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AIR PRODUCTS AND CHEMICALS, INC.

RESEARCH AND DEVELOPMENT DEPARTMENT

PART II. FINAL REPORT

CONTRACT NAS9-1782

HOUDRY PROCESS AND CHEMICAL CO.

FEASIBILITY STUDY OF A LOW TEMPERATURE CARBON DIOXIDE REMOVAL SUBSYSTEM

Technical Report No. 70

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Date

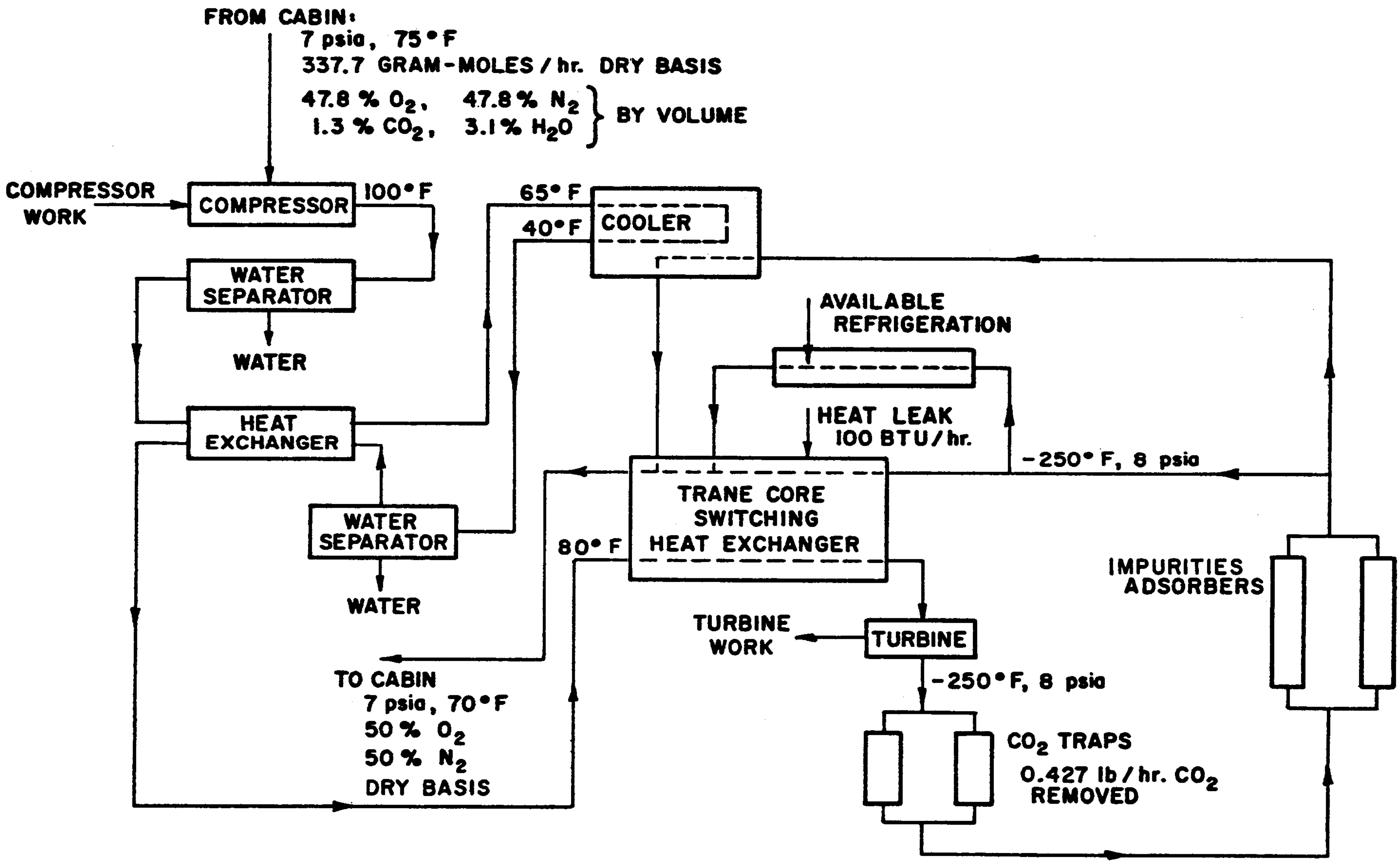
ABSTRACT

This study was conducted to determine the feasibility of removing metabolic carbon dioxide (CO_2) and water from the atmosphere of a four-man space cabin by means of a low-temperature freezeout process. The system is to run continuously for one year. The recovered carbon dioxide is passed to a reduction system where it is reduced to breathable oxygen and waste hydrocarbons. A portion of the recovered water is passed to an electrolytic cell and the remainder is used as drinking water.

The carbon dioxide is produced at a rate of 1 gram-mole metabolic CO_2/man-hour plus 0.4 gram-moles regenerated CO_2/hour. This is equal to 0.427 lb CO_2/hour. Metabolic water is produced at a rate of 1 lb H_2O/lb CO_2. The cabin atmosphere was composed of 47.8% oxygen, 47.8% nitrogen, 1.3% carbon dioxide, and 3.1% water by volume. The cabin atmosphere pressure is 7 psia.

The results of this study indicated that the CO_2 removal system shown on page ii is feasible for systems having a compressor discharge pressure equal to or greater than 70 psia. Power requirement was minimum for the 70 psia system and was 550 watts.

The equipment will require a volume of 7 ft^3 and will weigh 200 lbs.



CARBON DIOXIDE REMOVAL AND AIR CONDITIONING
SYSTEM FOR FOUR-MAN SPACE CABIN

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

I. INTRODUCTION

This study was conducted to determine the feasibility of removing metabolic carbon dioxide and water from the atmosphere of a four-man space cabin by means of a low-temperature freezeout process. The removed carbon dioxide is passed to a reduction system where it is reduced to breathable oxygen and waste hydrocarbons. The recovered water is used for electrolysis and human consumption. The integrated system is schematically shown in Figure 1.

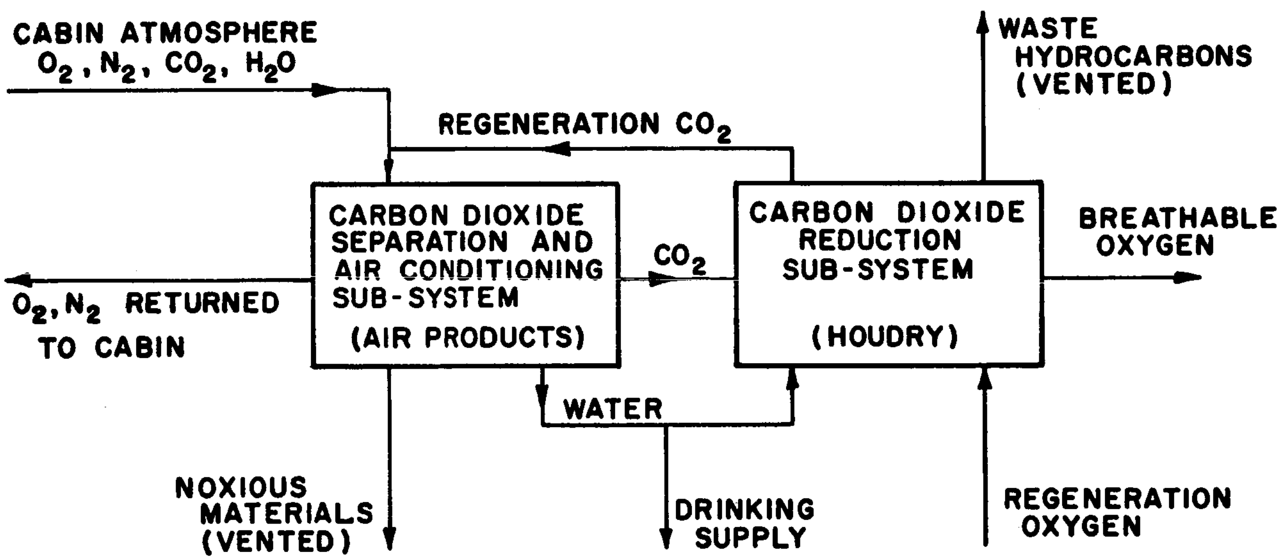


FIGURE I. INTEGRATED CARBON DIOXIDE REDUCTION SYSTEM. NAS 9 - 1782

The carbon dioxide removal subsystem is schematically shown and identified in Figure 2. Flow rates and compositions are also shown in Figure 2.

It was desired to determine the thermodynamic process which would result in the minimum net power consumption and minimum system weight. Power is available at a penalty of 300 pounds per kilowatt.

The following items were assumed constant for all calculations:

1. Compressor inlet conditions
2. Compressor outlet temperature
3. Cooler temperature
4. Inlet temperature of high pressure stream to switching heat exchanger
5. Temperature and pressure of turbine outlet
6. System heat leak
7. Temperature and pressure of low pressure stream leaving the switching heat exchanger
8. Carbon dioxide removed.

The criteria used in determining the constants are discussed under the Process Design Criteria.

Compressor discharge pressure was the independent variable of the system. The dependent variables were:

1. Net power consumption
2. Balancing heat load on switching heat exchanger
3. Outlet temperature of high pressure stream in the switching heat exchanger.

