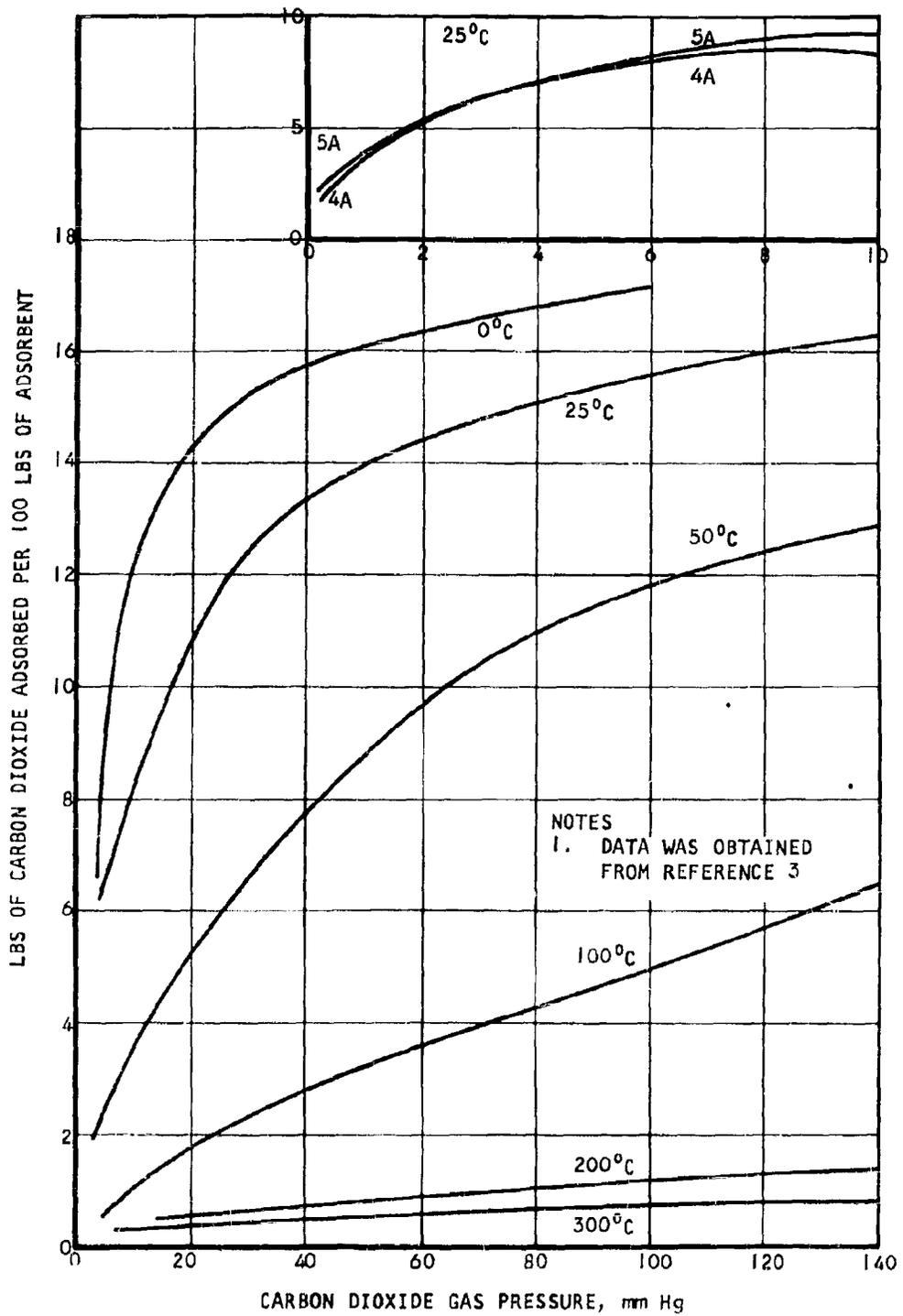


Figure 6. Carbon Dioxide Adsorption Isotherms for Molecular Sieves

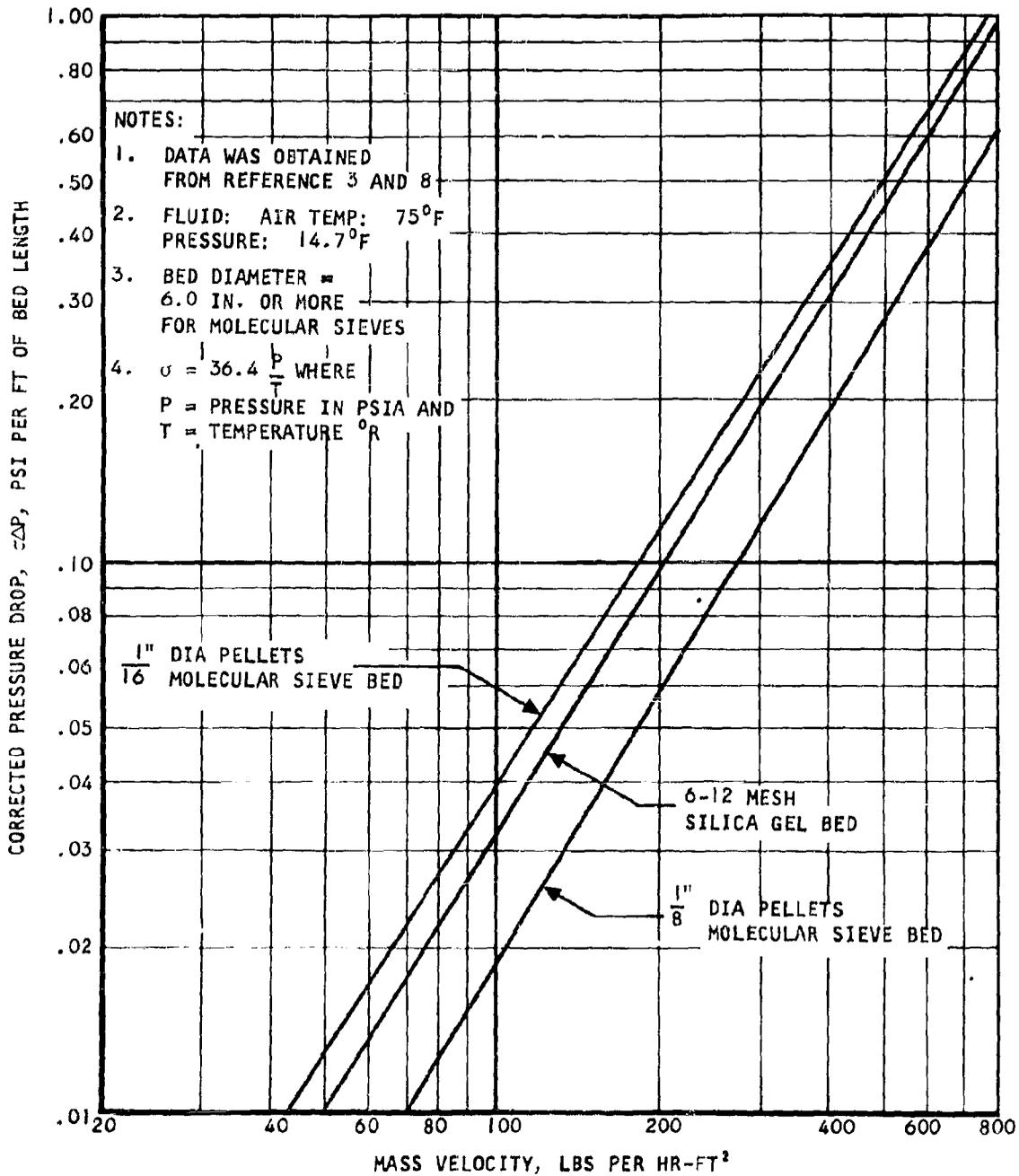


Figure 7. Pressure Drop Characteristics of Typical Molecular Sieve Beds and 6-12 Mesh Silica Gel Bed

7. Liquid Oxygen Converters and Storage Vessels - In contaminant freeze-out systems which use liquid oxygen as a heat sink, the conversion of the liquid oxygen to gaseous oxygen for breathing purposes must be accomplished within the freeze-out heat exchangers; therefore, the contaminant freeze-out system accomplishes the dual function of liquid oxygen to gaseous oxygen conversion and contaminant removal. A separate converter is not needed, but some form of weightless condition liquid delivery device is required. The heat transfer process inside the freeze-out heat exchangers is by the mechanism of forced convection boiling, which is not influenced by the absence of gravity. The liquid storage and delivery unit must also work in a weightless environment. One method of accomplishing this is to use a liquid storage vessel with positive expulsion. Positive expulsion may be obtained with a flexible bladder which is pressurized with helium gas. Low fluid pressures are used, so that, with proper design, the fluid is stored and expelled as a saturated or subcooled liquid.

The characteristics of several types of cryogenic converters for space flight applications are discussed in the Appendix. In the present study, the liquid oxygen will be assumed to be stored as a saturated liquid at a pressure of 75 psia in a flexible bladder type of positive expulsion storage vessel. Delivery will be accomplished by helium pressure and by throttling the liquid to the freeze-out system operating pressure as shown on the pressure vs enthalpy diagram of oxygen in Figure 8. At this time, no other form of storage appears to be feasible for freeze-out systems. See the Appendix, page 91, for a more complete discussion on liquid storage with positive expulsion.

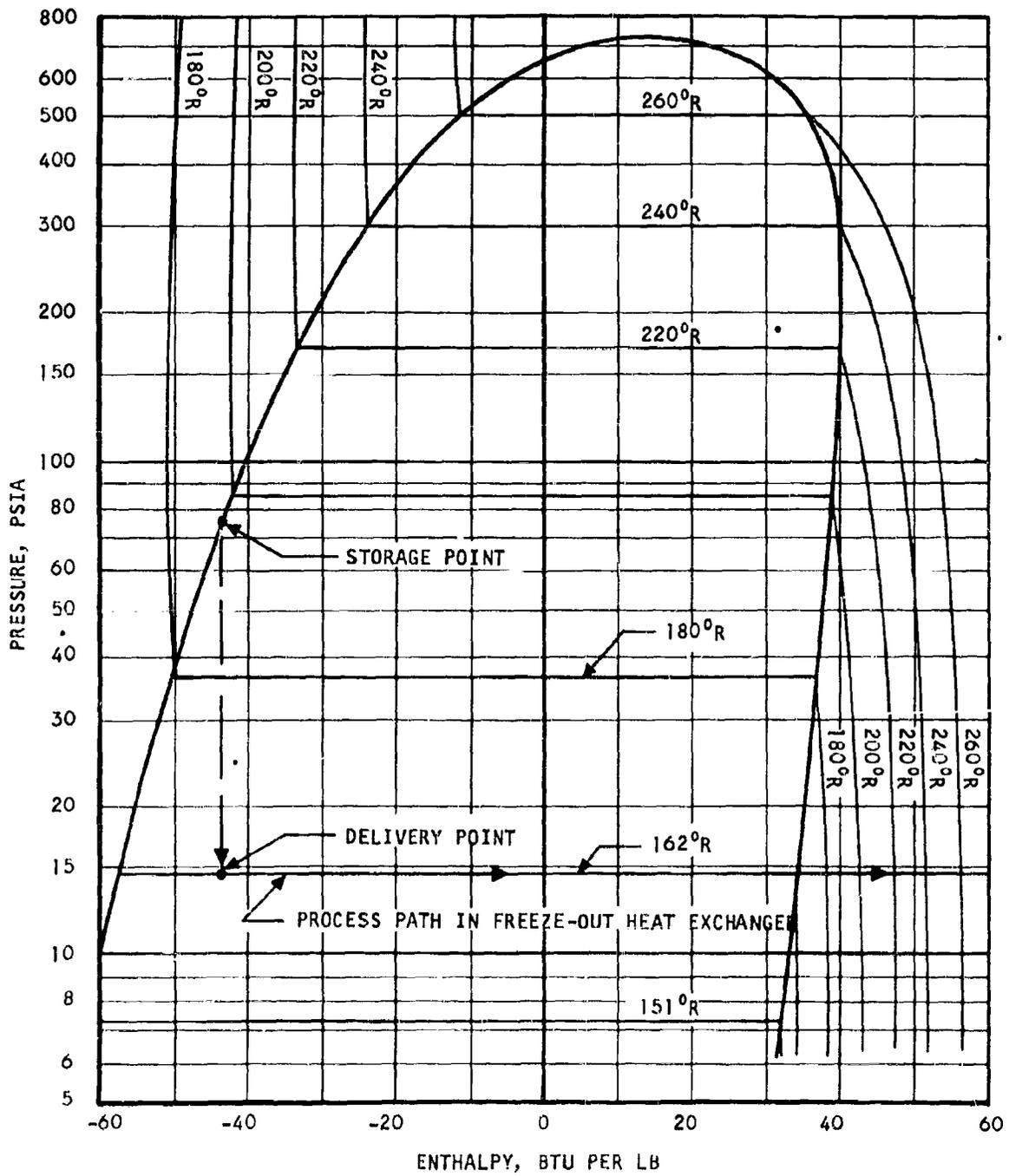


Figure 8. Pressure - Enthalpy Diagram for Oxygen Near the Two Phase Region

SECTION III

DESCRIPTION AND CYCLE ANALYSIS OF CONTAMINANT FREEZE-OUT SYSTEMS

- I. Preliminary Considerations - All of the calculations which follow are based on a "per lb of pure, dry air basis". The contaminant flow rate in the air stream is then equal to the air flow rate times the contaminant concentration by weight. According to Dalton's law of partial pressures for perfect gases the contaminant concentrations by weight for water vapor and carbon dioxide in air are given by

$$H = \frac{m_w P_w}{m_a P_a} \text{ lb water per lb air} \quad \text{III-1}$$

$$y = \frac{m_c P_c}{m_a P_a} \text{ lb CO}_2 \text{ per lb air} \quad \text{III-2}$$

where m is the molecular weight and P is the partial pressure. The subscripts a , c and w refer to air, carbon dioxide, and water vapor respectively. The partial pressure of the air is

$$P_a = P_t - P_c - P_w \quad \text{III-3}$$

where P_t is the total pressure. The ratio of partial pressures is also equal to the concentration by volume.

The carbon dioxide flow rate in the freeze-out system air stream is equal to $y_1 w_a$, and the amount of carbon dioxide removed by the freeze-out system is $w_c = \eta y_1 w_a$. Therefore, the air flow rate to the freeze-out system is

$$w_a = \frac{w_c}{\eta y_1} \quad \text{III-4}$$

where w_c is the metabolic production rate of CO_2 , y_1 is the inlet CO_2 concentration by weight, and η is the CO_2 removal efficiency. The removal efficiency includes the effects of incomplete precipitation of carbon dioxide because of saturation temperature limits and incomplete separation of the frozen-out solids from the air stream.

The following data was used to evaluate and compare the various freeze-out systems on a one man basis:

- 1) Inlet air temperature, $T_1 = 495^\circ\text{R}$
- 2) Inlet dew point 495°R

- 3) Carbon dioxide removal rate, $w_c = 2.25$ lb per day
- 4) Metabolic ratio of CO_2 to O_2 flow, $w_c/w_{om} = 1.125$
- 5) Specific heats in Btu per lb $^{\circ}R$: CO_2 , 0.19; water vapor, 0.45; oxygen, 0.22; air at 7.35 psia (45 per cent O_2 by vol), 0.235; and air at 14.7 psia (21 per cent O_2 by vol), 0.240.
- 6) Liquid oxygen heat sink data after being throttled from a saturated liquid at a storage pressure of 75 psia:

System pressure, psia	7.35	14.7
Sink temperature, $^{\circ}R$	151	162
Effective latent heat, λ_o , Btu per lb	75.8	77.8

- 7) Heat of adsorptions (latent heat plus heat of wetting) of silica gel and molecular sieves are 1200 Btu per lb and 300 Btu per lb respectively.
- 8) For preliminary comparison purposes, the relationship between effectiveness, E , and number of heat transfer units, NTU , in the freeze-out heat exchangers was based on conventional theory for heat transfer in counter-flow. That is

$$NTU = \frac{1}{1-Z} \ln \frac{1-EZ}{1-E} \quad \text{III-5}$$

where $E = \frac{\Delta T_h}{\Delta T_o}$ the warm side temperature effectiveness

$Z = \frac{(w_c)_h}{(w_c)_o}$ the capacity rate ratio

$NTU = \frac{UA}{(w_c)_h}$ the number of heat transfer units

When the frozen-out solids are both completely regenerated, as is the case with the simple freeze-out system with water removal, the oxygen flow rate is very small, and the value of Z may be taken as unity. When $Z = 1$, Equation III-5 reduces to

$$NTU = \frac{E}{1-E} \quad \text{III-6}$$

When water is frozen-out, but not regenerated, the capacity ratio for the water freeze-out heat exchanger may be taken as

$$Z = \frac{w_a (\Delta h/\Delta T)_h}{w_a c_{pa} + w_o c_{po}} \frac{(\Delta h/\Delta T)_h}{c_{pa} + \frac{w_o}{w_a} c_{po}} \quad \text{III-7}$$

$$\text{where } \Delta h = (c_{pa} + y_1 c_{pc}) \Delta T + H_1 \left[\lambda_i + \frac{c_{pi}}{2} (492 - T_2) + c_{pw} (T_1 - 492) \right]$$

The capacity rate ratio for the carbon dioxide freeze-out heat exchanger may be taken as

$$Z \approx \frac{w_a c_{pa}}{w_a c_{pa} + w_o c_{po}} = \frac{c_{pa}}{c_{pa} + \frac{w_o}{w_a} c_{po}} \quad \text{III-8}$$

2. Simple Freeze-Out - Simple freeze-out consists of freezing out the water and carbon dioxide with heat exchangers. Efficient operation (low liquid oxygen flow) can only be obtained by regenerating the water and the carbon dioxide in a freeze-out switching heat exchanger as indicated in Figure 9. The water and carbon dioxide are dumped overboard as vapors; both contaminants are removed from the system. If water recovery is desired a system such as the one shown in Figure 10 must be used. The water is recovered by heating and rehumidifying the outlet air while the carbon dioxide is frozen out regeneratively. A description and cycle analysis of each of these simple freeze-out systems are given below:

- a. System I. Simple Freeze-Out With Water Vapor Removal - This process takes place completely within a four channel switching heat exchanger, and the system is described schematically in Figure 9. The contaminated air enters the system and flows through a freeze-out channel where cooling and freeze-out of the water and carbon dioxide occurs. The heat transfer surface is designed to hold the frozen-out solids. The purified air then mixes with liquid oxygen and returns in the adjacent regenerative channel. A third sublimation channel (which was filled with solids during the previous cycle) is purged by venting to the vacuum of space while a fourth channel is empty during this half of the cycle. When the freeze-out channel becomes loaded with solids (and the sublimation channel is completely purged) all valves are reversed so that the sublimation channel now becomes the freeze-out channel, and the empty channel now becomes the regenerative channel. Clean out (purging) of the solids takes place by sublimation. Heat flows from the freeze-out channel to both the regenerative and sublimation channels. The cycle switching time is indicated by a specified rise in pressure drop in the freeze-out channel. The heat exchanger construction is described in Section IV.

The heat sink is liquid oxygen which is stored in a positive expulsion storage vessel. The operating pressure in the vessel is well below the critical pressure -- probably 75 psia. The liquid is throttled down to system operating pressure before entering the heat exchanger.

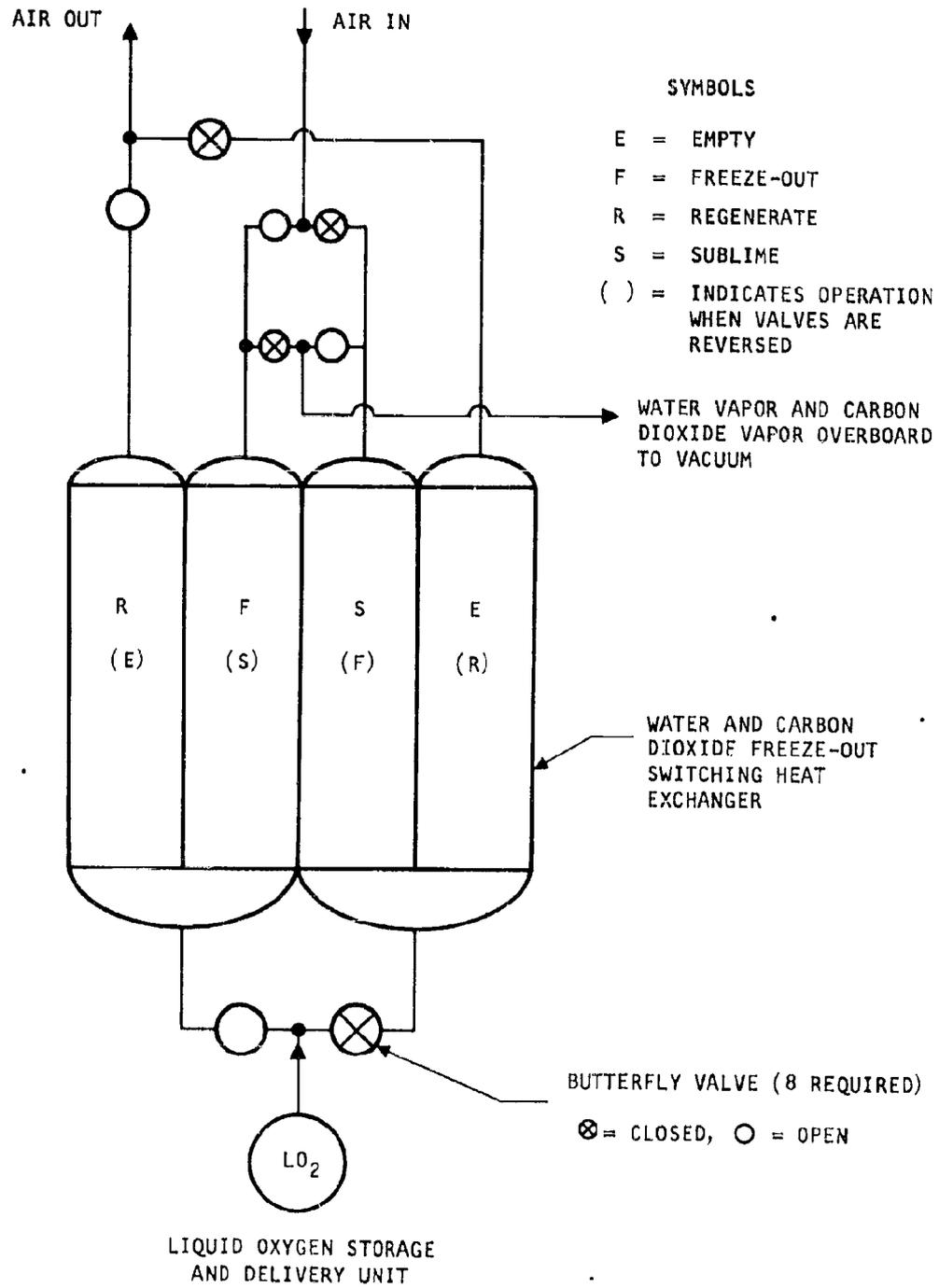


Figure 9. Schematic Diagram for System I
 (Simple Freeze-Out with Water Removal)

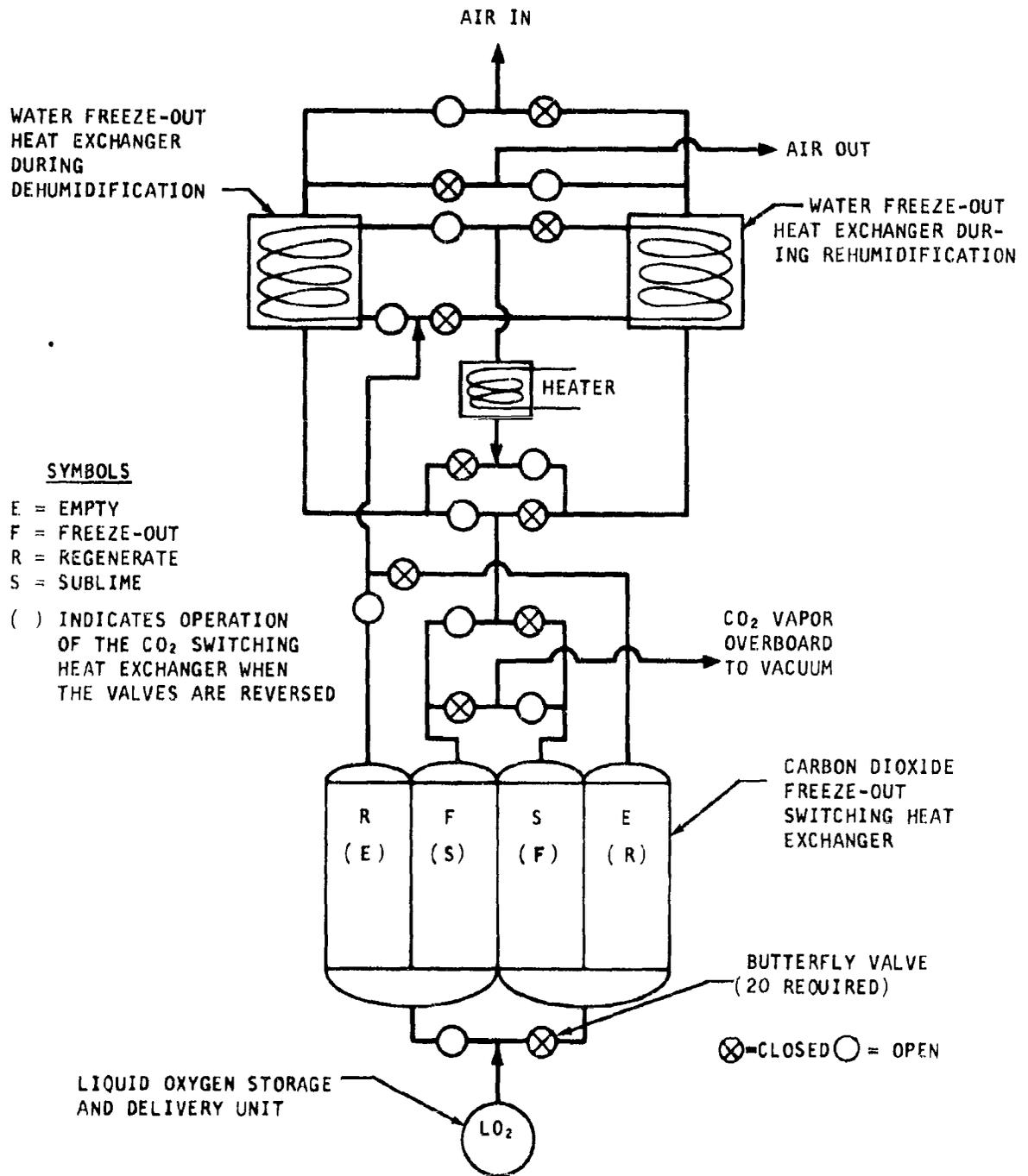


Figure 10. Schematic Diagram for System II
 (Simple Freeze-Out with Water Recovery)

