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USE OF REFRIGERANTS AS POSSIBLE WORKING FLUID IN A HANFORD POWER RECOVERY SYSTEM

by

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June 6, 1956

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USE OF REFRIGERANTS AS POSSIBLE WORKING FLUID IN A HANFORD POWER RECOVERY SYSTEM

INTRODUCTION

Interest in power recovery from the vast quantities of low temperature waste heat discharged from the Hanford reactors into the Columbia River has been stimulated by the recent proposed effluent temperature increases of the pressurization and recirculation studies for increased plutonium production[1,2,3]. Although power recovery from effluent temperatures appears feasible from an engineering standpoint[4], only low pressure steam can be generated from the waste heat. The operation of a steam cycle at atmospheric or sub-atmospheric pressures presents many problems: (1) vacuum, a considerable portion of the equipment will be operating at sub-atmospheric pressure requiring elaborate seals and vacuum pumps to maintain the vacuum in the system, (2) turbine efficiency, the low steam pressures would allow only two stages in the turbine reducing the plant efficiency over and above the low thermal efficiency of the cycle, and (3) blade erosion, to generate sufficient power to be feasible, the steam must be expanded through the turbine until the moisture content is about 10 to 12 per cent. Although this moisture content is not excessive, special blades would have to be incorporated in the turbine design to handle the wet steam.

A study was instigated to investigate the refrigerant fluids, mainly, Freons, for use as the working fluid of the power cycle. The Freon gases have lower boiling points than water and consequently would generate higher pressures allowing a more efficient turbine design with more stages. By using a Freon fluid a closed loop would be used, eliminating radiation contamination of the power generating equipment.

SUMMARY

A preliminary investigation was made of the use of refrigerants as possible working fluids in a power recovery cycle. Of the many refrigerants, one class (Freons) was considered; the other refrigerants were rejected from either their flammability and/or toxicity properties. Two Freon fluids (Freon 11 and 113) with atmospheric boiling points 10°F or more above average river water temperature were further investigated for probable use. Turbine pressures up to 60 psia can be generated for a possible power recovery of approximately 140,000 KW. Mass flow of a Freon cycle are increased seventeen-fold over a steam system of equal capacity. Cooling water requirements of the Freon power cycles are three to four times greater than for a

(1) Johnson, R. E., "Budget Study for Production Increases through Reactor Rear Face Pressurization", HW-40985, January 19, 1956.
(2) Heacock, H. W., "Adaption of a Recirculation Water System to 100-H Area", HW-40837, January 9, 1956.
(3) Johnson, R. E., "Rear Face Pressurization Applied to 100-H Area", HW-41298, February 6, 1956.
(4) Karnie, A. J., et. al., "Utilization of Waste Heat from Hanford Reactors for Production of Marketable Power", HW-38968, September 23, 1955.

[redacted]

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low pressure steam system of comparable capacity. A preliminary estimate of the cost of the equipment and fluid for a Freon closed power system as compared to a steam flash tank power system of equivalent capacity will be approximately double. Further study of the possible use of refrigerants as a working fluid of low temperature power cycles does not appear to be justified at this time.

DISCUSSION

Refrigerants

Refrigerants investigated to be used as the working fluid in the power cycle were the Freons. These fluids are halogenated hydrocarbons of the methane and ethane series with chlorine and/or fluorine atoms substituted for the hydrogen atoms. The fluids are chemically stable, non-corrosive, and moderately safe, i.e., non-flammable and low toxicity. Ammonia, sulfur dioxide and the straight hydrocarbons were not considered because of their high toxicity and/or high flammability.

The physical and thermodynamic properties of the Freon fluids can be adapted to power cycles with low temperature heat sources. Their low boiling temperatures are particularly suited to power recovery from low temperature heat sources as working pressures above atmospheric pressures can be generated. Their low specific heat and latent heat of vaporization are somewhat undesirable because larger mass flows are required to produce an equivalent amount of power as with steam at identical temperatures. Their high density of the liquid phase is desirable in sizing equipment and piping.

In this study, two Freon fluids, Freon 11 and 113, were considered as possible working fluids in the power cycle. The two were selected because their atmospheric boiling points are above the average temperature of water available for condenser cooling water. The remaining Freons fluids were rejected as their atmospheric boiling points are well below the temperature of available cooling water. To use these Freons in the power cycle, the gas after expansion through the turbine would necessitate compression to the liquid phase before the heat unavailable for work could be removed by the cooling water. The power required for compression work would be a considerable portion of the gross power generated.

Table I lists some physical properties and cost of Freon 11 and 113 as compared to water.

Power Cycles

Pile conditions selected to develop the power cycles are based on Case 4 of the pressurized water systems of the Pressurization and Recirculation Studies[5]. The flow and temperature of the effluent water are 89,100 gpm and 123°C (253°F).

Figure 1 shows a flow diagram of the proposed closed power loop using Freon as the working fluid. The fluid is evaporated at a designated working pressure, expanded through the turbine and condensed at 72°F. The condensate is pumped through the

(5) Etheridge, E. L., "Pressurization and Recirculation Study - Summary of Design and Development Data", HW-42529, April 16, 1956.