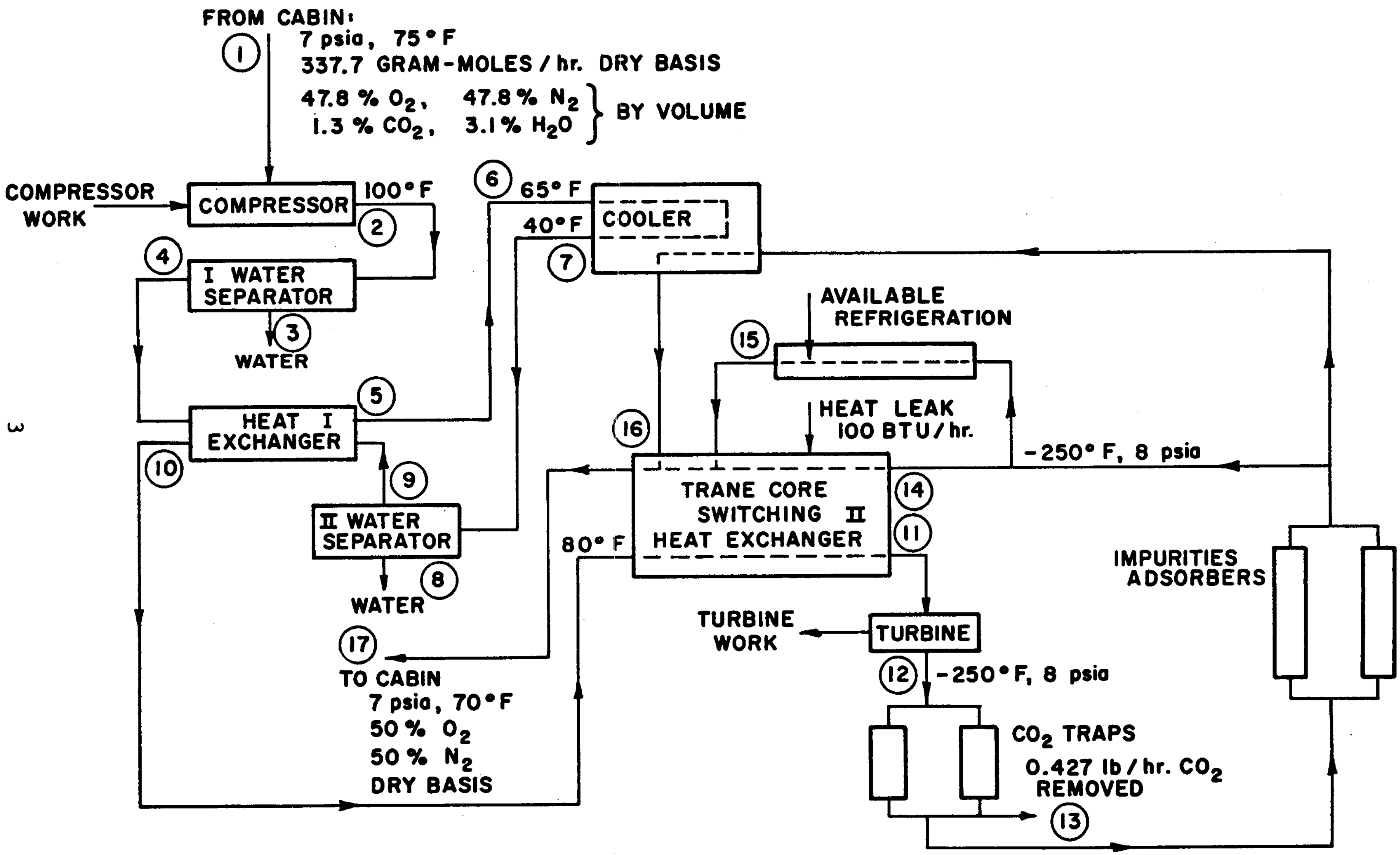


FIGURE 2. CARBON DIOXIDE REMOVAL AND AIR CONDITIONING SYSTEM FOR FOUR-MAN SPACE CABIN

The net power required by the system was calculated for compressor discharge pressures of 40, 75, 100, 150, 400, and 1000 psia. Results of these calculations are shown in Table I and Figures 3, 4, 5, 6, and 7.

All systems are feasible for discharge pressures equal to or greater than 70 psia. The minimum power requirement system occurred at a compressor discharge pressure of 70 psia. Power consumption of this system was 550 watts.

The equipment considerations are discussed under Equipment Criteria. A detailed set of calculations for the 100 psia system is presented under Calculations of CO_2 Removal Systems.

TABLE I.

CALCULATED RESULTS OF THE CARBON DIOXIDE REMOVAL SYSTEM

Comp. Discharge Pressure, Psia	40	75	100	150	400	1000
Comp. Work, Watts	623	790	887	1022	1342	1640
Number of Stages	1	2	2	2	3	4
Cooler Refrigeration, Watts	127.3	79.8	67.0	56.5	40.4	35.2
Switching Heat Exchanger Feed Stream Outlet Temperature, °F	-160	-150	-145	-140	-120	-105
Balancing Heat Load, Watts	104	120	128	134	157	164
Available Refrigeration Watts	- 52.7	10.8	31.6	48.1	87.2	99.4
Turbine Work, Watts	161	176	182	186	204	197
Required Turbine Efficiency, %	104	84	69	59	57	51
Recoverable Refrigeration from CO_2 Traps, Watts	40	40	40	40	40	40
Net Power Consumption, Watts	435	563	633	748	1011	1304
Water Removed, lb/hour						
(a) Compression	0.092	0.257	0.299	0.340	0.394	0.413
(b) Cooler	0.294	0.148	0.133	0.081	0.028	0.011
Total Water Removed, lb/hour	0.386	0.405	0.411	0.421	0.422	0.424

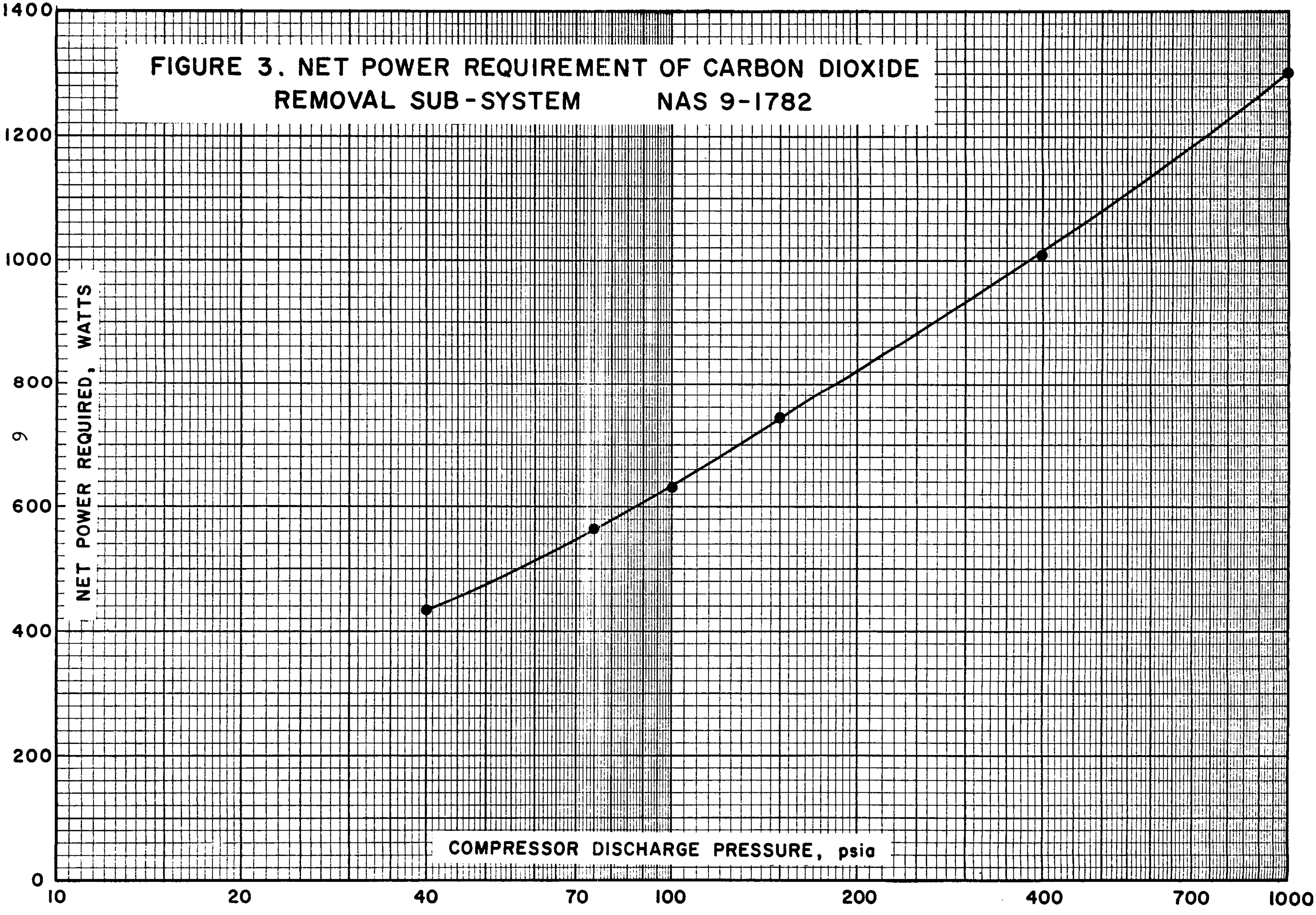


FIGURE 3. NET POWER REQUIREMENT OF CARBON DIOXIDE REMOVAL SUB-SYSTEM NAS 9-1782
NET POWER REQUIRED, WATTS
COMPRESSOR DISCHARGE PRESSURE, psia
1400
1200
1000
800
600
400
200
0
10
20
40
70
100
200
400
700
1000