A cycle analysis of the simple freeze-out system for zero cabin leakage may be visualized by referring to the flow chart in Figure 11. An energy balance for the system may be written as

$$Q_1 + Q_9 + Q_{10} + Q_{11} = Q_6 + Q_7 + Q_8 \quad \text{III-9}$$

where, on a dry air basis, the rate equations are

$$Q_1 = w_a (h_{a1} + y_1 h_{c1} + H_1 h_{w1}) \quad \text{III-10}$$

$$Q_6 = w_a [h_{a6} + (1-\eta) y_1 h_{c6}] + w_o h_{o6} \quad \text{III-11}$$

$$Q_7 = w_a H_1 h_{w7} \quad \text{III-12}$$

$$Q_8 = w_a \eta y_1 h_{c8} \quad \text{III-13}$$

$$Q_9 = w_o h_{o9}$$

Q_{10} , Q_{11} are heat leaks into the system

Substituting the rate equations into the energy balance equation and rearranging terms, the energy balance becomes

$$w_a \left[(h_{a1} - h_{a6}) + y_1 \left\{ h_{c1} - (1-\eta) h_{c6} - \eta h_{c8} \right\} + H_1 (h_{w1} - h_{w7}) \right] + Q_{10} + Q_{11} = w_o (h_{o6} - h_{o9}) \quad \text{III-14}$$

where, assuming perfect gas behavior,

$$h_{a1} - h_{a6} = c_{pa} (T_1 - T_6) \quad \text{III-15}$$

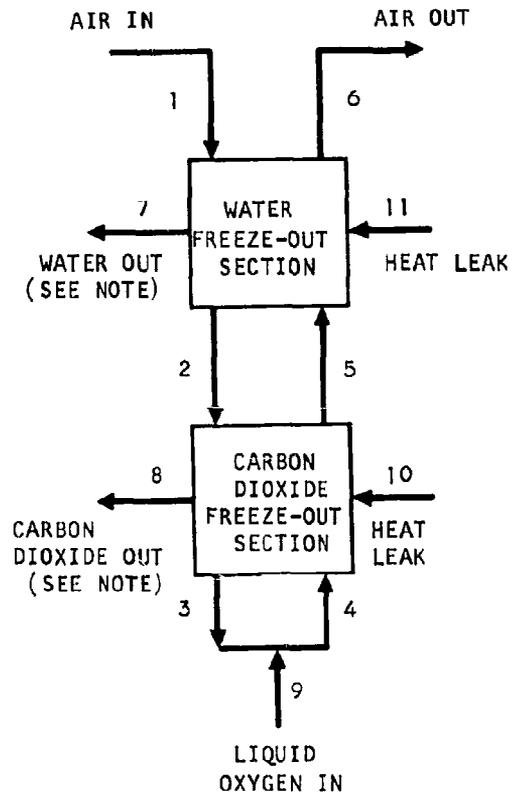
$$h_{c1} - (1-\eta) h_{c6} - \eta h_{c8} = h_{c1} - h_{c6} + \eta (h_{c6} - h_{c8}) = c_{pc} \left[(T_1 - T_6) + \eta (T_6 - T_8) \right] \quad \text{III-16}$$

$$h_{w1} - h_{w7} = c_{pw} (T_1 - T_7) \quad \text{III-17}$$

$$h_{o6} - h_{o9} = \lambda_o + c_{po} (T_6 - T_9) \quad \text{III-18}$$

Substituting Equations III-15 through III-18 into Equation III-14 and rearranging the terms the ratio of oxygen flow to air flow is found to be

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc})(T_1 - T_6) + H_1 c_{pw} (T_1 - T_7) + \eta y_1 (T_6 - T_8) + (Q_{10} + Q_{11})/w_a}{\lambda_o + c_{po} (T_6 - T_9)} \quad \text{III-19}$$



NOTE: THE FROZEN-OUT WATER AND CARBON DIOXIDE ARE REGENERATED TO TEMPERATURE T_6 AND DUMPED OVERBOARD AS VAPORS

Figure 11. Flow Chart for System I (Simple Freeze-Out With Water Removal)

If the water and carbon dioxide are regenerated within the heat exchanger to temperature T_6 ($T_7 = T_6$ and $T_8 = T_6$), and if the heat leaks are negligible ($Q_{10} = Q_{11} = 0$), the ratio of oxygen flow to air flow is

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc} + H_1 c_{pw})(T_1 - T_6)}{\lambda_o + c_{po}(T_6 - T_9)} \quad \text{III-20}$$

The amount of carbon dioxide which is removed by the freeze-out system equals the metabolic production rate; therefore,

$$w_c = w_{cm} = |y_1| w_a \quad \text{III-21}$$

This may be rearranged to give the required air flow as

$$w_a = \frac{w_c}{|y_1|} \quad \text{III-22}$$

The ratio of metabolic oxygen flow rate, w_{om} , to air flow rate, w_a , may be found by dividing w_{om} by Equation III-22. The result is

$$\frac{w_{om}}{w_a} = \frac{|y_1|}{\left(\frac{w_c}{w_o}\right)_m} \quad \text{III-23}$$

Finally, the ratio of oxygen flow required for freeze-out of water and carbon dioxide to oxygen flow required for metabolism is found by dividing Equation III-20 by Equation III-23. The result is

$$\frac{w_o}{w_{om}} = \frac{(c_{pa} + y_1 c_{pc} + H_1 c_{pw})(T_1 - T_6) \left(\frac{w_c}{w_o}\right)_m}{\left[\lambda_o + c_{po}(T_6 - T_9) \right] |y_1|} \quad \text{III-24}$$

where $y_1 = \frac{m_c P_{c1}}{m_a P_a}$ the inlet air carbon dioxide content by weight

$H_1 = \frac{m_w P_w}{m_a P_a}$ the inlet air water vapor content by weight

$\frac{w_c}{w_{om}}$ the metabolic ratio of carbon dioxide production to oxygen consumption 1.125

$$\eta = 1 - \frac{Y_3}{Y_1} = 1 - \frac{P_{c3}}{P_{c1}} = \text{the carbon dioxide removal}$$

efficiency. This includes the effects of incomplete holdup of the solid in the heat exchanger and incomplete precipitation of the solid (saturation temperature limits).

c_p = specific heat at constant pressure, Btu per lb °F

λ_o = effective latent heat of vaporization at freeze-out pressure

T = temperature, °R

The letters a, c, o, w and m are subscripts for air, carbon dioxide, oxygen, water and metabolic, respectively. The numbers 1, 6 and 9 refer to locations within the system as shown in Figure 11.

Note that regeneration of the water and carbon dioxide can only be obtained with regeneration within the heat exchanger by sublimation of the vapors in a vacuum. If recovery of the water (water returned to the air stream rather than dumping overboard to the vacuum of space) is desired, the ice must be vaporized by periodic purging of the freeze-out flow passage with hot air. The water recovery system is discussed in Paragraph (b) of this section.

The heat exchanger design temperatures may be computed from Equation III-24 and an energy balance on the low temperature end. Equation III-24 may be solved for the temperature difference $T_1 - T_6$ by noting that $T_6 - T_9 = (T_1 - T_9) - (T_1 - T_6)$. This substitution and rearrangement of Equation III-24 gives

$$T_1 - T_6 = \frac{\lambda_o + c_{po} (T_1 - T_9)}{c_{po} + \frac{(c_{pa} + Y_1 c_{pc} + H_1 c_{pw}) \left(\frac{w_c}{w_o} \right)_m}{\eta Y_1 \left(\frac{w_o}{w_{om}} \right)}} \quad \text{III-25}$$

An energy balance on the low temperature end of the heat exchanger gives

$$Q_3 + Q_9 = Q_4 \quad \text{III-26}$$

where

$$Q_3 = w_a \left[h_{a3} + Y_1 (1-\eta) h_{c3} \right] \quad \text{III-27}$$

$$Q_9 = w_o h_{o9} \quad \text{III-28}$$

$$Q_4 = w_a \left[h_{a4} + \gamma_1 (1-\eta) h_{c4} \right] + w_o h_{o4} \quad \text{III-29}$$

Substituting Equation III-27 through III-29 into Equation III-26 and rearranging the terms gives

$$\frac{w_o}{w_a} = \frac{(h_{a3} - h_{a4}) + \gamma_1 (1-\eta)(h_{c3} - h_{c4})}{(h_{o4} - h_{o9})} \quad \text{III-30}$$

where, assuming perfect gas behavior

$$h_{a3} - h_{a4} = c_{pa} (T_3 - T_4) \quad \text{III-31}$$

$$h_{c3} - h_{c4} = c_{pc} (T_3 - T_4) \quad \text{III-32}$$

$$h_{o4} - h_{o9} = \lambda_o + c_{po} (T_4 - T_9) = \lambda_o + c_{po} \left[(T_3 - T_9) - (T_3 - T_4) \right] \quad \text{III-33}$$

Equation III-30 may now be rewritten in terms of temperatures to give

$$T_3 - T_4 = \frac{\left(\frac{w_o}{w_a} \right) \left[\lambda_o + c_{po} (T_3 - T_9) \right]}{\left(\frac{w_o}{w_a} \right) c_{po} + c_{pa} + \gamma_1 (1-\eta) c_{pc}} \quad \text{III-34}$$

The cold end exit temperature, T_3 , equal to the carbon dioxide saturation temperature at a partial pressure of $p_{c3} = (1-\eta) p_{c1}$.

The range of operating pressures will probably be 0.5 to 1.0 atm for the total pressure and 3.8 to 7.6 mm Hg for the carbon dioxide partial pressure. The air inlet temperature and dew point to the freeze-out system will be about 35°F, and the liquid oxygen will probably be stored at a pressure of about 75 psia. The liquid will be throttled down to system operating pressure before entering the heat exchanger. The latent heat of vaporizations are 75.8 and 77.8 Btu per lb, and the heat sink temperatures are 151 and 162°R at the system pressures of 0.5 and 1.0 atm respectively. These numbers, together with Equation III-24 reveal that at high carbon dioxide concentrations and low temperature differences, the system is capable of operation at an oxygen flow to metabolic oxygen flow ratio of unity ($w_o/w_{om} = 1.0$).

A temperature difference of about 10°R is required for a reasonable system design. A lower temperature difference permits operation of low oxygen flow rates but requires very high heat exchanger effectiveness. If the required effectiveness is too high, the heat exchanger will not

only be very large, but will also be very susceptible to a reduction in performance because of heat leaks. A higher temperature difference decreases the heat exchanger size and heat leak problem, but greatly increases the oxygen flow requirements.

A thermodynamic cycle analysis of the system was carried out for a one man system by using Equations III-20, III-22, and III-34. A warm end temperature difference ($T_1 - T_6$) of 10°R , and metabolic flow rates of 2.25 lb of carbon dioxide and 2.0 lb of oxygen per man day were used. A summary of the calculations is given in Table I.

TABLE I

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM I
(SIMPLE FREEZE-OUT WITH WATER REMOVAL)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		166	81.6	325	161
Oxygen Flow/Air Flow, lb O ₂ per lb air		.0156	.0158	.0159	.0161
Oxygen Flow/Metabolic Oxygen Flow		1.295	0.645	2.58	1.295
Temperatures, °R	T ₁	495	495	495	495
	T ₂	360	360	360	360
	T ₃	238	245	238	245
	T ₄	231	238	232	238
	T ₅	352	352	352	352
	T ₆ , T ₇ , T ₈	485	485	485	485
	T ₉	151	151	162	162
Heat Exchanger Effectiveness, E		.975	.975	.975	.975
Number of Heat Transfer Units, NTU		39	39	39	39

Notes:

1. The carbon dioxide removal efficiency was assumed to be 90 per cent.
2. The heat exchanger capacity rate ratio was assumed to be unity.
3. See Figure 9 for a schematic diagram and Figure 11 for a flow chart of System I.

- b. System II. Simple Freeze-out with Water Vapor Recovery - This process is shown schematically in Figure 10. The system is composed of two water freeze-out heat exchangers in which the flow is alternated back and forth, a heater, one carbon dioxide freeze-out switching heat exchanger, and twenty butterfly control valves. Air enters one of the water freeze-out heat exchangers where the water is frozen out in the freeze-out passage and is held on the heat transfer surfaces. The air then flows through the freeze-out channel of the carbon dioxide freeze-out switching heat exchanger where the carbon dioxide is frozen out. The air then mixes with liquid oxygen and flows back through the switching heat exchanger in the regenerative channel. Sublimation of solid carbon dioxide from the previous cycle takes place simultaneously in the sublimation channel. The switching heat exchanger operates in the same manner as in the system with water removal (Figure 9). After leaving the switching heat exchanger, the air enters the regenerative passage of the water freeze-out heat exchanger. Next, the air is heated so that clean up of the ice in the second water freeze-out heat exchanger may take place. When the ice is vaporized, the valves are reversed so that the water freeze-out and clean up processes exchange places. The water freeze-out heat exchangers may be operated independently of the carbon dioxide freeze-out switching heat exchanger.

A cycle analysis of the system may be accomplished by referring to the flow chart in Figure 12, and by substituting the rate equations into several energy balance equations. Starting with the carbon dioxide freeze-out switching heat exchanger and the liquid oxygen supply, an energy balance gives

$$Q_2 + Q_9 = Q_5 + Q_{10} \quad \text{III-35}$$

where

$$Q_2 = w_a (h_{a2} + y_1 h_{c2}) \quad \text{III-36}$$

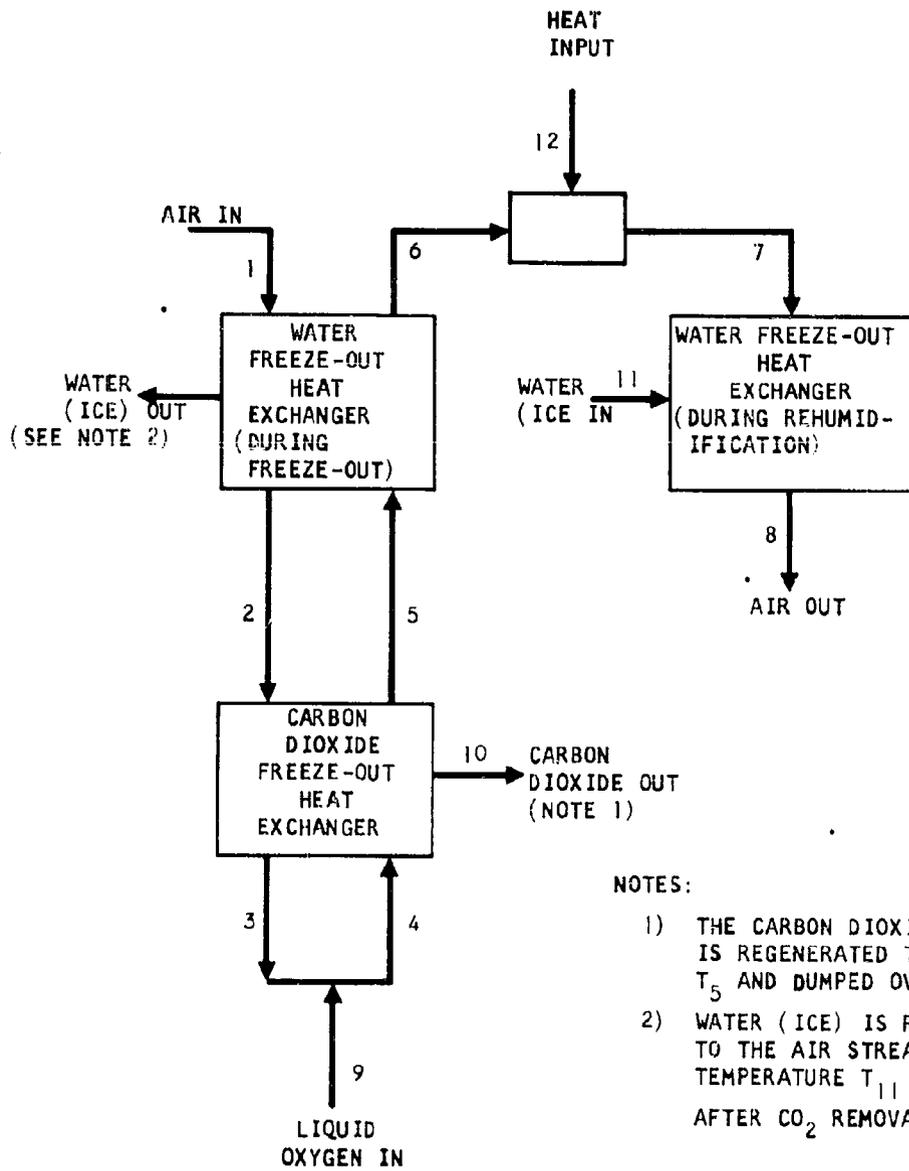
$$Q_5 = w_a [h_{a5} + (1 - \eta) y_1 h_{c5}] + w_o h_{o5} \quad \text{III-37}$$

$$Q_9 = w_o h_{o9} \quad \text{III-38}$$

$$Q_{10} = w_a \eta y_1 h_{c5} \quad (\text{CO}_2 \text{ is regenerated to } T_5) \quad \text{III-39}$$

Equation III-35 may now be written as

$$\frac{w_o}{w_a} = \frac{(h_{a2} - h_{a5}) + y_1 [h_{c2} - (1 - \eta) h_{c5} - \eta h_{c5}]}{h_{o5} - h_{o9}} \quad \text{III-40}$$



- NOTES:
- 1) THE CARBON DIOXIDE IS REGENERATED TO TEMPERATURE T_5 AND DUMPED OVERBOARD
 - 2) WATER (ICE) IS RETURNED TO THE AIR STREAM AT TEMPERATURE $T_{11} = (492 + T_2) / 2$ AFTER CO_2 REMOVAL

Figure 12. Flow Chart for Simple Freeze-out with Water Recovery

where, assuming perfect gas behavior,

$$h_{a2} - h_{a5} = c_{pa} (T_2 - T_5) \quad \text{III-41}$$

$$h_{c2} - (1 - \eta) h_{c5} - \eta h_{c5} = h_{c2} - h_{c5} = c_{pc} (T_2 - T_5) \quad \text{III-42}$$

$$h_{o5} - h_{o9} = \lambda_o + c_{po} (T_5 - T_9) \quad \text{III-43}$$

Substituting Equations III-41 through III-43 into III-40 and collecting the terms gives

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc})(T_2 - T_5)}{\lambda_o + c_{po} (T_5 - T_9)} \quad \text{III-44}$$

An energy balance on the water freeze-out heat exchanger during freeze-out is

$$Q_1 + Q_5 = Q_2 + Q_6 + Q_{11} \quad \text{III-45}$$

where

$$Q_1 = w_a [h_{a1} + y_1 h_{c1} + H_1 h_{w1}] \quad \text{III-46}$$

$$Q_2 = w_a (h_{a2} + y_1 h_{c2}) \quad \text{III-47}$$

$$Q_5 = w_a [h_{a5} + (1 - \eta) y_1 h_{c5}] + w_o h_{o5} \quad \text{III-48}$$

$$Q_6 = w_a [h_{a6} + (1 - \eta) y_1 h_{c6}] + w_o h_{o6} \quad \text{III-49}$$

$$Q_{11} = w_a H_1 h_{w11} \quad \text{III-50}$$

In equation III-50, the ice temperature varies from 492°R to T_2 . The average is about $(492 + T_2)/2$. Equation III-45 may now be written

$$w_a [(h_{a1} - h_{a2}) + y_1 (h_{c1} - h_{c2}) + H_1 (h_{w1} - h_{w11}) - (h_{a6} - h_{a5}) - (1 - \eta) y_1 (h_{c6} - h_{c5})] = w_o (h_{o6} - h_{o5}) \quad \text{III-51}$$

where, assuming perfect gas behavior

$$h_{a1} - h_{a2} = c_{pa} (T_1 - T_2) \quad \text{III-52}$$

$$h_{c1} - h_{c2} = c_{pc} (T_1 - T_2) \quad \text{III-53}$$

$$h_{w1} - h_{w11} = \lambda_1 + \frac{c_{p1}}{2} (492 - T_2) + c_{pw} (T_1 - 492) \quad \text{III-54}$$

$$h_{a6} - h_{a5} = c_{pa} (T_6 - T_5) \quad \text{III-55}$$

$$h_{o6} - h_{o5} = c_{po} (T_6 - T_5) \quad \text{III-56}$$

Equations III-52 through III-56 may be substituted into Equation III-51 to give

$$\frac{w_o}{w_a} = \frac{(c_{pa} + \gamma_1 c_{pc})(T_1 - T_2) + H_1 \left[\lambda_1 + \frac{c_{p1}}{2} (492 - T_2) + c_{pw} (492 - T_2) \right] - [c_{pa} + (1 - \eta) \gamma_1 c_{pc}] (T_6 - T_5)}{c_{po} (T_6 - T_5)} \quad \text{III-57}$$

An energy balance on the rehumidification process is

$$Q_7 + Q_{11} = Q_8 \quad \text{III-58}$$

where,

$$Q_7 = w_a [h_{a7} + (1 - \eta) \gamma_1 h_{c7}] + w_o h_{o7} \quad \text{III-59}$$

$$Q_8 = w_a [h_{a8} + (1 - \eta) \gamma_1 h_{c8} + H_1 h_{w8}] + w_o h_{w8} \quad \text{III-60}$$

Using the same procedure as above, the energy balance becomes

$$T_7 - T_8 = \frac{H_1 \left[\lambda_1 + \frac{c_{p1}}{2} (492 - T_2) + c_{pw} (T_8 - 492) \right]}{c_{pa} + \gamma_1 (1 - \eta) c_{pc} + c_{po} \left(\frac{w_o}{w_a} \right)} \quad \text{III-61}$$

The system may now be analyzed with the aid of the flow chart in Figure 12 and Equations III-34, III-44, III-57 and III-61. The procedure is as follows: assume $T_6 = 485^\circ\text{R}$ ($T_1 - T_6 = 10^\circ\text{R}$), assume T_5 , calculate w_o/w_a from Equation III-57, check w_o/w_a with Equation III-61. When agreement is obtained, calculate T_7 from Equation III-61. Finally, calculate the heater input $Q_{12} = (w_a c_{pa} + w_o c_{po}) (T_7 - T_6)$. The remaining temperatures may be determined as follows: Start with

$T_1 = 495^\circ\text{R}$, assume $T_2 = 360^\circ\text{R}$ (the air is essentially dry at this temperature), T_3 equals the carbon dioxide saturation temperature at $P_{c3} = (1 - \eta) P_{c1}$, calculate T_4 from Equation III-34, and assume $T_8 = 500^\circ\text{R}$. The outlet temperature, T_8 , must be equal to or greater than the inlet temperature, T_1 , to accomplish rehumidification. A reasonable outlet temperature is about 5°R higher than the inlet temperature. A summary of the thermodynamic cycle analysis calculations for the simple freeze-out system with water recovery is shown in Table II.

TABLE II

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM II
(SIMPLE FREEZE-OUT WITH WATER RECOVERY)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		166	81.6	325	161
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0867	.0911	.0525	.0515
Oxygen Flow/Metabolic Oxygen Flow		7.20	3.72	8.53	4.15
Heater Input, watts per man		29.4	14.5	35.9	18.0
Temperatures, °R	T ₁	495	495	495	495
	T ₂	360	360	360	360
	T ₃	238	245	238	245
	T ₄	204	208	217	224
	T ₅ , T ₁₀	317	316	334	335
	T ₆	485	485	485	485
	T ₇	542	542	521	521
	T ₈	500	500	500	500
	T ₉	151	151	162	162
	T ₁₁	426	426	426	426
	Water	Effectiveness	.759	.752	.838
Freeze-Out Heat Exchanger	Capacity Rate Ratio	1.24	1.25	1.12	1.13
	No. of Heat Transfer Units	5.84	5.68	8.00	9.23
Carbon Dioxide	Effectiveness	.782	.756	.853	.845
Freeze-Out Heat Exchanger	Capacity Rate Ratio	.925	.923	.955	.956
	No. of Heat Transfer Units	3.14	2.80	5.23	4.89

NOTES:

1. The carbon dioxide removal efficiency was assumed to be 90 percent.
2. See Figure 10 for a schematic diagram and Figure 12 for a flow chart of System II.

3. Freeze-Out Plus Adsorption - Freeze-out plus adsorption may be accomplished by either freezing out the water followed by adsorption of the carbon dioxide or by adsorption of the water followed by freeze-out of the carbon dioxide. If the water is frozen out, two systems are possible - one with water removal and the other with water recovery. If the water is adsorbed, only one system is necessary because water recovery may be accomplished without increasing the amount of the liquid oxygen heat sink. Many adsorbants are possible, but the only feasible materials appear to be silicon dioxide (silica gel) for the water adsorption and synthetic zeolites molecular sieves) for the carbon dioxide adsorption. Adsorption of contaminants (chemical reaction with the water and carbon dioxide) has not been considered. A description and cycle analysis of each of the three freeze-out plus adsorption systems are given below.

a. System III. Freeze-out and Removal of Water plus Adsorption of Carbon Dioxide - This system consists of one regenerative switching heat exchanger and two molecular sieve carbon dioxide adsorbers as shown schematically in Figure 13. Air enters the water freeze-out switching heat exchanger where the air is cooled to a temperature of 360°R and the water is frozen out. The operation of the heat exchanger is the same as in the simple freeze-out system. The air then enters one of the molecular sieve canisters where the carbon dioxide is removed. Two adsorbers are required--one to adsorb the carbon dioxide while the other is being desorbed of carbon dioxide from the previous cycle. The molecular sieve bed is desorbed by venting to the vacuum of space. The sieve bed has sufficient thermal capacity to vaporize the carbon dioxide provided the sieve is properly designed. The freeze-out heat exchanger may be switched independently of the molecular sieves.

