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SUBJECT: Thermodynamic Study of Ammonia-Water Heat Power Cycles

TO: A. M. Weinberg

FROM: James D. Maloney, Jr. and Roy C. Robertson

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II. NOMENCLATURE

The following nomenclature, symbols and units were adopted for this study:

b	Btu/lb	availability function: $b = h - T_o s$
B	Btu	availability function: $B = H - T_o S$
E	%	effectiveness
h	Btu/lb	enthalpy above 32°F datum
H	Btu/m lbs	enthalpy above 32°F datum
I	Btu	summation changes in available energy: $T_o \Sigma \Delta s$
m	lbs	flow
p	psia	absolute pressure
Q	Btu	heat transferred (+ = added, - = removed)
s	$\frac{\text{Btu}}{\text{lb-}^{\circ}\text{F abs}}$	entropy above 32°F datum
S	$\frac{\text{Btu}}{\text{m lbs-}^{\circ}\text{F abs}}$	entropy above 32°F datum
t	°F	temperature
T	°F abs	absolute temperature
v	ft^3/lb	specific volume
w	$\frac{\text{lbs } NH_3}{\text{lb sol.}}$	concentration of ammonia in mixture
W	Btu	mechanical work
x	%	percentage dry vapor per lb mixture (quality)

Subscripts

1,2, etc. refer to various state points in cycle

e = external medium

f = liquid phase

i = internal working substance

o = dead state (75 °F, 14.696 psia)

p = pump work

t = turbine work

v = vapor phase

III. SUMMARY

The purpose of this study is to determine whether an absorption-type power cycle using a mixture of ammonia and water as the working fluid, in a system such as that illustrated in Figure 1 of this Summary, would afford higher thermal efficiencies than those obtainable from a comparable steam power cycle, such as that shown in Figure 2.

It was found that the thermal efficiency of the ammonia-water power cycle is less than that of a typical steam power cycle. The efficiency of the ammonia-water cycle may slightly exceed that of a simple non-regenerative saturated steam cycle when boiler temperatures are above 350°F and the minimum exhaust quality is maintained at 83.5 percent, but the efficiency could never equal those of non-regenerative steam cycles operated with 30°F or more of superheat, but between the same overall temperature limits, and could never significantly exceed the efficiencies of a saturated steam cycle using two or more feedwater heaters. (See Figure 3 of this Summary). The above statement is true over the range of boiler temperatures investigated (280 to 380°F), and there is reason to believe that the same situation exists at higher temperatures, although the present lack of ammonia-water property data at the higher temperatures prevents computation of actual values.

The efficiency of the ammonia-water cycle is less than that of a steam cycle throughout the range of ammonia concentrations that could be employed. (See Figure 4 of this Summary). This is also true when other variables are introduced, such as permissible exhaust qualities, superheating of vapor to turbine, temperature of heat rejection, general arrangement of ammonia-water cycle equipment, and assumed actual turbine efficiencies.

A typical ammonia-water cycle efficiency is 20.7 percent while that of a saturated steam cycle operated between the same temperatures and permissible