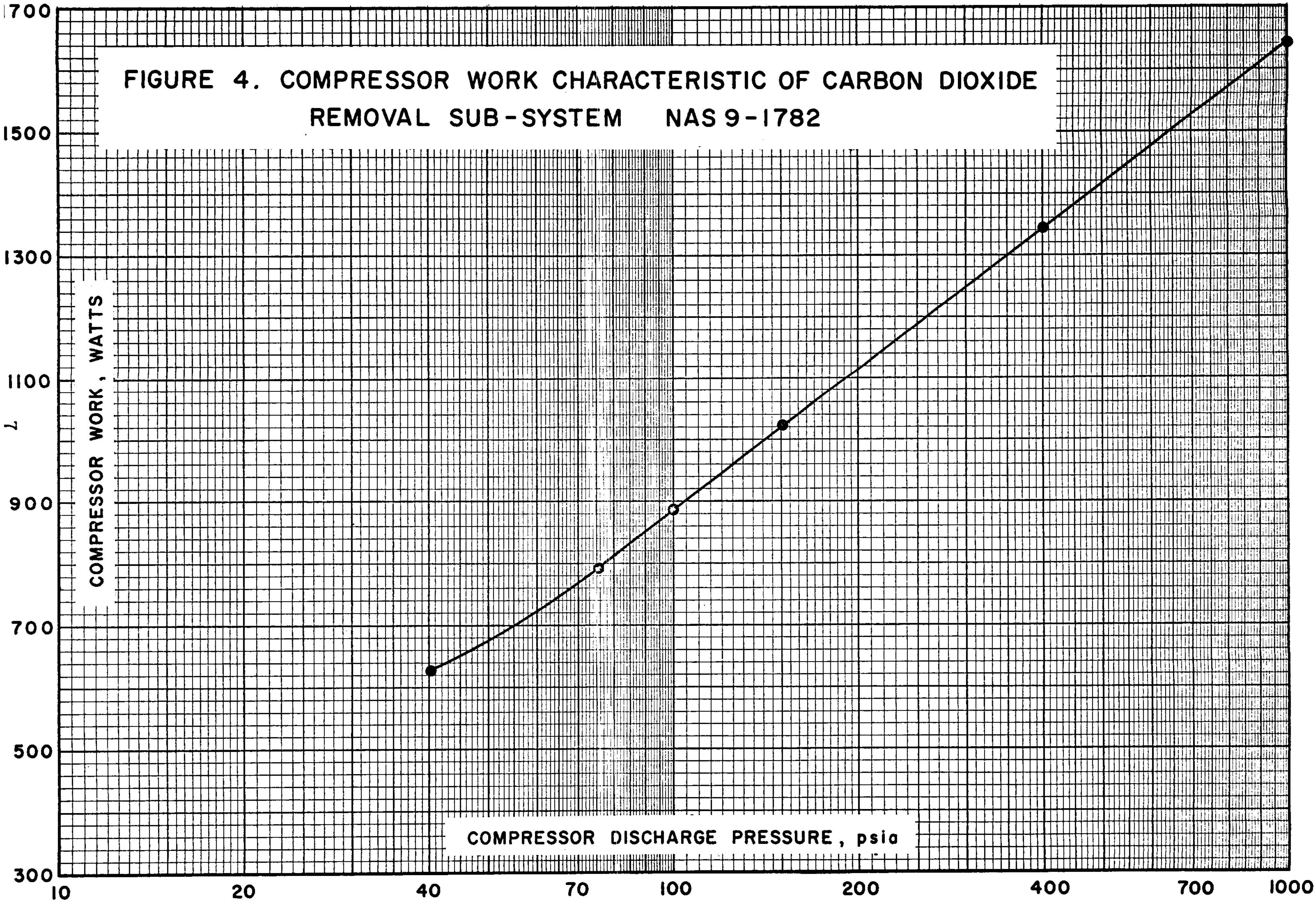


FIGURE 4. COMPRESSOR WORK CHARACTERISTIC OF CARBON DIOXIDE REMOVAL SUB-SYSTEM NAS 9-1782

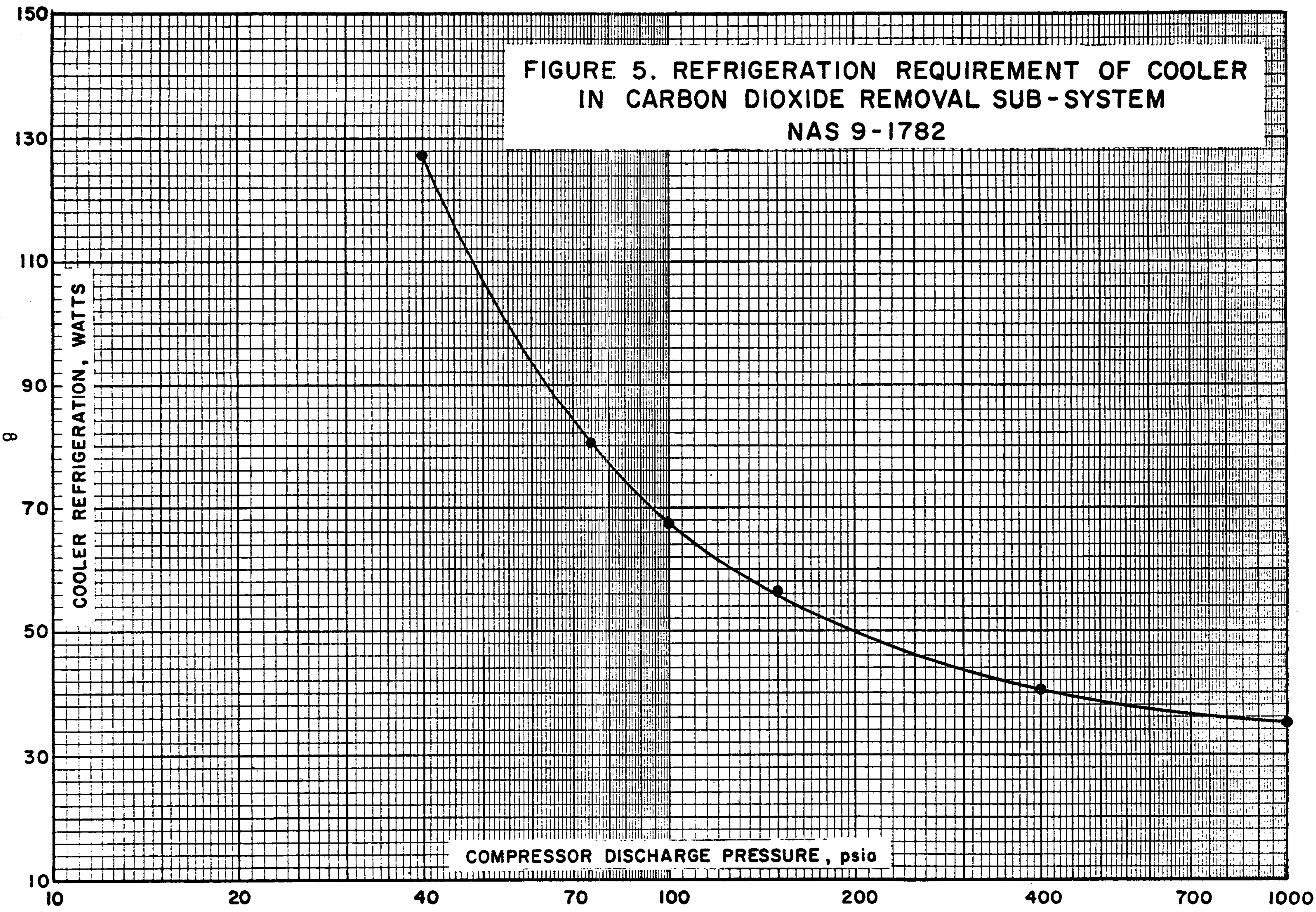
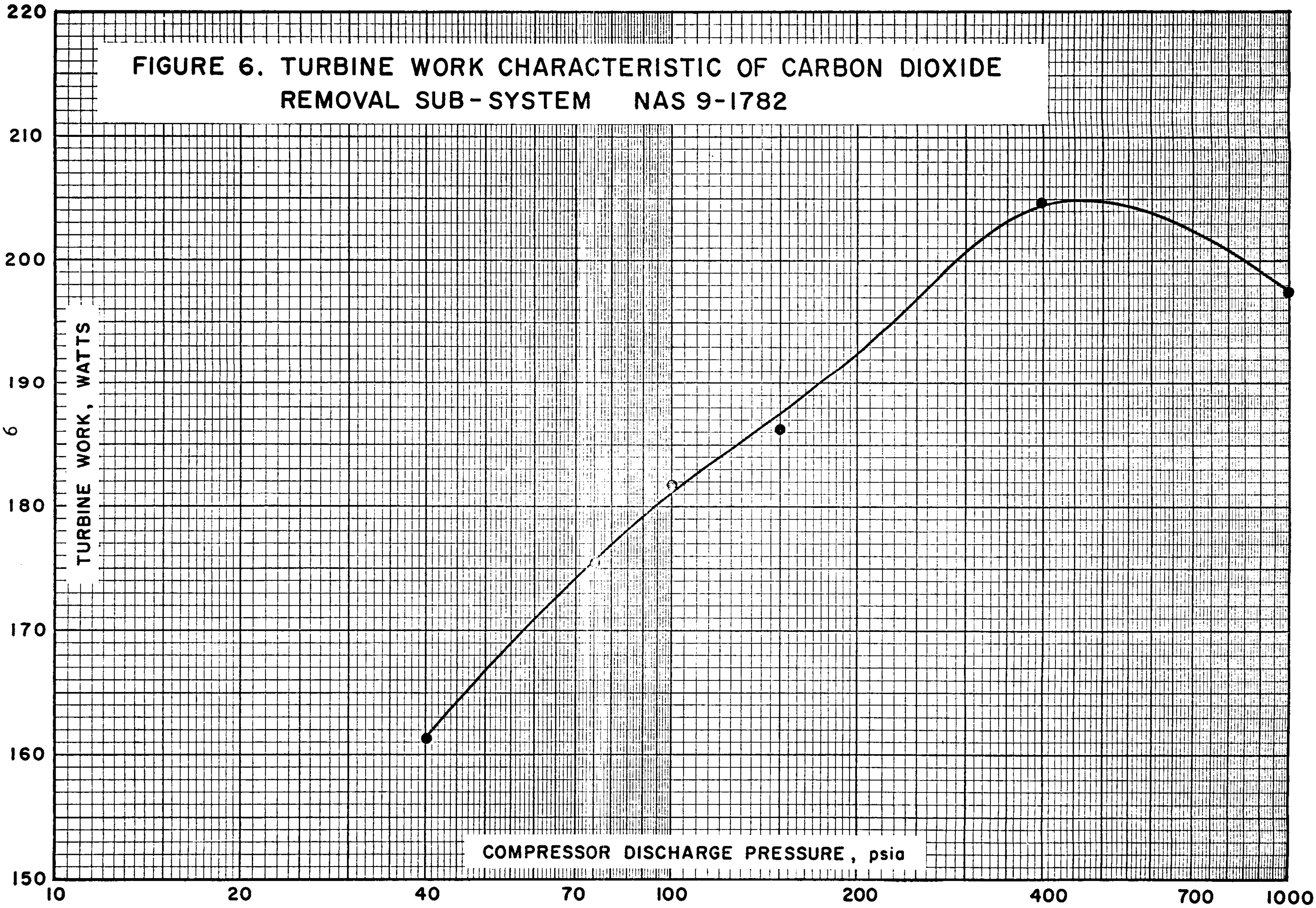


FIGURE 5. REFRIGERATION REQUIREMENT OF COOLER
IN CARBON DIOXIDE REMOVAL SUB-SYSTEM
NAS 9-1782
COOLER REFRIGERATION, WATTS
COMPRESSOR DISCHARGE PRESSURE, psia
150
130
110
90
70
50
30
10
10
20
40
70
100
200
400
700
1000