A cycle analysis of the system may be accomplished by referring to the flow chart in Figure 14. The procedure is to write the rate equations and substitute them into energy balances. The energy balance for the water freeze-out heat exchanger and the liquid oxygen supply is

$$Q_1 + Q_9 + Q_{10} = Q_4 + Q_8 \quad \text{III-62}$$

Neglecting the heat leak, Q_{10} , assuming that the water is regenerated to a vapor at temperature T_4 before being dumped overboard, and following the same procedure as used for the simple freeze-out system analysis, the energy balance may be written as

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc} + H_1 c_{pw})(T_1 - T_4)}{\lambda_o + c_{po}(T_4 - T_9)} \quad \text{III-63}$$

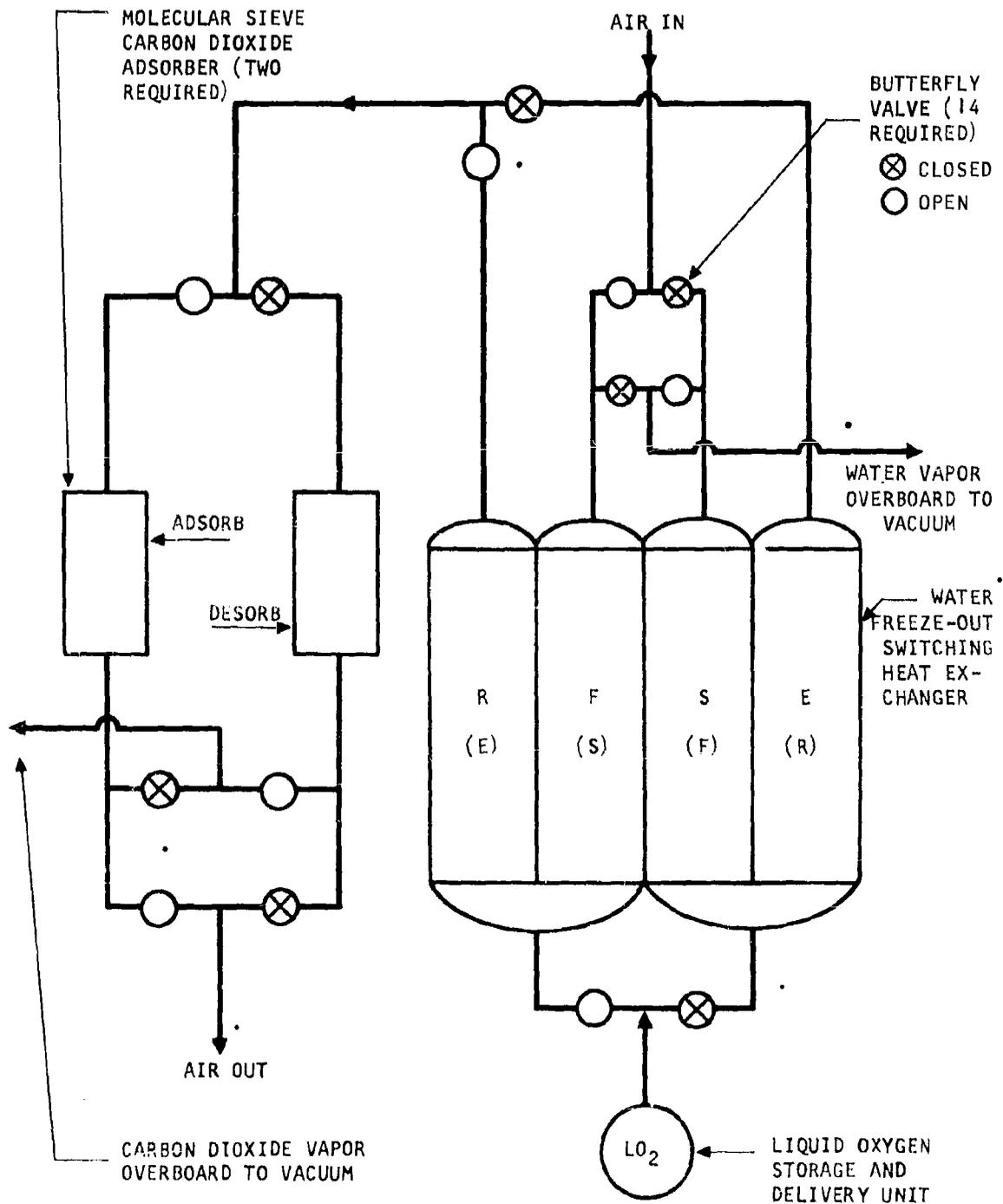


Figure 13. Schematic Diagram for System III (Water Freeze-Out Plus Carbon Dioxide Adsorption with Water Removal)

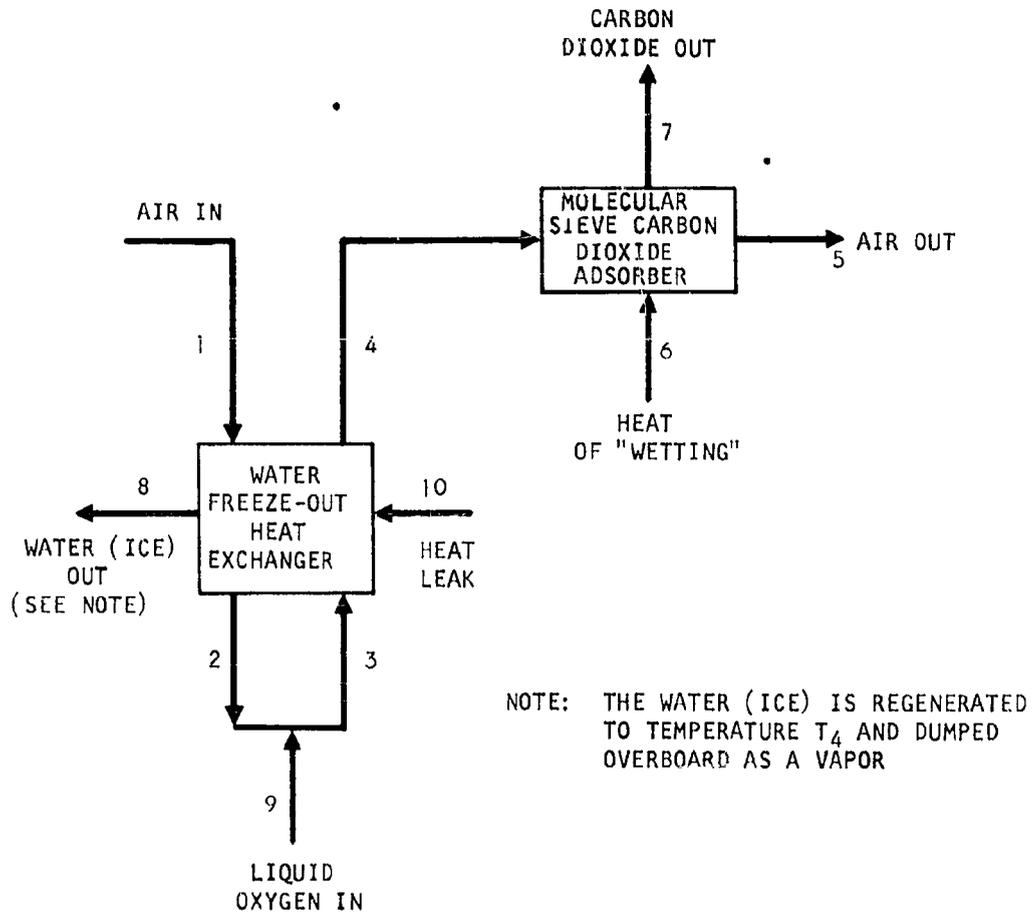


Figure 14. Flow Chart for System III
(Water Freeze-Out Plus Adsorption of Carbon Dioxide
With Water Removal)

An energy balance for the low temperature end of the heat exchanger gives

$$T_2 - T_3 = \frac{\frac{w_o}{w_a} [\lambda_o + c_{po} (T_2 - T_9)]}{\frac{w_o}{w_a} c_{po} + c_{pa} + y_1 c_{pc}} \quad \text{III-64}$$

An energy balance for the molecular sieve (neglecting the change in thermal capacity of the sieve) is

$$Q_4 + Q_6 = Q_5 + Q_7 \quad \text{III-65}$$

where

$$Q_4 = w_a (h_{a4} + y_1 h_{c4}) + w_o h_{o4} \quad \text{III-66}$$

$$Q_5 = w_a h_{a5} + w_o h_{o5} \quad \text{III-67}$$

$$Q_6 = Q_{\text{wetting}} - w_a y_1 q_c \quad \text{III-68}$$

$$Q_7 = w_a y_1 h_{c7} \quad \text{III-69}$$

Equation III-66 through III-69 may be substituted into Equation III-65 to give

$$w_a [(h_{a5} - h_{a4}) - y_1 (h_{c4} - h_{c7} + q_c)] + w_o (h_{o5} - h_{o4}) = 0 \quad \text{III-70}$$

where

$$h_{a5} - h_{a4} = c_{pa} (T_5 - T_4) \quad \text{III-71}$$

$$h_{c4} - h_{c7} + q_c = \lambda_c + \frac{Q_{\text{adsorption}}}{w_a} \quad \text{III-72}$$

$$h_{o5} - h_{o4} = c_{po} (T_5 - T_4) \quad \text{III-73}$$

Equations III-71 through III-73 may now be substituted into Equation III-70 to give

$$T_5 - T_4 = \frac{y_1 Q_{\text{adsorb}}}{c_{pa} + \frac{w_o}{w_a} c_{po}} \quad \text{III-74}$$

Equations III-63, III-64 and III-74 may now be used to calculate the temperature and flow rates in the system. As in the simple freeze-out systems, assume that the heat exchanger warm end temperature difference ($T_1 - T_4$) is 10°R . The heat of adsorption of the molecular sieve may be taken as 300 Btu per lb, and the carbon dioxide removal efficiency may be taken as 100 per cent provided the sieves are desorbed before their break-through point is reached. A summary of the cycle analysis calculations is given in Table III.

TABLE III

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM III
(WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION WITH WATER REMOVAL)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{cl} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		149	73.5	292	145
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0156	.0158	.0159	.0160
Oxygen Flow/Metabolic Oxygen Flow		1.16	0.581	2.32	1.16
Temperatures, °K	T ₁	495	495	495	495
	T ₂	360	360	360	360
	T ₃	352	352	352	352
	T ₄ , T ₈	485	485	485	485
	T ₅	504	524	495	504
	T ₈	485	485	485	485
	T ₉	151	151	162	162
Heat Exchanger Effectiveness		.944	.944	.944	.944
No. of Heat Transfer Units		16.9	16.9	16.9	16.9

NOTES:

- (1) The carbon dioxide removal efficiency was assumed to be 100 percent.
- (2) The heat exchanger capacity rate was assumed to be unity.
- (3) See Figure 13 for a schematic diagram and Figure 14 for a flow chart of System III.

- b. System III. Freeze-out and Recovery of Water plus Adsorption of Carbon Dioxide - This system consists of two water freeze-out heat exchangers, one heater, and two molecular sieves for the adsorption of carbon dioxide as shown in Figure 15. One heat exchanger is used for water freeze out while the other is used for rehumidification. When the rehumidifying heat exchanger is purged of ice the valves are reversed so that ice filled dehumidifying heat exchanger may be purged. The two molecular sieves may be operated independently of the heat exchangers. One sieve adsorbs carbon dioxide while the other is being desorbed to the vacuum of space.

A cycle analysis of the system may be accomplished by referring to the flow chart in Figure 16. By substituting the various rate equations into energy balance equations in the same manner as before, the following equations may be derived:

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc})(T_1 - T_4) + H_1 \left[\lambda_1 + \frac{c_{pi}}{2}(492 - T_2) + c_{pw}(T_7 - 492) \right]}{\lambda_o + c_{po}(T_4 - T_9)} \quad \text{III-75}$$

$$T_6 - T_7 = \frac{H_1 \left[\lambda_1 + \frac{c_{pi}}{2}(492 - T_2) + c_{pw}(T_7 - 492) \right]}{c_{pa} + c_{po} \left(\frac{w_o}{w_a} \right)} \quad \text{III-76}$$

$$Q_{12} = (w_a c_{pa} + w_o c_{po})(T_6 - T_5) \quad \text{III-77}$$

Equations III-64, III-74, III-75, III-76, and III-77 may be used to calculate the flow rates and temperatures for the system. As in previous systems the heat exchanger warm end temperature difference ($T_1 - T_4$) may be taken as 10°R . The ice freeze-out temperature (T_2) and the air outlet temperature may be taken as 360°R and 500°R respectively. A summary of the cycle analysis calculations are given in Table IV.

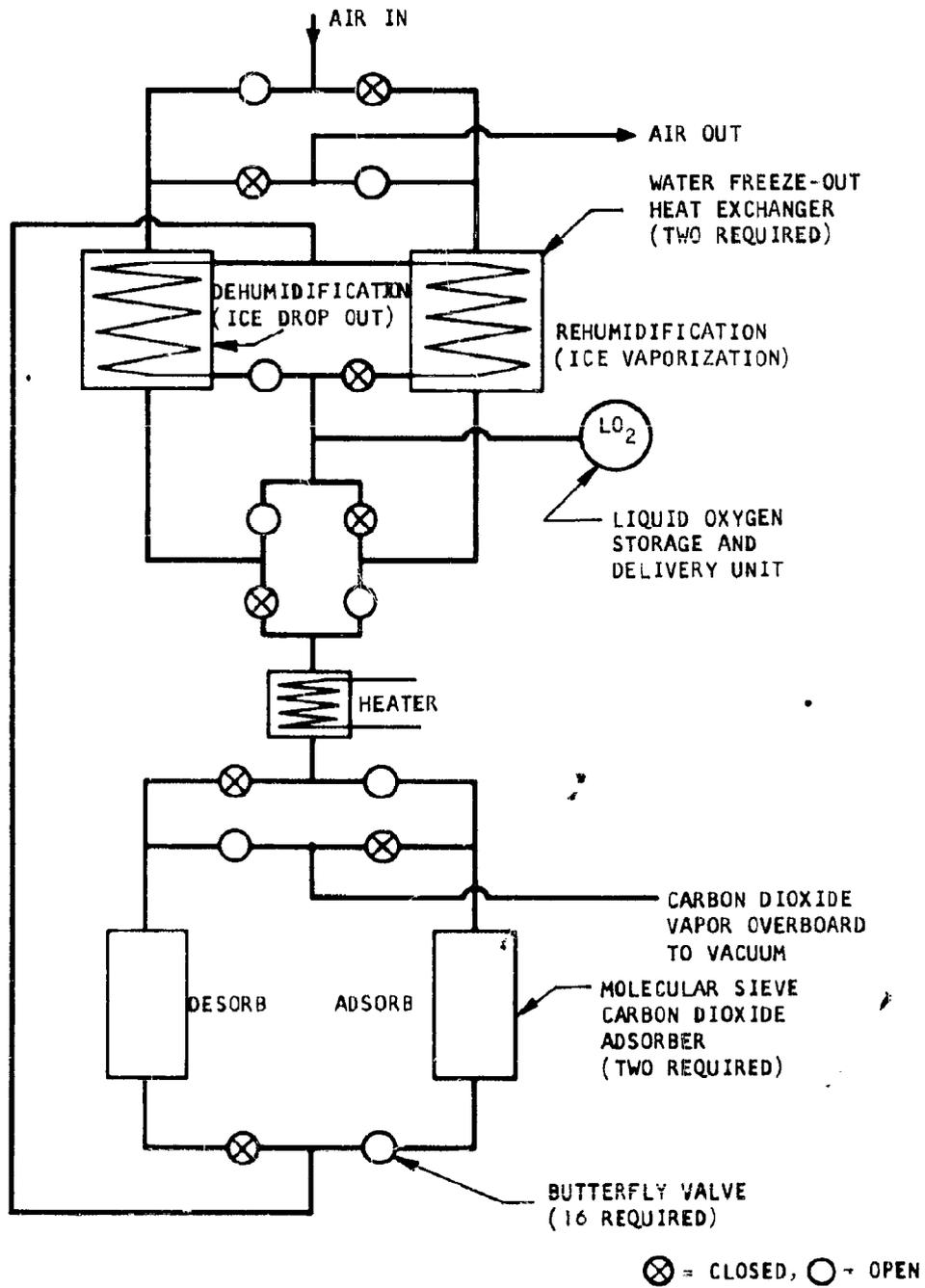


FIGURE 15. SCHEMATIC DIAGRAM FOR SYSTEM IV (WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION WITH WATER RECOVERY)

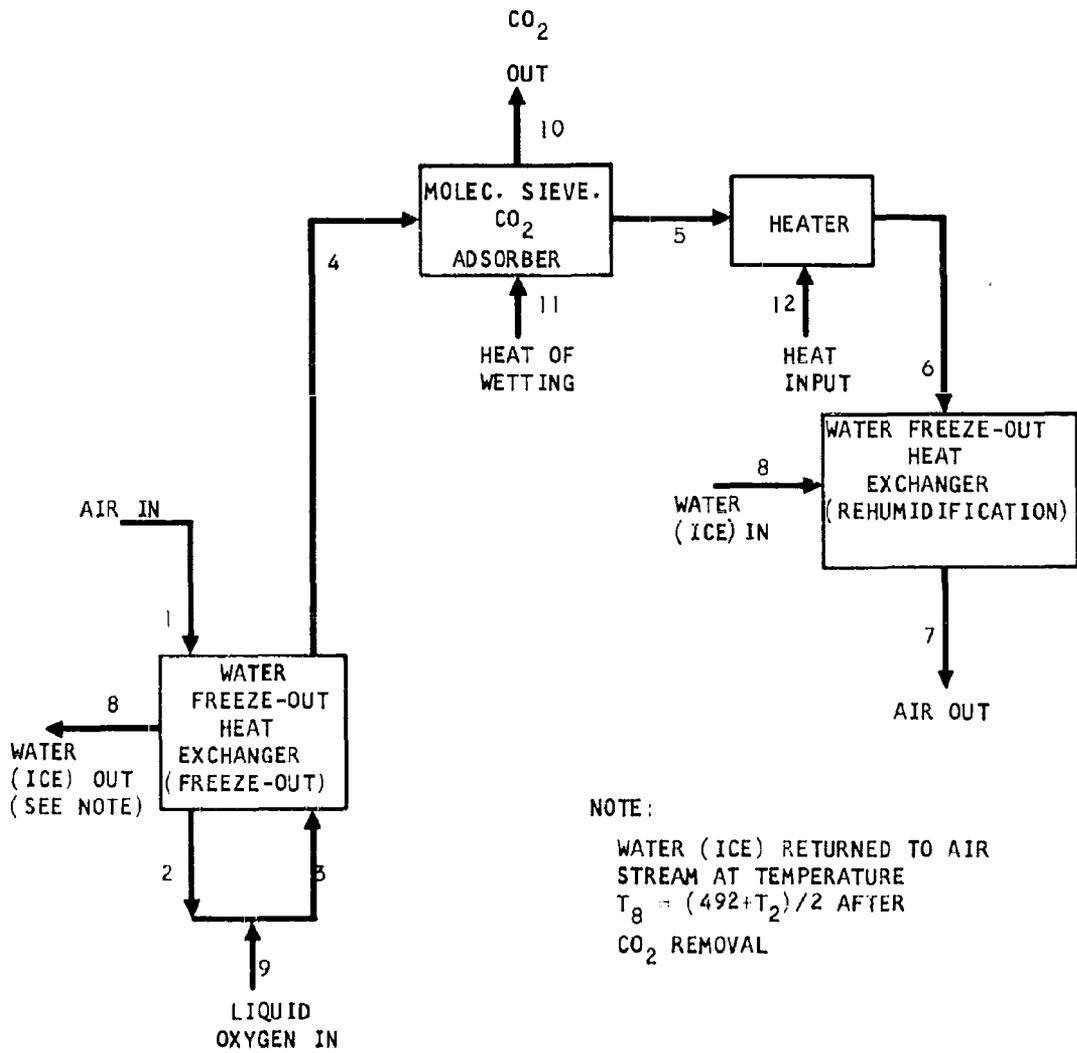


Figure 16. Flow Chart for System IV
(Water Freeze-Out Plus Carbon Dioxide Adsorption with Water Recovery)

TABLE IV

SUMMARY OF CYCLE ANALYSIS FOR A SYSTEM IV
(WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION WITH WATER RECOVERY)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		149	73.5	292	145
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0835	.0844	.0510	.0514
Oxygen Flow/Metabolic Oxygen Flow		6.29	3.10	7.44	3.73
Heater Input, watts per man		17.9	4.6	24.2	8.0
Temperature, °R	T ₁	495	495	495	495
	T ₂	360	360	360	360
	T ₃	318	318	335	335
	T ₄	485	485	485	485
	T ₅	503	521	494	504
	T ₆	542	542	521	522
	T ₇	500	500	500	500
	T ₈	426	426	426	426
	T ₉	151	151	162	162
Heat Exchanger Effectiveness		.762	.762	.843	.843
Capacity Rate Ratio		1.25	1.26	1.12	1.13
Number of Heat Transfer Units		6.40	6.85	8.45	9.15

NOTES:

- (1) The carbon dioxide removal efficiency was assumed to be 100%.
- (2) See Figure 15 for a schematic diagram and Figure 16 for a flow chart of System IV.

- c. System V. Adsorption of Water Plus Freeze-Out of Carbon Dioxide - This system consists of two silica gel water adsorbers, a carbon dioxide freeze-out switching heat exchanger and a heater as shown in Figure 17. The heat exchanger operates in the same manner as in the simple freeze-out systems, and the air flow to the silica gel beds are cycled to accomplish the adsorption and desorption. The silica gel beds and the freeze-out heat exchanger may be operated independently of each other. The air to the silica gel bed must be heated to a temperature of about 710°R (250°F) before desorption can take place. Also the heat of adsorption (heat of wetting plus latent heat of vaporization) is about 1400 Btu per lb of water adsorbed. Some of this heat is transferred to the silica gel bed so that the net amount of heat of adsorption which is dissipated to the air stream is about 1200 Btu per lb of water adsorbed.

At the start of the adsorption part of the cycle, the silica gel bed temperature is 710°R (250°F), and it would appear that cooling of the bed is necessary prior to starting the adsorption process. However, pre-cooling is not necessary because the water adsorption front moves through the bed at a slower rate than the air cooling front. Thus, at the beginning of the adsorption part of the cycle, the air outlet temperature will be 710°R. After a short time, the air temperature will drop to a steady-state energy balance value, and the bed will be cool during most of the adsorption part of the cycle. During the desorption part of the cycle, the hot inlet air is partly cooled by the vaporization of the water and partly by the heat-absorbing capacity of the cool bed.

A consequence of the high temperature of desorption is that the outlet air temperature will rise rapidly to 710°R at the end of each cycle. Then, when the silica gel bed control valves are reversed, the outlet air temperature will fall rapidly to the normal temperature of about 525°R (65°F). Although a cycle analysis is rather complex, because of the transient nature of the system, a simplified analysis may be accomplished by neglecting the thermal capacity of the silica gel beds and by supplementary cooling takes place at the outlet of the silica gel beds as shown in the flow chart in Figure 18.