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TABLE I

PHYSICAL PROPERTIES OF FREON 11 AND 113, AND WATER

Property	Freon-11	Freon-113	Water
Chemical Formula	$C\,Cl_3\,F$	$C_2Cl_3F_3$	H_2O
Molecular Weight	137.38	187.39	18.02
Boiling Point (1 ATM)	74.7°F	117.6°F	212.0°F
Freezing Point (1 ATM)	-168.0°F	-31.0°F	32.0°F
Specific Heat (100°F)	.21 Btu/lb	.22 Btu/lb	1.0 Btu/lb
Latent Heat of Vaporization (1 ATM)	78.31 Btu/lb	63.12 Btu/lb	970.3 Btu/lb
Density (liquid) 68°F	92.8 lbs/ft^3	98.4 lbs/ft^3	62.4 lbs/ft^3
Viscosity (liquid) 70°F	.439 centipoises	.697 centipoises	.978 centipoises
Color	Clear and water white	Clear and water white	Clear and water white
Odor	Ethereal and similar to Carbon Tetrachloride	Ethereal and similar to Carbon Tetrachloride	None
Toxicity	5(1)	6(1)	None
Flammability	Non-flammable & non-combustable	Non-flammable & non-combustable	Non-flammable & non-combustable
Cost(2)	$0.30/lb	$0.80/lb	Nil

(1) National Fire Underwriters Group, Number 1 - most toxic.

(2) Costs are 75 per cent of the price of fluids sold in 200 pound cylinders.

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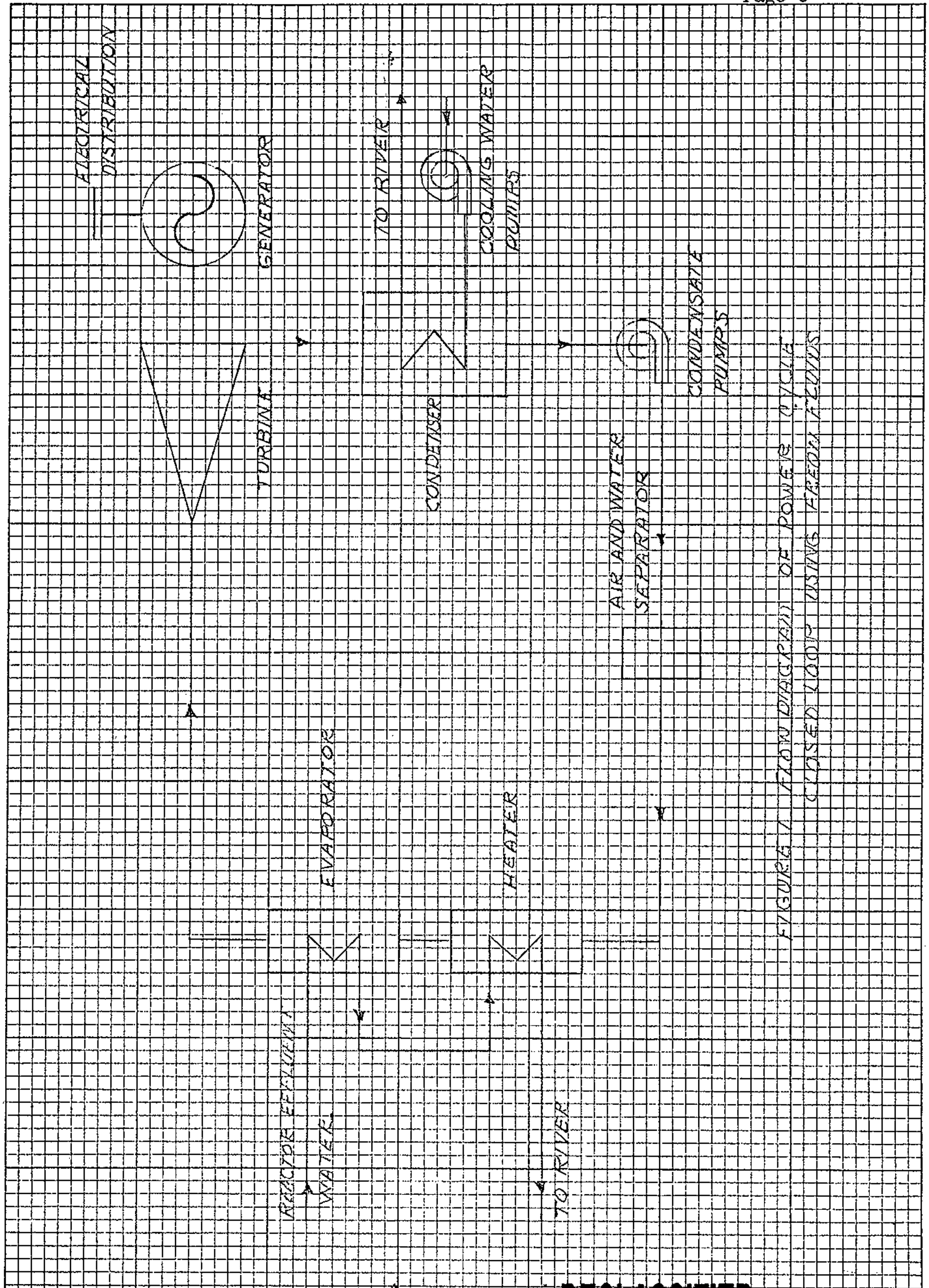

FIGURE 1 FLOW DIAGRAM OF POWER CYCLE
CLOSED LOOP USING FREON FLUIDS

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heater section where the temperature of the liquid is raised to within 4°F of the saturated temperature and then forced into the evaporator to complete the cycle. The pile effluent water is routed through the evaporator, into the heater section, and then discharged to the existing 107 basins or to the river directly. A moisture and air separator is included in the loop to maintain the air and moisture content at low level to prevent corrosion.