FIGURE 6. TURBINE WORK CHARACTERISTIC OF CARBON DIOXIDE REMOVAL SUB-SYSTEM NAS 9-1782



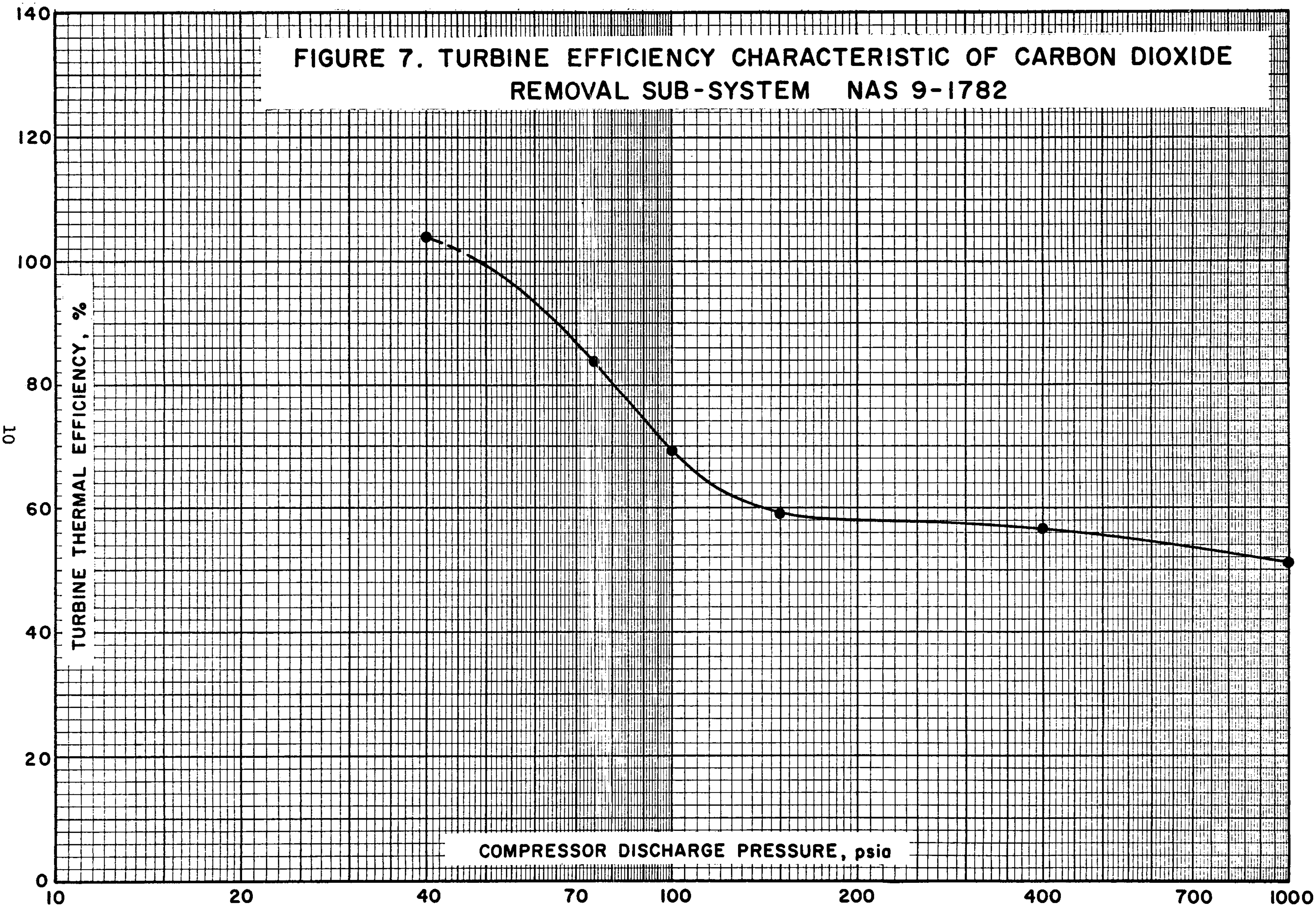


FIGURE 7. TURBINE EFFICIENCY CHARACTERISTIC OF CARBON DIOXIDE REMOVAL SUB-SYSTEM NAS 9-1782

II. PROCESS DESIGN CRITERIA

CONSTANT QUANTITIES

1. Compressor Inlet Conditions

The system feed stream is drawn from the cabin atmosphere. The cabin pressure and oxygen-nitrogen composition were specified by NASA. The CO_2 flow rate of 0.427 lb/hour was based upon a metabolic production rate of 1 gram-mole CO_2/man-hour plus a regeneration rate of 0.4 gram-mole CO_2/hour. Metabolic water production was taken as 1 lb H_2O/lb CO_2. The metabolic CO_2 and H_2O production rates were based upon a 3000 calorie diet.

2. Compressor Outlet Temperature

The compressor outlet temperature was selected as 100°F. This temperature can be readily attained by available commercial equipment.

3. Cooler Temperature

The design cooler temperature was taken as 40°F. This temperature was low enough to assure condensation of a major portion of water vapor, yet it was high enough to prevent freezing of the condensed water.

4. Inlet Temperature of Feed Stream to Switching Heat Exchanger

The counterflow return stream exit temperature was 70°F as determined by cabin conditions. The temperature difference at the warm end of the switching heat exchanger was selected as 10°F, a figure commonly used in exchanger design. The inlet temperature of the exchanger was, therefore, fixed at 80°F.

5. Temperature and Pressure of Turbine Outlet

The pressure at the turbine outlet was determined by the cabin pressure and the pressure drop between the turbine and cabin. The cabin pressure was 7 psia and the pressure drop was estimated at 1 psi. Therefore, the turbine outlet pressure was taken as 8 psia. The turbine outlet temperature was chosen as -250°F. This temperature was high enough to assure that oxygen would not be liquefied in the flow stream. The concentration of CO_2 in the stream to the cabin is 0.29 ppm.

6. Heat Leak

The heat leak was calculated based upon the use of evacuated multiple layer insulation. The generator used to load the turbine was assumed to have an efficiency of 95%. Since the generator is located inside the cold box, the 5% losses were considered as a heat leak into the system. The size of the cold box and equipment does not vary appreciably for the various systems considered, therefore, the heat leak was considered a constant.

7. Outlet Temperature and Pressure of Return Stream on Switching Heat Exchanger

The outlet temperature and pressure of the return stream in the switching heat exchanger were 70°F and 7 psia respectively, as determined by the desired cabin atmosphere.

8. Carbon Dioxide Removal

Carbon dioxide is removed at the rate of 0.427 lb/hour. This rate was determined as discussed previously. Since the design CO_2 rate is only 1.3% of the total flow, variations in the metabolic generation rate caused by periods of physical exertion will not produce appreciable effects in the system power requirements.

INDEPENDENT VARIABLE

Compressor Discharge Pressure

The first system calculations were made for a discharge pressure of 150 psia. When this system was found to be feasible, calculations were made for discharge pressures of 100, 75, and 40 psia. The 40 psia system was not feasible because of insufficient refrigeration and unachievable required turbine efficiencies. Calculations were then made for discharge pressures of 400 and 1000 psia. The net power requirement characteristics indicated that there was no advantage in making calculations at higher pressures.

DEPENDENT VARIABLE

Outlet Temperature of Feed Stream in the Switching Heat Exchanger

The outlet temperature of the feed stream in the switching heat exchanger was chosen as 10°F above the saturation temperature of CO_2, taken at the partial pressure of CO_2. A 10°F temperature margin will assure that CO_2 will not freeze on the exchanger walls for slight fluctuations of the turbine outlet temperature.

III. EQUIPMENT CRITERIA

COMPRESSOR

The compression ratio and working fluid used in this system are commonly handled by commercial compressors. Lubrication and weight are special areas which must be considered. The compressor weight can be minimized by using lightweight metals such as aluminum and titanium now used by the aircraft industry.

The flow stream should be maintained oil free because of oxygen incompatibility and contamination of drinking water and flow lines. The use of an oil-free compressor will eliminate the lubrication problem. Piston type oil-free compressors are available commercially[11], however, the volume flow rates are several times larger than the 348.5 gram-moles/hour of the CO_2 removal system. A development program would be required to scale down the existing compressors.

The heat of compression may be used to heat the cabin. If this function is already fulfilled, the compressor may be mounted externally to the cabin and the heat can be dissipated by radiators.

WATER SEPARATION IN A ZERO GRAVITY FIELD

The water can be removed by utilizing the kinetic energy of the flow stream in a vortex type separator. The flow stream is conducted through a spiral path and the water droplets are forced against the container wall by centrifugal force. The liquid is moved into a reservoir by fluid friction.

Separators have been developed in which the water droplets are impinged onto a rotating impeller. The impeller hurls the water radially outward. The use of moving parts makes this type of separation less desirable than the vortex type.

TURBINE

A turbine having the characteristics required by the CO_2 removal system is not available from commercial suppliers. Barbour-Stockwell[7], manufacturers of small air turbines, have indicated that modifications could be made on their model 1001 which would result in the thermodynamic expansion required by the CO_2 removal subsystem.

The selection of turbine bearings must also be considered. Three types of bearings are available:

1. Oil lubricated bearings
2. Gas lubricated bearings
3. Dry bearings.

Oil lubricated bearings require the use of an oxygen compatible oil such as a halocarbon. Dry bearings such as Graphalloy[9,10] have been operated at 100,000 RPM. This material is compatible with oxygen at low temperature.

It is recommended that a turbine development program be undertaken starting with the modified Barbour-Stockwell model 1001 turbine having Graphalloy bearings.

The device to load the turbine is conceived as an electrical generator mechanically coupled to the turbine. This turbo-generator would be sealed in a metal case located in the evacuated cold box. The pressure in the turbo-generator case can be maintained positive thus preventing any problems associated with generator lubrication in a vacuum.

NOXIOUS MATERIALS

The major noxious materials which may be found in the cabin atmosphere are sulfur dioxide, methane, freons, and bacteria[3].

Sulfur Dioxide, SO_2: The SO_2 will combine with the water to form sulfurous acid and will be removed in the drinking water.

Bacteria: Bacteria can be removed from the process stream by means of an ultrafine sterilization filter. One of this type of filters is the Ultipor[15].

Methane, CH_4: The melting point of methane is -296.5°F at 169 psia. Any methane entering the CO_2 removal subsystem will pass through the system unaffected since the lowest temperature in the system is -250°F. An adsorption column of silica gel located on the low pressure stream between the turbine and switching heat exchanger will remove the methane from the system.

Freon: The boiling and melting points of the Freons are shown in Table II[8].

TABLE II

BOILING AND MELTING TEMPERATURE OF VARIOUS FREONS AT 1 ATMOSPHERE

Freon	Boiling Temperature °F	Freezing Temperature °F
11	+ 74.8	-168
12	+ 21.6	-252
13	-114.6	-294
13B1	- 72.0	-270
14	-198.4	-299
21	+ 48.1	-211
22	- 41.4	-256
23	-115.7	-247.4
112	+199.0	+ 74.8
113	+117.6	- 31
114	+ 38.4	-137
114B2	+117.1	-166.8
115	- 37.7	-159
116	-108.8	-149.1
C318	+ 21.5	- 42.5
502	+ 50.1	----

Although most of the Freons theoretically would be in the solid state at the turbine outlet, at any particular time, the quantity of Freon present is insufficient to form an appreciable amount of solid particles. The random molecules of Freon passing through the system will be adsorbed in the silica gel bed located between the CO_2 traps and the switching heat exchanger[1]. The bed can be reactivated by venting it to the atmosphere outside the cabin.

IV. CALCULATION PROCEDURES

COMPRESSOR WORK

The required number of compression stages was calculated based upon equal work per stage. The equation from the literature[14] expressing this condition is

$$\frac{P_x}{P_1} = \sqrt[n]{\frac{P_2}{P_1}}$$

where:

P_x is the discharge pressure of the first stage

P_1 is the compressor inlet pressure

P_2 is the compressor discharge pressure

n is the number of stages.

The left side of the equation is constant for each stage. A value of 4 is used in most compressor designs.

A number of stages n was assumed and P_x/P_1 was calculated. If P_x/P_1 was greater than 4, a higher integer was chosen for n. If P_x/P_1 was less than 4, a lower integer was chosen for n.

When n was determined, the compressor work was obtained[6].

BALANCING HEAT LOAD ON THE SWITCHING HEAT EXCHANGER

The outlet temperature of the feed stream T11 was determined as discussed previously. When T11 was determined, the balancing heat load was calculated by writing an energy balance on the heat exchanger as shown in Figure 8.