An energy balance on the silica gel bed during the adsorption process gives

$$T_2 - T_1 = \frac{H_1 Q_{\text{adsorb}}}{(c_{pa} + y_1 c_{pc})} \quad \text{III-78}$$

and an energy balance during the desorption process (neglecting the small amount of residual carbon dioxide in the air) gives

$$T_6 - T_7 = \frac{H_1 \left[Q_{\text{adsorb}} + c_{pw} \left(T_7 - \frac{T_1 + T_2}{2} \right) \right]}{c_{pa} + \frac{w_o}{w_a} c_{po}} \quad \text{III-79}$$

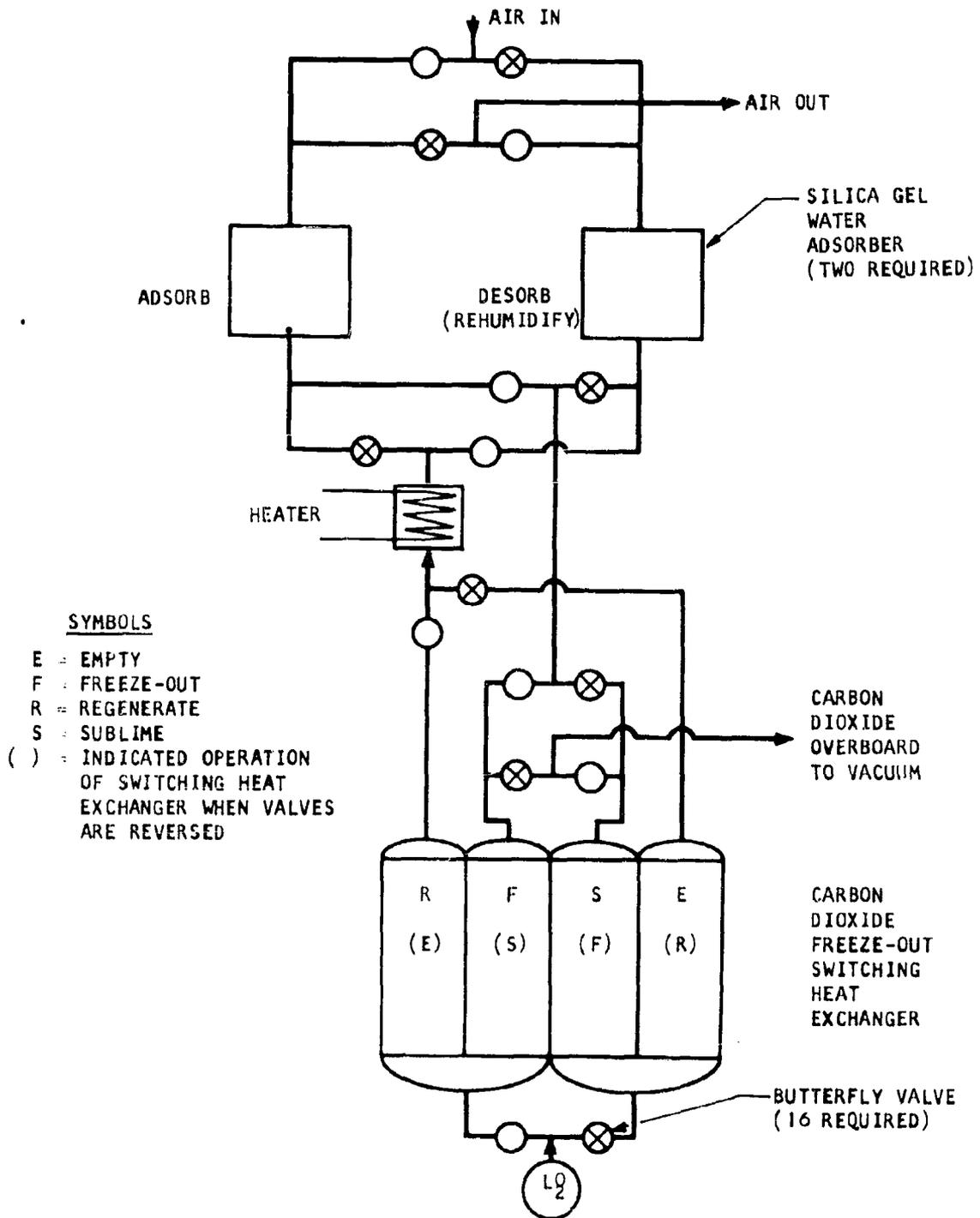


FIGURE 17. SCHEMATIC DIAGRAM FOR SYSTEM V
(WATER ADSORPTION PLUS CARBON DIOXIDE FREEZE-OUT)

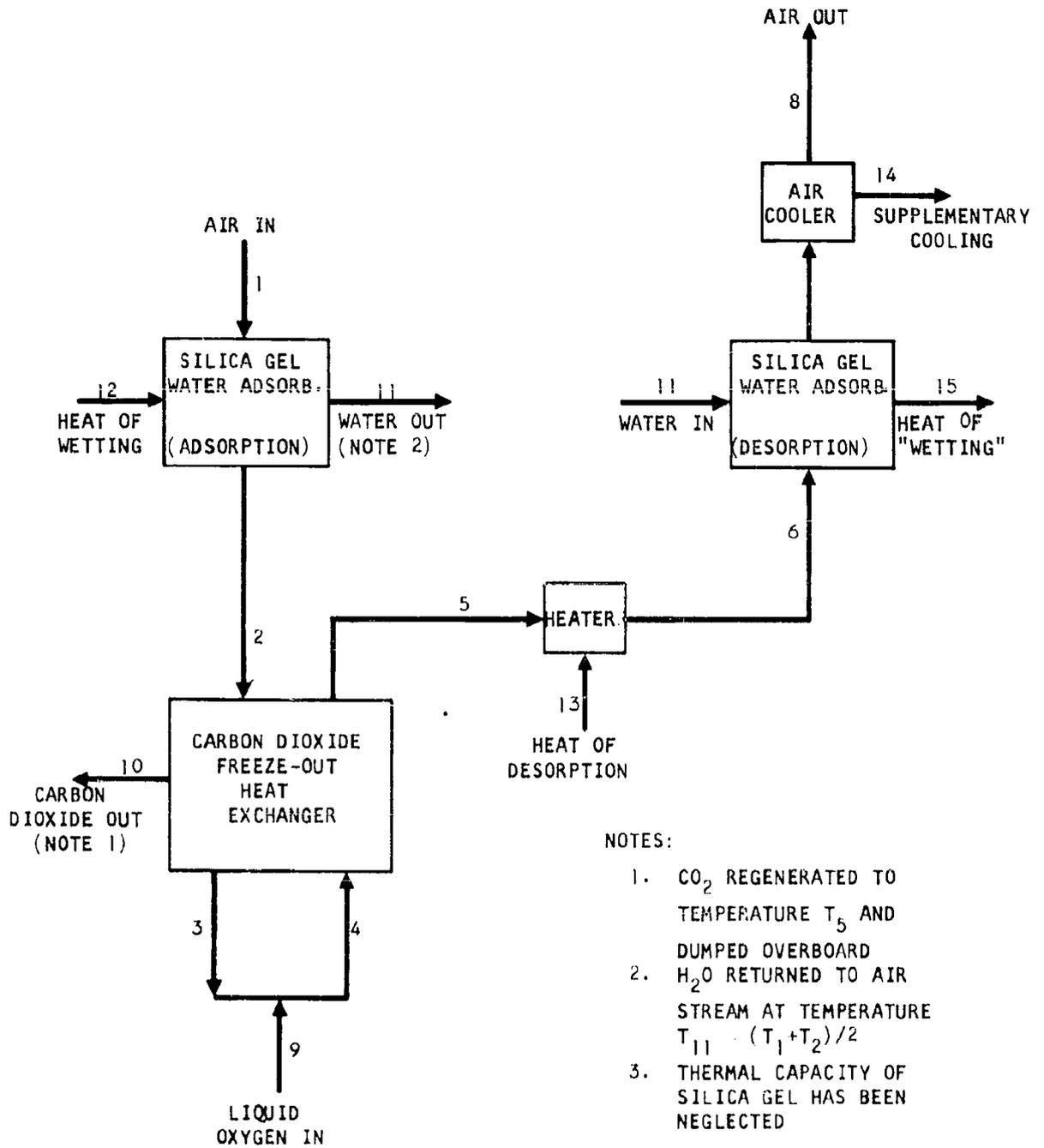


Figure 18. Flow Chart (Simplified) for System V (Water Adsorption Plus Carbon Dioxide Freeze-Out)

The heater (Q_{13}) and supplementary cooling (Q_{14}) loads may be expressed as

$$Q_{13} = (w_a c_{pa} + w_o c_{po})(T_6 - T_5) \quad \text{III-80}$$

$$Q_{14} = \left[w_a (c_{pa} + H_l c_{pw}) + w_o c_{po} \right] (T_7 - T_8) \quad \text{III-81}$$

The ratio of oxygen flow to air flow (w_o/w_a) may be computed from Equation III-44, and temperature difference ($T_3 - T_4$) at the low temperature end of the carbon dioxide freeze-out heat exchanger may be computed from Equation III-34. The following temperatures were assumed:

An inlet (T_1) = 495°R, temperature difference ($T_2 - T_5$) and the warm end of the carbon dioxide freeze-out heat exchanger = 10°R, cold temperature (T_3) out of the carbon dioxide freeze-out heat exchanger = T_{sat} at

$P_{c3} = (1-\eta) P_{c1}$, silica gel desorbing temperature (T_6) = 710°R, air outlet temperature = 500°R and liquid oxygen heat sink temperature (T_9) = 151°R at 7.35 psia and 162 at 14.7 psia. The carbon dioxide removal efficiency was taken as 90 per cent, and the heat of adsorption (also desorption) was taken as 1200 Btu per lb.

TABLE V

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM V
(WATER ADSORPTION PLUS CARBON DIOXIDE FREEZE-OUT)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		166	81.6	325	161
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0143	.0145	.0152	.0154
Oxygen Flow/Metabolic Oxygen Flow		1.19	0.583	2.47	1.24
Heater Input, watts per man		87.5	43.2	201	100
Supplementary Cooling, watts per man		80.2	39.5	182	90
Temperatures, °R	T ₁	495	495	495	495
	T ₂	538	537	511	511
	T ₃	238	245	238	245
	T ₄	232	239	232	239
	T ₅ , T ₁₀	528	527	501	501
	T ₆	710	710	710	710
	T ₇	664	664	687	687
	T ₈	500	500	500	500
	T ₉	151	151	162	162
	T ₁₁	517	516	503	503
	Heat Exchanger Effectiveness		.980	.980	.978
No. of Heat Transfer Units		49	49	44.5	44.5

Notes:

1. The carbon dioxide removal efficiency was assumed to be 90 per cent.
2. The heat exchanger capacity rate ratio was assumed to be unity.
3. See Figure 17 for a schematic diagram and Figure 18 for a flow chart of System V.

4. Systems With Auxillary Cooling - All of the systems which have been described could use some form of auxillary cooling. The advantages of auxillary cooling are a reduction in heat exchanger size and a reduction in the amount of liquid oxygen required for freeze-out, especially when water recovery is desired. Auxillary cooling may be obtained with expanders, mechanical refrigeration (vapor cycle or gas cycle) or auxillary heat sinks. Auxillary cooling, which is obtained by use of expanders or mechanical refrigeration should be accomplished as near to the warm temperature end of the freeze-out system as possible. The reason for this is the auxillary equipment weight and power requirements increase as the temperature level of cooling is decreased. Unfortunately, the presence of frozen out contaminants complicates the problem. Piston expanders will become clogged with or damaged by the frozen out solids if the expanders are located where freeze-out occurs. Therefore, piston expanders should be placed in the system where frozen out solids are not present. For example, in simple freeze-out systems, the solids free region is located between the ice and carbon dioxide freeze-out heat exchangers where water is completely frozen out at a temperature of about 360°R , and carbon dioxide does not start to freeze-out until a temperature of about 270°R is reached. In water freeze-out plus carbon dioxide adsorption systems the expander should be located between the water freeze-out heat exchanger and the carbon dioxide adsorber.

Turbine expanders, however, may be designed to operate anywhere in the freeze-out system without becoming clogged with solids. A separator must then be installed downstream of the turbine to remove the very small entrained particles from the air. This introduces another complication, namely, how to make best use of the removed solids. At the warm temperature end of the system ice or snow particles may be removed and returned to the air-stream by the heat transfer matrix of a rotary heat exchanger. At the low temperature end of the system, solid carbon dioxide particles may be removed by a fine steel wool matrix which, when filled, may be replaced by a clean matrix. However, the use of expanders at the low temperature end is not desirable because of excessive power and weight penalties. Also, regeneration of the solid carbon dioxide may be a very difficult mechanical problem.

In systems in which auxillary cooling is obtained by an auxillary heat sink, the location of the sink would depend upon the type of freeze-out system and the type of heat sink which is used. For example, a liquid hydrogen or cryogenic hydrogen (such as vapors from a liquid hydrogen storage tank) heat sink may be located anywhere in the system because their temperatures are lower than that required for freeze-out of carbon dioxide.

Auxillary cooling is most useful in those systems in which water is frozen out and recovered (returned to the air stream after CO_2 removal). The reason for this is that freeze-out water recovery systems require an excessive amount of liquid oxygen for freeze-out (seven to nine times the metabolic rate). The liquid oxygen requirements may be reduced

considerably by making use of auxiliary cooling. On long duration missions, the saving in weight of liquid oxygen may more than compensate for the weight and power penalty of the auxiliary cooling system. An analysis of several methods of applying auxiliary cooling to systems in which water is frozen out and recovered and carbon dioxide is absorbed is given below. The effects of cooling by expansion will be investigated only at the cold end of the water freeze-out heat exchanger. Auxiliary cooling by mechanical vapor cycle refrigeration and by an auxiliary hydrogen heat sink will be investigated at the warm end of the ice freeze-out heat exchanger. Mechanical gas cycle refrigeration will not be considered because it is too inefficient for this type of work.

- a. System VI. Auxiliary Cooling by Use of Expanders - A schematic diagram of this system is shown in Figure 19. The inlet air is compressed and cooled before entering the water freeze-out heat exchanger. Part of the heat of compression is transferred to the outlet stream so that rehumidification may be accomplished. The remainder of the heat of compression is removed from the air stream by supplementary cooling from the cabin refrigeration system. The supplementary cooling heat exchanger is made integral with the water freeze-out heat exchanger because, at higher pressures, water condenses out at a temperature higher than 495°R. This arrangement also makes complete water recovery possible. After leaving the water freeze-out heat exchanger the air is cooled by expansion through a piston or turbine expander. The small amount of expander work may be absorbed by a friction device. The remainder of the system is the same as in the system without auxiliary cooling except the heater is replaced by the compressor heat exchanger.

A flow chart for the system is shown in Figure 20. For comparison purpose, the temperatures, as far as possible, are designated by the same numbers as in the system without auxiliary cooling (See Figure 15). The purpose of the expander is to reduce the cold end temperature (T_2) so that the amount of liquid oxygen required for water freeze-out and recovery is equal to the metabolic consumption rate (2.0 lb per man day). The required amount of auxiliary cooling with expanders, the supplementary cooling load and the power input requirements may be determined as follows:

Assume the water freeze-out inlet temperature (T_1) is 495°R and the warm end temperature difference is 10°R as in all other water freeze-out systems. Calculate the cold end inlet temperature (T_3). An energy balance on the ice freeze-out heat exchanger during freeze-out gives

$$T_3 - T_4 = \frac{(c_{pa} + Y_1 c_{pc})(T_1 - T_2') + H_1' \left[\lambda_1 + \frac{c_{pl}}{2}(492 - T_2') + c_{pw}(T_1 - 492) \right]}{\frac{w_o}{w_a} c_{po} + c_{pa} + Y_1 c_{pc}}$$

III-82

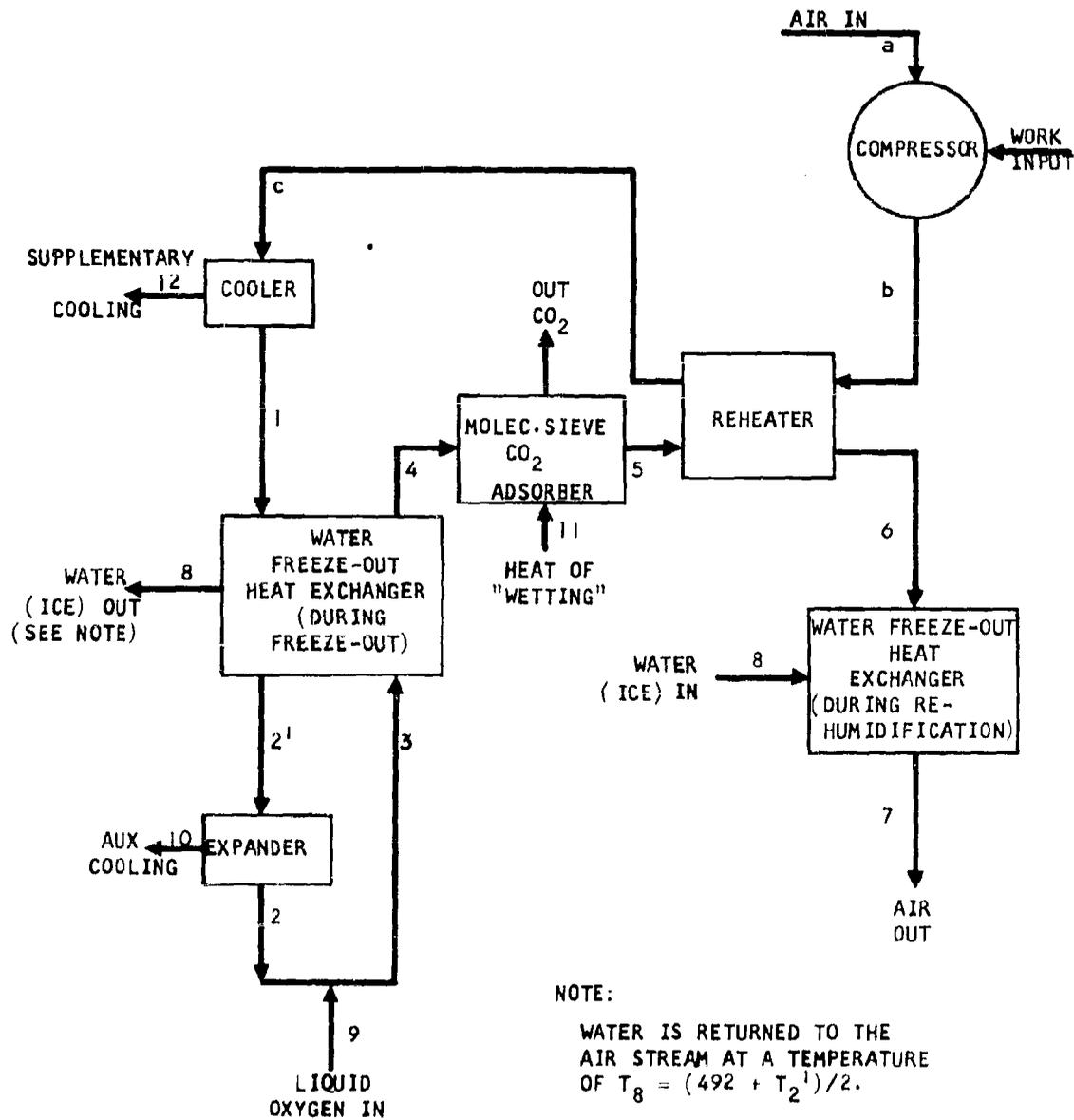


Figure 20. Flow Chart for System VI
 (Water Freeze-Out Plus Carbon Dioxide Adsorption
 With Water Recovery and With Aux. Cooling by
 use of an Expander)

where $T_2^1 = 360^\circ\text{R}$ (Temperature required for complete water freeze-out)

$$H_1^1 = H_1 \frac{p_a}{p_b} \text{ (Water content of the air as it leaves the supplementary cooler) and } w_o/w_a = 2/w_a$$

The expander outlet temperature (T_2) may be calculated by rewriting Equation III-64 to give

$$T_2 = \frac{T_3 \left(\frac{w_o}{w_a} c_{po} + c_{pa} + y_1 c_{pc} \right) + \frac{w_o}{w_a} (\lambda_o - c_{po} T_9)}{c_{pa} + y_1 c_{pc}} \quad \text{III-83}$$

The amount of auxiliary cooling may be calculated from

$$Q_{aux} = w_a (c_{pa} + y_1 c_{pc}) (T_2^1 - T_2) = w_a \bar{c}_p (T_2^1 - T_2) \quad \text{III-84}$$

The corresponding expander pressure ratio may be computed by noting that the auxiliary cooling is also equal to

$$Q_{aux} = w_a \eta_e (h_2^1 - h_{2s}) \quad \text{III-85}$$

where $h_2^1 - h_{2s}$ is the isentropic enthalpy change and η_e is the expander efficiency. Assuming perfect gas behavior

$$h_2^1 - h_{2s} = \bar{c}_p (T_2^1 - T_{2s}) = \bar{c}_p T_2^1 \left(1 - \frac{T_{2s}}{T_2^1} \right) = \bar{c}_p T_2^1 \left[1 - \left(\frac{p_2}{p_2^1} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad \text{III-86}$$

Equations III-84, 85 and 86 may be combined to give the expander pressure ratio as

$$\frac{p_2^1}{p_2} = \frac{1}{\left(1 - \frac{T_2^1 - T_2}{\eta_e T_2^1} \right)^{\frac{\gamma}{\gamma-1}}} \quad \text{III-87}$$

Neglecting the small pressure drop in the heat exchangers and plumbing, the corresponding compressor work is

$$WK_c = \frac{w_a \bar{c}_p T_a}{\eta_c} \left[\left(\frac{p_2^1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad \text{III-88}$$

where η_c is the compressor efficiency, and the compressor outlet temperature is

$$T_b = T_a + \frac{WK_c}{w_a c_p} = T_a \left[1 + \frac{\left(\frac{p_2'}{p_2} \right)^{\frac{\gamma-1}{\gamma}} - 1}{\eta_c} \right] \quad \text{III-89}$$

The outlet temperature (T_5) from the molecular sieve may be computed from Equation III-74, the air outlet temperature (T_7) may be assumed to be 500°R , the inlet temperature (T_6) to the ice freeze-out heat exchanger during rehumidification may be computed from Equation III-76, (with T_2 replaced by T_2'), and the inlet temperature (T_c) to the supplementary cooling heat exchanger may be calculated from

$$T_c = T_b - \frac{(c_{pa} + \frac{w_o}{w_a} c_{po})(T_6 - T_5)}{c_{pa} + \gamma_1 c_{pc} + H_1 c_{pw}} \quad \text{III-90}$$

The amount of supplementary cooling is

$$Q_{\text{sup}} = w_a \left[(c_{pa} + \gamma_1 c_{pc} + H_1 c_{pw})(T_c - T_1) + H_1 \lambda_w \left(1 - \frac{p_2}{p_2'} \right) \right] \quad \text{III-91}$$

The efficiencies of expanders and compressors will be quite low because the air flow rates in the freeze-out system are very small. In a one man system, the flow rates at one half an atmosphere will be within a range of about 0.05 to 0.10 lb per min. The corresponding volume flow is only 1.3 to 2.6 standard CFM. This flow is too low for high efficiencies, and an accurate determination of efficiency cannot be made at this time. However, for preliminary calculation purposes, a rough approximation is adequate, and the compressor and expander efficiencies should be about 30 per cent.

A summary of the cycle analysis calculations is given in Table VI. At the beginning of the analysis the compressor pressure ratio is not known but must be assumed. A trial and error procedure is required. Note also that it was assumed that the compressor and expander pressure ratios are equal.

TABLE VI
 SUMMARY OF CYCLE ANALYSIS FOR SYSTEM VI
 (WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION WITH WATER RECOVERY
 AND WITH AUXILIARY COOLING BY USE OF AN EXPANDER).

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H _a , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, y ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		149	73.5	292	145
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0134	.0272	.00685	.0138
Oxygen Flow/Metabolic Oxygen Flow		1.0	1.0	1.0	1.0
Compressor Input, watts per man		179	68	268	111
Auxiliary Cooling Obtained, watts per man		9.3	3.7	14.7	6.2
Supplementary Cooling Required, watts per man		170	67	253	111
Expander Pressure Ratio		2.16	1.82	1.82	1.65
TEMPERATURE, °R	T _a	495	495	495	495
	T _b	900	804	804	762
	T _c	860	783	777	743
	T ₁	495	495	495	495
	T ₂ ¹	360	360	360	360
	T ₂	338.5	343	343	345.5
	T ₃	331.5	329.5	339	338.5
	T ₄	485	485	485	485
	T ₅	504	523	495	504
	T ₆	545	544	522	522
T ₇	500	500	500	500	
T ₈	426	426	426	426	
T ₉	151	151	162	162	
Water Freeze-Out Heat Exchanger	Effectiveness	.825	.815	.865	.863
	Capacity Rate Ratio	1.15	1.18	1.09	1.10
Reheater (Z=1.0)	No. of Heat Transfer Units	8.07	8.84	9.58	9.70
	Effectiveness	.127	.075	.087	.074
Supplementary Cooler (Z=0)	No. of Heat Transfer Units	0.146	0.081	0.095	0.080
	Effectiveness	.973	.967	.965	.962
		3.60	3.45	3.40	3.30

NOTES:

- (1) The carbon dioxide removal efficiency was assumed to be 100 per cent.
- (2) The compressor and expander efficiencies were assumed to be 30 per cent.
- (3) See Figure 19 for a schematic diagram and Figure 20 for a flow chart of System VI.

b. System VII. Auxiliary Cooling by use of Vapor Cycle Mechanical Refrigeration

This type of auxiliary cooling may be accomplished by using a Freon-12 vapor cycle type of mechanical refrigeration at the warm end of the water freeze-out heat exchanger. A schematic diagram of the system is shown in Figure 21, and the corresponding flow chart is shown in Figure 22. The Freon evaporator in which the auxiliary cooling takes place is made integral with the water freeze-out heat exchanger. This arrangement facilitates the re-humidification process during the ice clean-up operation because most of the water is frozen out by the auxiliary cooling. The auxiliary cooling sub-system operates continuously while the freeze-out plus adsorption system is switched back and forth as indicated by the butterfly valve positions. The amount of liquid oxygen which is required for freeze-out of the remaining water may be made equal to the metabolic rate of 2.0 lb per man day except when the carbon dioxide partial pressure is 3.8 mm Hg and the system pressure is 14.7 psia. In this case the liquid oxygen flow must be increased to 3.0 lb per man day. A cycle analysis of the system may be performed as follows:

Assume the temperature out of the auxiliary cooler (T_1'), and calculate the oxygen flow to air flow ratio. An energy balance on the ice freeze-out heat exchanger gives

$$\frac{w_o}{w_a} = \frac{(c_{pa} + y_1 c_{pc})(T_1' - T_4) + H_1' \left[\lambda_i + \frac{c_{pi}}{2}(T_1' - T_2) \right]}{\lambda_o + c_{po}(T_4 - T_9)} \quad \text{III-92}$$

where the moisture content is

$$H_1' = \frac{P_{w1}' H_1}{P_{w1}} = \frac{P_{w1}' H_1}{0.1} \quad \text{III-93}$$

The temperature difference ($T_1' - T_4$) may be taken as 5°R . The cold end outlet temperature (T_2) is 360°R as in all other systems.

If Equation III-92 does not give the oxygen flow rate, assume a new value of T_1' , and repeat the calculation until agreement is obtained.