Net power produced using the Freons as the working fluids in a power cycle with the available heat source is shown in Figure 2. Rankine power cycles over a range of saturation temperatures and pressures were calculated to determine the maximum power that could be produced. Net power was assumed as 80 per cent of the Rankine cycle, allowing 20 per cent for turbine and exhaust losses. Also included on the chart are steam cycles operating at almost identical conditions. Steam is flashed directly from the pile effluent water, expanded through the turbine, condensed, and discharged directly to the river. Pressures generated at temperatures indicated by the dashed line section of the steam curve are too low to operate a turbine. Table II lists the pressures, mass flow rates, cooling water requirements, etc., at designated temperatures of the working fluids for operation of a power cycle at the specified output.

Cycle Analysis

The available work which may be obtained from a heat source is limited by the absolute temperatures of the heat source and sink regardless of the working fluid. A glance at the net power output of the power cycles shows approximately equivalent amounts of energy can be recovered by any of the three different working fluids. However, the conditions under which power is recovered is vastly different. Pressures up to 60 psia can be generated by the Freons whereas steam produced would be under atmospheric pressure. The specific volume of the Freon vapors is at least 50 times less than steam, yet due to the low specific heat and latent heat of vaporization approximately 17 times more mass flow is required for the Freon fluids than steam. These conditions effect the physical size of the turbine and piping. It is probable that smaller turbines can be designed for a Freon system than is possible for a steam system at low pressures.

Superheating cannot increase the cycle efficiency of the Freon fluids and thus reduce the required large mass flow. To illustrate this point, power cycles are plotted on temperature-entropy charts of the respective fluids on Figures 3, 4 and 5. Line 1-2 represents isentropic expansion of the vapor through the turbine, line 2-3, condensation of the vapor by the abstraction of latent heat of vaporization with cooling water, line 3-4, heating the liquid to the boiling point, and line 4-1, evaporation of the liquid to the vapor. As can be seen with the saturated vapor cycles, the heat converted to work includes the specific heat of the fluid from t_1 to t_2 and with Freon-11 and steam about 2 and 11 per cent of the latent heat of vaporization. Freon-113, after expansion, is superheated approximately 20°F above the temperature of condensation. By superheating the Freon fluids 30°F at the beginning of the cycle (line 5-6 on Figures 3 and 4) it can be seen that the fluids will be superheated about 20°F above the temperature of condensation after expansion with Freon-11 and about 46°F with Freon-113. In the case of Freon-113, as shown on Figure 4, about 5 Btu/lb is added to the enthalpy of the fluid by superheating 30°F whereas about an additional 7 Btu/lb of fluid must be abstracted during the

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TABLE II (A)

FREON 11

tura- on ess. ia	Specific Volume Vapor $ft^3/\#$	Mass Flow #/hr	Condenser Temp. °F	Condenser Press. psia	Gross, KW	Net, KW	Work Rate # Fluid/ KW-hr	Turbine Cycle Heat Rate Btu/KW-hr	Cooling Water Required, gpm	Thermal Efficiency %
5.3	1.652	8.12×10^7	72	13.9	101,500	81,500	1035	108,500	2.0×10^6	3.1
0.0	1.407	7.68×10^7	72	13.9	128,000	102,500	750	79,500	1.87×10^6	4.3
5.4	1.206	7.10×10^7	72	13.9	146,500	117,000	605	65,000	1.73×10^6	5.2
0.2	1.068	6.7×10^7	72	13.9	157,500	125,500	530	57,000	1.62×10^6	6.0
5.5	0.951	6.3×10^7	72	13.9	165,000	132,000	475	51,500	1.50×10^6	6.6
9.8	0.872	5.97×10^7	72	13.9	169,000	135,000	417	45,600	1.45×10^6	7.5
6.0	0.779	5.55×10^7	72	13.9	173,000	138,500	402	44,500	1.35×10^6	7.7
1.0	0.718	5.22×10^7	72	13.9	172,500	138,000	378	42,000	1.26×10^6	8.1
5.0	0.676	4.49×10^7	72	13.9	170,000	136,000	363	40,000	1.20×10^6	8.5

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TABLE II (B)