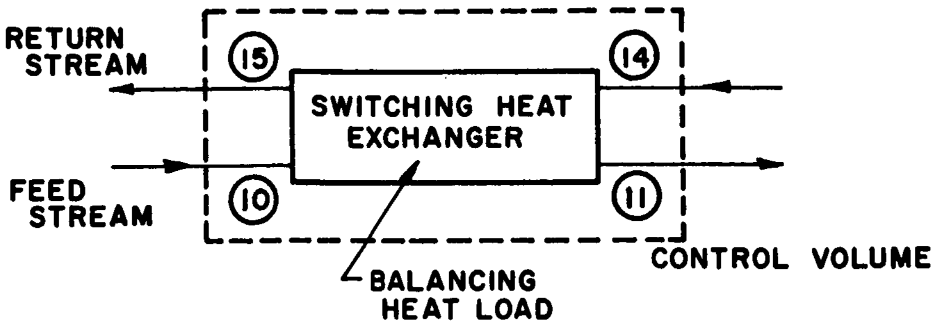


FIGURE 8. ENERGY FLOW DIAGRAM OF SWITCHING HEAT EXCHANGER

The heat exchanger size was calculated as shown on page 33. Calculations indicated that a heat leak of 100 BTU/hour would be obtained using evacuated multiple layer insulation. For systems having a compressor discharge pressure above 40 psia, the balancing heat load was greater than 100 BTU/hour. Therefore, an additional heat load would have to be provided. This additional heat load would have to be equal to the balancing heat load minus 100 BTU/hour. The cooler refrigeration will provide a portion of the load. The available refrigeration shown in Figure 2 represents the remainder of the required load and may be used for cooling equipment anywhere in the cabin. If this refrigeration cannot be used in the cabin, the flow stream can be passed to a radiator external to the cabin.

For systems having a compressor discharge pressure less than 66 psia, the total heat load is greater than the balancing heat load, and auxiliary refrigeration would be required.

TURBINE

The turbine work represents the amount of energy removed from the flow stream in order to maintain the system in thermal equilibrium. The turbine work was calculated by writing an energy balance on the heat exchanger-turbine circuit as shown in Figure 9.

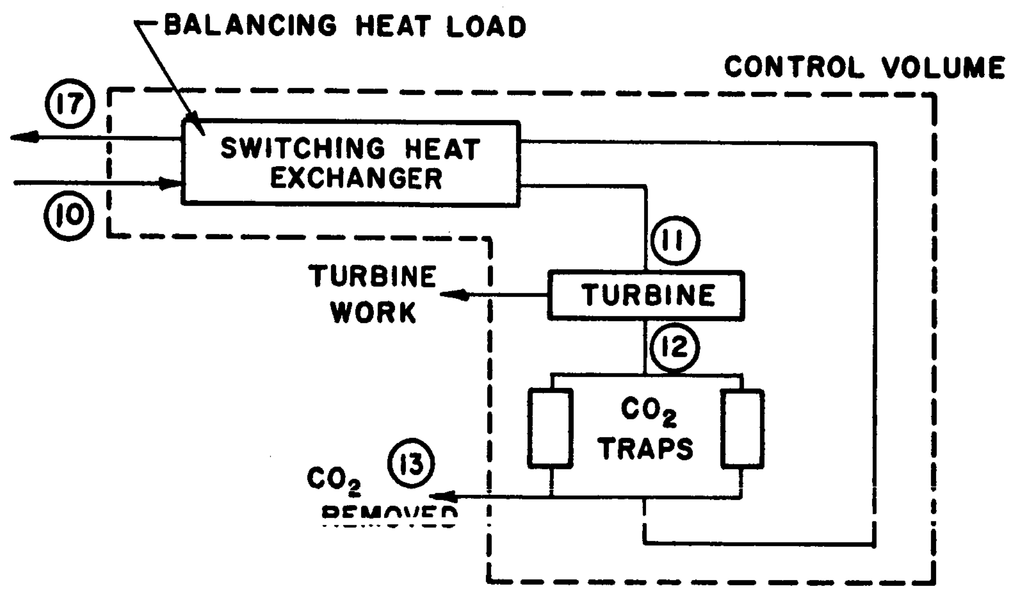


FIGURE 9. ENERGY FLOW DIAGRAM OF TURBINE-SWITCHING HEAT EXCHANGER GROUP

Once the turbine work was computed, the required turbine efficiency could be calculated. The efficiency was defined as $\frac{\text{Actual Turbine Work}}{\text{Isentropic Turbine Work}} \times 100$.

The isentropic turbine work was equal to m $(h_{11}-h_{12s})$, where m is the system flow rate and $(h_{11}-h_{12s})$ is the isentropic enthalpy change between points 11 and 12.

The use of Joule-Thomson expansion was considered instead of the turbine. In order to attain 7 psia and -250°F at the valve outlet, the valve inlet temperature T11 would be such that CO_2 would freeze onto the walls of the switching heat exchanger. Thus, the use of Joule-Thomson expansion is not feasible in this system.

V. CALCULATIONS OF CO_2 REMOVAL SYSTEM HAVING 100 PSIA COMPRESSOR DISCHARGE PRESSURE

Cabin Atmosphere Analysis:

Given: $\frac{4.4 \text{ gram-mole } CO_2}{\text{hour}}$ and $\frac{1 \text{ lb } H_2O}{\text{lb } CO_2}$

$$\left(\frac{453 \text{ grams } H_2O}{453 \text{ grams } CO_2}\right)\left(\frac{1 \text{ gram-mole } H_2O}{18 \text{ grams } H_2O}\right)\left(\frac{44 \text{ grams } CO_2}{1 \text{ gram-mole } CO_2}\right) = \frac{2.44 \text{ gram-mole } H_2O}{\text{gram-mole } CO_2}$$

$$\left(\frac{2.44 \text{ gram-mole } H_2O}{\text{gram-mole } CO_2}\right)\left(\frac{4.4 \text{ gram-mole } CO_2}{\text{hour}}\right) = \frac{10.8 \text{ gram-mole } H_2O}{\text{hour}}$$

Given: $7568 \frac{\text{standard liters}}{\text{hour}}$ of CO_2, O_2, N_2

$$\left(\frac{7568 \text{ standard liters}}{\text{hour}}\right)\left(\frac{\text{gram-mole}}{22.4 \text{ standard liters}}\right) = \frac{337.9 \text{ gram-mole } CO_2, O_2, N_2}{\text{hour}}$$

Total Flow Rate = 337.9 + 10.8 = 348.7 $\frac{\text{gram-mole}}{\text{hour}}$

$$\% \; CO_2 = \frac{4.4 \times 100}{348.7} = 1.26\%$$

$$\% \; H_2O = \frac{10.8 \times 100}{348.7} = 3.10\%$$

$$100.00\% - 4.36 = 95.64\% \; O_2, N_2$$

Given: O_2 and N_2 are present in equal volume percent.

$$\% \; O_2 = \frac{95.64}{2} = 47.82\%$$

$$\% \; N_2 = \frac{95.64}{2} = 47.82\%$$

Analysis of Cabin Atmosphere:

O_2 = 47.82%

N_2 = 47.82%

CO_2 = 1.26%

H_2O = $\underline{3.10\%}$

100.00%

COMPRESSOR WORK

Compression Ratio $\frac{100}{7} = 14.29$

For a two-stage reciprocating machine with equal work per stage, the compression ratio per stage is $\frac{P_x}{P_1}$ where P_x and P_1 is the interstage and inlet pressure respectively.

$$\frac{P_x}{P_1} = \sqrt{\frac{100}{7}} = 3.78$$

K = 1.4 (ref. 6)

The brake horsepower is $\frac{12.8 \text{ HP}}{(100 \text{ SCFM})(\text{Stage})}$

The total flow was shown to be 348.7 $\frac{\text{gram-mole}}{\text{hour}}$

$$\left(\frac{348.7 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{22.4 \text{ standard liters}}{\text{gram-mole}}\right)\left(\frac{.0353 \text{ ft}^3}{\text{liter}}\right)\left(\frac{\text{hour}}{60 \text{ min.}}\right) = 4.6 \text{ SCFM}$$

$$\text{Brake horsepower} = \left(\frac{12.8 \text{ HP}}{100 \text{ SCFM x stage}}\right)\underline{(4.6 \text{ SCFM})\ (2 \text{ stages})} = 1.2 \text{ HP}$$

Compressor work = 1.2 HP = 887 watts

WATER OF COMPRESSION

Saturation pressure of H_2O at 100°F is 0.9492 psia (ref. 12).

The volume per cent of water in the stream leaving the compressor is

$$\left(\frac{0.949}{100}\right)\frac{100}{} = 0.95\%.$$

The CO_2, O_2, and N_2 thus will account for 100.00-0.95 or 99.05%.

The percent composition of the CO_2, O_2, and N_2 thus are

$$\%\ CO_2 = \frac{4.4}{337.9} \times 99.05 = 1.29\%$$

$$\%\ O_2 = \frac{166.8}{337.9} \times 99.05 = 48.88$$

$$\%\ N_2 = \frac{166.8}{337.9} \times 99.05 = 48.88$$

$$\%\ H_2O = 0.95$$

The amount of water vapor in the stream leaving the No. 1 separator is

$$\left(\frac{0.95 \text{ gram-mole } H_2O}{1.29 \text{ gram-mole } CO_2}\right)\left(\frac{4.4 \text{ gram-mole } CO_2}{\text{hour}}\right) = 3.24 \text{ gram-mole } H_2O/\text{hour}$$

The amount of water removed in the first water separator is

$$(10.75 - 3.24)\ \frac{\text{gram-mole}}{\text{hour}} = \frac{7.51 \text{ gram-mole}}{\text{hour}}$$

COOLER REFRIGERATION

The heat exchanger, cooler, configuration is shown in Figure 10.

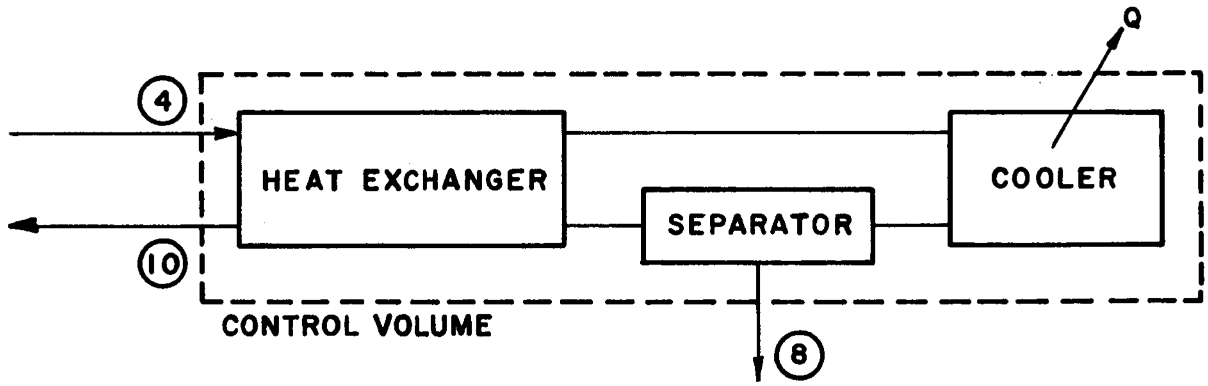


FIGURE 10. ENERGY FLOW DIAGRAM OF HEAT EXCHANGER-COOLER UNIT

Writing an energy balance across the control volume:

$$M_4h_4 = Q + M_8h_8 + M_{10}h_{10} \tag{1}$$

$$Q = M_4h_4 - M_8h_8 - M_{10}h_{10} \tag{2}$$

$$Q = M_{4_{O_2}} h_{4_{O_2}} + M_{4_{N_2}} h_{4_{N_2}} + M_{4_{CO_2}} h_{4_{CO_2}} + M_{4_{H_2O}} h_{4_{H_2O}} - M_{8_{H_2O}} h_{8_{H_2O}}$$

$$- M_{10_{O_2}} h_{10_{O_2}} - M_{10_{N_2}} h_{10_{N_2}} - M_{10_{CO_2}} h_{10_{CO_2}} - M_{10_{H_2O}} h_{10_{H_2O}} \tag{3}$$