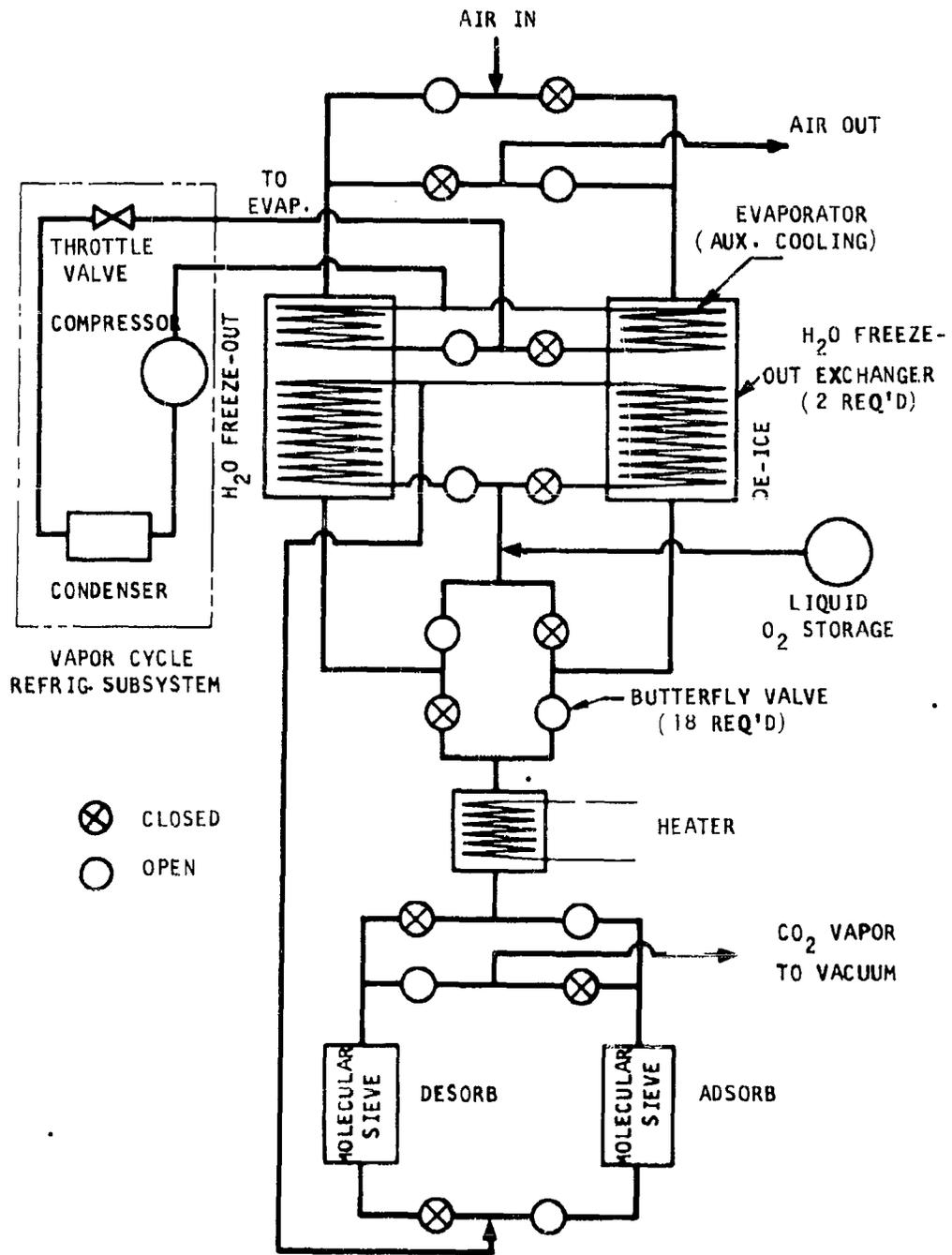
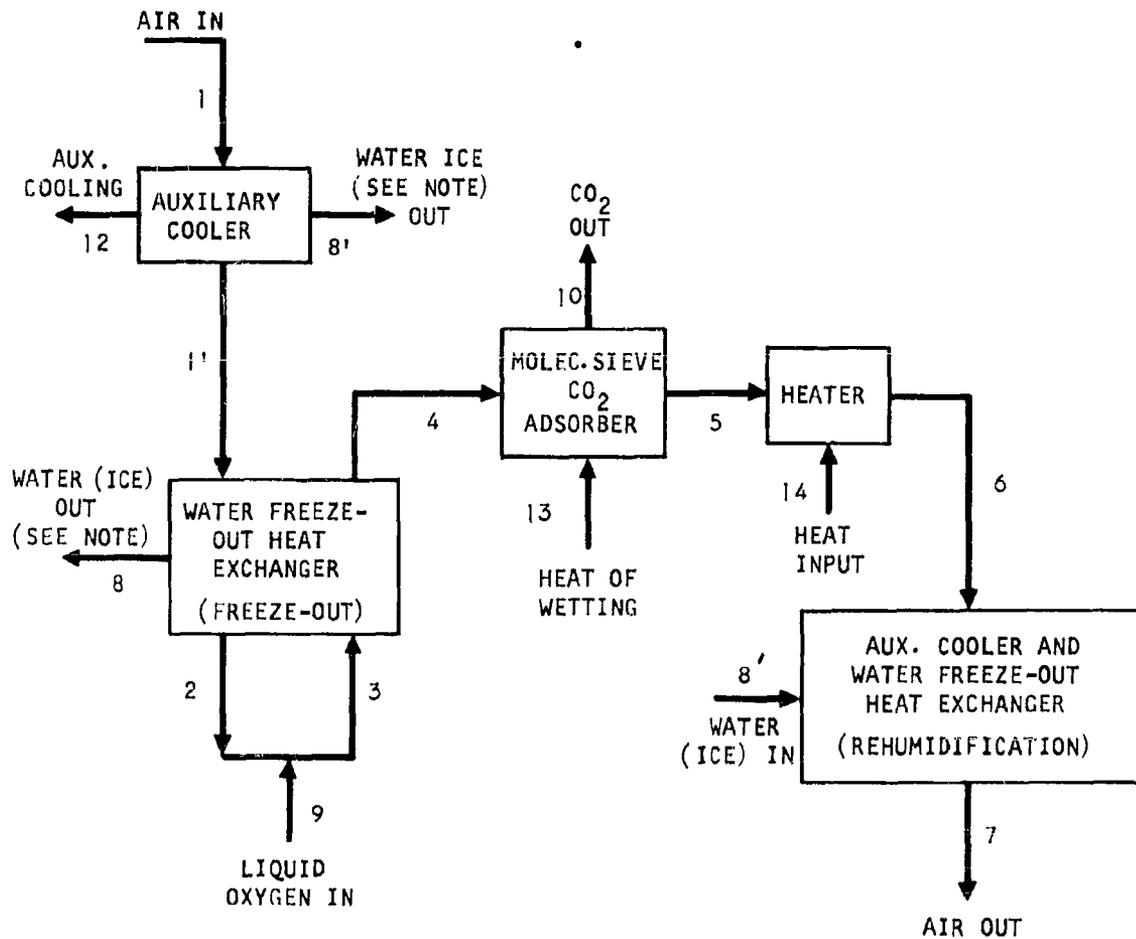


Figure 21. Schematic Diagram for System VII
 (Water Freeze-Out Plus Carbon Dioxide Adsorption
 With Water Recovery and With Auxiliary Cooling By
 Use of Vapor Cycle Refrigeration)



NOTE: WATER (ICE) IS RETURNED TO AIR STREAM AFTER CO₂ REMOVAL AT A TEMPERATURE OF
 $T_8 = T_{8'} = (492 + T_2) / 2$

Figure 22. Flow Chart for System VII (Water Freeze-Out Plus Carbon Dioxide Adsorption With Water Recovery and With Auxiliary Cooling by use of Vapor Cycle Mechanical Refrigeration)

Temperatures T_3 , T_5 , and T_6 may be calculated from Equations III-64, III-74, and III-76 respectively. The air outlet temperature (T_7) may be taken as 500°R . The amount of auxiliary cooling from the refrigeration system is

$$Q_{\text{aux}} = w_a \left[(c_{pa} + y_1 c_{pc})(T_1 - T_1') + (H_1 - H_1') \lambda_1 \right] \quad \text{III-94}$$

and the heat input may be computed from Equation III-80.

A summary of the calculations is given in Table VII.

TABLE VII

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM VII
(WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION WITH WATER RECOVERY AND WITH
AUXILIARY COOLING BY USE OF VAPOR CYCLE MECHANICAL REFRIGERATION)

System Pressure, psia		7.35		14.7	
CO ₂ Partial Pressure, P _{c1} , mm Hg		3.8	7.6	3.8	7.6
H ₂ O Content, H ₁ , lb H ₂ O per lb air		.00842	.00851	.00428	.00431
CO ₂ Content, γ ₁ , lb CO ₂ per lb air		.0151	.0306	.0077	.0155
Air Flow Rate, lb per man day		149	73.5	292	145
Oxygen Flow/Air Flow, lb O ₂ /lb air		.0134	.0272	.0103	.0138
Oxygen Flow/Metabolic Oxygen Flow		1.0	1.0	1.5	1.0
Auxiliary Cooling, Watts per man		39.8	12.7	74.0	26.2
Heater Input, Watts per man		38.1	9.3	75.0	23.8
Temperatures, °R	T ₁	495	495	495	495
	T ₁ '	443	467.5	430	453
	T ₂	360	360	360	360
	T ₃	353	346	355	353
	T ₄	438	462.5	425	448
	T ₅	457	501	435	467
	T ₆	545	544	522	522
	T ₇	500	500	500	500
	T ₈	426	426	426	426
	T ₉	151	151	162	162
Auxiliary Cooler (Z=0)	Effectiveness	.912	.846	.929	.893
	No. of Heat Transfer units	2.43	1.70	2.64	2.25
Water Freeze-Out Heat Exchanger (Z=1)	Effectiveness	.922	.885	.934	.930
	No. of Heat Transfer Units	11.8	7.70	14.1	13.3

NOTES:

- (1) The carbon dioxide removal efficiency was assumed to be 100 per cent.
- (2) The evaporating Freon-12 temperature in the auxiliary cooler was assumed to be equal to T₁' minus 5°R.
- (3) See Figure 21 for a schematic diagram and Figure 22 for a flow chart of System VII.

- c. System VIII. Auxiliary Cooling by Use of an Auxiliary Heat Sink - In a water freeze-out and recovery plus carbon dioxide adsorption system an auxiliary heat sink may be used either at the low temperature end or in front of the water freeze-out heat exchanger. If the auxiliary heat sink is located in front of the water freeze-out heat exchanger, the effects are the same as for auxiliary cooling by the vapor cycle mechanical refrigerator process described in paragraph (b) above. If the vapors from a liquid hydrogen storage tank are used, the amount of hydrogen required is:

$$W_h = \frac{Q_{aux}}{\Delta h}$$

III-95

where Q_{aux} is the auxiliary cooling listed in Table VII and Δh is the hydrogen enthalpy change. If the hydrogen is stored at a pressure of 25 psia (40°R) and is heated to 360°R, the change in enthalpy (in the para form) is 1062 Btu per lb. The hydrogen flow rates and auxiliary cooling heat exchanger requirements are summarized in Table VIII. All other data is the same as in Table VII.

TABLE VIII

SUMMARY OF CYCLE ANALYSIS FOR SYSTEM VIII.
 (WATER FREEZE-OUT PLUS CARBON DIOXIDE ADSORPTION
 WITH WATER RECOVERY AND WITH AUXILIARY COOLING
 BY USE OF A HYDROGEN VAPOR AUXILIARY HEAT SINK)

Pressure, psia	7.35		14.7			
CO ₂ Partial Pressure, P _{cl} , mm Hg	3.8	7.6	3.8	7.6		
Auxiliary Cooling, Btu per man day	3260	1040	6060	2145		
Hydrogen Flow, $W_h = \frac{Q_{aux}}{1062}$, lb per man day	3.07	0.98	5.70	2.02		
Auxiliary Cooling Heat Exchanger	Capacity Rate Ratio $Z = \frac{(wc)_h}{(wc)_a}$		0.285	0.184	0.265	0.192
$E_h = \frac{360-40}{495-40} = 0.7$	No. of Heat Transfer Units, NTU _h		1.37	1.30	1.36	1.32

NOTES

- 1) Hydrogen vapor inlet temperature = 40°R, outlet = 360°R
- 2) All other figures are the same as on Table VII.
- 3) See Figure 21 (with vapor cycle subsystem replaced by the hydrogen heat sink) for a schematic diagram and Figure 22 for a flow chart of the system.

SECTION IV

SYSTEM COMPONENT REQUIREMENTS

- I. Heat Exchangers - All of the contaminant freeze-out systems described in Section III require the use of heat exchange equipment. The most important types are regenerative freeze-out and partially regenerative freeze-out heat exchangers. The other types include air heaters and various types of auxiliary devices such as an air to freon evaporator and an air to cryogenic hydrogen heat exchanger. A description of each type of heat exchanger and an approximate method of determining its weight and volume are given below.
 - a. Regenerative Freeze-Out Heat Exchanger - A schematic sketch which depicts the operation of this heat exchanger is shown on Figure 23. Notice that four flow channels are required and that the frozen out solids are removed by sublimation to a vacuum. The contaminated air enters the freeze-out channel where the contaminants are frozen out and heat is transferred in two directions--one to the regenerative (air outlet) channel and the other to the sublimation channel. A fourth channel is empty. The purified air returns through the regenerative channel where reheating to nearly inlet conditions takes place. When the freeze-out channel is partly filled (and the sublimation channel is nearly free of solids) the butterfly control valves are reversed so that freeze-out and sublimation processes exchange places. At the same time the empty and regenerative channels also exchange places. Although this type of regenerative heat exchanger is a four channel switching unit, it is essentially a three passage heat exchanger. The fourth passage is necessary because it permits continuous operation by switching channels. Three channels are not sufficient for continuous operation unless two heat exchangers are used in parallel. The overall heat transfer process is rather complex because single phase heating of a gas takes place in one channel, cooling of a gas which contains condensible impurities occurs in the second channel, and sublimation of a solid takes place in the third channel. The heat exchanger must also be designed so that the sublimation rate equals the freeze-out rate. Inspection of Figure 23 also reveals that an extended type of heat transfer surface (such as plate-fin) must be used in the freeze-out and sublimation channels to provide a heat conduction path to the sublimating solids. The surfaces must also be designed to hold the frozen out solids inside the heat exchanger. If this is not done the return manifolds will become clogged and regeneration will not be possible. For preliminary design purposes, a very compact offset plate-fin type of heat transfer surface will be used in all four channels. The core geometry of this surface, together

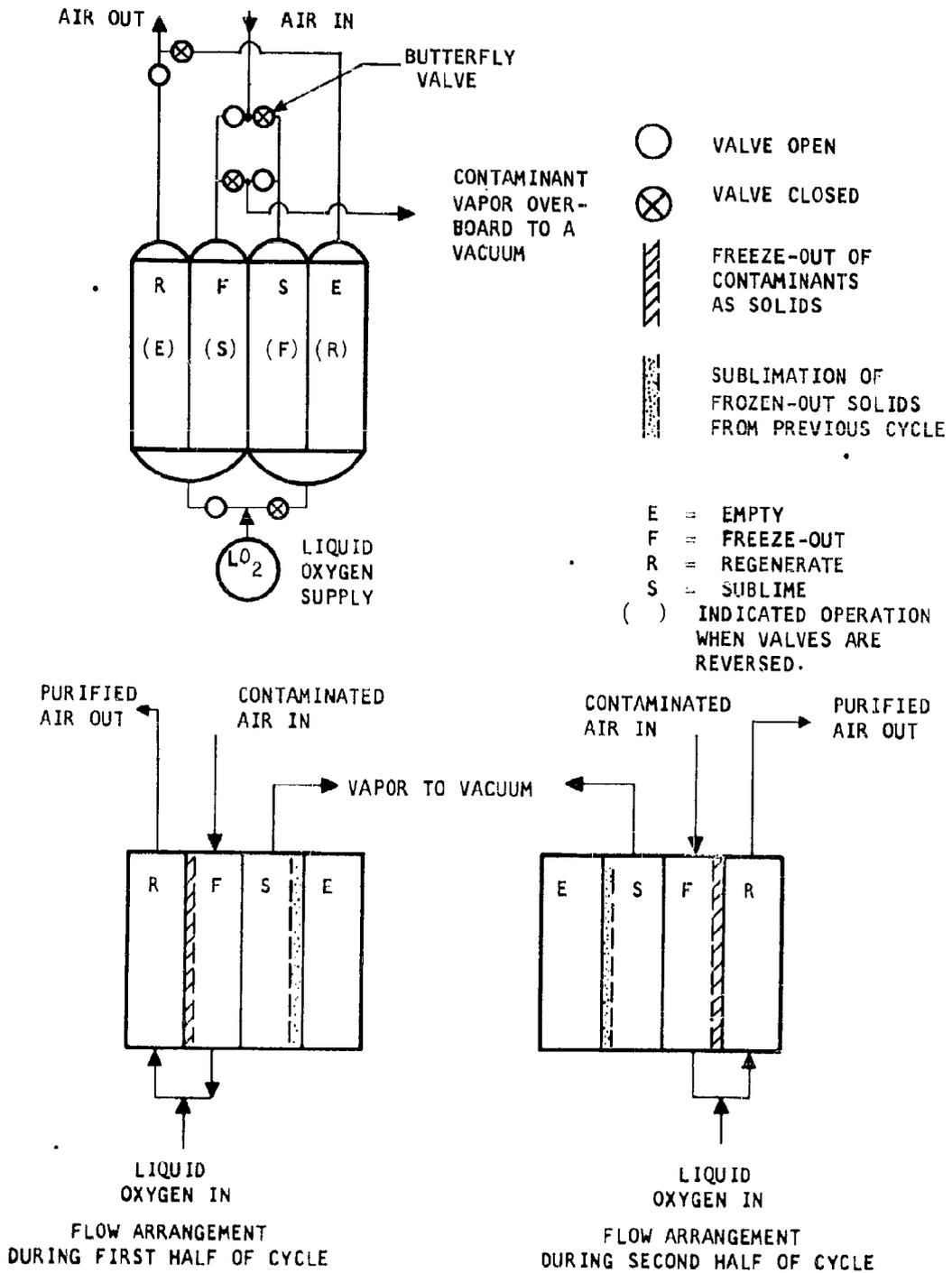


Figure 23. Schematic Drawing of a Typical Regenerative Freeze-out Heat Exchanger

with its basic performance characteristics, are given in Figure 24. This surface is manufactured by AiResearch and has been tested at Stanford University (See Reference 5, page 16). The surface is made of aluminum and has 20 fins per inch, is 0.1 inch high, is offset at 1/8 inch intervals and has 0.004 inch thick fins. The total heat transfer area of 698 ft² per cubic foot of volume. An approximate method of calculating the size of the regenerative freeze-out heat exchangers is given below.

The hydraulic radius of the plate-fin flow passages is defined by the

$$\frac{A}{A_c} = \frac{L}{r} \quad \text{IV-1}$$

friction pressure drop is given by the basic flow friction equation

$$\Delta P = \frac{G^2 f L}{2g \rho r} \quad \text{IV-2}$$

and the heat transfer characteristics are given by

$$j = St Pr^{2/3} = \frac{h}{G c_p} Pr^{2/3} = \frac{h \alpha}{G} \quad \text{IV-3}$$

The friction factor, f , and the Colburn modulus, j , are properties of the heat transfer surface and are functions of the Reynolds number (See Figure 24)

$$Re = \frac{4 r G}{\mu} \quad \text{IV-4}$$

$$\text{where } G = \frac{w}{A_c} \quad \text{IV-5}$$

Equations IV-1, IV-2, IV-3, and IV-5 may be combined to give the free flow area of a heat transfer flow passage as

$$A_c = \sqrt{\frac{\alpha f w h A}{2 g j \rho \Delta P}} \quad \text{IV-6}$$

Neglecting the thermal resistance of the frozen out solids, and neglecting the heat absorbed by the sublimating solids, the overall thermal conductance between the freeze-out and regenerative channels may be expressed in the conventional manner for heat transfer between two flow streams. That is

$$\frac{1}{UA} = \frac{1}{h_1 A_1} + \frac{1}{h_2 A_2} \quad \text{IV-7}$$

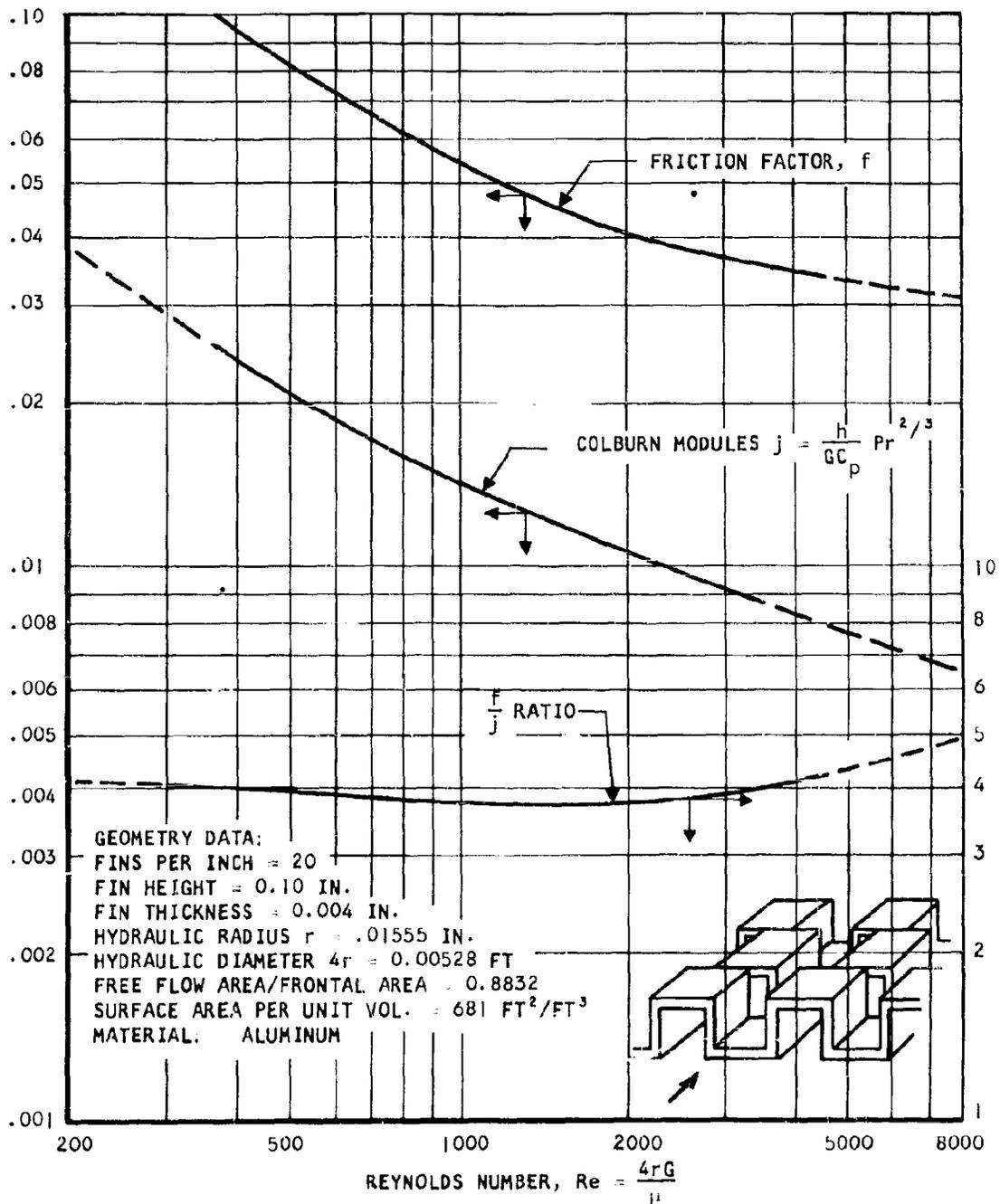


Figure 24. Basic Performance Characteristics and Geometry Data of Rectangular Offset Plate-Fin Heat Transfer Surface 20R-0.1-1/8-.004 (A)

which may be rearranged to give

$$UA = \frac{\eta_o hA_1}{1 + \frac{\eta_o hA_1}{\eta_o hA_2}} \approx \frac{hA_1}{1 + \frac{hA_1}{hA_2}} \quad \text{IV-8}$$

where $\eta_o hA_1$ and $\eta_o hA_2$ are the individual side thermal conductances. The overall fin effectiveness, η_o , is nearly unity for this type of heat exchanger.

The two flow streams have about the same thermal conductance; therefore, in either the regenerative or the freeze-out channel the thermal conductance is about

$$hA = 2 UA \quad \text{IV-9}$$

The number of heat transfer units is defined by

$$NTU = \frac{UA}{w c_p} \quad \text{IV-10}$$

where the NTU's for the various systems which use regenerative heat exchangers are as tabulated in Tables I, II, III and V. Equations IV-9 and IV-10 may be combined to give

$$hA = 2 w c_p (NTU) \quad \text{IV-11}$$

which may be substituted into Equation IV-6 to give the flow passage free flow area as

$$A_c = w \sqrt{\frac{(\alpha f c_p (NTU))}{g j \mu \Delta P}} = w \sqrt{\frac{Pr^{2/3} f NTU}{g j \mu \Delta P}} \quad \text{IV-12}$$

Equations IV-1, IV-3, and IV-5 may be combined to give the flow passage length as

$$L = \frac{Pr^{2/3} r hA}{w j c_p} \quad \text{IV-13}$$

which, upon substitution of Equation IV-11, is also equal to

$$L = \frac{2 r Pr^{2/3} NTU}{j} \quad \text{IV-14}$$

Note that all passages have the same flow length because the flow arrangement is counter-flow.

The volume of each flow passage may be expressed as

$$V_{\text{passage}} = L A_{\text{face}} = \frac{L A_c}{K} \quad \text{IV-15}$$

where K is a core geometry constant that is equal to the ratio of free flow area to frontal area for the plate-fin surface. The total volume between plates of the heat exchanger may not be calculated by recalling that a total of four flow passages are required to accomplish the regenerative heat transfer process. Also the pressure drop and density during the regenerative and freeze-out processes are very nearly equal. This information, together with Equations IV-12 and IV-14, may be substituted into Equation IV-15 to give the total volume between plates as

$$V^t = \frac{4 L A_c}{K} = \frac{8 r Pr (NTU)^{3/2} w}{K j} \sqrt{\frac{l}{g \rho \Delta P} \left(\frac{f}{j}\right)} \quad \text{IV-16}$$

where ΔP is the friction pressure drop in each passage. The Colburn modulus, j , and the f/j ratio are functions of the Reynolds number ($4rG/i$) as shown in Figure 24. Notice that the f/j ratio is nearly constant at a value of about 3.9 over a range of Reynolds number from about 200 to 4000. Even over a range of Reynolds numbers from 100 to 10,000 the f/j ratio varies only between 3.7 and 5.0. The total volume of the heat exchanger core depends upon the plate thickness, and, if multiple sandwich construction is used, the splitter plate thickness. The splitter plate is a thin sheet which is placed between fins and is used for brazing purposes. The total core volume may be expressed as

$$V = \left(1 + \frac{t_s + t_p}{N b}\right) V^t \quad \text{IV-17}$$

where t_p is the plate thickness, t_s is the splitter thickness, b is the fin height, and N is the number of fin sandwiches per flow passage. For purposes of this study a double sandwich construction will be assumed ($N = 2$), the plate and splitter thicknesses will be assumed to be 0.016 and 0.006 inch respectively. The fin height is 0.10 inch and the ratio of free flow to frontal area is $K = 0.795$. The total core volume is

$$V = 1.10 V^t = \frac{1.10 \times 4 L A_c}{.795} = 5.54 L A_c \quad \text{IV-18}$$

The corresponding weight of the heat exchanger core may be expressed by

$$W_{\text{core}} = \rho_{\text{core}} V \quad \text{IV-19}$$

where ρ_{core} is, for the type of aluminum plate-fin heat exchanger being considered, equal to 0.025 lb per cu in. of core volume. The overall heat exchanger weight, which includes manifolds and support structure is

difficult to estimate without going into considerable detail. However, the overall heat exchanger weight should not be more than 50 per cent greater than the core weight. The total heat exchanger weight is about

$$Wt_{Hx} = 1.5 \rho_{core} V$$

IV-20

The above analysis assumes that the rate of sublimation in the vacuum vented channel does not control the design. This assumption appears to be plausible, but it has not been checked either analytically or experimentally. A rigorous theoretical analysis of the overall heat transfer process in the three channels (the fourth channel is required for switching purposes, but does not enter into the heat transfer calculations) has not been worked out. The best approach appears to be an experimental verification of the above simplified analysis.