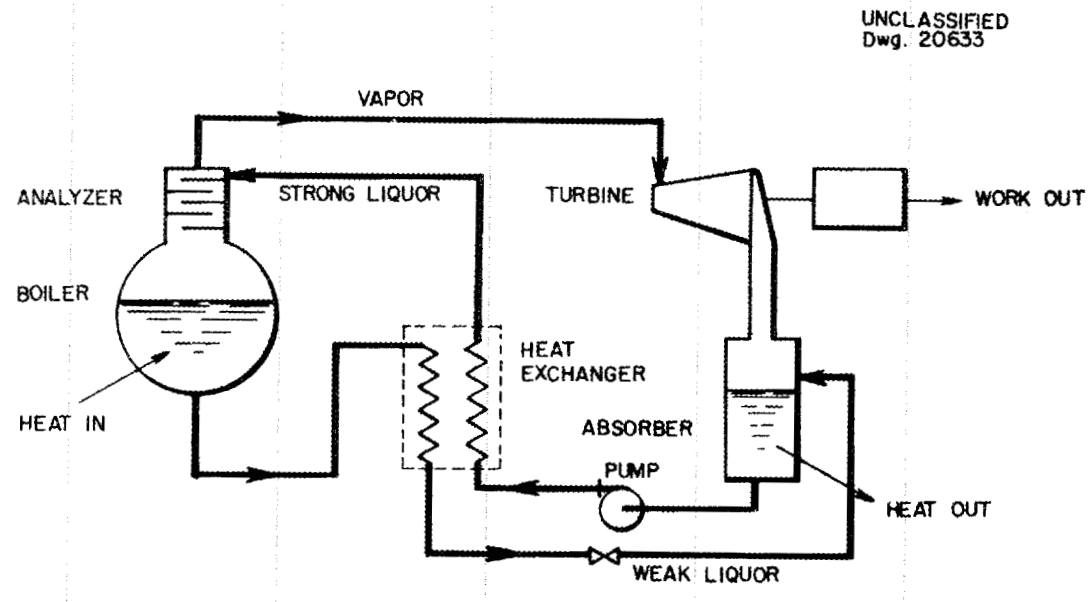

Figure 1. Ammonia-Water Power Cycle

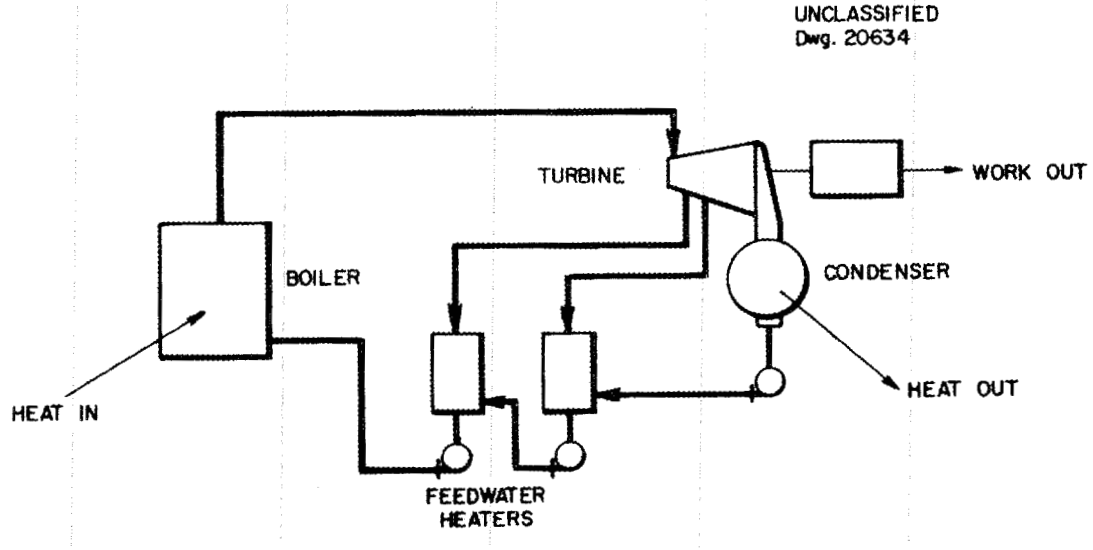

Figure 2. Simple Regenerative Steam Power Cycle

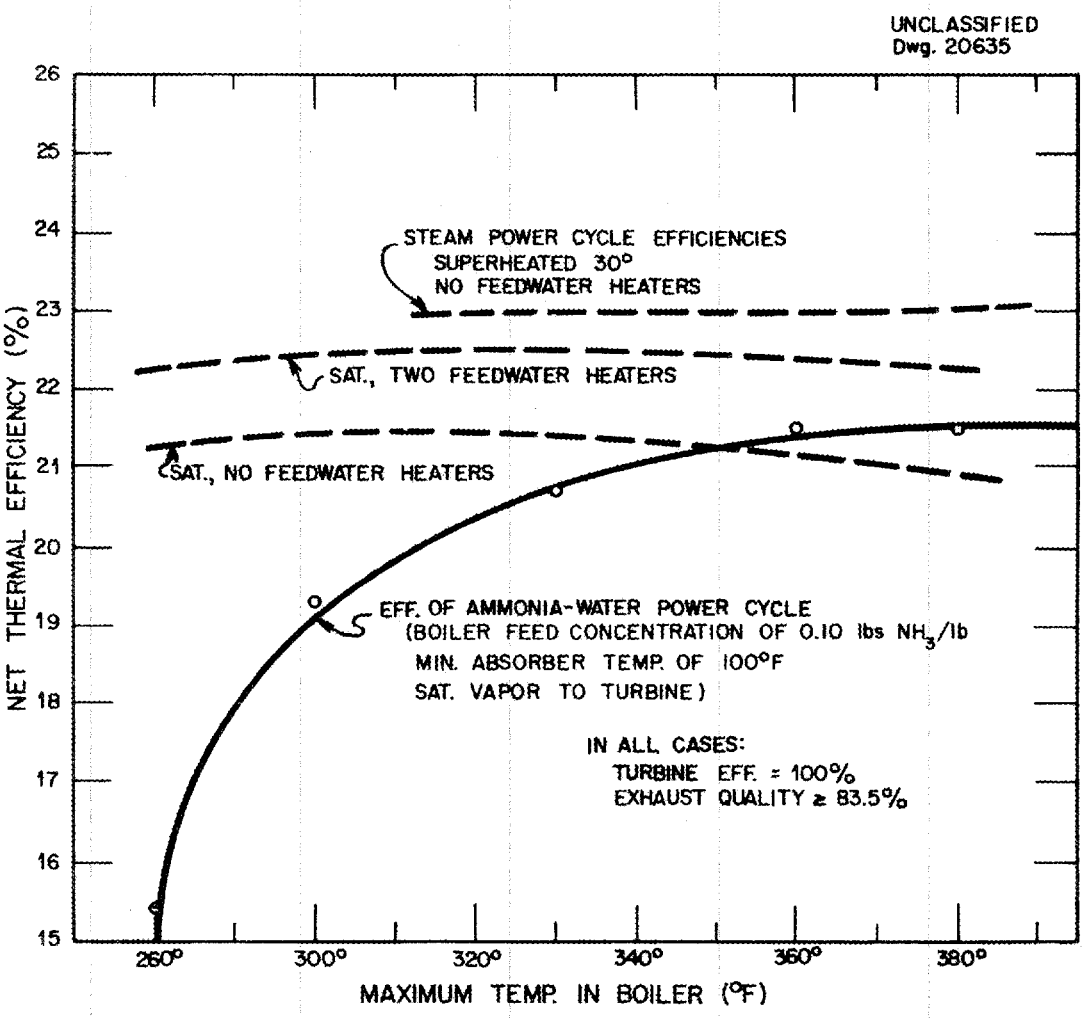

Figure 3. Effect of Boiler Temperature on Net Thermal Efficiency in Ammonia-Water and Conventional Saturated Steam Power Cycle

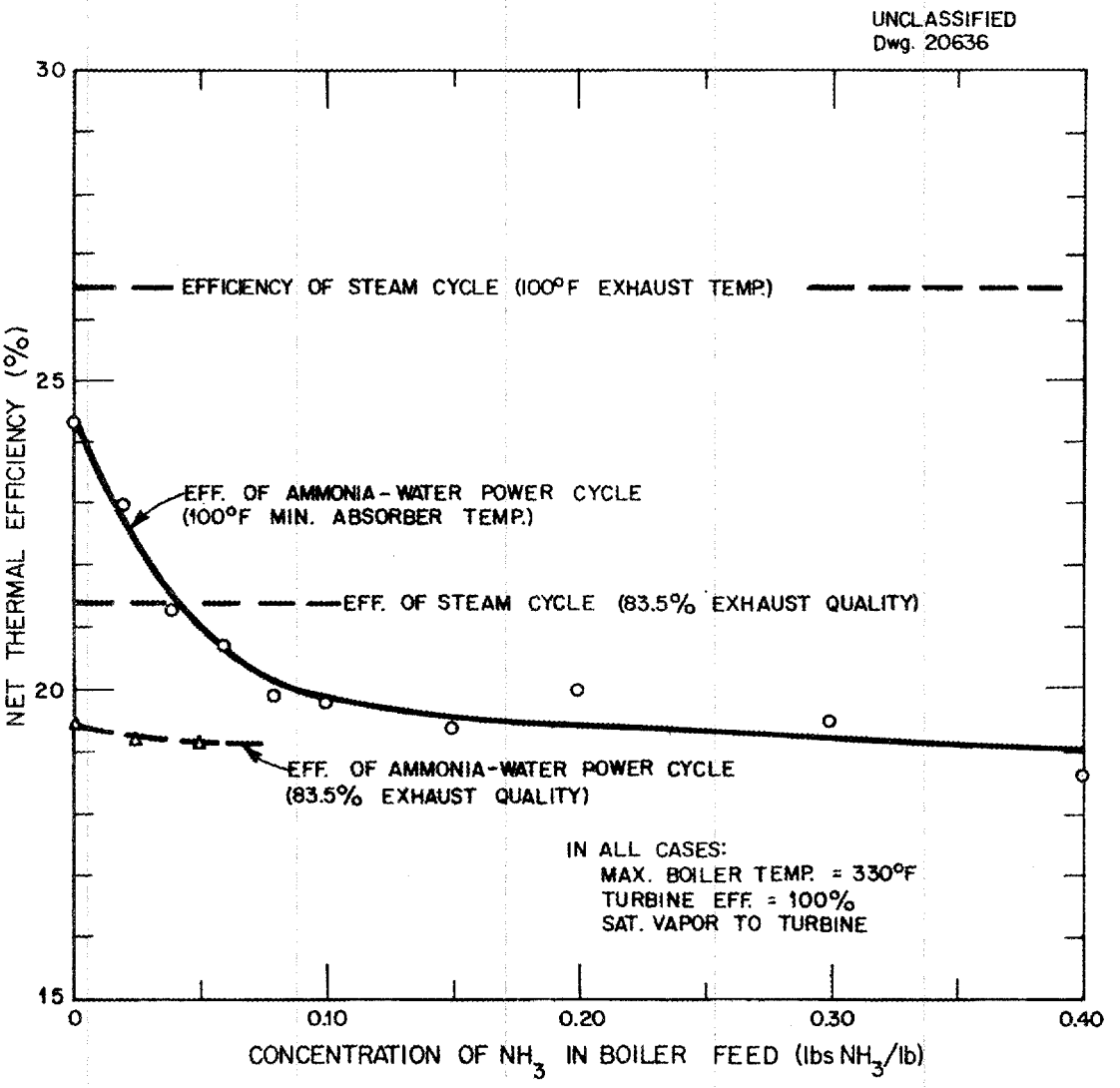

Figure 4. Effect of Ammonia Concentration on Net Thermal Efficiency of Ammonia-Water Power Cycle

exhaust conditions is 21.4 percent without regeneration, and 22.5 percent with two open feedwater heaters. A steam cycle with 30°F superheat, but maintaining the same overall temperature limits, would have an efficiency of 22.9 percent non-regenerative, and 24.1 percent using two feedwater heaters.

It is concluded that an absorption-type ammonia-water power cycle has no significant thermodynamic advantage as a means of converting heat into useful work.

A two-substance working fluid in which water is the more volatile component, such as a lithium chloride-water combination, may hold more promise, but to make studies of systems using such mixtures would require a great deal of preliminary effort to determine the thermodynamic properties.