FREON 113

tura- on ess. ia	Specific Volume Vapor $ft^3/\#$	Mass Flow #/hr	Condenser Temp. °F	Condenser Press. psia	Gross, KW	Net, KW	Work Rate # Fluid/ KW-hr	Turbine Cycle Heat Rate Btu/KW-hr	Cooling Water Required, gpm	Thermal Efficiency %
6.0	2.008	8.46×10^7	72	5.8	152,500	122,000	692	67,000	1.84×10^6	5.1
9.8	1.642	7.74×10^7	72	5.8	164,000	131,000	591	58,000	1.63×10^6	5.9
5.1	1.313	6.78×10^7	72	5.8	173,500	139,000	489	49,300	1.5×10^6	6.9
9.5	1.128	6.24×10^7	72	5.8	178,500	143,000	436	44,500	1.37×10^6	7.7
5.5	0.944	5.45×10^7	72	5.8	176,500	141,000	388	40,200	1.2×10^6	8.5
0.0	0.844	4.91×10^7	72	5.8	172,000	137,500	358	37,500	1.09×10^6	9.1
7.6	0.713	4.10×10^7	72	5.8	155,500	124,500	329	35,000	9.12×10^5	9.7
0.0	0.681	3.76×10^7	72	5.8	147,000	117,500	320	34,300	8.5×10^5	9.9

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TABLE II (C)

STEAM

atura- ion ress. sia	Specific Volume Vapor ft3/#	Mass Flow #/hr	Condenser Temp. °F	Condenser Press. 1" Hg	Gross, KW	Net, KW	Work Rate # Fluid/ KW-hr	Turbine Cycle Heat Rate Btu/KW-hr	Cooling Water Required, gpm	Thermal Efficiency %
.9	123.01	5.0×10^6	80	1	152,000	121,500	41.2	46,200	8.67×10^5	7.4
.6	97.07	4.61×10^6	80	1	164,500	131,500	35.0	39,400	7.87×10^5	8.7
.74	77.29	4.18×10^6	80	1	165,500	132,000	31.6	35,700	7.09×10^5	9.5
.0	62.06	3.88×10^6	80	1	167,000	134,000	29.0	32,900	6.52×10^5	10.3
.5	50.23	3.32×10^6	80	1	160,000	128,000	26.0	29,600	5.50×10^5	11.5
.34	40.96	2.88×10^6	80	1	150,000	120,000	24.0	27,400	4.72×10^5	12.4
1.53	33.64	2.44×10^6	80	1	135,000	108,500	22.6	25,700	3.98×10^5	13.2