A sample calculation for the heat exchanger size and weight determination is given below. Consider the conditions where the total pressure in the cabin is 14.7 psia and the carbon dioxide partial pressure is 7.6 mm Hg. Referring to Table I for the simple regenerative freeze-out system with water removal, the air flow rate, w_a , is 161 lb per man day and the heat exchanger NTU is 39.0. Assume that the allowable freeze-out system pressure drop is 2.5 per cent of the total pressure, that is $\Delta P = 0.025 \times 14.7 = 0.367$ psi (10.2 in. H₂O). Assume also that 20 per cent of the allowable ΔP is charged to the valves and plumbing. The remaining pressure drop is 0.30 psi which is divided equally between the freeze-out and regenerative channels. Nearly all of this will be caused by friction. Flow acceleration or deceleration and end effects may be neglected. The average temperature in the heat exchanger is (see Table I) $T_m = (T_1 + T_3)/2 = (495 + 245)/2 = 370^\circ R$, the average pressure is $14.7 - 0.367/2 = 14.5$ psia, and the average density is $\rho = P/RT = (14.5 \times 144)/(53.3 \times 370) = 0.106$ lb per cu ft. The average values for the thermal properties of air (see Reference 6) are specific heat = 0.240 Btu per lb $^\circ R$, viscosity = 0.914×10^{-5} lb per ft, sec and Prandtl number = 0.738. A procedure for estimating the regenerative heat exchanger weight is as follows:

Calculate the free flow area per passage from Equation IV-12. Some trial and error is required because the f/j ratio must first be assumed and then checked after the Reynolds number is evaluated.

Assume $f/j = 3.9$

$$w = \frac{161}{24 \times 3600} = 0.00186 \text{ lb/sec}$$

$$\Delta P = \frac{.30}{2} \times 144 = 21.6 \text{ lb/ft}^2$$

$$A_c = w \sqrt{\frac{Pr^{2/3} f \text{ NTU}}{g j \rho \Delta P}} = .00186 \sqrt{\frac{(0.738)^{2/3}}{32.2} \times 3.9 \times \frac{39.0}{.106 \times 21.6}}$$

$$A_c = 0.00242 \text{ ft}^2 (0.348 \text{ in}^2)$$

$$G = \frac{w}{A_c} = \frac{.00186}{.00242} = 0.770 \text{ lb/sec ft}^2$$

$$Re = \frac{4 r G}{\mu} = \frac{.00528 \times .770}{.914 \times 10^{-5}} = 445$$

Referring to Figure 24; $f = .0870$ $j = .0222$ and $f/j = 3.9$, which is equal to the assumed value of 3.9. A second approximation is not necessary.

The flow length, from Equation IV-14, is

$$L = \frac{2 r Pr^{2/3} \text{ NTU}}{j} = \frac{2 \times .001295 \times (.738)^{.667} \times 39.0}{.0222}$$

$$L = 3.72 \text{ ft (44.6 in.)}$$

The heat exchanger volume, from Equation IV-18 is

$$V = 5.54 LA_c = 5.54 \times 3.72 \times .00242 = .0499 \text{ ft}^3 (86.0 \text{ in}^3)$$

The heat exchanger weight from Equation IV-20, is

$$Wt_{Hx} = 1.5 \rho_{\text{core}} V = 1.5 \times 0.25 \times 86.0 = 3.22 \text{ lbs}$$

The shape of the heat exchanger may be determined by noting that the total face area is

$$A_{\text{face}} = \frac{V}{L} = \frac{86.0}{44.6} = 1.93 \text{ in}^2$$

The length perpendicular to the fins is

$$L_b = 4 (Nb + t_s + t_p) = 4(2 \times .1 + .016 + .006) = 0.888 \text{ in.}$$

the width is

$$L_w = \frac{A_{\text{face}}}{L_b} = \frac{1.93}{.888} = 2.17 \text{ in.}$$

and the flow length (see above calculations) is

$$L = 44.6 \text{ in.}$$

The overall heat exchanger length, which includes headers and manifolds, will be about 46 inches.

- b. Partially Regenerative (Air Only) Freeze-Out Heat Exchangers - In systems where water is first frozen-out as a solid prior to carbon dioxide removal, water recovery (water returned to the effluent air stream) may only be accomplished by heating the effluent air and passing it over the ice. This operation requires two heat exchangers in parallel. Each one is a conventional two fluid heat exchanger. During freeze-out the warm moist air is cooled by the cold effluent, oxygen enriched air. The moisture is deposited in the unit air passages and regeneration of the ice does not take place. When the heat exchanger becomes loaded with ice the control valves are reversed, and the heated, dry effluent air is directed through the flow passages to vaporize the ice and rehumidify (return the water to the air stream) the air. The other heat exchanger which has been previously purged of ice takes over the freeze-out operation.

The design of this type of heat exchanger may be done by applying single phase heat transfer theory to the counter-flow arrangement as was done in the case for the regenerative heat exchanger. Only two flow channels are needed instead of four. For preliminary sizing purposes the effect of the ice on the heat transfer rates may be neglected.

- c. Miscellaneous Heat Exchangers - All of these units are conventional cross-flow type of heat exchangers. Their weight and size may be determined by conventional methods such as given in Reference 7. The heat transfer surfaces are of the compact type such as described in Reference 5.
2. Adsorbers - In this study, all water adsorption is accomplished with silica gel, and all carbon dioxide adsorption is done with molecular sieves. Although molecular sieves may be used to adsorb either water or carbon dioxide, silica gel is used for water adsorption rather than molecular sieves because of desorption temperature requirements. Molecular sieves have such a strong affinity for water that they require a water desorption temperature of about 400 to 600°F, whereas a silica gel bed may be desorbed of water by heating to only about 250°F. High desorption temperatures impose a severe heating power penalty on closed respiratory systems for space vehicles.

The theory of adsorption and desorption is a very complex matter, and a detailed, precise treatment is not justified for a system feasibility study. Therefore, the size and weight of adsorption beds has been determined by using approximate techniques. The adsorbate content of an adsorber bed at saturation is B lb of adsorbate per lb of adsorbent, where B is 0.1 for silica gel and .05 for molecular sieves under the conditions of this study. The apparent bed density for each adsorber is 45 lb per cu ft. This saturation relationship may be expressed by

$$\frac{w_a X U}{A_f L \rho_{bed}}$$

IV-21

where X = the contaminant concentration by weight in the air

The pressure drop through the beds (See Figure 7) is a straight line on log-log coordinates, therefore,

$$\frac{u \Delta P}{L} = C \left(\frac{w_a}{A_f} \right)^n$$

IV-22

The values of C and n are 2.7×10^{-5} and 1.58 respectively for 1/16 inch pellet molecular sieve beds. For 6-12 mesh silica gel the values of C and n are 1.8×10^{-5} and 1.63 respectively.

The data in References 3, 4 and 8 indicate that the face velocity ($V_f = w_a / A_f \rho_a$) should be less than 100 ft per min, and the contact time ($t = L / V_f$) should be equal to or greater than 0.5 sec, for a silica gel bed and 1.5 sec for a molecular sieve bed. The flow length may be expressed in terms of the contact time

$$L = t V_f = \frac{t G}{\rho_a}$$

IV-23

Equations IV-22 and IV-23 may be combined to give an expression for the allowable mass velocity. The result is

$$G = \left(\frac{\rho_a u \Delta P}{C t} \right)^{\frac{1}{1+n}}$$

IV-24

The flow length may now be calculated from Equation IV-23, the face area is $A_f = w_a / G$, and the time, t , to reach saturation (which is the cycling time) may be computed from Equation IV-21.

The adsorber material may be packed into a canister which fits into the air stream duct. The total weight of adsorber material and canister is probably about 1.5 times the adsorber material weight. Sample calculations for a silica gel bed and a molecular sieve bed are given below.

For sample adsorber bed calculations consider a cabin pressure of 14.7 psia and a carbon dioxide partial pressure of 7.6 mm Hg. The contaminant concentration in the air are 0.00431 lb water vapor per lb air and 0.0155 lb carbon dioxide per lb air. The air flow rate for the one system which uses silica gel (see Table V) is 161 lb per man day and the air flow rate for the systems which use a molecular sieve (see, for example, Table III) is 145 lbs per man day. The overall allowable

system pressure drop is 2.5 per cent of the system pressure, or $.025 \times 14.7 = 0.378$ psi. The portion of this which is available to the adsorbers during adsorption is about 20 per cent or $0.20 \times 0.378 = 0.075$ psi. The remainder is used by other components and the system plumbing. The average temperature in the adsorber beds is about 500°R , and the density correction factor is $\rho = 36.4 \times 14.5/500 = 1.07$. The corrected pressure drop is $\rho \Delta P = 1.07 \times .075 = 0.0803$ psi.

The above cabin conditions may now be substituted into Equations IV-21, IV-23 and IV-24 to give the size and weight of the adsorbers.

For the silica gel bed, the mass velocity is

$$G = \left(\frac{\rho_a \rho \Delta P}{C t} \right)^{\frac{1}{1+n}} = \left(\frac{.0794 \times .0803 \times 3600}{1.8 \times 10^{-5} \times .5} \right)^{\frac{1}{2.63}}$$

$$G = 280 \text{ lb per hr-ft}^2$$

The face area is

$$A_f = \frac{w_a}{G} = \frac{161}{24 \times 280} = 0.024 \text{ ft}^2 = 3.45 \text{ in}^2$$

The bed diameter is

$$D = \sqrt{\frac{4 A_f}{\pi}} = \sqrt{\frac{4 \times 3.45}{\pi}} = 2.10 \text{ in.}$$

The bed flow length is

$$L = \frac{L G}{\rho_a} = \frac{.5 \times 280}{3600 \times .0794} = 0.49 \text{ ft} = 5.88 \text{ in.}$$

The overall canister length is about 7.0 in.

The cycling time is

$$t = \frac{L \rho_{\text{bed}} B}{G \pi} = \frac{.49 \times 45 \times .1}{280 \times .00431} = 1.83 \text{ hrs} = 110 \text{ min}$$

The face velocity is

$$V_f = \frac{G}{\rho_a} = \frac{280}{.0794 \times 60} = 58.7 \text{ ft per min}$$

The adsorber (bed and canister) weight is about

$$Wt_{\text{adsorber}} = 1.5 \times A_f \times L \times \rho_{\text{bed}} = 1.5 \times .024 \times .49 \times 45 = 0.79 \text{ lbs}$$

For the 1/16 inch pellet size molecular sieve bed, the mass velocity is

$$G = \left(\frac{.0794 \times .0803 \times 3600}{2.7 \times 10^{-5} \times 1.5} \right)^{\frac{1}{2.58}} = 172 \text{ lb per hr ft}^2$$

The face area is

$$A_f = \frac{145}{24 \times 172} = 0.0351 \text{ ft}^2 = 5.05 \text{ in}^2$$

The bed diameter is

$$D = \sqrt{\frac{4 \times 5.05}{\pi}} = 2.54 \text{ in.}$$

The bed flow length is

$$L = \frac{1.5 \times 172}{3600 \times .0794} = 0.904 \text{ ft} = 10.8 \text{ in.}$$

The overall canister length is about 12.0 in.

The cycling time is

$$t = \frac{.904 \times 45 \times .05}{172 \times .0155} = 0.763 \text{ hrs} = 46 \text{ min}$$

The adsorber (bed and canister) weight is about

$$Wt_{\text{adsorber}} = 1.5 \times .0351 \times .904 \times 45 = 2.14 \text{ lbs}$$

SECTION V

SYSTEM COMPARISONS

1. Weight and Size - A total of eight systems have been analyzed in a preliminary manner for the purpose of determining approximate weights and sizes. The heat exchangers and adsorbers were sized by the methods given in Section IV. The weight of each butterfly valve and its actuator was taken as 0.25 lb, and the corresponding volume was taken as 5 cu in. each. The weights and sizes of auxiliary equipment, such as compressors, expanders and refrigeration equipment was estimated by extrapolating data from larger size equipment.

A summary of the weights and sizes, together with power penalties, water losses and liquid oxygen requirements, for each system is given in Table IX. Each system was analyzed at carbon dioxide partial pressures of 3.8 and 7.6 mm Hg and cabin pressures of 0.5 and 1.0 atm. These pressures were chosen because they probably represent the maximum and minimum conditions for long time exposure in closed respiratory systems. All systems remove the same amount of carbon dioxide from the air, namely 2.25 lb per man day. All systems, where carbon dioxide is frozen out, have a removal efficiency of 90 per cent. The adsorption systems have a removal efficiency of 100 per cent.

The most probable operating conditions for a contaminant freeze-out system, in a closed cycle respiratory system, are a cabin pressure of 0.5 atm and a carbon dioxide partial pressure of 3.8 mm Hg. The weight and volume data for this condition are summarized in Table X. Notice that System I (Simple Freeze-out with Water Removal) has the lowest total equivalent weight, where the equivalent weight is defined as the sum of the fixed weight, power penalty and heat rejection penalty.

The effect of the variable weights (water loss and excess oxygen) is shown in Figure 25, which is a plot of total weight chargeable to the system vs operating time. Notice that Systems I and II are the lightest weight systems up to an operating time of about 25 days. At this time, System I, III, V and VII are all about equal. Systems II, IV, VI and VIII are not competitive.

System II has the lowest volume, but its weight is excessive because of the high liquid oxygen requirements. System I appears to be the best system from a combined volume and weight point of view.

TABLE IX
WEIGHT AND SIZE COMPARISON OF CONTAMINANT FREEZE-OUT SYSTEMS ON A ONE MAN BASIS

SYSTEM	I SIMPLE FREEZE-OUT WITH WATER REMOVAL			II SIMPLE FREEZE-OUT WITH WATER RECOVERY		
	0.5	1.0	1.0	0.5	0.5	1.0
TOTAL SYSTEM PRESSURE, Atm.	3.8	7.6	7.6	3.8	3.8	7.6
CO ₂ PARTIAL PRESSURE, mm Hg	166	81.6	161	166	166	81.6
AIR FLOW RATE, LB PER MAN DAY	4.2	2.1	3.2	0.8	0.4	1.8
FIXED WEIGHTS, LBS:						
HEAT EXCHANGERS	0	0	0	0	0	0
ADSORBERS	2.0	2.0	2.0	5.0	5.0	5.0
VALVES & ACTUATORS (0.25 LB EACH)	1.0	1.0	1.0	2.5	2.5	2.5
PLUMBING, FAN, INSULATION	0	0	0	0.7	0.4	0.9
AUXILIARY EQUIPMENT	1.4	1.0	1.2	1.8	1.7	2.0
MISCELLANEOUS (20 PER CENT)	8.6	6.1	7.4	10.8	10.0	12.2
TOTAL FIXED WEIGHT LBS	6.9	3.4	6.7	36.3	27.9	49.4
POWER REQUIREMENTS, WATTS	0.7	0.3	0.7	3.6	2.8	4.9
POWER PENALTY (0.1 LB/WATT, LBS)	0	0	0	0	0	0
SUPPLEMENTARY COOLING, WATTS	0	0	0	0	0	0
HEAT REJECTION PENALTY (0.3 LB/WATT)	0	0	0	0	0	0
VARIABLE WEIGHTS, LBS						
WATER LOSS, LB PER MAN DAY	1.40	0.69	0.69	0	0	0
EXCESS LIQ. O ₂ , LB PER MAN DAY	0.59	-0.71	0.59	12.4	5.45	15.1
VOLUMES, IN ³						
HEAT EXCHANGERS	150	80	125	30	20	70
ADSORBERS	0	0	0	0	0	0
VALVES & ACTUATORS (5 IN ³ EACH)	40	40	40	100	100	100
AUXILIARY EQUIPMENT	0	0	0	10	10	10
MISCELLANEOUS (20 PER CENT)	40	25	35	30	25	35
TOTAL VOLUME, IN ³	230	145	200	170	155	215
OVERALL PACKAGE SIZE, IN	3x5x30	3x2.7x30	3x6.2x46	3x2.7x46	3x1.5x12	3x3.1x22
						3x1.8x23

NOTES: 1. INLET DOW. POINT = 35°F

2. CO₂ REMOVAL RATE = 2.25 LB PER MAN DAY

3. LIQUID O₂ STORAGE PRESSURE = 75 PSIA

4. LIQUID O₂ IS THROTTLED TO SYSTEM PRESSURE

5. SYSTEM PRESSURE DROP = 2.5% OF TOTAL PRESSURE

6. HEAT SINK TEMPERATURES = 151°R AT 0.5 ATM

AND 162°R AT 1.0 ATM

7. METABOLIC O₂ RATE = 2.0 LB PER MAN DAY

TABLE IX (CONTINUED):
WEIGHT AND SIZE COMPARISON OF CONTAMINANT FREEZE-CUT SYSTEMS ON A ONE MAN BASIS

SYSTEM	III H ₂ O FREEZE-OUT AND REMOVAL PLUS CO ₂ ADSORPTION		IV H ₂ O FREEZE-OUT AND RECOVERY PLUS CO ₂ ADSORPTION	
	0.5	1.0	0.5	1.0
TOTAL SYSTEM PRESSURE, Atm.				
CO ₂ PARTIAL PRESSURE, mm. Hg	3.8	3.8	3.8	3.8
AIR FLOW RATE, LB PER MAN DAY	149	292	149	292
FIXED WEIGHTS, LBS:				
HEAT EXCHANGERS	1.3	2.1	0.7	0.4
ADSORBERS	8.4	8.6	4.2	4.2
VALVES & ACTUATORS (0.25 LB EACH)	3.5	3.5	3.5	4.0
PLUMBING, FAN, INSULATION	1.8	1.8	1.8	2.0
AUXILIARY EQUIPMENT	0	0	0	0.6
MISCELLANEOUS (20 PER CENT)	3.0	3.2	2.0	2.1
TOTAL FIXED WEIGHT LBS	18.0	19.2	12.2	12.8
POWER REQUIREMENTS, WATTS	6.2	12.1	6.0	8.0
POWER PENALTY (0.1 LB/WATT, LBS)	0.6	1.2	0.6	0.8
SUPPLEMENTARY COOLING, WATTS	0	0	0	0
HEAT REJECTION PENALTY (0.3 LB/WATT)	0	0	0	0
VARIABLE WEIGHTS, LBS				
WATER LOSS, LB PER MAN DAY	1.25	1.25	0.63	0
EXCESS LIQ. O ₂ , LB PER MAN DAY	0.32	2.64	-0.82	4.2
VOLUMES, IN ³				
HEAT EXCHANGERS	55	80	30	20
ADSORBERS	280	290	140	140
VALVES & ACTUATORS (5 IN ³ EACH)	70	70	70	80
AUXILIARY EQUIPMENT	0	0	0	10
MISCELLANEOUS (20 PER CENT)	80	90	50	50
TOTAL VOLUME, IN ³	485	530	290	300
OVERALL PACKAGE SIZE, IN	4.2x12.2x14	3.9x11.5x21	4.2x10.8x12	3.9x10.6x22

- NOTES:
1. INLET DEW POINT = 35°F
 2. CO₂ REMOVAL RATE = 2.25 LB PER MAN DAY
 3. LIQUID O₂ STORAGE PRESSURE = 75 PSIA
 4. LIQUID O₂ IS THROTTLED TO SYSTEM PRESSURE
 5. SYSTEM PRESSURE DROP = 2.5% OF TOTAL PRESSURE
 6. HEAT SINK TEMPERATURES = 151°R AT 0.5 ATM AND 162°R AT 1.0 ATM
 7. METABOLIC O₂ RATE = 2.0 LB PER MAN DAY

TABLE IX (CONTINUED)
WEIGHT AND SIZE COMPARISON OF CONTAMINANT FREEZE-OUT SYSTEMS ON A ONE MAN BASIS

SYSTEM	V WATER ADSORPTION PLUS CARBON DIOXIDE FREEZE-OUT			VI SYSTEM IV WITH AUXILIARY COOLING BY AN EXPANDER		
	0.5	1.0	1.0	0.5	0.5	1.0
TOTAL SYSTEM PRESSURE, Atm.	3-8	7-6	3-8	3-8	7-6	7-6
CO ₂ PARTIAL PRESSURE, mm Hg	166	81-6	325	149	73-5	292
AIR FLOW RATE, LB PER MAN DAY						
FIXED WEIGHTS, LBS:						
HEAT EXCHANGERS	6-2	3-0	8-3	1-6	1-0	2-9
ADSORBERS	3-4	1-7	3-3	8-4	4-2	8-6
VALVES & ACTUATORS (0.25 LB EACH)	4-0	4-0	4-0	4-5	4-5	4-5
PLUMBING, FAN, INSULATION	2-0	2-0	2-0	2-3	2-3	2-3
AUXILIARY EQUIPMENT	2-2	1-1	5-0	5-0	5-0	2-0
MISCELLANEOUS (20 PER CENT)	2-8	2-4	4-5	5-4	3-4	4-1
TOTAL FIXED WEIGHT LBS	16-6	14-2	27-1	27-2	20-4	24-4
POWER REQUIREMENTS, WATTS	94	47	215	179	68	268
POWER PENALTY (0.1 LB/WATT) LBS	9-4	4-7	21-5	17-9	6-8	26-8
SUPPLEMENTARY COOLING, WATTS	80	40	182	170	67	253
HEAT REJECTION PENALTY(0.3 LB/WATT)	24	12	54-6	51	20	76
VARIABLE WEIGHTS, LBS						
WATER LOSS, LB PER MAN DAY	0	0	0	0	0	0
EXCESS LIQ. O ₂ , LB PER MAN DAY	0-36	-0-83	2-94	0	0	0
VOLUMES, IN ³						
HEAT EXCHANGERS	210	110	280	60	35	105
ADSORBERS	125	65	120	280	140	290
VALVES & ACTUATORS (5 IN ³ EACH)	80	80	80	90	90	90
AUXILIARY EQUIPMENT	40	40	40	50	20	20
MISCELLANEOUS (20 PER CENT)	90	60	105	95	65	100
TOTAL VOLUME, IN ³	545	355	625	575	380	605
OVERALL PACKAGE SIZE, IN	4-5x8-1x26	3-5x5x26	4-3x7-4x38	4-2x10-8x24	3-8x25	3-9x10-6x37
			3-4x4-6x38			3x7-6x36

- NOTES:
1. INLET DEW POINT = 35°F
 2. CO₂ REMOVAL RATE = 2.25 LB PER MAN DAY
 3. LIQUID O₂ STORAGE PRESSURE = 75 PSIA
 4. LIQUID O₂ IS THROTTLED TO SYSTEM PRESSURE
 5. SYSTEM PRESSURE DROP = 2.5% OF TOTAL PRESSURE
 6. HEAT SINK TEMPERATURES = 151°R AT 0.5 ATM AND 162°R AT 1.0 ATM
 7. METABOLIC O₂ RATE = 2.0 LB PER MAN DAY

TABLE IX (CONTINUED)
WEIGHT AND SIZE COMPARISON OF CONTAMINANT FREEZE-OUT SYSTEMS ON A ONE MAN BASIS

SYSTEM	VII SYSTEM IV WITH AUX COOLING BY VAPOR CYCLE REFRIGERATION		VIII SYSTEM IV WITH AUX COOLING WITH CRYOGENIC H ₂ GAS	
	0.5	1.0	0.5	1.0
TOTAL SYSTEM PRESSURE, Atm	3.8	7.6	3.8	7.6
CO ₂ PARTIAL PRESSURE, mm Hg	149	73.5	149	73.5
AIR FLOW RATE, LB PER MAN DAY	2.0	1.3	2.0	1.3
FIXED WEIGHTS, LBS:				
HEAT EXCHANGERS	8.4	4.2	8.4	4.2
ADSORBERS	4.5	4.5	4.5	4.5
VALVES & ACTUATORS (0.25 LB EACH,	2.3	2.3	2.3	2.3
PLUMBING, FAN, INSULATION	4.0	1.3	0.5	0.1
AUXILIARY EQUIPMENT	4.2	2.7	3.5	3.8
MISCELLANEOUS (20 PER CENT)	25.4	16.3	21.2	14.9
TOTAL FIXED WEIGHT LBS	84	25	44.9	12.7
POWER REQUIREMENTS, WATTS	8.4	2.5	4.5	1.3
POWER PENALTY (0.1 LB/WATT) LBS	80	25	0	0
SUPPLEMENTARY COOLING, WATTS	24	8	0	0
HEAT REJECTION PENALTY(0.3 LB/WATT)				
VARIABLE WEIGHTS, LBS				
WATER LOSS, LB PER MAN DAY	0	0	0	0
EXCESS LIQ. O ₂ , LB PER MAN DAY	0	0	0	0
VOLUMES, IN ³				
HEAT EXCHANGERS	75	45	75	45
ADSORBERS	280	140	280	140
VALVES & ACTUATORS (5 IN ³ EACH)	90	90	90	90
AUXILIARY EQUIPMENT	100	40	10	10
MISCELLANEOUS (20 PER CENT)	110	65	90	60
TOTAL VOLUME, IN ³	655	380	545	345
OVERALL PACKAGE SIZE, IN	4.2x10.8x29	3x8x33	4.2x10.8x29	3x8x33
			4.2x10.2x38	4.2x10.2x38
				3x7.6x43
				3x7.6x43

- NOTES:
1. INLET DEW POINT = 35°F
 2. CO₂ REMOVAL RATE = 2.25 LB PER MAN DAY
 3. LIQUID O₂ STORAGE PRESSURE = 75 PSIA
 4. LIQUID O₂ IS THROTTLED TO SYSTEM PRESSURE
 5. SYSTEM PRESSURE DROP = 2.5% OF TOTAL PRESSURE
 6. HEAT SINK TEMPERATURES = 151°R AT 0.5 ATM AND 162°R AT 1.0 ATM
 7. METABOLIC O₂ RATE = 2.0 LB PER MAN DAY

TABLE X

WEIGHT AND VOLUME COMPARISON OF POSSIBLE CONTAMINANT
FREEZE-OUT SYSTEMS OPERATING AT 0.5 ATM PRESSURE
AND 3.8 mm Hg CARBON DIOXIDE PARTIAL PRESSURE

SYSTEM	I	II	III	IV	V	VI	VII	VIII
Fixed Weight, lbs	8.6	10.8	18.0	18.7	16.6	27.2	25.4	21.2
Power Penalty, lbs	0.7	3.6	0.6	2.5	9.4	17.9	8.4	4.4
Heat Rejection Penalty, lbs	0	0	0	0	24.0	51.0	24.0	0
Total Equivalent Weight, lbs	9.3	14.4	18.6	21.2	50.0	96.1	57.8	25.6
Water Loss, lb per day	1.40	0	1.25	0	0	0	0	0
Excess Oxygen, lb per day	0.59	12.4	0.32	10.6	0.36	0	0	See Note 3
Total Volume, cu in.	230	170	485	430	545	575	655	545

Notes:

- (1) The above data was obtained from Table IX.
- (2) Total equivalent weight equals the sum of the fixed weight, the power penalty and the heat rejection penalty.
- (3) System VIII has zero excess oxygen but require 3.07 lbs of hydrogen vapor per day.