IV. BACKGROUND AND INTRODUCTION

Although the search for a better working fluid has continued since the thermodynmaic heat-power cycle had its inception, no fluid has been found to be as practical as pure water. In modern high-temperature steam power cycles, however, there are two important disadvantages of the use of water as the working fluid. First, its relatively low critical point ($705^{\circ}F$) requires subcritical boiling and then superheating to the higher temperatures (about $1000^{\circ}F$) desirable for turbine operation. If the critical temperature were higher the heat could be added to the cycle at a higher average temperature and thus the heating process of the Carnot cycle could be more nearly realized. Secondly, the relatively high specific heat of water causes a proportionately large loss in available energy when the sub-cooled liquid is heated to saturation conditions, a loss which necessitates use of regenerative feedwater heating to realize the efficiencies now prevalent in modern plants.

Despite these shortcomings of water, no fluid has better characteristics over the entire range of temperature between 100° and $1000^{\circ}F$ now employed in heat power cycles. This is illustrated in Figure 5 which compares on the T-s plane cycles using pure water (b) and pure ammonia (c). Also shown in Figure 5 (a), are the properties of a hypothetical fluid which would permit the ideal Carnot cycle efficiency to be realized. The areas inclosed in the cycle represent the amount of work available from the cycle, per pound of working fluid. In the case of the ammonia cycle, however, additional work is required to compress the exhaust vapor to the saturation pressure to enable it to be condensed. This compression work must be subtracted from the available work so that the efficiency of the pure ammonia cycle is lower than indicated in the diagram, (Figure 5). As boiling and condensing afford practical means

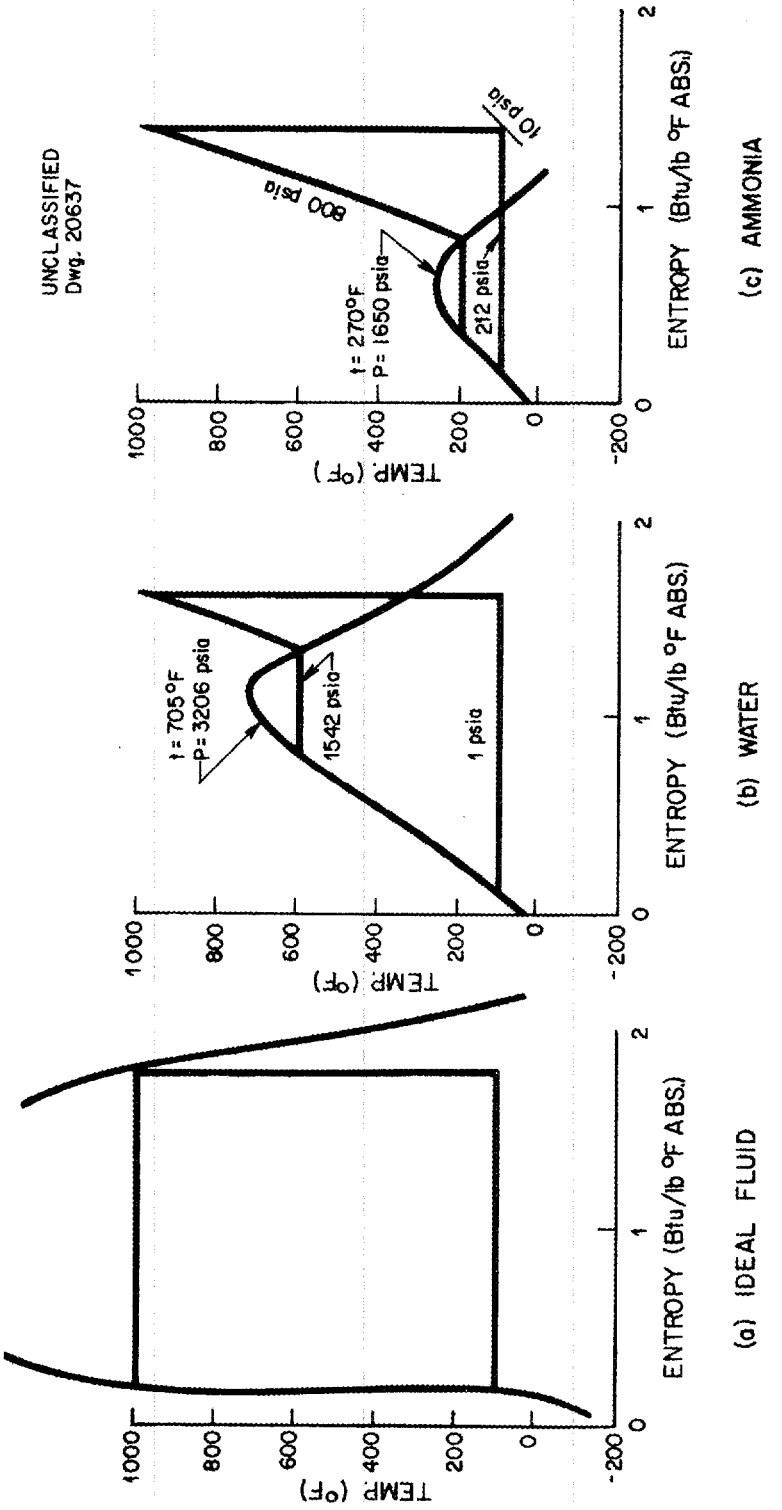

Figure 5. Temperature - Entropy Diagrams of Heat Power Cycles Operating Between 100°F and 1000°F

whereby heat can be added and subtracted from the cycle at constant temperature, it can be seen why it is desirable to have a fluid with saturated liquid and vapor conditions such as those shown in Figure 5 (a).

One of the most successful means of overcoming the disadvantages of the pure water cycle at high temperatures has been to use a two-fluid system of the binary type, wherein a fluid more suitable for the high temperature ranges rejects heat into a conventional steam cycle. Several binary plants using mercury as the high-temperature fluid are now in operation.

Absorption-type systems, using a mixture of two fluids, have been proposed. In such a system the boiler delivers a vapor rich in the more volatile component and an absorber is utilized at the turbine exhaust to reunite the weak liquor from the boiler with the vapor from the turbine. This type of cycle appears attractive in that, with suitable solutions, the heat will be added at a higher average temperature in the boiler. One heat exchanger located between the strong and weak liquor flows, replaces the several feedwater heaters in the conventional regenerative steam cycle.

Since water has good thermodynamic properties, a mixture in which water is the more volatile of the two components would seem to have the best possibilities for a multisubstance cycle. The soda machines of Honigmann and Zander,[1] proposed over fifty years ago, utilized steam from a boiling mixture of sodium hydroxide to drive a steam engine. In the references consulted, no data were given as to the actual efficiencies realized. The corrosion problem existing at that time discouraged further development.

An absorption-type binary cycle was proposed by Koenneman[2] in 1930. A solution of zinc ammonium chloride was to be heated in a boiler, for the purpose of driving off ammonia vapor which was to be passed through a turbine, then

reabsorbed in the weak solution, the heat of absorption being used to evaporate water. Initial superheat was added to the steam by the flue gases, and after partial expansion in the steam turbine, the steam was re-superheated and returned to the turbine. Further cycle improvement proposed the use of three stages of feedwater heating. An overall thermal efficiency of about 35 percent was estimated for the cycle. This value appears attractive in that the maximum cycle temperature was only 854 °F. No system has been constructed using this principle, probably because of the complexity of the equipment, the tendency of ammonia to disassociate at the proposed temperatures, corrosion problems, and the considerable uncertainty as to the actual thermodynamic properties of the mixtures used.

An exhaustive search of the literature has not been made for heat power cycles using multisubstance fluids, but insofar as is known, no study has been published covering an ammonia-water absorption-type power cycle. This report indicates the thermal efficiencies realizable in such an ammonia-water cycle when the concentration of ammonia, boiler temperature, etc., are varied. Comparisons are made to equivalent steam cycles. This study also includes an analysis of the ammonia-water cycle to show wherein the major losses of available energy occur.