Collecting terms according to components

$$Q = M_{4_{O_2}} (h_{4_{O_2}} - h_{10_{O_2}}) + M_{4_{N_2}} (h_{4_{N_2}} - h_{10_{N_2}}) + M_{4_{CO_2}} (h_{4_{CO_2}} - h_{10_{CO_2}})$$

$$+ M_{4_{H_2O}} h_{4_{H_2O}} - M_{8_{H_2O}} h_{8_{H_2O}} - M_{10_{H_2O}} h_{10_{H_2O}} \tag{4}$$

Enthalpies for O_2, N_2, and CO_2 are taken from reference 5.

$$M_{4_{O_2}} (h_{4_{O_2}} - h_{10_{O_2}}) = \left(\frac{166.8 \text{ gram-mole } O_2}{\text{hour}}\right) \left(\frac{1 \text{ lb}}{453 \text{ grams}}\right) (5890-5740) \frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{4_{O_2}} (h_{4_{O_2}} - h_{10_{O_2}}) = 55.2 \frac{\text{BTU}}{\text{hour}} \tag{5}$$

$$M_{4_{N_2}} (h_{4_{N_2}} - h_{10_{N_2}}) = \left(\frac{166.8 \text{ gram-mole } N_2}{\text{hour}}\right) \left(\frac{1 \text{ lb}}{453 \text{ grams}}\right) (5885-5750) \frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{4_{N_2}} (h_{4_{N_2}} - h_{10_{N_2}}) = 49.7 \frac{\text{BTU}}{\text{hour}} \tag{6}$$

$$M_{4_{CO_2}} (h_{4_{CO_2}} - h_{10_{CO_2}}) = \frac{4.4 \text{ gram-mole } CO_2}{\text{hour}} \; \frac{1 \text{ lb}}{453 \text{ grams}} \; (6220-6050) \frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{4_{CO_2}} (h_{4_{CO_2}} - h_{10_{CO_2}}) = 1.7 \frac{\text{BTU}}{\text{hour}} \tag{7}$$

$h_{4_{H_2O}} = 1105.2 \frac{\text{BTU}}{\text{lb}}$ taken from reference 11.

$$M_{4_{H_2O}} h_{4_{H_2O}} = \left(\frac{3.24 \text{ gram-mole } H_2O}{\text{hour}}\right) \left(\frac{1 \text{ lb}}{453 \text{ grams}}\right) \left(1105.2 \frac{\text{BTU}}{\text{lb}}\right) \left(\frac{18 \text{ grams } H_2O}{\text{gram-mole } H_2O}\right)$$

$$M_{4_{H_2O}} h_{4_{H_2O}} = 142.3 \frac{\text{BTU}}{\text{hour}} \tag{8}$$

CALCULATION OF M_8

The return stream in the heat exchanger is saturated with water at 40°F. The saturation pressure of water at 40°F is 0.1217 psia. Thus, the volume per cent of water in the return stream leaving the No. 1 heat exchanger is $\frac{0.1217 \times 100}{95} = 0.13\%$.

The O_2, N_2, CO_2 account for 100.00 - 0.13 = 99.87%.

Thus, the analysis of the stream leaving No. 1 heat exchanger and travelling toward the switching heat exchanger is

$$\% \; O_2 = \frac{166.8}{337.8} \times 99.87 = 49.31$$

$$\% \; N_2 = \frac{166.8}{337.8} \times 99.87 = 49.31$$

$$\% \; CO_2 = \frac{4.4}{337.8} \times 99.87 = 1.30$$

$$\% \; H_2O = 0.13$$

The water leaving No. 1 exchanger is $M_{10_{H_2O}}$

$$\left(\frac{0.13 \text{ gram-mole } H_2O}{1.13 \text{ gram-mole } CO_2}\right)\left(\frac{4.4 \text{ gram-mole } CO_2}{\text{hour}}\right) = \frac{0.44 \text{ gram-mole}}{\text{hour}} = M_{10_{H_2O}}$$

The water removed in No. 2 separator is $M_{8_{H_2O}}$

$$(3.24 - 0.44) \frac{\text{gram-mole}}{\text{hour}} = 2.80 \frac{\text{gram-mole}}{\text{hour}} = M_{8_{H_2O}}$$

$$M_{8_{H_2O}} h_{8_{H_2O}} = \left(\frac{2.8 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{18 \text{ grams}}{\text{gram-mole}}\right)\left(\frac{1 \text{ lb}}{453 \text{ grams}}\right)\left(\frac{8.05 \text{ BTU}}{\text{lb}}\right)$$

$$M_{8_{H_2O}} h_{8_{H_2O}} = 0.9 \frac{\text{BTU}}{\text{hour}} \qquad (9)$$

$$M_{10_{H_2O}} h_{10_{H_2O}} = \left(\frac{0.44 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{18 \text{ grams}}{\text{gram-mole}}\right)\left(\frac{1 \text{ lb}}{453 \text{ grams}}\right)\left(1095 \frac{\text{BTU}}{\text{hour}}\right)$$

$$M_{10_{H_2O}} h_{10_{H_2O}} = 19.1 \frac{\text{BTU}}{\text{hour}} \qquad (10)$$

Substituting Equations (5, 6, 7, 8, 9, and 10) into Equation (4)

$$Q = (55.2 + 49.7 + 1.7 + 142.3 - 0.9 - 19.1)\ \frac{BTU}{hour}$$

$$Q = 228.9\ \frac{BTU}{hour} \tag{11}$$

$$Q = \left(228.9\ \frac{BTU}{hour}\right)\left(\frac{hour}{60\ min}\right)\left(\frac{Watt}{0.05692\ BTU/min}\right) = 67.0\ Watts$$

SWITCHING HEAT EXCHANGER

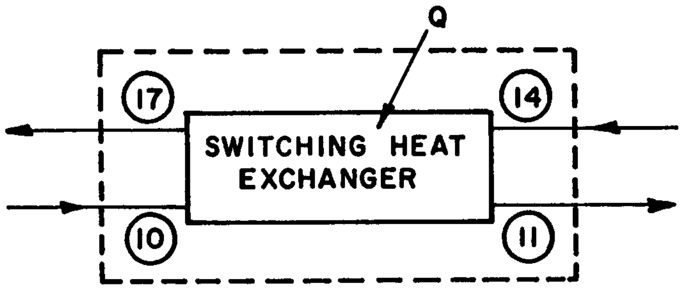


FIGURE 11. ENERGY FLOW DIAGRAM OF SWITCHING HEAT EXCHANGER

The pressure of CO_2 in the exchanger is $\left(\frac{1.3}{100}\right)$ x 100 = 1.3 psia. The saturation temperature of CO_2 at 1.3 psia is -155°F. In order to prevent any CO_2 freezeout in the exchanger, let Q be such that $T_{11} = -145^\circ$F.

Q, the balancing heat load, is comprised as follows:

Q_a = heat leak into switching heat exchanger = 100 $\frac{BTU}{hour}$

Q_b = cooler refrigeration

Q_c = available refrigeration = $Q - (Q_a + Q_b)$

Writing an energy balance for the exchanger:

$$M_{10}h_{10} + M_{14}h_{14} + Q = M_{11}h_{11} + M_{17}h_{17} \tag{12}$$

$$Q = M_{11}h_{11} + M_{17}h_{17} - M_{10}h_{10} - M_{14}h_{14} \tag{13}$$

$$Q = M_{11_{O_2}} h_{11_{O_2}} + M_{11_{N_2}} h_{11_{N_2}} + M_{11_{CO_2}} h_{11_{CO_2}} + M_{17_{O_2}} h_{17_{O_2}} + M_{17_{N_2}} h_{17_{N_2}}$$

$$+ M_{17_{H_2O}} h_{17_{H_2O}} - M_{10_{O_2}} h_{10_{O_2}} - M_{10_{N_2}} h_{10_{N_2}} - M_{10_{CO_2}} h_{10_{CO_2}} - M_{10_{H_2O}} h_{10_{H_2O}}$$

$$- M_{14_{O_2}} h_{14_{O_2}} - M_{14_{N_2}} h_{14_{N_2}}$$

Collecting the terms according to components

$$Q = M_{11_{O_2}} (h_{11_{O_2}} - h_{10_{O_2}}) + M_{11_{N_2}} (h_{11_{N_2}} - h_{10_{N_2}}) + M_{11_{CO_2}} (h_{11_{CO_2}} - h_{10_{CO_2}})$$

$$+ M_{17_{O_2}} (h_{17_{O_2}} - h_{14_{O_2}}) + M_{17_{N_2}} (h_{17_{N_2}} - h_{14_{N_2}}) + M_{17_{H_2O}} (h_{17_{H_2O}} - h_{10_{H_2O}}) \quad (14)$$

$$M_{11_{O_2}} (h_{11_{O_2}} - h_{10_{O_2}}) = \left(\frac{166.8 \text{ gram-moles}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)(4160-5740)\,\frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{11_{O_2}} (H_{11_{O_2}} - h_{10_{O_2}}) = -581.8\ \frac{\text{BTU}}{\text{hour}} \quad (15)$$

$$M_{11_{N_2}} (h_{11_{N_2}} - h_{10_{N_2}}) = \left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)(4155-5750)\,\frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{11_{N_2}} (h_{11_{N_2}} - h_{10_{N_2}}) = -587.3\ \frac{\text{BTU}}{\text{hour}} \quad (16)$$

$$M_{11_{CO_2}} (h_{11_{CO_2}} - h_{10_{CO_2}}) = \left(\frac{4.4 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)(4155-6050)\,\frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{11_{CO_2}} (h_{11_{CO_2}} - h_{10_{CO_2}}) = -18.4\ \frac{\text{BTU}}{\text{hour}} \quad (17)$$

$$M_{17_{O_2}} (h_{17_{O_2}} - h_{14_{O_2}}) = \left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)(5680-3470)\,\frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{17_{O_2}} (h_{17_{O_2}} - h_{14_{O_2}}) = 813.7\ \frac{\text{BTU}}{\text{hour}} \quad (18)$$

$$M_{17_{N_2}}\left(h_{17_{N_2}} - h_{14_{N_2}}\right) = \left(\frac{166.8\ \text{gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453\ \text{grams}}\right)(5680-3480)\ \frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{17_{N_2}}\left(h_{17_{N_2}} - h_{14_{N_2}}\right) = 810.0\ \frac{\text{BTU}}{\text{hour}} \qquad (19)$$

$$M_{17_{H_2O}}\left(h_{17_{H_2O}} - h_{10_{H_2O}}\right) = \left(\frac{0.44\ \text{gram-mole}}{\text{hour}}\right)(1093-1095)\ \frac{\text{BTU}}{\text{lb}}\left(\frac{18\ \text{lb}}{\text{lb-mole}}\right)\ \frac{\text{lb}}{453\ \text{grams}}$$

$$M_{17_{H_2O}}\left(h_{17_{H_2O}} - h_{10_{H_2O}}\right) = .03\ \frac{\text{BTU}}{\text{hour}} \quad \text{This term may be ignored.} \qquad (20)$$