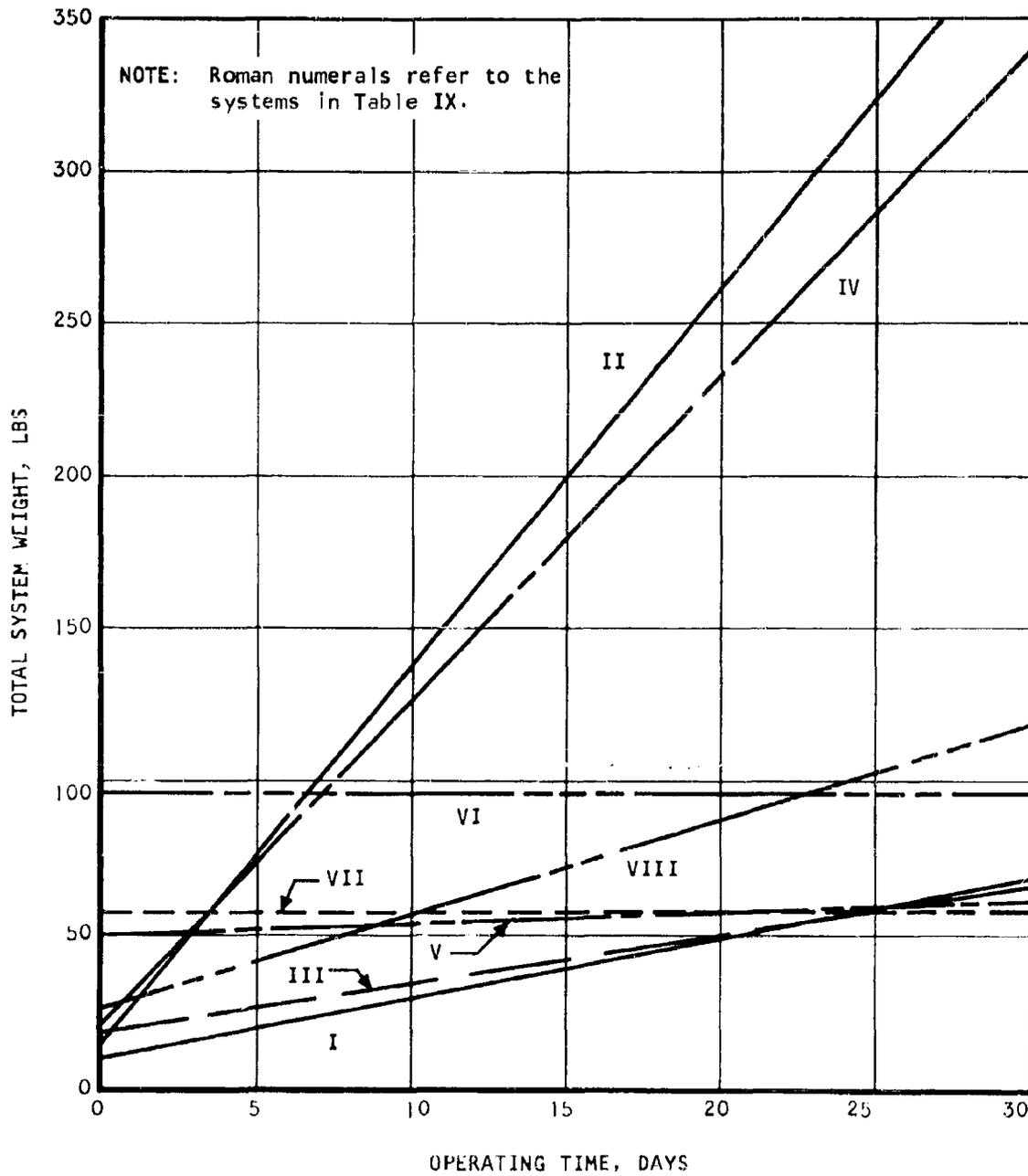


Figure 25. Total System Weight vs Operating Time for Possible Contaminant Freeze-Out Systems Operating at 0.5 Atm Pressure and 3.8 mm Hg Carbon Dioxide Partial Pressure

2. Qualitative Comparisons - The eight possible freeze-out systems may also be compared in a qualitative manner in regard to feasibility, reliability, complexity, ease of operation, development time, cost, and safety. Several of the quantitative items, such as total system weight, volume, and power requirements may also be considered in a qualitative manner. One method of making a qualitative comparison, which is somewhat quantitative in nature, consists of assigning a value and a grade to each of the items of comparison. The highest total grade then gives an indication of which freeze-out system is best. This was done for the eight possible freeze-out systems, and the results are summarized in Table XI.

Inspection of Table XI reveals that, from a qualitative point of view, System I (Simple Freeze-out with Water Removal) is the best system because it has the highest total grade for the ten items of comparison. The assignment of different values to each of the items would still rate System I as best.

3. Selection of an Optimum System - The optimum contaminant freeze-out system appears to be the simple freeze-out system with water removal (System I). This system is a regenerative switching heat exchanger in which both carbon dioxide and water are dumped overboard as vapors. The amount of water which is dumped overboard is a slight disadvantage, but this penalty can only be assessed when a complete water balance for the closed respiratory system is worked out. The regenerative switching heat exchanger is a static device, and it is the simplest of all systems considered. It also requires the least number of valves. In view of the above weight and size comparisons and the qualitative comparisons, the simple freeze-out system with water removal has been selected as the optimum contaminant freeze-out system.

TABLE XI
 QUALITATIVE COMPARISON OF POSSIBLE
 CONTAMINANT FREEZE-OUT SYSTEMS

SYSTEM	I	II	III	IV	V	VI	VII	VIII
Feasibility	10	3	9	3	8	4	5	5
Reliability	9	8	8	10	8	5	6	7
Complexity	10	8	8	9	8	4	5	5
Ease of Operation	10	6	8	8	8	5	6	6
Development Time	8	7	7	10	7	3	6	7
Cost	10	8	8	8	7	2	4	8
Safety	9	8	10	8	7	6	8	5
Weight (15 day mission)	10	1	9	2	7	3	6	4
Volume	8	10	6	6	5	5	4	5
Power Requirements	10	8	10	9	5	3	6	7
Total Grade (100 possible)	94	67	83	73	70	40	56	59

NOTES:

1. Each item of comparison was assigned a value of 10 points. The numbers in the table represent the grade given to each system.
2. A grade of 10 indicates that the most desirable feature of a particular item has been obtained relative to the other systems. For example, a grade of 10 in complexity means the system is the simplest.
3. The Roman numerals refer to the systems in Table IX.

SECTION VI

DEVELOPMENT OF AN OPTIMUM SYSTEM

1. Preliminary Design of a Model System - The system which is considered to be the most worthy of development is the Simple Freeze-Out System with Water Removal (System I). A model version of the system may be considered to be a one man system because the size of the apparatus is suitable for small scale laboratory testing.

During the course of the study most of the preliminary design of a model system has been completed. For the conditions which are easiest to obtain in the laboratory (1.0 atm pressure and 7.6 mm CO₂ partial pressure) the core of the regenerative freeze-out switching heat exchanger for a one man system is about 2.2 inches high, 0.9 inch wide and 45 inches long. A sketch of this unit is shown in Figure 26, and a schematic drawing which depicts the operation is given in Figure 23. The heat exchanger is controlled by eight butterfly valves, four of which must be capable of sealing against a vacuum. These valves need not be the minimum weight type for testing of the switching heat exchanger.

Note that the shape of the flow passage introduces a difficult manifolding problem. The transition from the air ducts to the core of the heat exchanger must guide the air stream from a round pipe to a thin, pencil shaped flow passage.

2. Additional Effort - The amount and type of additional effort required to prove feasibility and to reduce the design approach to hardware practice consists of preliminary heat transfer testing, heat exchanger construction, testing of the heat exchanger with standard valves, light weight valve development, testing of the heat exchanger with light weight valves, analysis of the test work, and report writing. A schematic diagram of a test setup is shown in Figure 27. All of this apparatus, except the heat exchanger and light weight valves, is presently available for testing in the AirResearch cryogenic test laboratory. A summary of the additional effort, together with elapsed time estimates is given in Table XII. The total estimated development time is about 8 months.

Preliminary heat transfer testing is necessary to check out the overall heat transfer process in which cooling of a gas with freeze-out occurs in one passage, heating of a gas occurs in one adjacent passage, and sublimation of frozen-out solids occurs in the other adjacent passage. This testing will also check the ability of the heat transfer surfaces to hold up the frozen out solids. If the ice and solid CO₂ do not adhere to the surfaces, a new surface will be required.

Light weight valve development is necessary because present valves are too large and heavy. Also, it would be desirable to combine all valves into a single unit which is operated by one actuator.

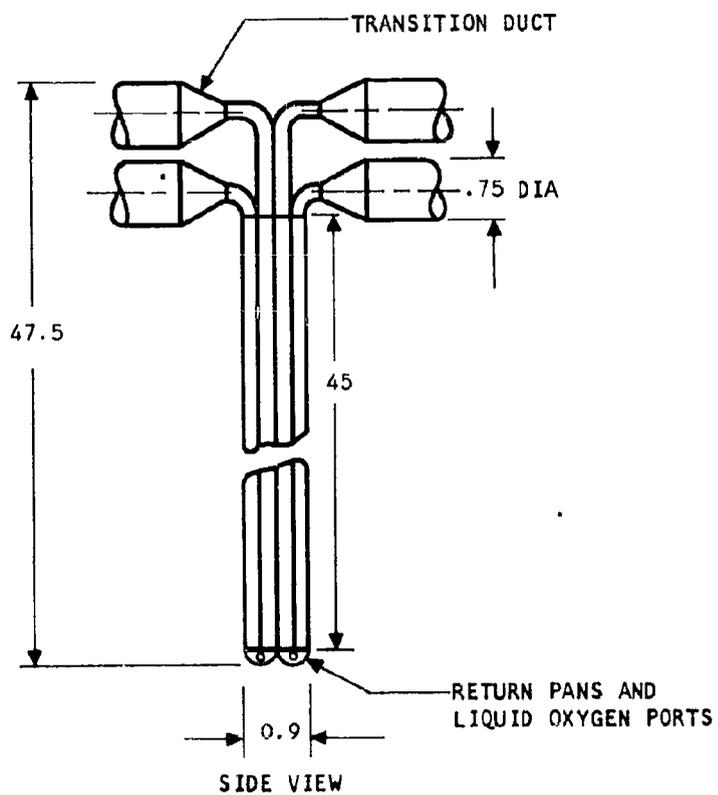
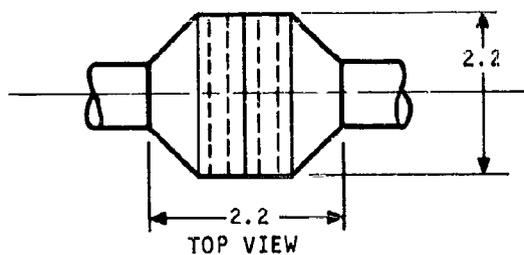


Figure 26. Sketch of a Model Regenerative Freeze-Out Switching Heat Exchanger

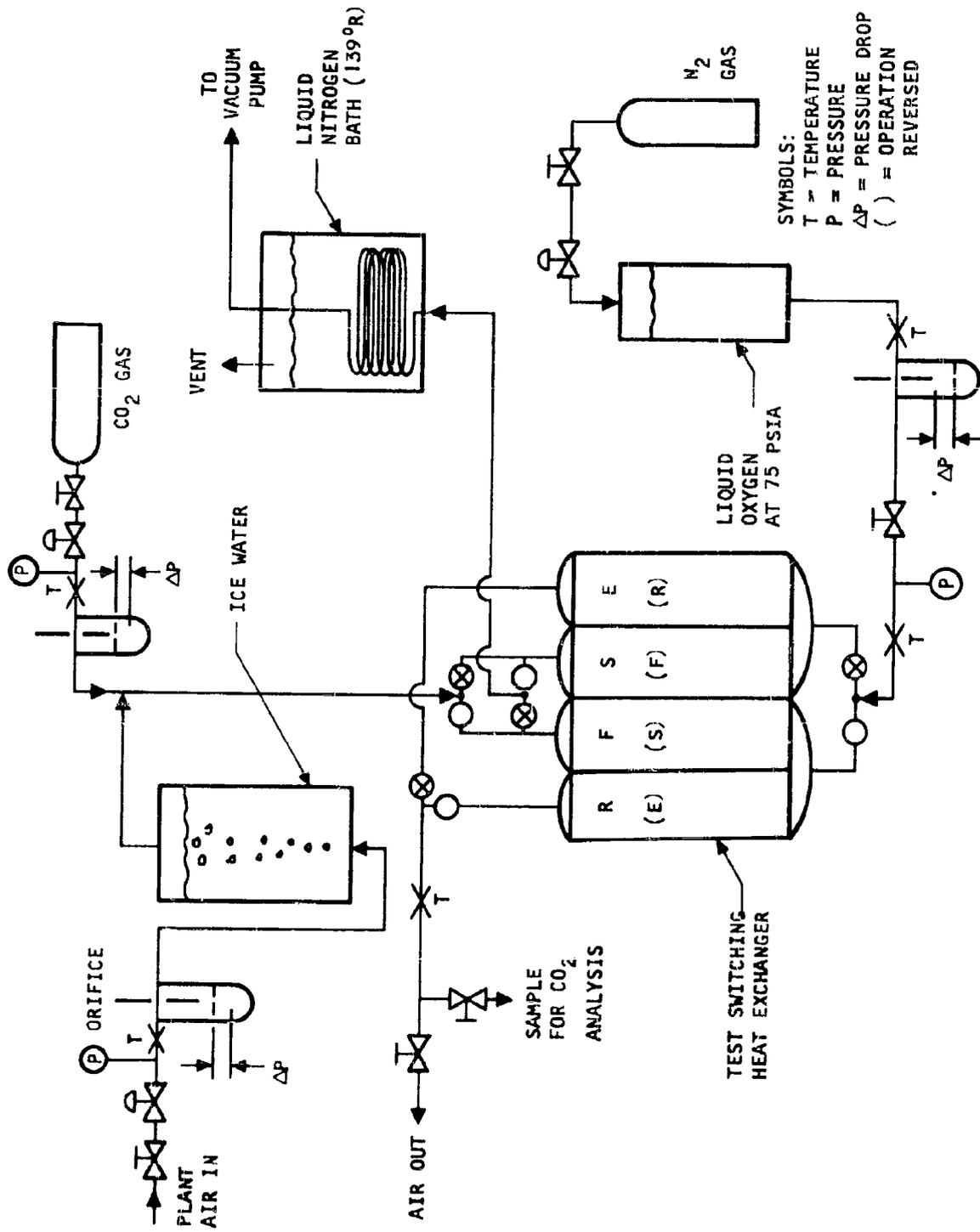


Figure 27. Schematic Diagram of a Test Set Up for a Regenerative Freeze-out System (Simple Freeze-Out with Water and CO₂ Removal)

TABLE XII

SUMMARY OF ADDITIONAL EFFORT REQUIRED TO PROVE THE FEASIBILITY OF THE REGENERATIVE FREEZE-OUT SWITCHING HEAT EXCHANGER AND TO REDUCE THE DESIGN TO HARDWARE PRACTICE

TYPE OF EFFORT	0	1	2	3	4	5	6	7	8
Preliminary Heat Transfer Tests				■					
Heat Exchanger Construction			■						
System Testing with Existing Valves				■					
Light Weight Valve Development			■	■					
System Testing with Light Weight Valves					■				
Analysis of Test Work			■	■	■				
Final Report							■	■	■

SECTION VII

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APPENDIX

CRYOGENIC STORAGE VESSEL CHARACTERISTICS

INTRODUCTION

A number of types of vessels are being developed for storage of fluids in a weightless environment for use in space vehicle environmental systems. Several such vessels are described in Reference 9. In this section, storage vessel characteristics will be reviewed briefly as they affect application to contaminant freeze-out systems.

Weightless environment fluid containers may be classified by the mode of fluid storage and the method of container pressurization. Fluid storage may be either as a single-phase fluid or as a two-phase mixture of liquid and vapor. Storage vessel pressurization may be accomplished by the use of an externally supplied gas to provide positive expulsion, or by the use of thermal energy added to the storage space. The thermal energy required for pressurization may be supplied by either electric power, container wall heat leaks, or by a heat exchanger mounted internally in the storage vessel.

Single-phase fluid storage is possible at both subcritical and supercritical pressures. At subcritical pressures, this is accomplished by the use of a positive expulsion type system while supercritical storage uses thermal pressurization for storage and expulsion.

Cryogenic containers using two-phase storage present design difficulties for weightless environment applications. In this class of vessels, the fluid is stored at subcritical pressure and exists as a mixture of liquid and vapor. For weightless environments, special phase separation provisions are therefore required to permit pressure stabilization during delivery and to prevent the accidental loss of liquid when venting is necessary. A number of phase separation techniques are being considered; these include the use of capillaries or semipermeable membranes or even rotation of the storage vessel to create an artificial gravity field. Magnetic fields may be used for liquid-vapor separation in cryogenic oxygen vessels, in view of the paramagnetic properties of oxygen. The heat sink capability of a vapor delivery system is equivalent to the attainable degree of superheat of the vapor; the heat sink capability of a liquid delivery system is equivalent to the latent heat of vaporization plus the superheat.

Several types of cryogenic storage systems, which may be used in a weightless environment, are described below. Here, for application to contaminant freeze-out systems, particular emphasis is placed upon the fact that the coolant supply temperature must be lower than that required for carbon dioxide freeze-out; this value is of the order of 240°R.

Liquid Storage with Positive Expulsion

A system using this storage technique is the simplest to visualize and is shown schematically on Figure 28, which also illustrates system thermodynamic operation on a pressure-enthalpy diagram. Here, helium from an external high pressure source is used to pressurize a flexible bladder within the fluid storage vessel and thereby to expel oxygen from the tank. Low fluid storage pressures are used, so that with proper design, the fluid masses are stored and expelled as a liquid.

For the considered application, it is assumed that the fluid storage space is filled completely on the ground with saturated liquid at atmospheric pressure and that the storage pressure is to be regulated at some higher operating pressure. The fluid fill state is shown as Point 1 on Figure 28. With the tank completely filled, a slight heat leak will result in compression of the liquid at constant density until tank regulated pressure is reached (Point 2). This initial compression takes place with a very small fluid temperature rise so that the liquid becomes substantially subcooled.

Heat leak into the storage tank during standby results in venting of liquid at constant pressure, with standby and operation occurring between Points 2 and 3 on Figure 28. This process is accompanied by a rise in the temperature of the fluid mass in the tank, with the result that the degree of subcooling attainable during use is diminished. For stable operation, it is desirable that the storage fluid be kept as a single-phase liquid during use. This requires that the tank liquid temperature be kept below the saturation temperature at the operating pressure. This factor often determines the design operating pressure.

Oxygen demand in the space vehicle will be at a pressure of approximately one atmosphere. Therefore, the oxygen supply is throttled to one atmosphere delivery pressure between Points 3 and 4 in Figure 28. The oxygen heat sink available will have a lower temperature limit of T_4 at Point 4 and an upper temperature limit T_5 assuming a constant pressure delivery. The available heat sink, Δh , is plotted in Figure 29 as a function of the upper temperature limit T_5 for oxygen storage pressures of 75 and 150 psia respectively. Also shown on Figure 29 for comparison are the heat sink capabilities of systems delivering saturated liquid and saturated vapor at a pressure of one atmosphere.

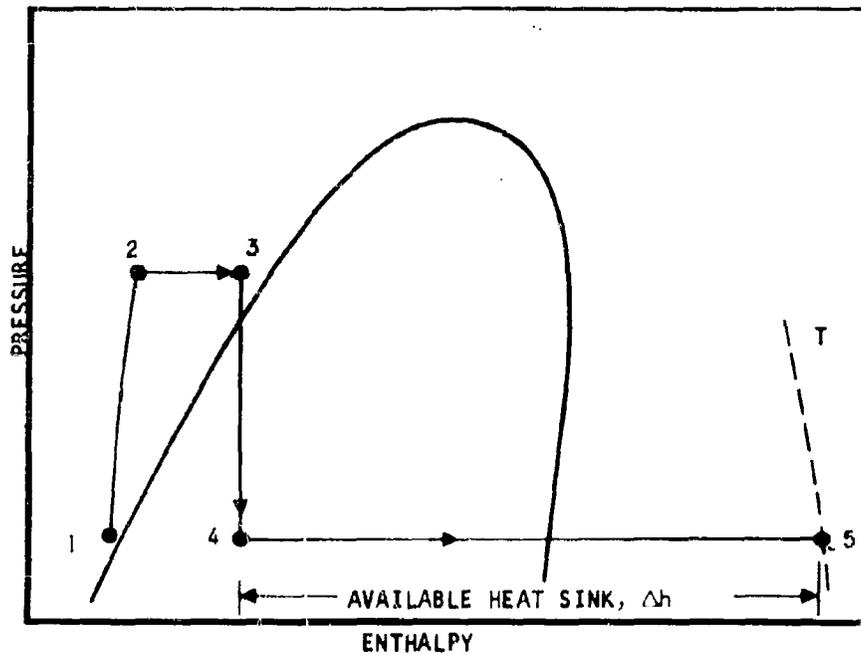
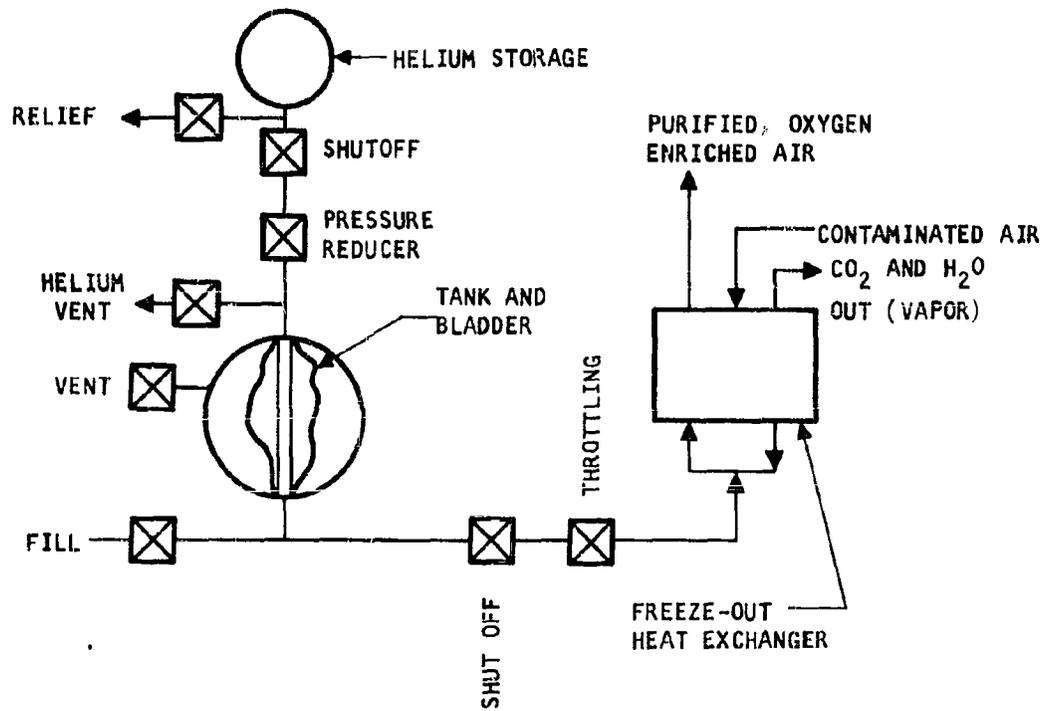


Figure 28. Liquid Storage with Positive Expulsion

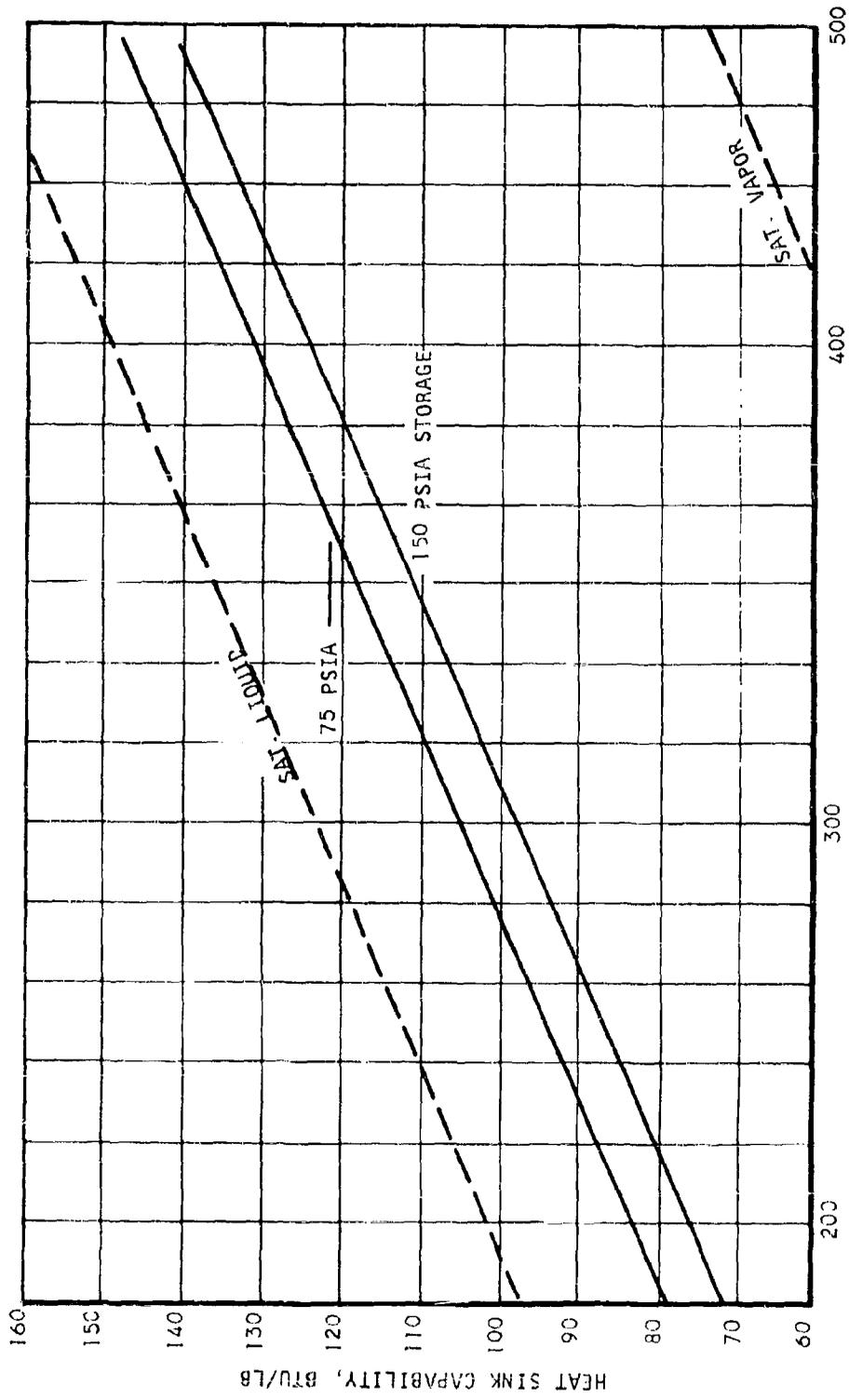


Figure 29. Heat Sink Capability of Oxygen Positive Expulsion System

Two-Phase Storage With Thermal Pressurization

There are a number of possible vessel designs utilizing two-phase storage. In this section, emphasis will be placed upon a particular type of vessel where the fluid is stored as a liquid-vapor mixture but designed for automatic vapor delivery. Figure 30 illustrates the fluid withdrawal process used on a schematic pressure-enthalpy diagram and shows one method of tank heat addition for pressurization. Pre-use pressurization is similar to the processes described previously. Here, tank operation is of most interest. It is most significant that the storage space fluid is a mixture of liquid and vapor. In the absence of gravity, the mass sampled at any point in the tank may consist of liquid and vapor in any proportion. Sampling states may thus range from Point 2 to Point 2' on Figure 30. Withdrawal system operation is described below.