V. DESCRIPTION OF AN AMMONIA-WATER POWER CYCLE

A power cycle utilizing a boiler to generate a vapor strong in ammonia, a turbine through which the vapor is expanded, and an absorber for absorbing the vapor from the turbine exhaust into the weak liquor, is shown in Figure 6. The efficiency of the cycle has been increased by the addition of a heat exchanger to heat the strong liquor with the flow of weak liquor. An analyzer is shown at the vapor outlet of the boiler wherein the saturated vapor leaving the boiler is increased in concentration to reach equilibrium with the strong liquor entering the analyzer.

The relationship of the properties of the working fluid as it passes through the various state points of the cycle is best shown by the enthalpy-concentration (h-w) diagram, (Figure 7). Starting at the throttle (1), the saturated vapor, rich in ammonia, is expanded through the turbine to a pressure fixed by the concentration and temperature of the strong liquor leaving the absorber. The temperature of the turbine exhaust (2 or 2') is determined by the efficiency of the expansion process and the exhaust pressure. The wet mixture leaving the turbine is absorbed in the weak solution in the absorber, giving up heat to the cooling medium. The weak solution leaves the boiler (5), passes through the heat exchanger and expansion valve to enter the absorber slightly subcooled (6). In the absorber the remaining heat of vaporization contained in the exhaust and the heat of absorption are removed from the cycle by the cooling medium and the ensuing liquor is cooled to within a few degrees of the available cooling water supply. The saturated strong liquor (3) leaving the absorber is then pumped to boiler pressure, passed through the heat exchanger and into the analyzer (4) slightly subcooled. The strong liquor cascading

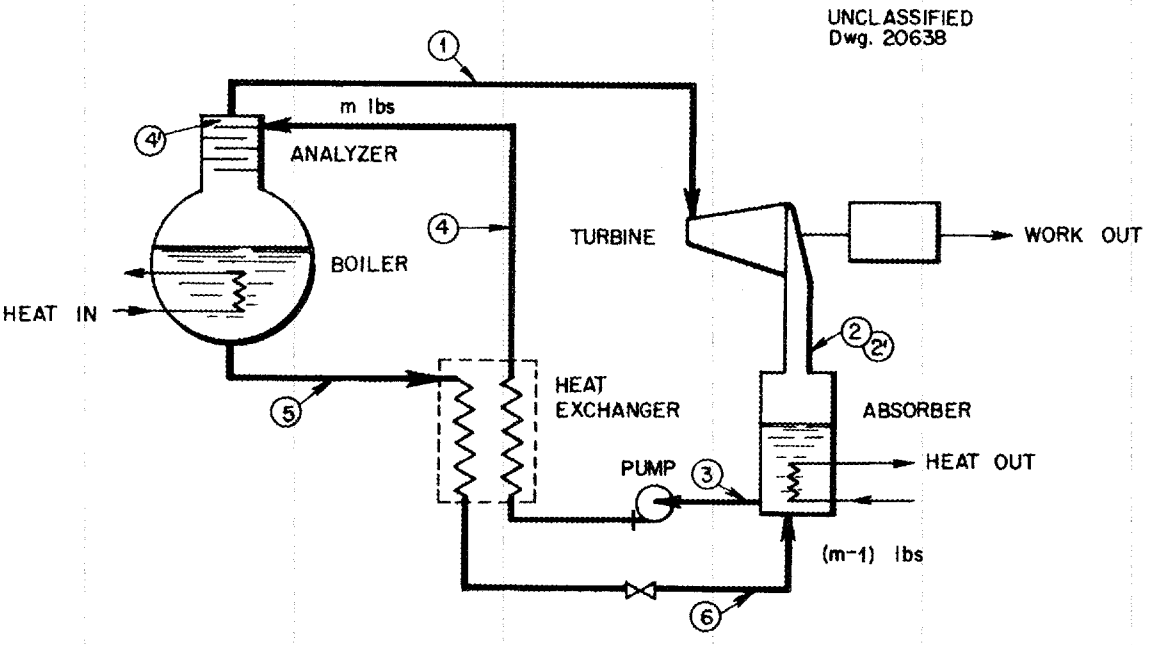

Figure 6. Ammonia-Water Power Cycle (Case I, II and III)

Table I

Properties of State Points in An Ammonia-Water Power Cycle

State Point	t	p	h	w	s	condition
1	300	248	799	0.748	1.313	sat. vapor, x = 100%
2 *	119	10	584	0.748	1.313	wet mixture, x = 84.8%
2' **	125	10	627	0.748	1.385	wet mixture, x = 89.4%
3	100	10	6	0.200	0.115	sat. liquid
4	296	248	242	0.200	0.467	subcooled liquid
4'	300	248	249	0.200	0.474	sat. liquid
5	330	248	300	0.134	0.523	sat. liquid
6	110	10	35	0.134	0.142	subcooled liquid

* Values assume turbine efficiency of 100%
** Values assume turbine efficiency of 80%

Assumed Conditions:

1 lb flow to the turbine throttle ; m = 9.3 lbs

$w_{f_3} = w_{f_1} = w_{f_4}$ $\quad t_1 = t_5 - 30°$ (except at low values of w)