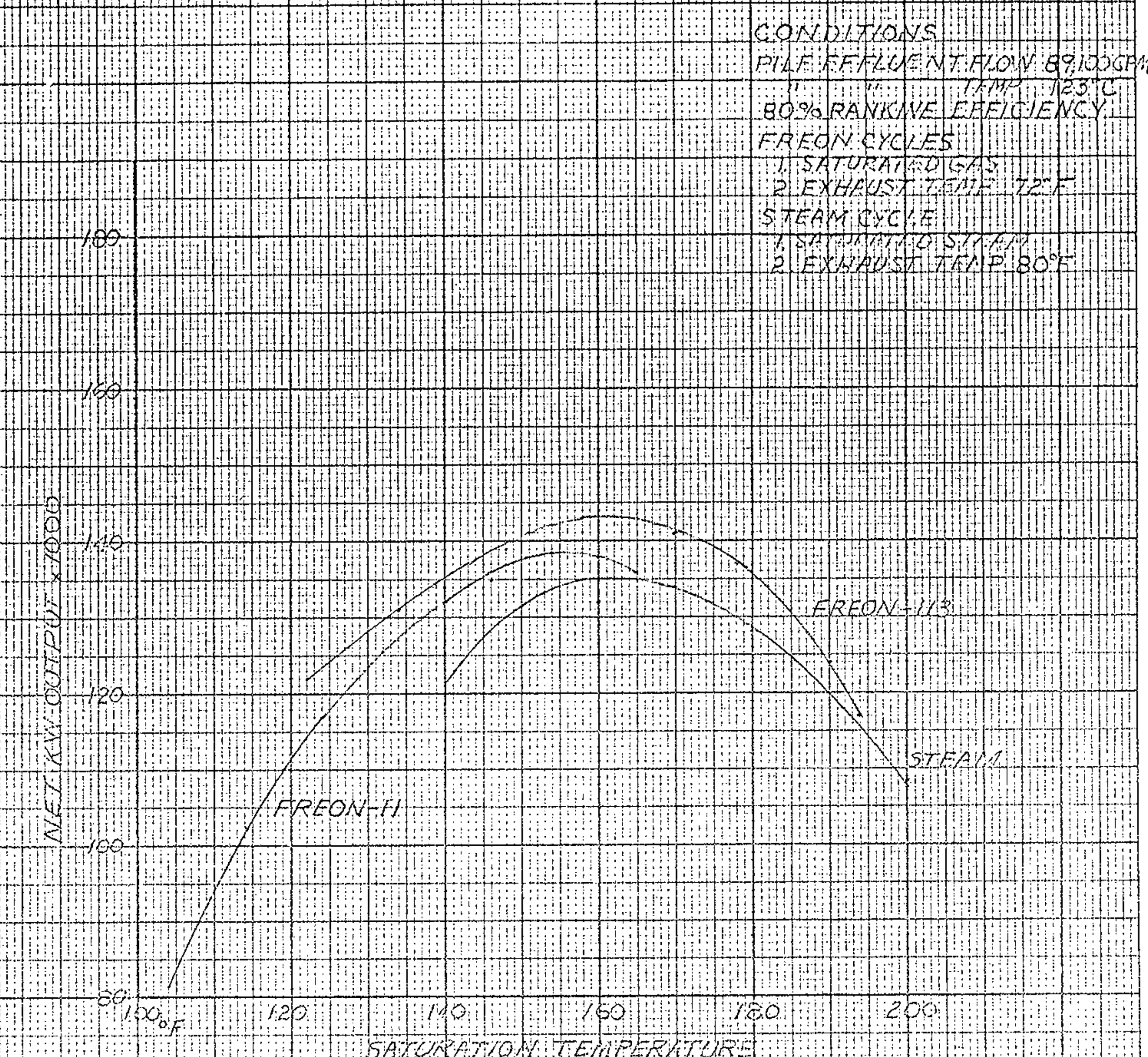

FIGURE 2 NET POWER PRODUCED USING FREON-11 AND 113, AND STEAM

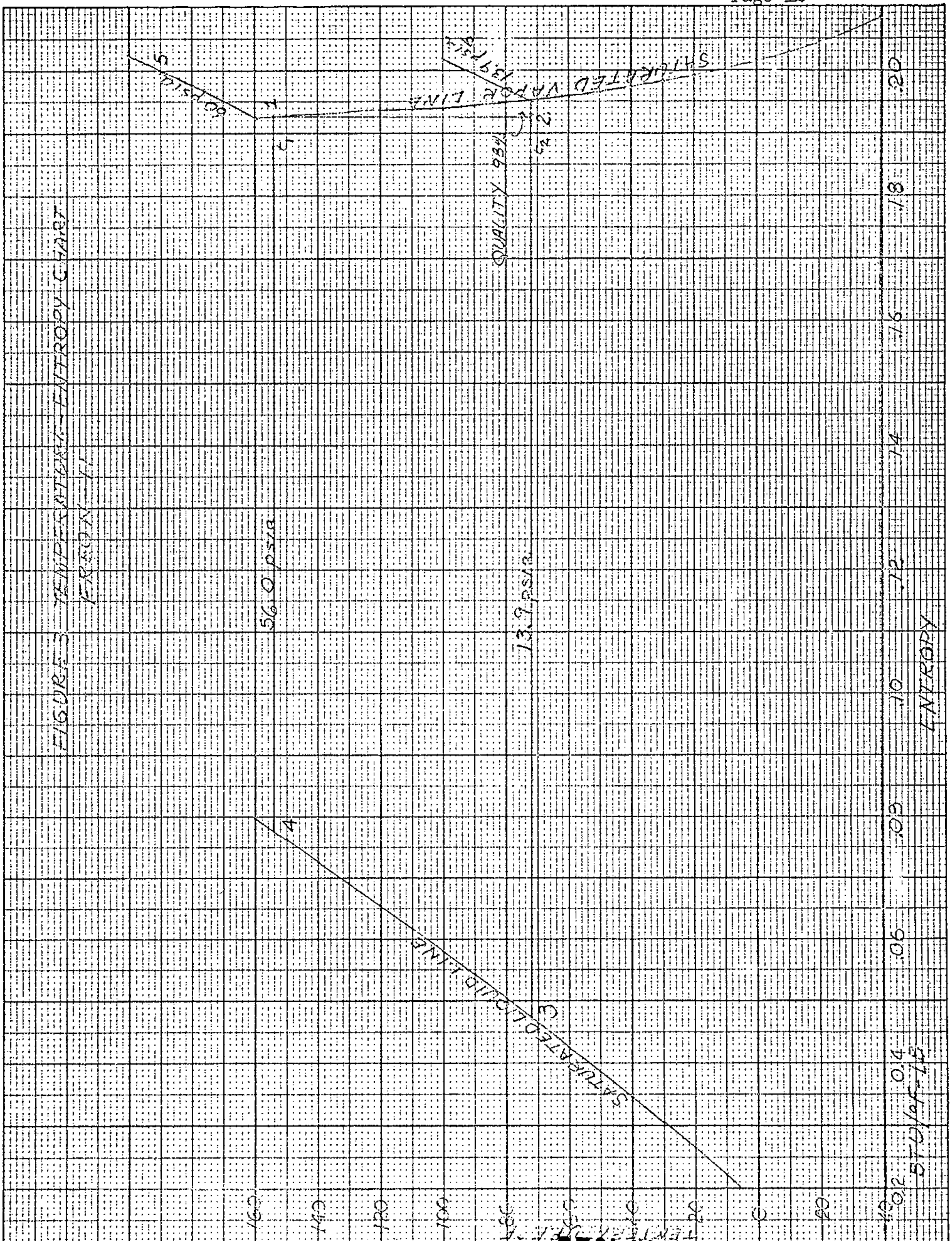
FIGURE 3 TEMPERATURE-ENTROPY CHART
FREON-11
SATURATED VAPOR LINE
SATURATED LIQUID LINE
56.0 psia
13.9 psia
QUALITY 93%
ENTROPY
TEMPERATURE DEG. F
.02
.04
.06
.08
.10
.12
.14
.16
.18
.20
160
140
120
100
80
60
40
20
0
-20
-40