Substituting Equations (15, 16, 17, 18, and 19) into Equation (14)

$$Q = (-581.8 - 587.3 - 18.4 + 813.7 + 810.0)\ \text{BTU/hour}$$

$$Q = 436.2\ \frac{\text{BTU}}{\text{hour}} \qquad (21)$$

$$Q = 436.2\ \frac{\text{BTU}}{\text{hour}}\ \frac{\text{hour}}{60\ \text{min}}\ \frac{\text{Watts}}{0.05692\ \text{BTU/min}} = 128\ \text{Watts}$$

<u>TURBINE WORK</u>

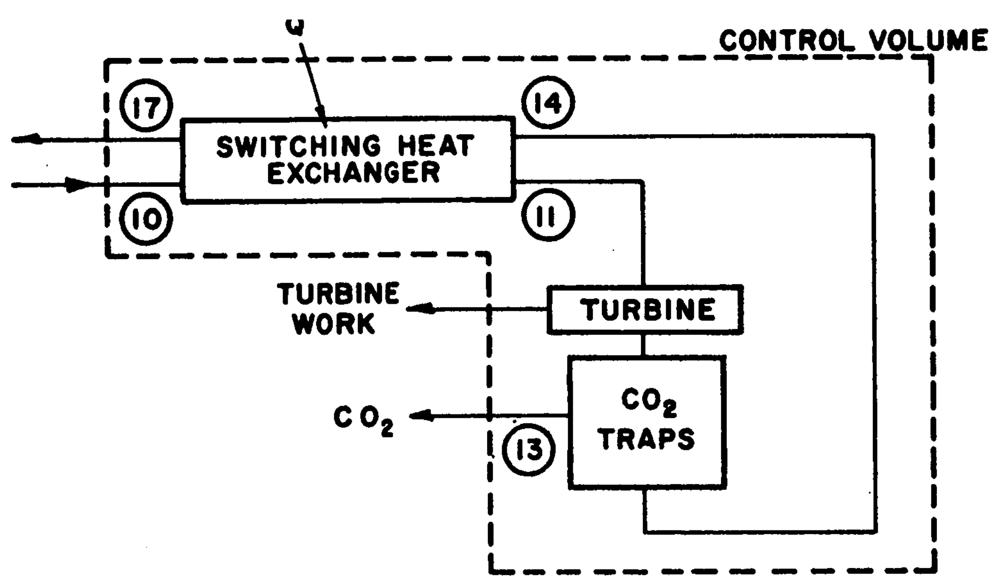


FIGURE 12. ENERGY FLOW DIAGRAM OF HEAT EXCHANGER-TURBINE-CO_2 TRAP UNIT

Writing an energy balance across the control volume

$$M_{10}h_{10} + Q = M_{17}h_{17} + W_{Turbine} + M_{13}h_{13}$$

$$W_{Turbine} = M_{10}h_{10} + Q - M_{17}h_{17} - M_{13}h_{13} \tag{22}$$

$$W_{Turbine} = M_{10_{O_2}} h_{10_{O_2}} + M_{10_{N_2}} h_{10_{N_2}} + M_{10_{CO_2}} h_{10_{CO_2}} + 436.2 \frac{\text{BTU}}{\text{hour}}$$

$$- M_{17_{O_2}} h_{17_{O_2}} - M_{17_{N_2}} h_{17_{N_2}} - M_{13}h_{13}$$

Collecting terms according to component

$$W_{Turbine} = M_{10_{O_2}} (h_{10_{O_2}} - h_{17_{O_2}}) + M_{10_{N_2}} (h_{10_{N_2}} - h_{17_{N_2}})$$

$$+ 436.2 \frac{\text{BTU}}{\text{hour}} + M_{10_{CO_2}} (h_{10_{CO_2}} - h_{13}) \tag{23}$$

$$M_{10_{O_2}} (h_{10_{O_2}} - h_{17_{O_2}}) = \left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right) \left(\frac{\text{lb}}{453 \text{ gram}}\right) (5740-5680) \frac{\text{BTU}}{\text{lb}}$$

$$M_{10_{O_2}} (h_{10_{O_2}} - h_{17_{O_2}}) = 22.1 \frac{\text{BTU}}{\text{hour}} \tag{24}$$

$$M_{10_{N_2}} (h_{10_{N_2}} - h_{17_{N_2}}) = \left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right) \left(\frac{\text{lb}}{453 \text{ grams}}\right) (5750-5680) \frac{\text{BTU}}{\text{lb}}$$

$$M_{10_{N_2}} (h_{10_{N_2}} - h_{17_{N_2}}) = 25.8 \frac{\text{BTU}}{\text{hour}} \tag{25}$$

Assuming the CO_2 is removed in the solid state at -250°F at point 13

$$h_{13_{CO_2}} = h_x + C_p (T_{-250} - T_x)$$

where point x is the saturated solid state point at 7 psia.

$$C_p \text{ for solid } CO_2 = 11 \frac{\text{BTU}}{\text{lb-mole } ^\circ\text{F}} \; [\,-250 - (-125)]\; ^\circ\text{F}$$

$$h_{13_{CO_2}} = (\,-6{,}600 - 1{,}375)\, \frac{\text{BTU}}{\text{lb-mole}}$$

$$h_{13_{CO_2}} = -\,7975 \,\frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{10_{CO_2}} \left(h_{10_{CO_2}} - h_{13_{CO_2}}\right) = \left(\frac{4.4 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right) (6050 + 7975)\, \frac{\text{BTU}}{\text{lb-mole}}$$

$$M_{10_{CO_2}} \left(h_{10_{CO_2}} - h_{13_{CO_2}}\right) = 136 \,\frac{\text{BTU}}{\text{hour}} \qquad (26)$$

Substituting Equations (24, 25, and 26) into Equation (23)

$$W_{\text{Turbine}} = 620.1 \,\frac{\text{BTU}}{\text{hour}} \qquad (27)$$

$$W_{\text{Turbine}} = \left(\frac{620.1 \text{ BTU}}{\text{hour}}\right)\left(\frac{\text{hour}}{60 \text{ min}}\right)\left(\frac{\text{Watts}}{0.05692 \text{ BTU/min}}\right) = 182 \text{ Watts}$$

TURBINE EFFICIENCY

$$\% \text{ Efficiency} = \frac{\text{Actual Turbine Work}}{\text{Isentropic Turbine Work}} \times 100 \qquad (28)$$

$$\% \text{ EFF} = \frac{(M_{11} - h_{14})\text{actual}}{(M_{11} - h_{14})\text{isentropic}} \times 100 \qquad (29)$$

$$M_{14}h_{14} \text{ isentropic} = M_{11_{O_2}} h_{14i_{O_2}} + M_{14_{N_2}} h_{14i_{N_2}} + M_{14_{CO_2}} h_{14i_{CO_2}} \qquad (30)$$

$$M_{14}h_{14} \text{ isentropic} = \left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)\left(3090 \,\frac{\text{BTU}}{\text{lb-mole}}\right)$$

$$+\left(\frac{166.8 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ grams}}\right)\left(\frac{3090 \text{ BTU}}{\text{lb-mole}}\right) + \left(\frac{4.4 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{\text{lb}}{453 \text{ gram}}\right)$$

$$\left(-\,\frac{7975 \text{ BTU}}{\text{lb-mole}}\right)$$

$$M_{14}h_{14} \text{ isentropic} = 1137.8 \frac{BTU}{hour} + 1137.8 \frac{BTU}{hour} - 77.5 \frac{BTU}{hour}$$

$$M_{14}h_{14} \text{ isentropic} = 2198.1 \frac{BTU}{hour} \tag{31}$$

$$M_{11}h_{11} = (1531.7 + 1529.9 + 40.4) \frac{BTU}{hour}$$

$$M_{11}h_{11} = 3102.0 \frac{BTU}{hour} \tag{32}$$

$$M_{14}h_{14} = 1277.8 \frac{BTU}{hour} + 1281.4 \frac{BTU}{hour} - 77.5 \frac{BTU}{hour}$$

$$M_{14}h_{14} = 2481.8 \frac{BTU}{hour} \tag{33}$$

Substituting 31, 32, and 33 into Equation 29

$$\% \text{ Efficiency} = \frac{3102.0 - 2481.8}{3102.0 - 2198.1} (100) = \frac{620.2}{903.9} (100)$$

$$\% \text{ Efficiency} = 69\% \tag{34}$$

SWITCHING HEAT EXCHANGER SIZE

$$Q_{Exch} = h_o A \Delta T \tag{35}$$

$$Q_{Exch} = M_{10}h_{10} - M_{11}h_{11} + Q_{Heat\ Leak} \tag{36}$$

$$Q_{Exch} = (581.8 + 587.3 + 18.4 + 436.2) \text{ BTU/hour}$$

$$Q_{Exch} = 1623.7 \text{ BTU/hour} \tag{37}$$

Assume ΔT is a log mean temperature difference

$$\Delta T = \frac{(-145 - [-250]) - (80-70)}{\ln \frac{(-145 - [-250])}{(80-70)}} = \frac{105-10}{\ln \frac{105}{10}} \tag{38}$$

$$\Delta T = 41°F \tag{39}$$

$$h_o A = \frac{Q}{\Delta T} = \frac{1623.7}{41} = 40 \text{ BTU/hour-°F} \tag{40}$$

For a Trane core heat exchanger type 50-375 from reference (17)

Flow Area = 0.037 ft^2/passage

The mass velocity G is equal to $\frac{\text{Mass flow rate}}{\text{Flow Area}}$

$$G = \left(\frac{337.9 \text{ gram-mole}}{\text{hour}}\right)\left(\frac{1}{0.037 \text{ ft}^2 \text{ passage}}\right) \qquad (41)$$

All of the Trane data are given in terms of $\frac{\text{lb}}{\text{hour ft}^2}$, therefore, the molecular weight of the flow stream must be found.