$t_6 = t_3 + 10°$ $\quad t_3 =$ min. of 100 °F, or, $x_2 \geq 83.5\%$

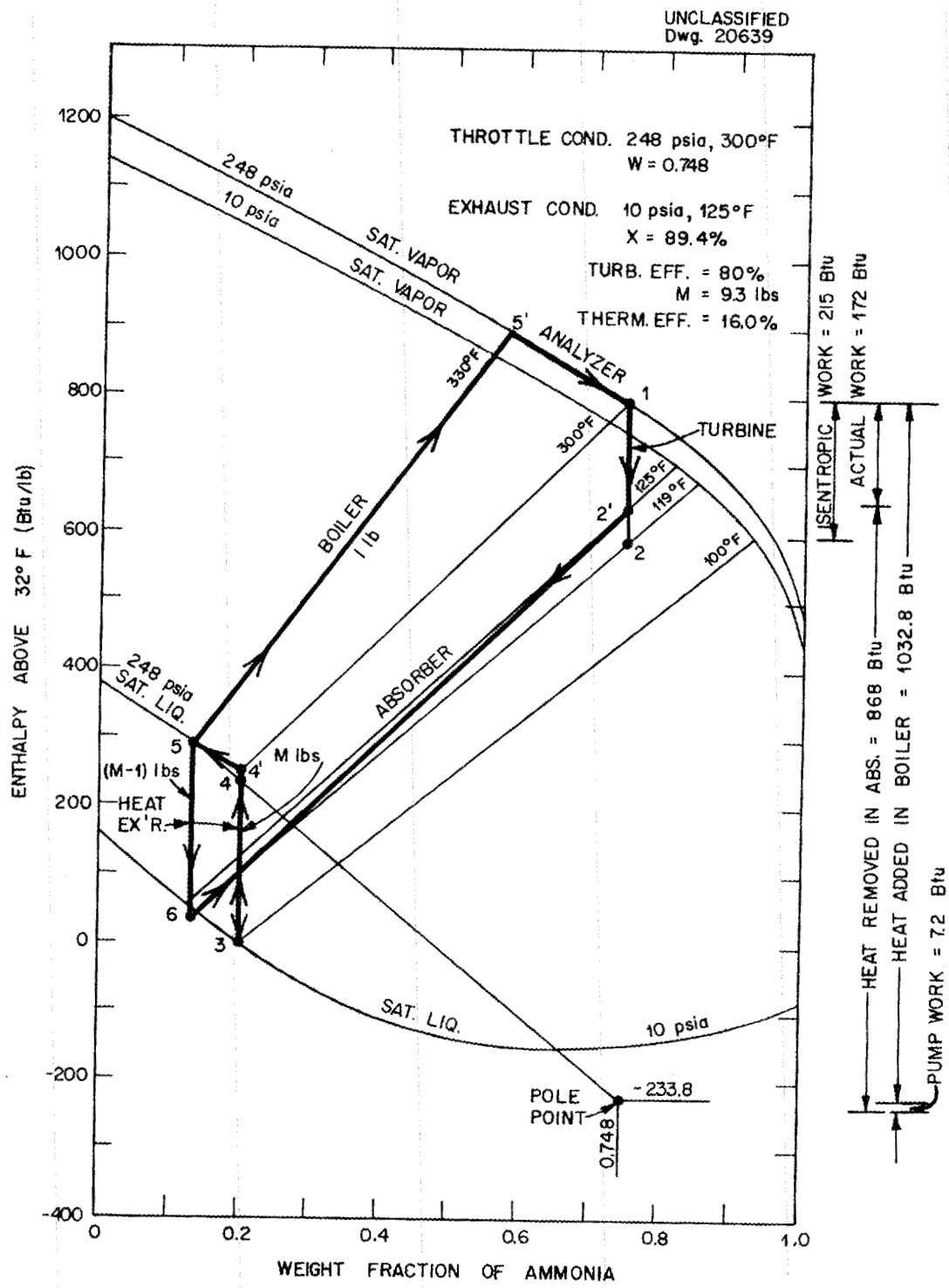

Figure 7. Enthalpy-Concentration Diagram, Ammonia-Water Power Cycle (Case III)

down through the upcoming saturated vapor gives up ammonia as it increases in temperature. A perfect analyzer is assumed, wherein the vapor leaving is at equilibrium concentration with the saturated strong liquor (4') and the cascading liquor leaves the analyzer in equilibrium with the vapor leaving the boiler, and is thus at weak liquor conditions (5).

A terminal difference of $10^{o}F$ has been assumed reasonable for the heat exchanger ($t_6 = t_3 + 10$), and this assumption has been used throughout this study. The difference in equilibrium temperature at boiler pressure between the strong and weak liquor has been maintained at $30^{o}F$, except in the cases where the weak liquor concentration is less than 0.10 lbs NH_3 per lb. In this region the difference must be less than $30^{o}F$ if equilibrium is to exist in the boiler and analyzer.

The absorber is assumed to be cooled with water at $75^{o}F$, and on assumption of reasonable Δt's, a minimum attainable absorber temperature of $100^{o}F$ has been assumed throughout this study. In the cases where an assumed minimum permissible quality of the wet mixture in the turbine exhaust is taken as the limiting factor, the absorber temperature may exceed $100^{o}F$.

The problem of erosion by moisture in the low-pressure blading of a steam turbine is well known. In the cases where 100 percent turbine efficiency has been assumed, a maximum allowable moisture content of 16.5 percent has been used, since at this condition the resulting exhaust pressure is the same as that reached by a turbine having 80 percent turbine efficiency and limited to 12 percent moisture in its exhaust. For simplicity 100 percent turbine efficiency has been assumed when studying the effects of boiler temperature and boiler feed concentration. The moisture consideration is not of concern

except at low ammonia concentrations, because at strong liquor concentrations greater than about 0.10 the exhaust end point is fixed by the 100°F minimum absorber temperature and the moisture in the exhaust is well below the allowable amount.

VI. ANALYSIS OF AN AMMONIA-WATER POWER CYCLE

A. Sources of Ammonia-Water Thermodynamic Properties Data

Practically all the data published on the properties of ammonia-water mixtures are based on the German investigations by Zinner[3] and Wucherer[4]. The range of temperatures covered is only that which is of interest in absorption refrigeration cycle studies. The Stickney[5] tables and charts were published in 1935, the Jennings[6] tables in 1938, and the Scatchard[7] values in 1947. There is fairly close agreement between the Stickney and Jennings data but the Scatchard values, particularly at high ammonia concentrations, departs markedly from the others, including the original Wurcherer data. Stickney writes[8] "... engineers should use the Scatchard tables with caution, if at all." Scatchard acknowledges the discrepancy but questions the original Wurcherer data.

In studying the ammonia-water power cycle, the properties of entropy, free energy and availability were necessary. Scatchard's tables are the only source of these properties, the Jennings and Stickney tables giving only t, h_f, h_v and w_v as a function of w_f and pressure, up to 300 psia. The Jennings data are tabulated in smaller increments of concentration between 0 and 10 percent ammonia and were found to be very useful in the low concentration regions of investigation. The Scatchard tables were used to determine entropy in all cases.

Although the values of h_f at higher temperatures may have differed by as much as 25 percent between the Scatchard and Jennings values (see Figure 16 in Appendix) comparative calculations of thermal efficiencies based on each source indicate that the efficiencies were not affected sufficiently ($\pm$ 0.5 percent thermal efficiency) to cause question as to the general findings of this study.

B. Determination of Properties and Method of Calculation

It was found that interpolation within the tables of properties introduced considerable error, particularly in the low concentration ranges because the properties in this region are affected greatly by the addition of only small amounts of ammonia. It was therefore found necessary to plot all the data and to read the values from these graphs. Some of the more important of the graphs of properties are shown in reduced scale in the Appendix as Figures 14 through 18.

In studying the effect of ammonia concentration in the system on the cycle thermal efficiency it was necessary to investigate the regions of low concentration. First attempts gave a scatter of $\pm$ 1 percent thermal efficiency at these low values, making placement of the efficiency curve doubtful. Trial calculations indicated that this was due to the latitude with which the property data could be read. For this portion of the study it was found necessary to plot all the values found in each step of the calculations against the concentration of the strong liquor, it being assumed that these values would vary along smooth curves, and by adjusting the property data within the range of possible readings, bring the plotted values into line. Two of these plots of calculated values have been included in the Appendix as Figures 19 and 20. Since the efficiency was more sensitive to the enthalpy in the turbine exhaust than to any other calculated value more attention had to be directed to adjustment of this value than to any other.

Although the absolute accuracy of the original property data is questionable, and some adjustments were necessary, as explained above, it is believed that the calculated thermal efficiencies are reasonably correct. In practically all

cases the efficiency of the steam system is enough greater to prevent small errors in the ammonia-water cycle efficiency from changing the overall conclusions presented in this report.

Determination of the turbine exhaust conditions presented the greatest difficulty in making the calculations because the state point in the exhaust is fixed by several variables not easily equated. Trial and error methods were used. The turbine exhaust pressure must be that established by the desired concentration and temperature in the bottom of the absorber and, at the same time, the exhaust temperature must satisfy the requirement of constant entropy expansion to the exhaust pressure. Even though the overall concentration in the flow through the turbine is constant, the vapor becomes richer in ammonia as the quality decreases.

Two different situations were presented, (1) when the minimum permissible quality of turbine exhaust established the end point of the expansion process, and (2) when the minimum absorber temperature of 100°F controlled the end point. It was found that the quality controlled at low concentrations and the absorber temperature at the higher values (over 0.10).

The general procedure followed in calculating the net thermal efficiency of the ammonia-water cycle, with 100°F minimum absorber temperature controlling, is briefly as follows: On assuming the strong liquor concentration entering the boiler and the maximum boiler temperature to be investigated, the entropy and concentration of the saturated vapor entering the turbine were obtained from the properties. At 100°F and the known concentration of strong liquor leaving the absorber, the exhaust pressure was taken from the properties. A trial turbine exhaust temperature was then assumed which, with the known

exhaust pressure, affords values from the properties for liquid and vapor concentrations in the exhaust. Knowing these values, the quality was then calculated. The values of entropy of saturated liquid and saturated vapor were taken from the properties and, using the calculated quality, the entropy determined. If the assumed exhaust temperature were correct, this calculated entropy would agree with the entering entropy. Three or four trials were usually sufficient to effect a solution. After the exhaust conditions were established, the absorber was then analyzed on the basis of an ammonia balance and the weight of weak liquor needed per pound of vapor through the turbine calculated. An energy balance of the heat exchanger could then be made for the determination of exit enthalpies. Enough data were then available to determine the enthalpies at all points of the cycle, from which the ratio of net useful work to the heat input required could be obtained.