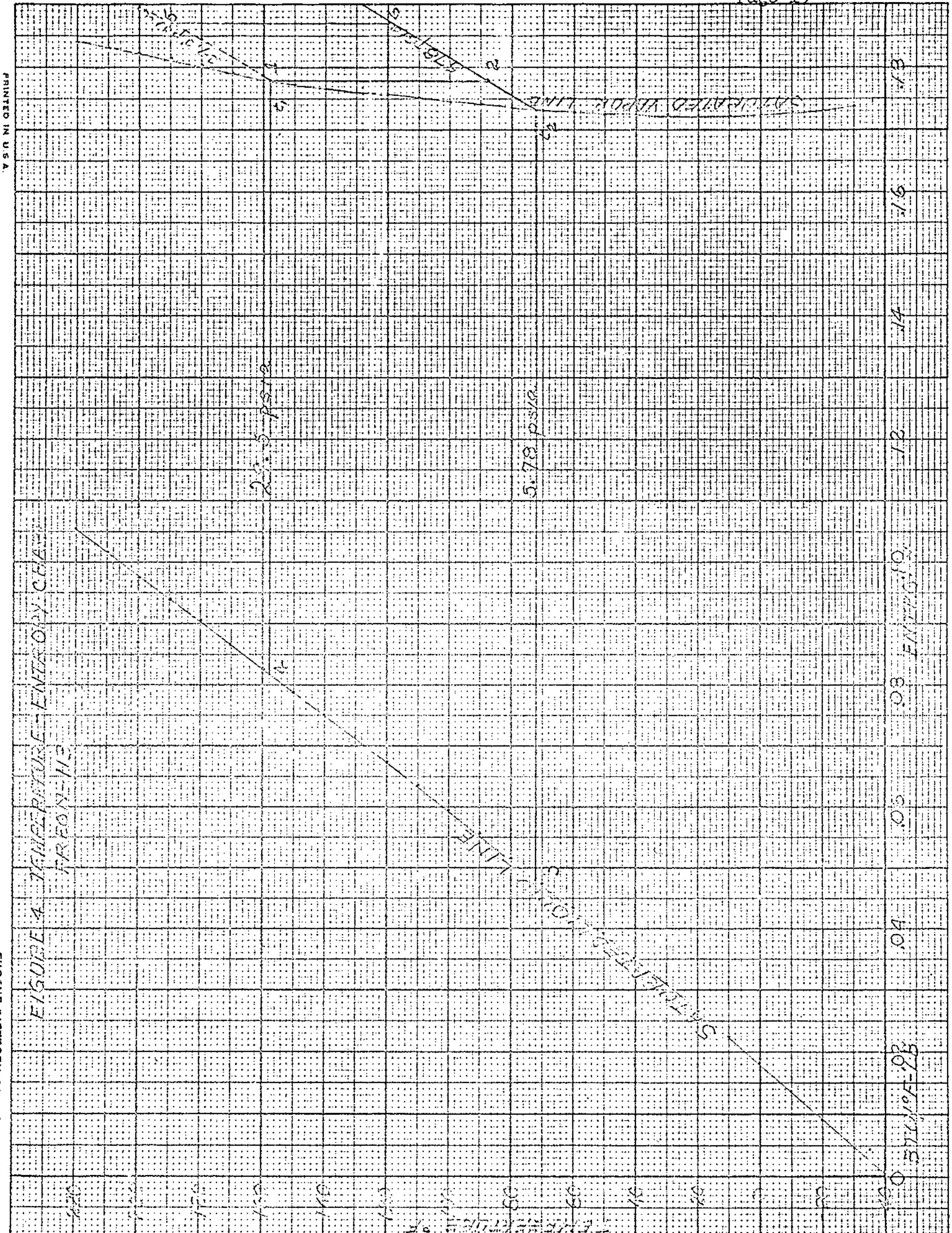

FIGURE 4. TEMPERATURE - ENTROPY CHART FREON-113

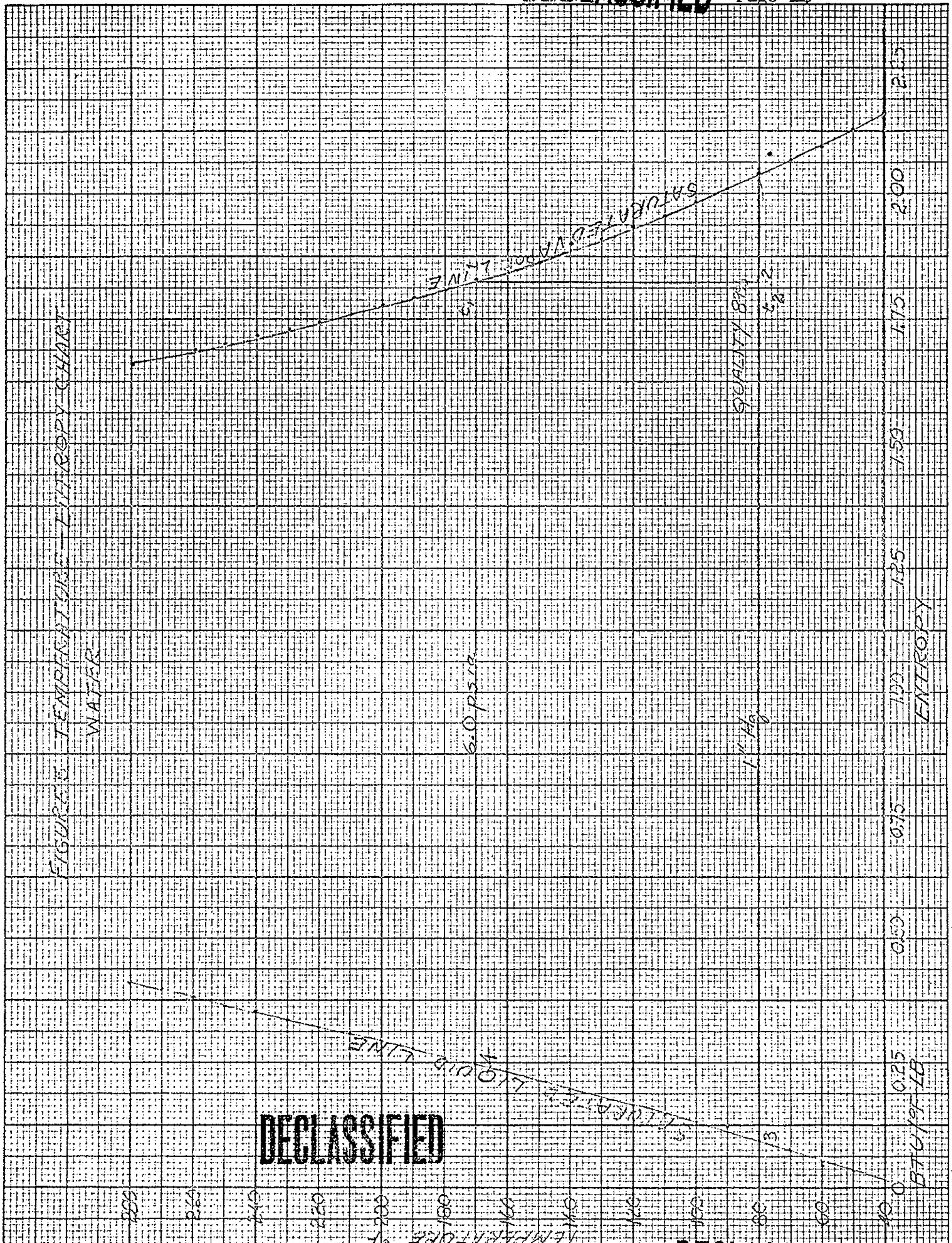
FIGURE 5 TEMPERATURE—ENTROPY CHART
WATER
TEMPERATURE °F
280
260
240
220
200
180
160
140
120
100
80
60
40
ENTROPY
BTU/°F LB
0
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
2.25
SATURATED LIQUID LINE
SATURATED VAPOR LINE
6.0 psia
1" Hg
QUALITY 8?%
t_2
2
3
4
5

condensing process. There is actually a loss in power recovery per pound of fluid. This condition is true as long as the saturated vapor line of a fluid slopes to the right. With water where the saturated vapor line slopes to the left superheating improves the cycle efficiency. Then, the heat source determines whether superheating is feasible which in this case it is not feasible.

Cooling water requirements of the various cycles of the fluids are shown on Figure 6. The difference between steam and the Freon rates is due to the condensing temperature of steam being 80°F instead of 72°F. A one inch Hg condenser pressure is the lowest practical pressure which can be maintained.

Plant Cost and Equipment

The first real cost which can be easily calculated is the power cycle working fluid cost. A time cycle in a closed loop would at the minimum be at least five minutes. Working fluid cost for maximum power recovery of Freon 11 and 113 respecitvely are $56,000 and $167,000.

Heat transfer equipment required in the closed loop are evaporators, condensers, and preheaters. Two factors will increase the size of the equipment above the size needed in a closed steam cycle, (1) mass flow rate, 17 times greater than steam, and (2) heat transfer, Freon fluids have lower film coefficients than water. While similar type surface condensers are used in both the Freon and steam cycles, the evaporators of the systems are different. The evaporator in an open steam cycle is a flash tank which is simple in construction as compared to a tube and shell type required for the Freon cycle.

The condensate pumping stations of the Freon systems will be large. Approximately 120,000 to 140,000 gpm capacities with heads up 45 psig are required.

As was mentioned before turbines for the Freon systems are a more conventional design and are smaller in size for equal power output rating than turbines needed for a sub-atmospheric pressure steam cycle, but a large turbine manufacturer has stated that a turbine using 6 psia steam could be designed and operated[6]. However, the costs of turbines for either Freon or steam systems would probably be about equal.

A preliminary estimate of construction cost of a Freon power recovery system as compared to a sub-atmospheric steam system of comparable power output would be approximately double in cost.