The fluid to be delivered is first passed through a valve and throttled to a pressure lower than tank pressure. Referring to Figure 30 states before throttling may range from Point 2 to 2', and after throttling from Points 3 to 3'. The temperature of the fluid after throttling is lower than storage temperature, however. Thus, passage of the vent fluid through a heat exchanger within the storage space permits a transfer of energy to take place along paths 3 to 4 or 3' to 4 (essentially constant pressure). The fluid can thus be evaporated and superheated slightly before being discharged at a temperature close to that of the storage fluid but at a lower pressure.

Tank heat addition for pressurization is provided by heat leak through the insulating walls of the tank and by using the heated delivery fluid as an energy source as shown in the schematic on Figure 30. However, for most environmental system designs, the oxygen requirement is such that the tank pressurization is accomplished by the vessel heat leak only. Therefore, the heat sink capability of this type of system will, in most cases, only include the enthalpy increase of the delivery fluid as a vapor from Point 4 to some cabin temperature T_5 . The available heat sink for this system Δh is plotted in Figure 31 as a function of the upper oxygen temperature limit T_5 assuming $h_4 = h_{sv}$ at one atmosphere. Notice that the heat sink capability is much smaller than that of the liquid storage positive expulsion system. Therefore, two phase storage of liquid oxygen is not suitable for contaminant freeze-out systems unless an auxiliary heat sink, such as cryogenic hydrogen, is available at the cold end of CO₂ freeze-out section.

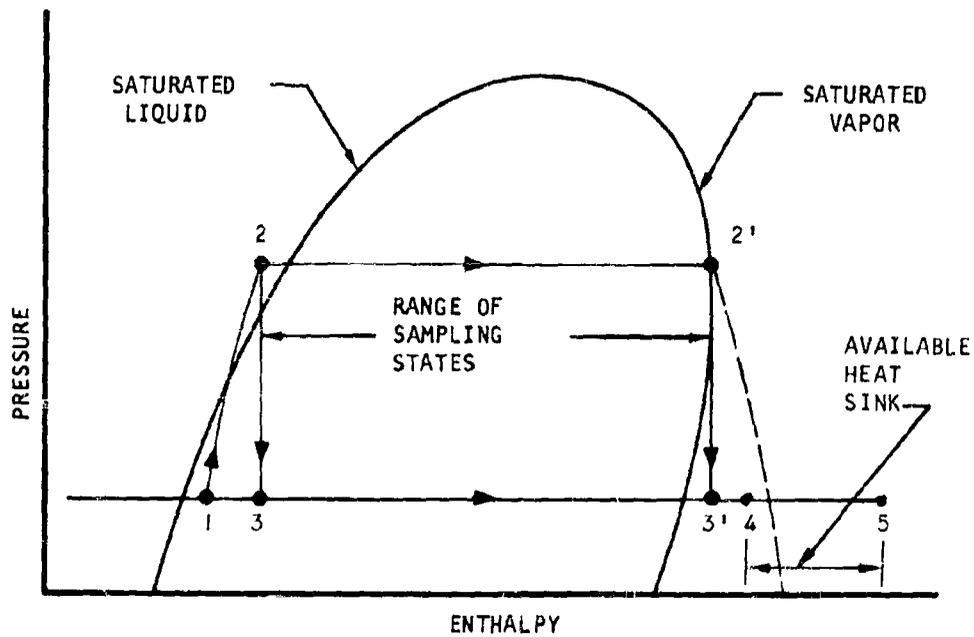
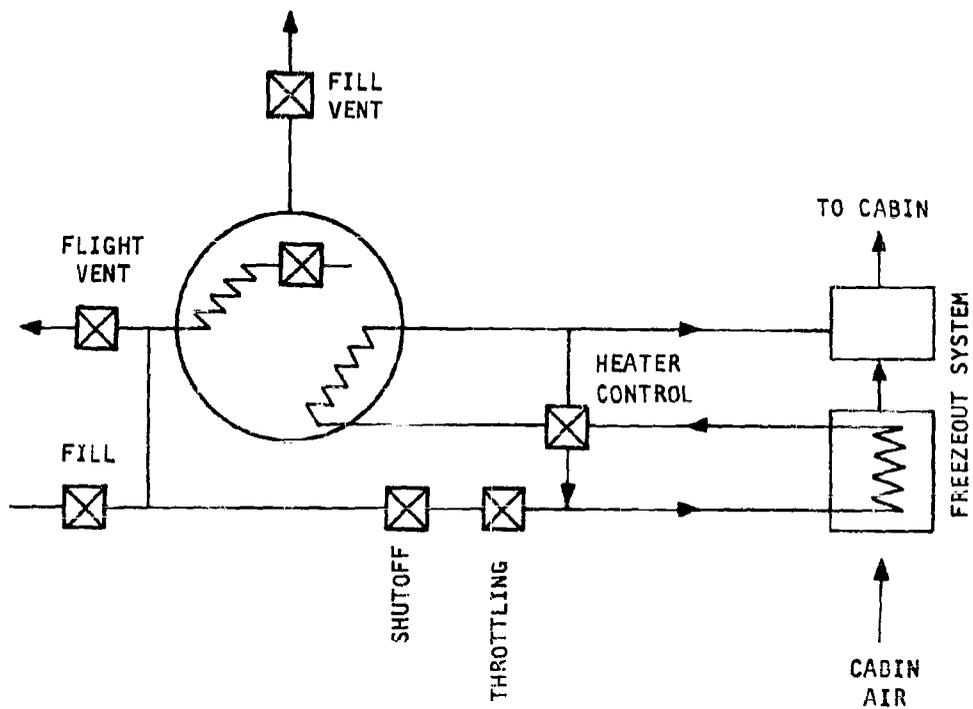


FIGURE 30. TWO-PHASE STORAGE WITH VAPOR DELIVERY, THERMALLY PRESSURIZED

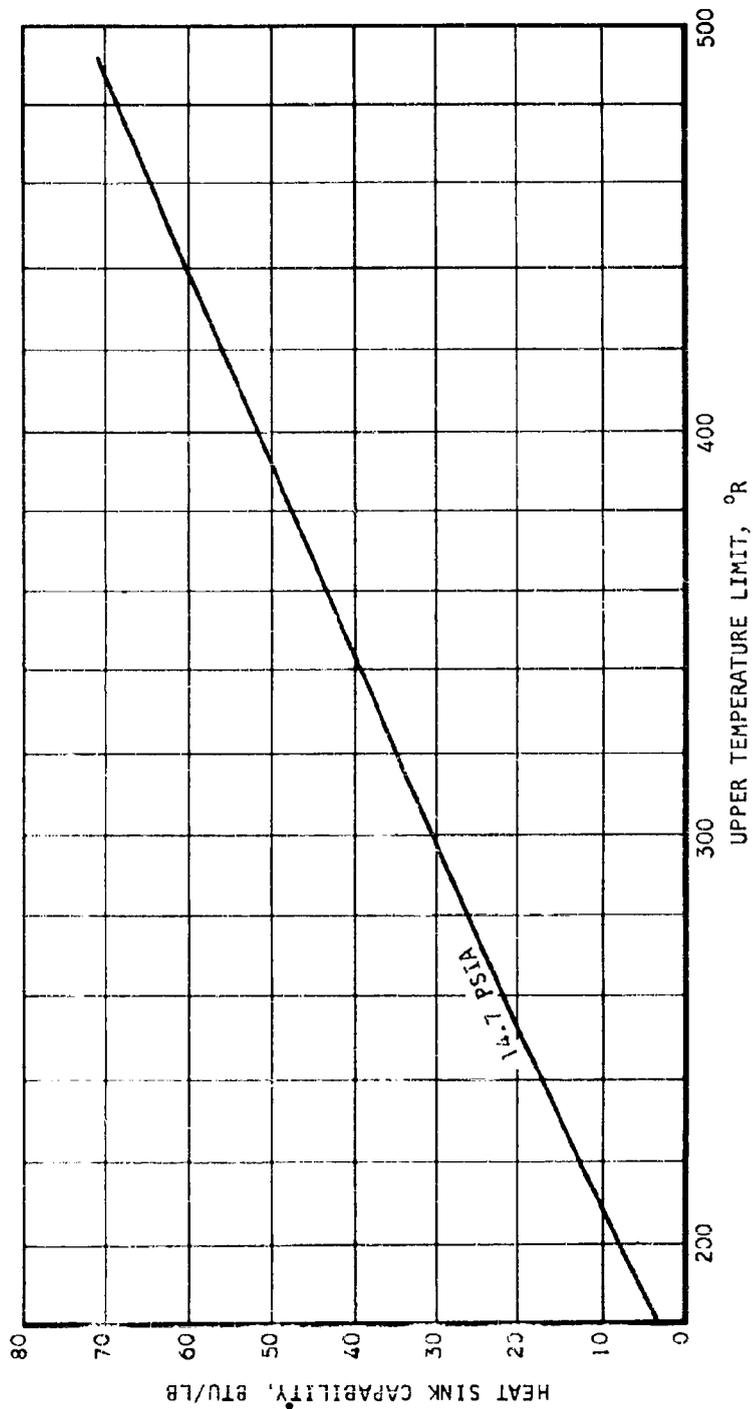


Figure 31. Heat Sink Capability - Two-Phase Storage With Vapor Delivery

Supercritical Storage with Thermal Pressurization

Figure 32 shows thermodynamic process operation on a schematic pressure-enthalpy diagram and illustrates one method of tank heat addition for supercritical oxygen storage. Tank fill conditions are indicated by Point 1. Heating before use results in pressurization at constant average density. In practice, the tank is heated prior to use until the storage pressure is higher than critical (Point 2). Tank temperature rises slightly during this process, but is below critical at Point 2.

Fluid delivery can start once supercritical pressure is reached, with pressure being maintained by heat addition to the storage space. Constant pressure operation in the storage vessel with the inherent temperature increase in the fluid is indicated by the path 2 to 2' on Figure 32. The delivery fluid is throttled through a valve to a lower delivery pressure at Point 3. Referring to Figure 32, state points before throttling may range from 2 to 2', and after throttling from 3 to 3'. The point 3' is the maximum possible oxygen delivery temperature for the contaminant freeze-out system (about 240°R) when the two systems are combined.

Pressurization of the storage vessel is accomplished by heat addition to the fluid. A portion of the heat required for pressurization is supplied by heat leak through the tank walls. The remainder must be supplied by another source such as the waste heat from the space vehicle.

Figure 33 presents a graph of the fluid storage temperature as a function of the weight per cent of fluid remaining in the tank for operating pressures of 875 and 1500 psia respectively. Using a pressure-enthalpy chart for oxygen, it is found that the maximum fluid storage temperature is 320°R for an 875 psia operating pressure for the case where the delivery temperature is 240°R after throttling to 14.7 psia. Referring back to Figure 32, one finds that only 85 per cent of the tank fluid capacity can be used with a containment freeze-out system.

The heat sink capability of the supercritical system varies with the weight per cent of fluid remaining in the tank. Figure 34 presents the heat input requirements for thermal pressurization as a function of the amount of fluid remaining in the tank and Figure 35 presents the heat sink capability of the delivery fluid for the 100 per cent and 15 per cent tank operating conditions.

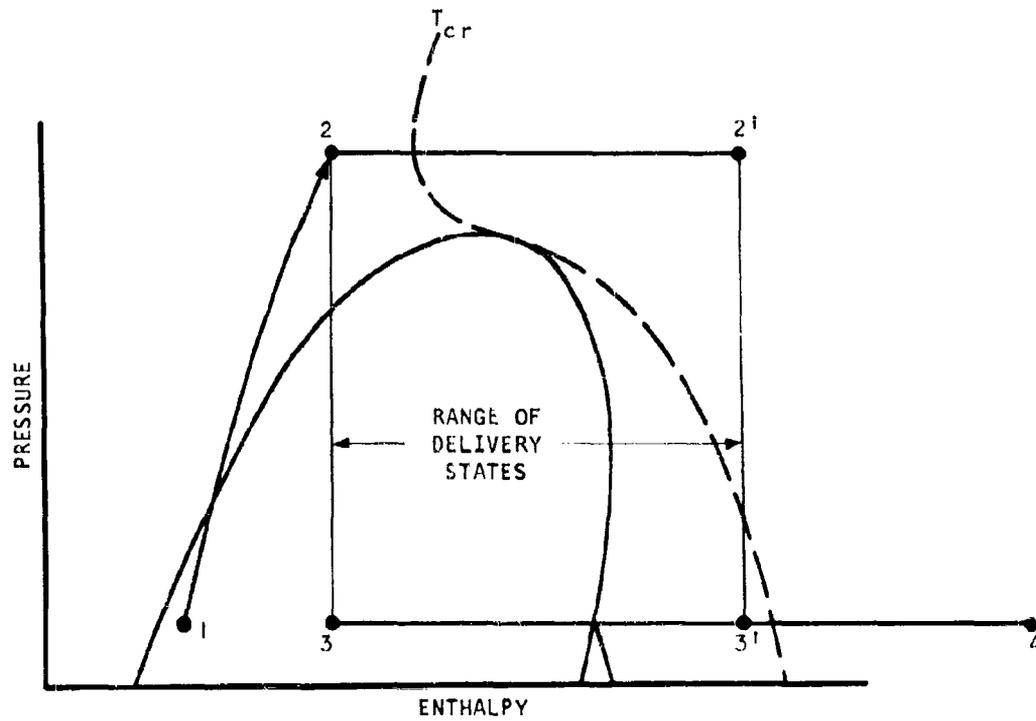
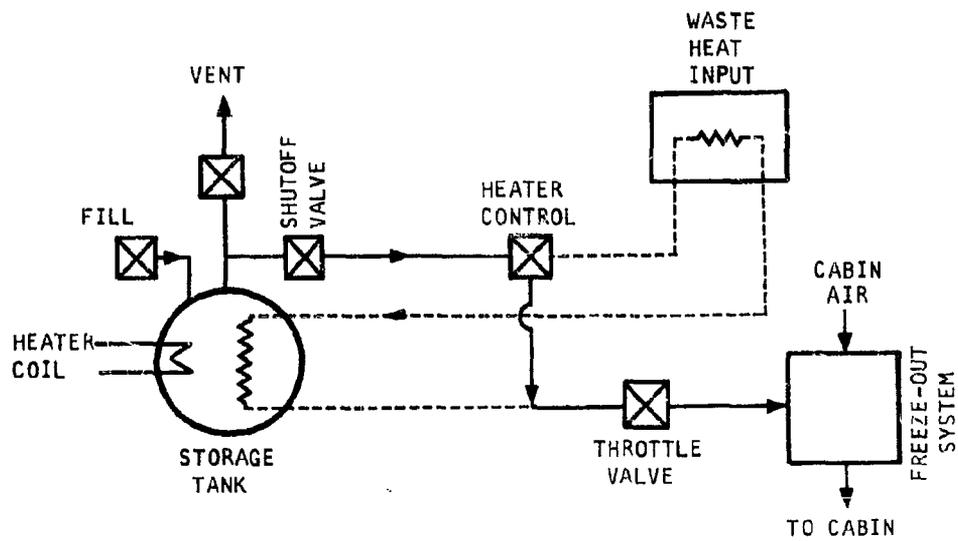


Figure 32. Thermally Pressurized Supercritical Storage

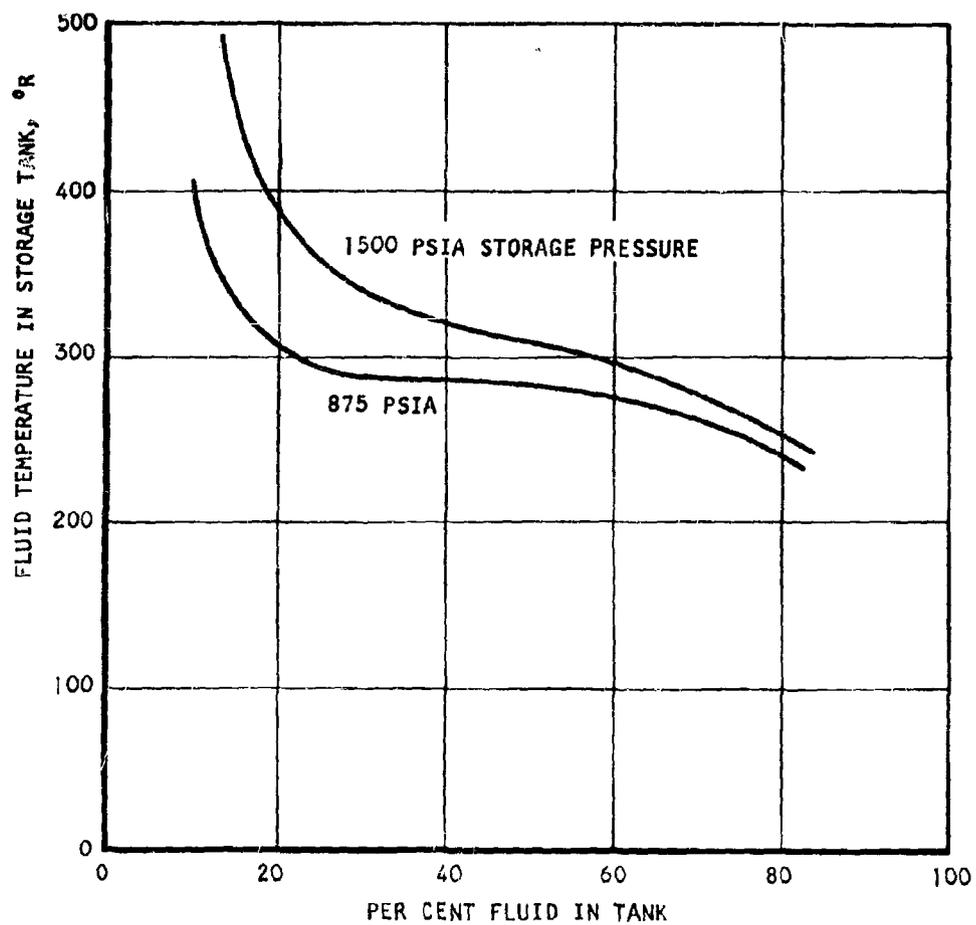


Figure 33. Fluid Temperature for Constant Pressure Delivery - Supercritical Oxygen Storage

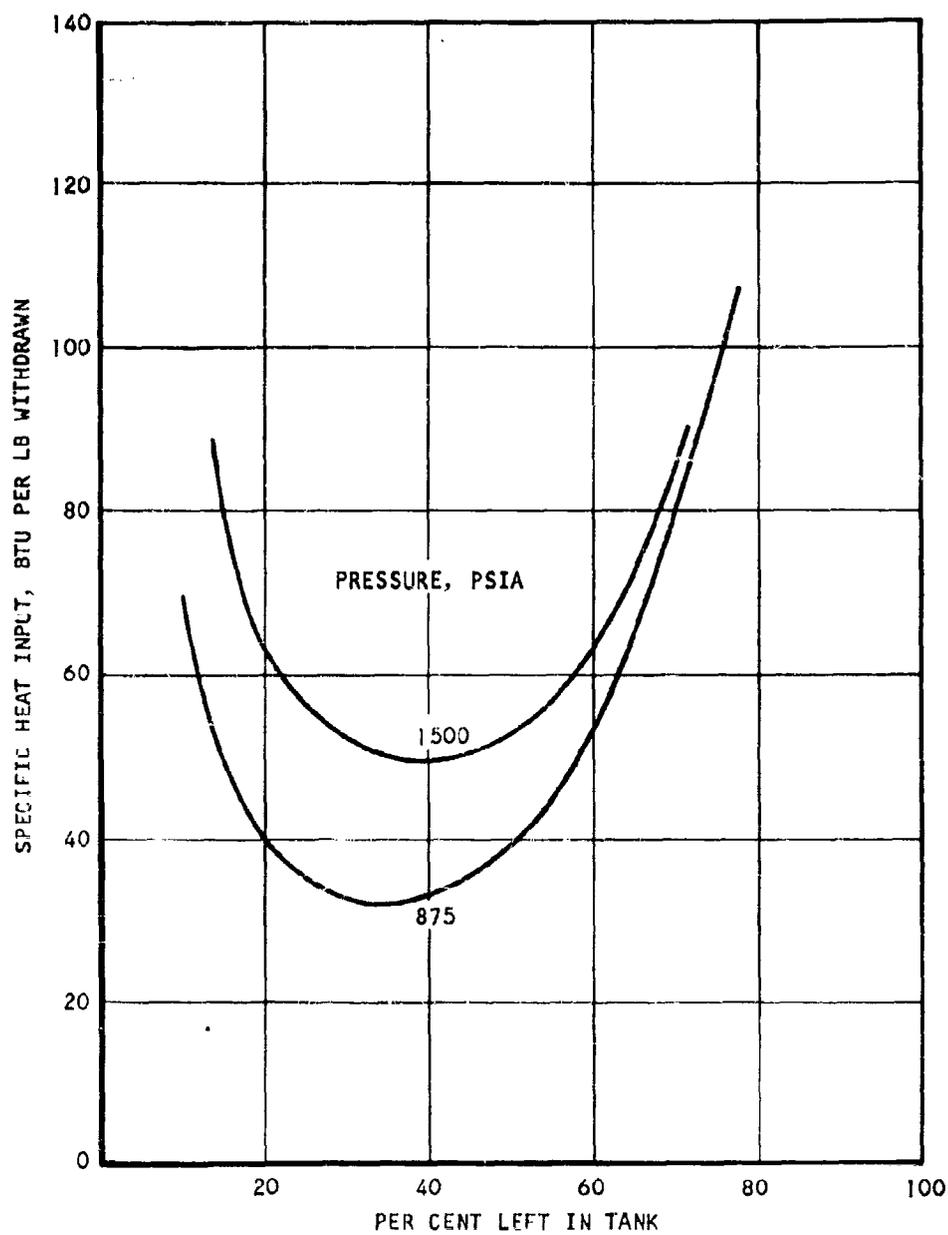


Figure 34. Specific Heat Input for Constant Pressure Delivery - Supercritical Oxygen Storage

CONCLUSIONS

A liquid oxygen heat sink with positive expulsion storage is feasible for CO_2 freeze-out from an overall thermodynamic point of view provided the frozen-out solids are regenerated and dumped overboard as vapors. Positive expulsion storage may be easily integrated with the freeze-out system.

Two phase storage of liquid oxygen is not suitable for CO_2 freeze-out unless the frozen-out solids are regenerated, and an auxiliary heat sink, such as cryogenic hydrogen, is available at the cold end of the CO_2 freeze-out heat exchanger. The reason for this is the heat sink capability of two phase storage is only about one-half of the heat sink capability of positive expulsion storage.

Supercritical storage of oxygen is not suitable for CO_2 freeze-out because the heat sink capability decreases as fluid is drawn out of the tank. The heat sink temperature (which is the temperature of the fluid after throttling from tank pressure to freeze-out system pressure) is constant for about 70 per cent of the storage vessel contents. Then, when liquid is no longer produced by the throttling process, the heat sink temperature rises as more fluid is withdrawn from the tank. The heat sink temperature exceeds the minimum temperature required for CO_2 freeze-out when the tank is about 15 per cent full.

<p>UNCLASSIFIED</p> <p>I. Space Biophysics, Closed-Cycle Ecological Systems</p> <p>I. AFSC Project 6373, Task 637302</p> <p>II. Contract AF 33(616)-7768</p> <p>III. AiResearch Manufacturing Co., Los Angeles 45, Calif.</p> <p>IV. Wright, C. C.</p> <p>V. Secondary Rpt No. SS-596-R, Rev. 1</p> <p>UNCLASSIFIED</p>	<p>Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio</p> <p>Rpt. No. NRL-TDR-62-7. CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS. Final report, Feb. 1962. 118p. incl illus., tables, 9 refs.</p> <p>Unclassified report</p> <p>An analytical feasibility study was conducted on contaminant freeze-out (water vapor and CO₂) systems employing the endothermic conversion of liquid to gaseous oxygen as a heat sink. Simple freeze-out systems, combination freeze-out and adsorption systems, and systems with auxiliary cooling were studied and compared. For minimum</p> <p>(over)</p>	<p>UNCLASSIFIED</p> <p>I. Space Biophysics, Closed-Cycle Ecological Systems</p> <p>I. AFSC Project 6373, Task 637302</p> <p>II. Contract AF 33(616)-7768</p> <p>III. AiResearch Manufacturing Co., Los Angeles 45, Calif.</p> <p>IV. Wright, C. C.</p> <p>V. Secondary Rpt No. SS-596-R, Rev. 1</p> <p>UNCLASSIFIED</p>	<p>Aerospace Medical Division, 6570th Aerospace Medical Research Laboratories, Wright-Patterson Air Force Base, Ohio</p> <p>Rpt. No. NRL-TDR-62-7. CONTAMINANT FREEZE-OUT STUDY FOR CLOSED RESPIRATORY SYSTEMS. Final report, Feb. 1962. 118p. incl illus., tables, 9 refs.</p> <p>Unclassified report</p> <p>An analytical feasibility study was conducted on contaminant freeze-out (water vapor and CO₂) systems employing the endothermic conversion of liquid to gaseous oxygen as a heat sink. Simple freeze-out systems, combination freeze-out and adsorption systems, and systems with auxiliary cooling were studied and compared. For minimum</p> <p>(over)</p>
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