Sample Calculations - Case II; (Item 10, Table II)

Conditions:

Maximum boiler temperature = t_5 = 330°F

Minimum absorber temperature = t_3 = 100°F

$t_6 = t_3 + 10°F = 110°F$

$t_1 = t_5 - 30°F = 300°F$

$w_{f_1} = w_{f_3} = w_{f_4} = 0.20$ lbs NH_3/lb sol.

From the property data with, $t_1 = 300°F$, $w_{f_4} = 0.20$, known

$p_1 = p_4 = p_5 = 248$ psia

$s_{v_1} = 1.313$

$w_{v_1} = 0.748$

$h_{v_1} = 799$ Btu/lb.

From the property data with, $t_3 = 100°F$, $w_{f_3} = 0.20$, known

$p_3 = p_2 = p_6 = 10$ psia

$h_{f_3} = 6$ Btu/lb.

From the property data with, $t_6 = 110°F$, $p_6 = 10$ psia, known

$w_{f_6} = 0.134$

$h_{f_6} = 35$ Btu/lb.

Since $w_{f_5} = w_{f_6}$, from the property data (Scatchard):

$h_{f_5} = 300$ Btu/lb.

Assuming a turbine exhaust temperature of 119°F, and knowing p_2, from the property data:

$w_{f_2} = 0.147$

$w_{v_2} = 0.857$

$$\text{and,} \quad x_2 = \frac{w_{v1} - w_{f_2}}{w_{v_2} - w_{f_2}}$$

$$\text{then} \quad x_2 = \frac{0.748 - 0.147}{0.857 - 0.147} = 0.848$$

$$\text{also} \quad s_{f_2} = 0.158$$

$$s_{v_2} = 1.510$$

Then with the quality, x_2

$$s_2 = x_2(s_{v_2} - s_{f_2}) + s_{f_2}$$

$$s_2 = 0.848(1.510 - 0.158) + 0.158 = 1.312$$

This value checks closely with $s_1 = 1.313$ and indicates a reasonably correct assumption of the turbine exhaust temperature.

Knowing t_2, p_2, w_{f_2}, and x_2

$$h_2 = x_2 h_{v2} + (1 - x_2)h_{f_2}$$

$$h_2 = 0.848(682) + 0.152(40) = 584.1 \text{ Btu/lb}$$

The gross work from the turbine is then

$$w_t = 799 - 584 = 215 \text{ Btu/lb to throttle}$$

By making a mass balance around the absorber

$$m = \frac{(w_{v1} - w_{f_6})}{(w_{f_3} - w_{f_6})}$$

$$m = \frac{(0.748 - 0.134)}{(0.20 - 0.134)}$$

$$m = 9.3 \text{ lbs/lb through the turbine}$$

By making a heat balance of the heat exchanger

$$h_4 = \frac{[(m - 1)(h_5 - h_6) + mh_3]}{m}$$

$$h_4 = \frac{8.3(300 - 35) + 9.3(6)}{9.3}$$

$$h_4 = 242 \text{ Btu/lb}$$

From a heat balance of the boiler and analyzer

$$Q_{in} = h_1 + (m - 1)h_5 - mh_4$$

$$Q_{in} = 799 + 8.3(300) - 9.3(242) = 1033 \text{ Btu}$$

The ideal pump work is approximately

$$W_p = \frac{144(p_4 - p_3)(v_{f_3})(m)}{778}$$

$$W_p = \frac{144(248 - 10)\ (0.0175\ (9.3)}{778} = 7.2 \text{ Btu}$$

The net work of the cycle is then

$$W = W_t - W_p = 215 - 7.2 = 208 \text{ Btu}$$

and the net thermal efficiency of the cycle is

$$\text{T.E.} = W/Q_{in} = \frac{208}{1033} = 20.0\ \%$$

Sample Calculations - Case III

Conditions:

> Same as for case II except that a turbine efficiency of 80 percent is assumed. The turbine exhaust condition is therefore changed and is represented by point 2' in Figures 6 and 7. The properties of this state point are fixed by the absorber pressure, overall concentration, and the enthalpy.

From the property data with, $p_{2'} = 10$ psia, $w_{2'} = 0.748$, turbine efficiency = 80 percent, known,

$$h_{2'} = h_1 - 0.8(h_1 - h_2)$$

$$h_{2'} = 799 - 0.8(799 - 584) = 627 \text{ Btu/lb}$$

Now graphically from the enthalpy-concentration diagram or from calculations,

$$t_{2'} = 125^{\circ}F$$

$$x_{2'} = 89.4\ \%$$

The gross turbine work is then,

$$W_t = 0.8 \text{ (ideal work) or } h_1 - h_{2'}$$

$$W_t = 0.8\ (215) = 172 \text{ Btu}$$

As the pump work is the same as in case II, the net turbine work is,

$$W = W_t - W_p = 172 - 7 = 165$$

As Q_{in} is also the same as in case II, the net thermal efficiency is then

$$\text{Thermal efficiency} = W/Q_{in} = 165/1033 = 16.0\ \%$$

C. Cases Studied in the Ammonia-Water Power Cycle

In order to determine whether or not any circumstances might exist wherein the thermal efficiency of the ammonia-water cycle would be significantly greater than that of a conventional steam cycle operating under comparable conditions, the following cases were selected for study:

Case I. Effect of boiler temperature on thermal efficiency, assuming 100 percent turbine efficiency. Following Carnot's principle, the thermal efficiency should increase with boiler temperature, but it is conceivable that the ammonia-water system might have better characteristics in this respect and might at some range of temperatures give better efficiencies. For simplicity, a turbine efficiency of 100 percent was assumed.

Case II. Effect of concentration of ammonia in the system on thermal efficiency, assuming 100 percent turbine efficiency. The higher efficiency of the pure water system over that of a pure ammonia system was illustrated on the T-s plane (Figure 5), but the question arises as to whether there is some optimum concentration in between that will afford better efficiencies than for the pure substance.

Case III. Effect of actual turbine efficiency on the thermal efficiency of the cycle. If an actual turbine efficiency of 80 percent is assumed, a value which approximates that of most turbines now in operation, the expansion can proceed to a lower pressure and temperature before the limiting amount of moisture is reached. Although this effect has already been taken into account by permitting 16.5 percent moisture for the 100 percent-efficient turbine as compared to 12 percent for the actual turbine, it was realized that the ammonia-water system, and the steam system might respond differently to the introduction to this variable.

Case IV. Effect of the analyzer in the ammonia-water power cycle on the thermal efficiency, assuming 80 percent turbine efficiency. If the analyzer were not used in the cycle, the vapor supplied to the turbine would be 30°F hotter (but at lower concentration), indicating a possibility that the cycle would have a better thermal efficiency without its use.

Case V. Effect of superheating the vapor from the analyzer, assuming 80 percent turbine efficiency. Since the advantage of superheat in a conventional steam cycle is that the turbine expansion can proceed to a lower pressure before the moisture limitation is reached, this effect on the ammonia-water cycle turbine is also of interest. The overall temperature limits were kept the same as in the other cases studied.

D. Results of Studies of Ammonia-Water Cycle

Case I. The effect of increasing the temperature in the boiler for a given strong liquor concentration of 0.10 has been shown in Figure 3 of the Summary. A summary of the calculated values for this case is shown in Table II (following). The thermal efficiency increased as the boiler temperature was increased, as was expected, but it should be noted in Figure 3 that the thermal efficiency curve has begun to level off at boiler temperatures above 380°F, indicating that if the temperature were increased further, little improvement in efficiency could be expected. This statement can be substantiated by examination of the effect boiler temperature has on turbine exhaust temperature (Figure 8).