Cycle Operation

With the available pressure which can be generated at low temperatures, Freon systems could be operated according to standard power plant practices. Provisions must be made to prevent air and water leakage into the system. Some difficulty will be experienced as Freon fluids are not easily contained because of their low surface tension. Corrosion in the system will practically be non-existent if the moisture and air content is below the minimum allowable.

(6) Letter, G. B. Warren from C. W. Elston, Manager, Turbine Engineering, Large Steam Turbine Generator Department, July 2, 1954.

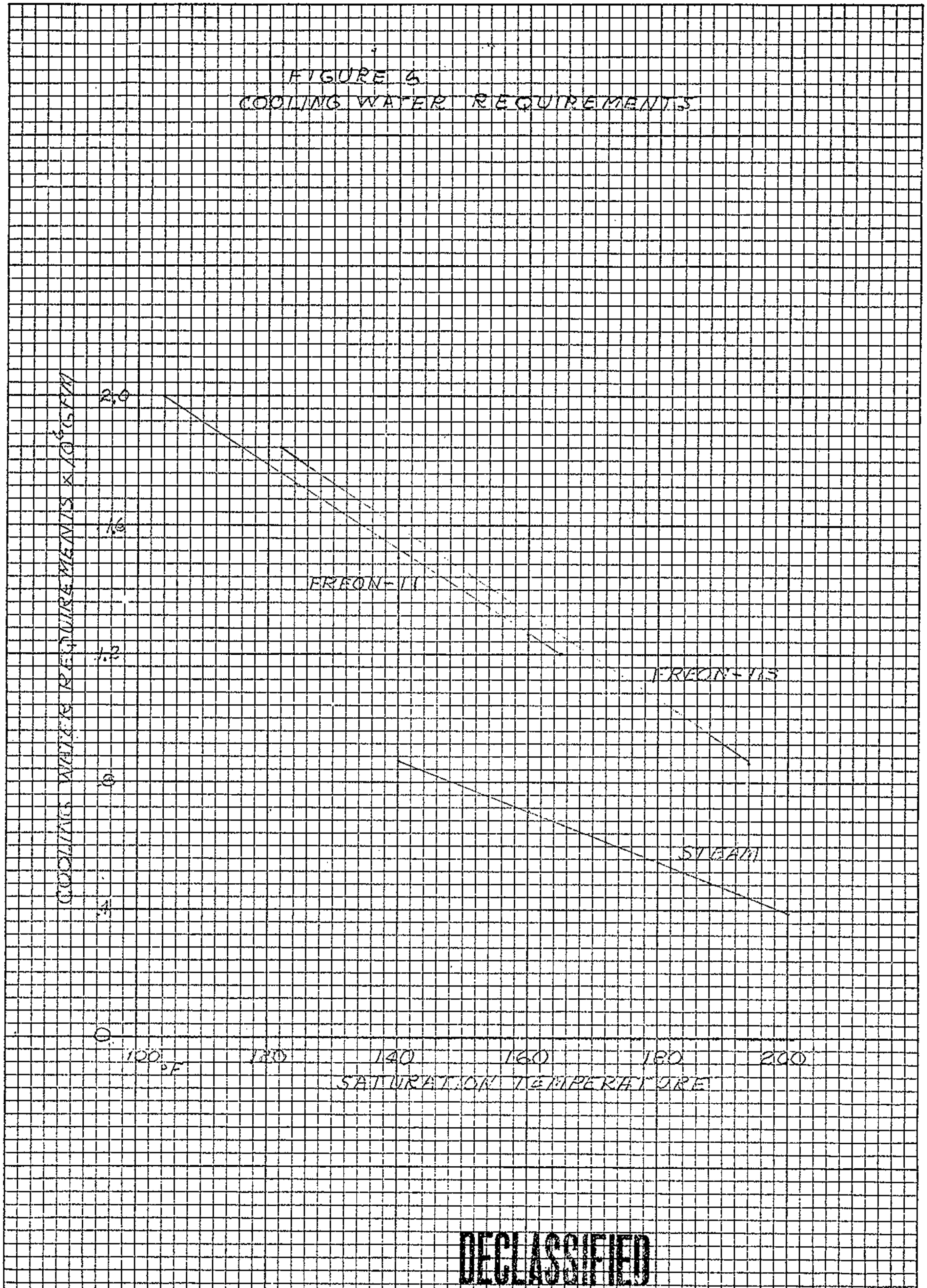

FIGURE 6
COOLING WATER REQUIREMENTS

CONCLUSIONS

The investigation of the possible use of refrigerants as the working fluids for low temperature power system have shown the following results:

1. The use of refrigerants as the working fluid in low temperature power cycles is technically feasible.

2. Net power recovery are approximately equivalent from reactor effluent water at 253°F using either Freon or steam.

3. Working pressures up to 60 psia can be generated from the reactor effluent water by substituting Freon fluids in the power cycle.

4. Larger mass flow rates are required with Freon fluids.

5. Larger cooling water rates are required for Freon systems than steam.

6. Substitution of Freon fluids for steam in a low temperature power recovery system does not appear feasible economically.

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