The volumetric analysis of the high pressure flow stream in the switching heat exchanger was shown to be

49.31% O_2
49.31% N_2
1.30% CO_2
0.13% H_2O

This can be converted to mass per cent by multiplying each component by its molecular weight

$$\left(\frac{0.493 \text{ gram-mole } O_2}{\text{gram-mole total}}\right)\left(\frac{32 \text{ grams } O_2}{\text{gram-mole } O_2}\right) = 15.78$$

$$\left(\frac{0.493 \text{ gram-mole } N_2}{\text{gram-mole total}}\right)\left(\frac{28 \text{ grams } N_2}{\text{gram-mole } N_2}\right) = 13.80$$

$$\left(\frac{0.013 \text{ gram-mole } CO_2}{\text{gram-mole total}}\right)\left(\frac{44 \text{ grams } CO_2}{\text{gram-mole } CO_2}\right) = 0.57$$

$$\left(\frac{0.0013 \text{ gram-mole } H_2O}{\text{gram-mole total}}\right)\left(\frac{18 \text{ grams } H_2O}{\text{gram-mole } H_2O}\right) = \underline{0.02}$$

$$30.17 \ \frac{\text{grams total}}{\text{gram-mole total}} \qquad (42)$$

$$G = \left(\frac{337.9 \text{ gram-mole dry gas}}{\text{hour}}\right)\left(\frac{\text{gram-mole total gas}}{.9987 \text{ gram-mole dry gas}}\right)\left(\frac{30.17 \text{ grams}}{\text{gram-mole}}\right)\frac{1}{0.037 \text{ ft}^2 \text{ passage}}$$

Assuming the use of one passage per stream, then

$$G = 2.75 \times 10^5 \ \frac{\text{grams}}{\text{hour ft}^2 \text{ passage}}$$

$$G = (2.75 \times 10^5)\left(\frac{1}{453}\right)\left(\frac{\text{lb}}{\text{hour ft}^2}\right) = 609 \ \frac{\text{lb}}{\text{hour ft}^2} \qquad (43)$$

$\mu = 0.039$ lb/ft-hr taken at the bulk temp. $\frac{(80-145)}{2} = -33°F$

$$\frac{G}{\mu} = \frac{609}{0.039} = 15{,}600 \text{ lbm/hour ft}^2$$

From curve A-A,

$$j = 0.025 \qquad (44)$$

$$j = \frac{h_o (Pr)^{2/3}}{G \ C_p} \qquad (45)$$

$$h_o = \frac{jGC_p}{(Pr)^{2/3}}$$

$$C_p = 7.1 \text{ BTU/lb-mole-°F}$$

$$Pr = \frac{C_p \mu}{k} \qquad (46)$$

$$\mu = 0.039 \text{ lb/ft-hour}$$

$$k = 0.0124 \text{ BTU/hour-ft-°F}$$

$$Pr = \left(\frac{7.1 \text{ BTU}}{\text{lb-Mole °F}}\right)\left(\frac{0.039 \text{ lbM}}{\text{ft-hour}}\right)\left(\frac{\text{Hour ft °F}}{0.0124 \text{ BTU}}\right)\left(\frac{\text{lb-Mole}}{30.17 \text{ lb}}\right) \qquad (47)$$

$$Pr = 0.765$$

$$Pr^{2/3} = 0.836 \qquad (48)$$

$$h_o = \left(\frac{1}{.836}\right) \frac{(0.025)}{} \left(\frac{609\ \text{lbM}}{\text{hour ft}^2}\right) \left(\frac{7.1\ \text{BTU}}{\text{lb-mole}\ ^\circ\text{F}}\right) \left(\frac{\text{lb-mole}}{30.17\ \text{lbs}}\right) \tag{49}$$

$$h_o = 4.29\ \text{BTU/hour-ft}^2\text{-}^\circ\text{F}$$

Substituting Equation (49) into Equation (40)

$$h_oA = 40\ \text{BTU/hour}\ ^\circ\text{F}$$

$$A = \frac{40\ \text{BTU}}{\text{Hour}\ ^\circ\text{F}} \frac{\text{Hour ft}^2\ ^\circ\text{F}}{4.29\ \text{BTU}}$$

$$A = 9.1\ \text{ft}^2\ \text{heat transfer area}$$

Since there is 17.55 ft^2/ft passage the length of the heat exchanger should be 0.52 ft long.

Make exchanger 30" long to assure H_2O removal. The exchanger will be comprised of two passages. The high pressure and low pressure streams will switch at regular intervals in order to remove the water from the exchanger.

For the low pressure side of the exchanger:

$$\text{Bulk Temperature} = \frac{-250 + 70}{2} = -90^\circ\text{F}$$

$$\mu = 0.034\ \text{lb/ft-hour}$$

$$\frac{G}{\mu} = \frac{609}{0.034} = 17{,}900$$

j = 0.02 from curve A-A (reference 17)

$$C_p = 7.0\ \text{BTU/lb-mole}\ ^\circ\text{F}$$

$$k = 0.0105\ \text{BTU/hour ft}\ ^\circ\text{F}$$

$$Pr = \frac{C_p \mu}{k} = \left(\frac{7.0 \text{ BTU}}{\text{lb-mole } ^\circ F}\right)\left(\frac{0.034 \text{ lbm}}{\text{ft-hour}}\right)\left(\frac{\text{Hour ft } ^\circ F}{0.0105 \text{ BTU}}\right)\left(\frac{\text{lb-mole}}{30.17 \text{ lbm}}\right)$$

$$Pr = 0.756$$

$$Pr^{2/3} = 0.83$$

$$h_o = \frac{jGC_p}{(Pr)^{2/3}} = (0.020)\left(\frac{609 \text{ lbm}}{\text{hour ft}^2}\right)\left(\frac{7.0 \text{ BTU}}{\text{lb-mole } ^\circ F}\right)\left(\frac{1}{0.83}\right)\left(\frac{\text{lb-mole}}{30.17 \text{ lbm}}\right)$$

$$h_o = 3.43 \text{ BTU/hour ft}^2 \ ^\circ F$$

$$h_o A = 40 \text{ BTU/hour } ^\circ F$$

$$A = \left(\frac{40 \text{ BTU}}{\text{hour } ^\circ F}\right)\left(\frac{\text{Hour ft}^2 \ ^\circ F}{3.43 \text{ BTU}}\right) = 11.8 \text{ ft}^2$$

11.8 ft^2 Heat Transfer Area

$$11.8 \text{ ft}^2 \left(\frac{\text{ft of passage}}{17.55 \text{ ft}^2}\right) = 0.673 \text{ ft}$$

30" long exchanger will be sufficient.

PRESSURE DROP ACROSS EXCHANGER

The following pressure drop correlation is given in reference 17.

$$\Delta P = \frac{G^2}{2g}\left(\frac{8fL}{\rho_1 + \rho_2} - \frac{0.864}{\rho_1} + \frac{1.507}{\rho_2}\right)$$

where:

ΔP = pressure drop, lb_f/ft^2

G = flow stream mass velocity lbm/ft^2 hour

g = gravity constant 4.17×10^8 $ft/hour^2$

f = fanning friction factor

L = exchanger flow length ft

ρ_1 = entrance density lbm/ft^3

ρ_2 = exit density lbm/ft^3

For the low pressure stream:

$f = 0.25$ (reference 17)

$\rho_1 = 0.11 \text{ lbm/ft}^3$

$\rho_2 = 0.045 \text{ lbm/ft}^3$

$$\Delta P = \frac{(609)^2}{2(4.17 \times 10^8)} \left(\frac{8(0.25)(2.5)}{0.155} - \frac{0.864}{0.11} + \frac{1.507}{0.045} \right)$$

$$\Delta P = 4.4 \times 10^{-4} \, [14.2 - 7.85 + 33.5]$$

$$\Delta P = 4.4 \times 10^{-4} \, (3267)$$

$$\Delta P = 1.4 \frac{\text{lb}}{\text{ft}^2}$$

$$\Delta P = 0.097 \frac{\text{lb}_f}{\text{in}^2} = 2.7 \text{ in } H_2O$$

SYSTEM HEAT LEAK

Since the heat exchanger will be 30 inches long, the height of the cold box is estimated to be 36 inches. In order to contain the various components, the cold box is estimated to be 18 inches wide and 18 inches deep. The conductive heat leak through the walls of the cold box will be

$$Q_1 = \frac{kA\Delta T}{t} \qquad \text{where } t = \text{insulation thickness.} \qquad (50)$$

The apparent conductivity k of multiple layer insulation 2" thick at a pressure of 10^{-4} mm Hg is $3 \times 10^{-5} \frac{\text{BTU}}{\text{hour ft}^2 \frac{°F}{\text{ft}}}$ (51)

For an average internal cold box temperature of $\frac{80 + (-250)}{2} = -85°F$,

the temperature ΔT will be $70°F - (-85) = 155°F$ (52)

The heat transfer area will be:

$$4 \, (3 \times 1.5) \text{ ft}^2 + 2 \, (1.5 \times 1.5) \text{ ft}^2 = 18 + 4.5 = 22.5 \text{ ft}^2 \qquad (53)$$

$$t = \frac{2}{12}\ ft = .167\ ft \tag{54}$$

Substituting 51, 52, 53, and 54 into Equation (50)

$$Q_1 = \frac{3 \times 10^{-5}\ BTU}{hour\ ft^2\ \frac{°F}{ft}}\ (22.5\ ft^2)\ (85°F)\ \frac{1}{.167\ ft}$$

$$Q_1 = 0.3\ \frac{BTU}{hour}$$

For a typical support member such as a 3" stainless steel channel

$$Q_2 = \frac{kA\Delta T}{t}$$

$$A = \frac{1.5\ in^2}{144} = 0.01\ ft^2$$

$$k = 8\ \frac{BTU}{hour\ ft^2\ \frac{°F}{ft}}$$

If support members are attached to the warm pieces of equipment such as the No. 1 heat exchanger, cooler, and water separator a ΔT of $(70-40)°F = 30°F$ will be obtained

$$t = \frac{2}{12}\ ft = 0.167\ ft.$$

$$Q_2 = \left(\frac{8\ BTU}{hour\ ft^2\ \frac{°F}{ft}}\right)\ (0.01\ ft^2)\ (30°F)\ \frac{1}{0.167\ ft}$$

$$Q_2 = \frac{14.4\ BTU}{hour}\ \text{per structural member}$$

Assuming a 95% efficient power conversion in the turbine loading device, a heat load of (0.05 x turbine work) will be placed on the system.

$$Q_3 = (0.05)\ (620.1\ \frac{BTU}{hour}) = 31\ \frac{BTU}{hour}$$

The total heat leak = $Q = Q_1 + Q_2 + Q_3$

$$Q = 0.3 + (4 \text{ structural members}) \left(14.4 \frac{\text{BTU}}{\text{hour structural member}}\right) + 31 \frac{\text{BTU}}{\text{hour}}$$

$$Q = 89 \frac{\text{BTU}}{\text{hour}}$$

Let $Q_{\text{heat leak}} = 100 \frac{\text{BTU}}{\text{hour}}$

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   a. Oxygen

(1)	Pressure-Enthalpy	101.144 (Rev. 2)
(2)	Density	104.144
(3)	Viscosity	103.144 (Rev. 1)
(4)	Specific Heat	105.144
(5)	Thermal Conductivity	106.144 (Rev. 1)

   b. Nitrogen

(1)	Pressure-Enthalpy	101.141 (Rev. 2)
(2)	Density	104.141 (Rev. 1)
(3)	Viscosity	103.141 (Rev. 1)
(4)	Thermal Conductivity	106.141 (Rev. 1)

   c. Carbon Dioxide

(1)	Pressure-Enthalpy	101.207
(2)	Vapor Pressure	107.207 (Rev. 1)
(3)	Specific Heat	105.207

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