TABLE II

SUMMARY CALCULATED VALUES

AMMONIA-WATER POWER CYCLE STUDIES

Cases I and II

Item No.	Max. Boiler Temp. °F	Boiler Concen. w_1	Turbine Exhaust Quality %	Turbine Exhaust Temp. °F	Temp. Leaving Absorber °F	Weak Liquor Circu. lbs.	Heat Added Btu	Pump Work Btu	Turbine Work gross Btu	Net Thermal Eff. %
1	280	0.10	87.8	112	100	9.7	1096	2.1	171	15.4
2	280	0.20	87.8	112	100	10.3	1021	4.1	170	16.3
3	300	0.10	84.8	115	100	9.6	1076	2.7	210	19.3
4	330	0.00	79.4	100	100	10.0	1210	3.0	297	24.3
5	330	0.02	81.0	106	100	7.6	1165	2.4	270	23.0
6	330	0.04	81.8	110	100	7.0	1130	2.0	243	21.3
7	330	0.06	82.8	114	100	7.2	1106	2.4	231	20.7
8	330	0.08	83.5	119	100	7.7	1111	3.4	225	19.9
9	330	0.10	83.5	119	100	8.4	1091	3.6	219	19.8
10	330	0.20	84.8	119	100	9.3	1038	7.2	215	20.0
11	330	0.30	86.1	115	100	9.0	982	12.0	203	19.5
12	330	0.40	88.0	105	100	7.4	843	15.1	172	18.6
13	330	0.00	83.5	147	147	10.0	1167	3.0	229	19.4
14	330	0.025	83.5	135	125	7.2	1111	2.2	215	19.2
15	330	0.05	83.5	122	111	6.6	1113	2.1	213	19.1
16	360	0.10	83.5	140	117	7.5	1074	7.5	236	21.5
17	380	0.10	83.5	155	133	6.7	1065	4.9	234	21.5

All of the above values based on 1 lb. vapor to turbine.

For values of h and w, Case II, see Figures 19 and 20, Appendix.

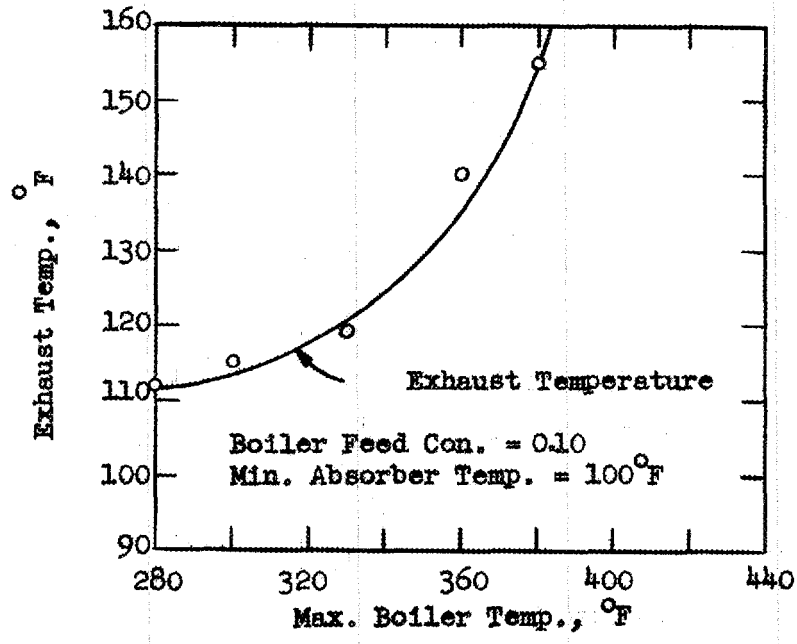

Fig. 8 Effect of Boiler Temperature on Turbine Exhaust Temperature Ammonia-Water Power Cycle.

At concentrations of 0.10 and above, the limiting factor on turbine expansion is the 100°F minimum absorber temperature; therefore, the turbine exhaust temperature increases very rapidly with increases in boiler temperature and it is indicated that if ammonia-water property data had been available for studies above 380°F, the thermal efficiency would show little or no improvement. This effect of boiler temperature was briefly investigated at several different boiler concentrations and found to be substantially the same, so only one boiler concentration is reported in detail.

Some caution must be exercised in explaining the influence of temperature on the ammonia-water cycle using Carnot's efficiency and taking turbine exhaust temperature as a criterion. This exhaust temperature is not the minimum temperature at which heat is rejected and, furthermore, the enthalpies in the exhaust are a function not only of temperature but also of concentration.

The thermal efficiency of the ammonia-water system can exceed that of a saturated steam system without feedwater heaters at temperatures above 360°F, if 30°F or more of superheat is used for the steam cycle, but with the same overall temperature limits, the efficiency of the ammonia-water cycle can not equal that of the steam cycle. If two stages of feedwater heating are used in the saturated steam cycle the efficiency is higher than that of the ammonia-water cycle, and if both superheat and regeneration are employed the steam cycle is significantly higher in efficiency than the ammonia-water cycle.

Case II. The effect of decreasing the concentration of ammonia in the cycle is to increase the thermal efficiency, reaching a maximum efficiency at zero ammonia, as was shown in Figure 4 in the Summary. The results of this study are also summarized in Table II. At zero concentration the circulation of boiler liquid through the heat exchanger is no longer necessary, but even with this irreversible process left in the cycle the thermal efficiency is still better than when any ammonia is in solution. Elimination of the heat exhanger and the circulation of the boiler liquid, converting the cycle to that of a simple steam cycle, improves the efficiency still further. The effect of concentration at several different boiler temperatures was briefly investigated and found to be about the same; therefore, only one boiler temperature, 330° F, was studied in detail. As shown in Figure 4, no concentration exists between 0 and 0.4 for 330 °F boiler temperature that affords thermal efficiencies as high as that of a simple, non-regenerative, saturated steam cycle operated with the same turbine exhaust limitations. There could be no workable concentration greater than 0.4 because at this value the turbine exhaust temperature is essentially that of the 100 °F minimum allowable in the absorber; further increases in

strong liquor concentration would result in exhaust pressures lower than that required in the absorber, a situation that for practical purposes could not exist.

Case III. If a turbine efficiency of 80 percent is assumed, the thermal efficiencies of the ammonia-water and the steam cycles are reduced in about the same proportion. When the maximum boiler temperature is 330 °F and the minimum absorber temperature is 100 °F, changing the turbine efficiency from 100 percent to 80 percent causes the exhaust temperature to increase from 119 °F to 125 °F, raises the exhaust quality from 84.8 percent to 89.4 percent, and reduces the thermal efficiency from 20.0 percent to 16.0 percent. This study has been summarized in Figures 6 and 7, with the state point leaving the 80 percent-efficient turbine designated as 2'. A similar reduction in turbine efficiency in a steam cycle reduces the cycle thermal efficiency from 21.4 percent to 17.1 percent, as shown by Items 6 and 12, Table VI.

It is concluded that if actual turbine efficiencies are used, the thermal efficiency of the ammonia-water cycle will be reduced about the same amount as will the efficiency of a steam cycle, and the ammonia-water cycle shows no superiority in this respect. With regard to the turbine efficiencies that could be expected from an ammonia turbine, it is not known what effects the differences in molecular weight, viscosities, etc. would have since ammonia turbines have never been developed.

Case IV. When the analyzer is removed from the basic cycle and the system is arranged as shown in Figures 9 and 10, the thermal efficiency is not improved. Without the analyzer the saturated vapor to the turbine is in equilibrium with the weak liquor rather than with the strong liquor but

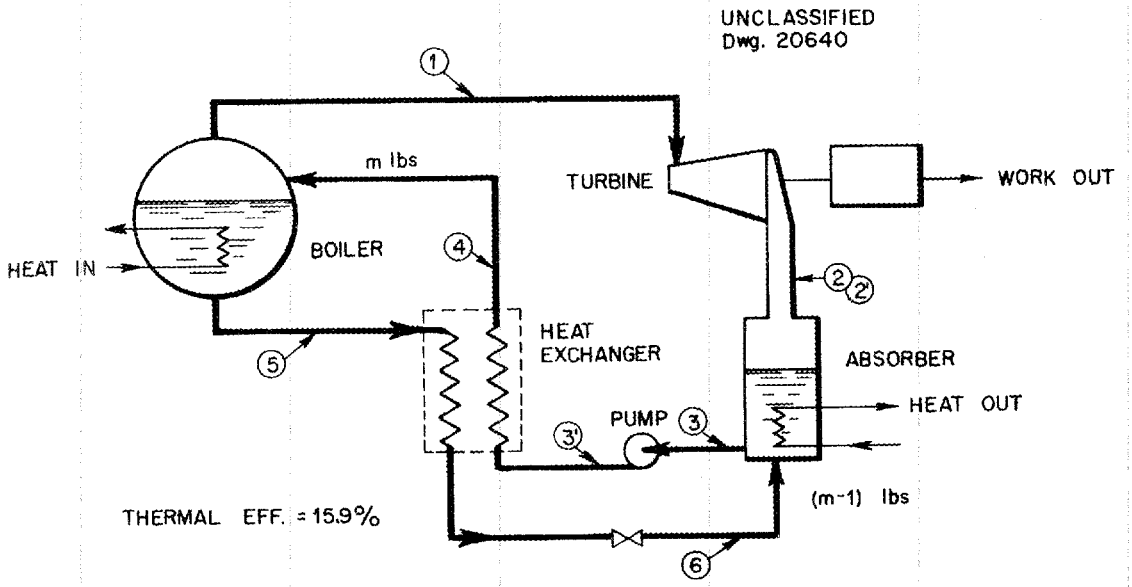

Figure 9. Ammonia-Water Power Cycle (Without Analyzer, Case IV)

Table III

Properties of State Points in An Ammonia-Water Power cycle (without analyzer)

State Point		t	p	h	w	s	Condition
1		330	248	895	0.58	1.392	sat. vapor, x = 100%
2	*	148.5	10	680	0.58	1.392	wet mixture, x = 85.1%
2'	**	154	10	723	0.58	1.47	wet mixture, x = 89.1%
3		100	10	5.6	0.20	0.1154	sat. liquid
3'		100	248	6.3	0.20	0.1154	subcooled liquid
4		288	248	230	0.20	0.445	subcooled liquid
5		330	248	300	0.134	0.523	sat. liquid
6		110	10	35.5	0.134	0.142	subcooled liquid

* Values assume turbine efficiency of 100%
** Values assume turbine efficiency of 80%

Assumed Conditions:

1 lb flow to the turbine throttle; m = 6.67 lbs

$w_{f_1} = w_{f_5}$ $\quad t_6 = t_3 + 10°$ $\quad t_3$ = min. of 100 °F

$w_{f_3} = w_{f_4}$ $\quad t_1 = t_5$ = max. of 330 °F

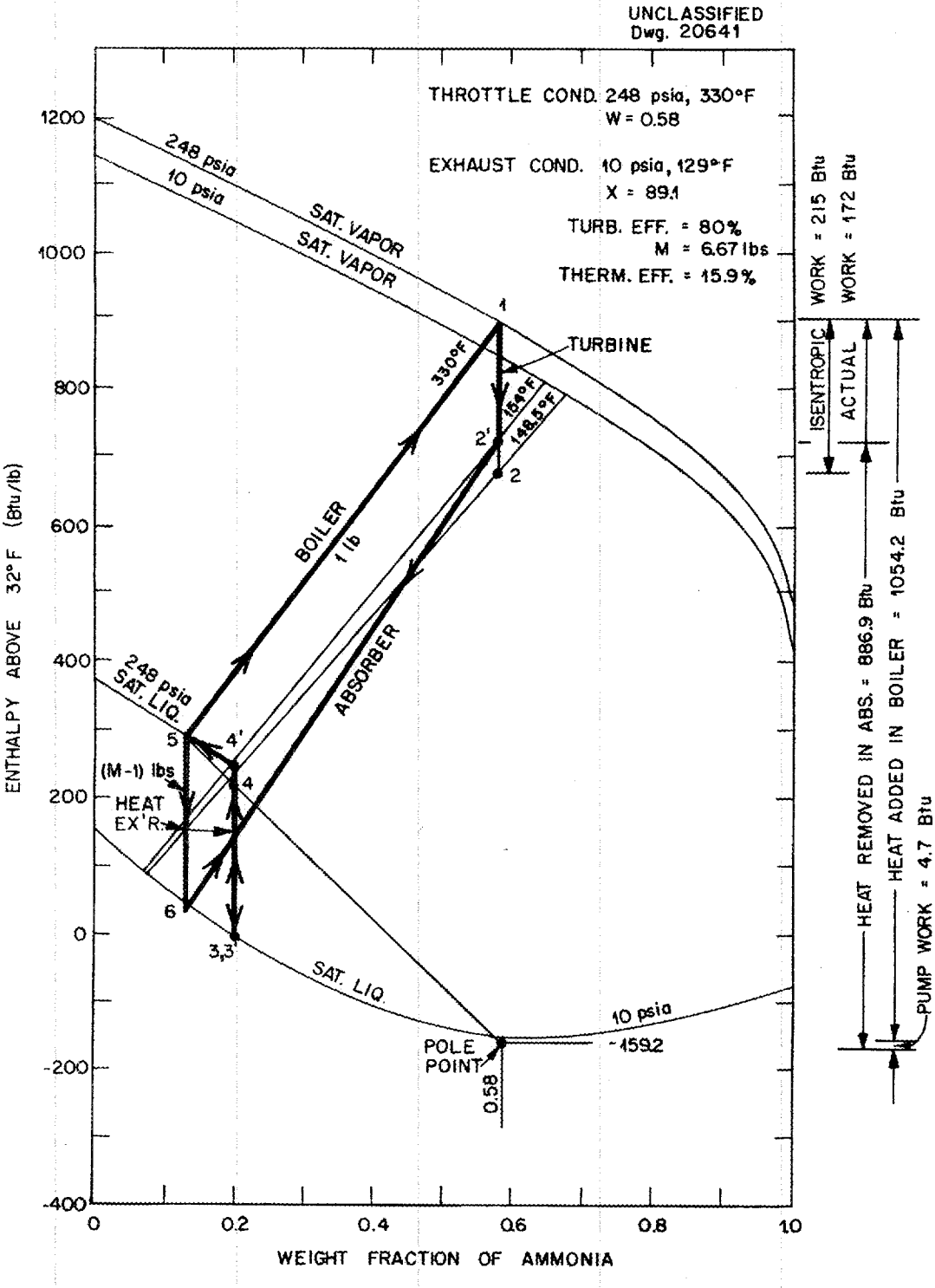

Figure 10. Enthalpy Concentration Diagram, Ammonia-Water Power Cycle (Case IV)

is at 30 °F higher temperature. It was found that the exhaust temperature is also higher, and in the case studied (0.20 concentration and 330 °F boiler temperature) with 80 percent turbine efficiency the thermal efficiency is 15.9 percent without the analyzer and 16.0 percent with it. Although the analyzer apparently does not increase the efficiency of the cycle, its use does simplify the cycle studies because the relationship between the saturated vapor and the absorber pressure is easily established.

Case V. When 30 °F of superheat is added to the 300 °F vapor leaving the analyzer, as shown in Figures 11 and 12, there is an accompanying increase in enthalpy to the throttle. However, the 100 °F minimum absorber temperature limits the expansion in the turbine (when turbine efficiency is 80 percent) rather than the moisture limitation, so that the expansion is only to 93.7 percent quality as compared with 89.4 percent without the application of superheat, and the thermal efficiency is only 15.5 percent with superheat as compared with 15.9 percent without it. The same situation wherein the use of superheat does not improve the cycle efficiency can also exist in the steam system, as shown in Items 1 and 3 of Table VI. In the situations where superheat is desirable in one cycle it would also probably be advantageous for the other one, so that it is concluded that use of superheat would not change the general findings of this study of ammonia-water heat-power systems.

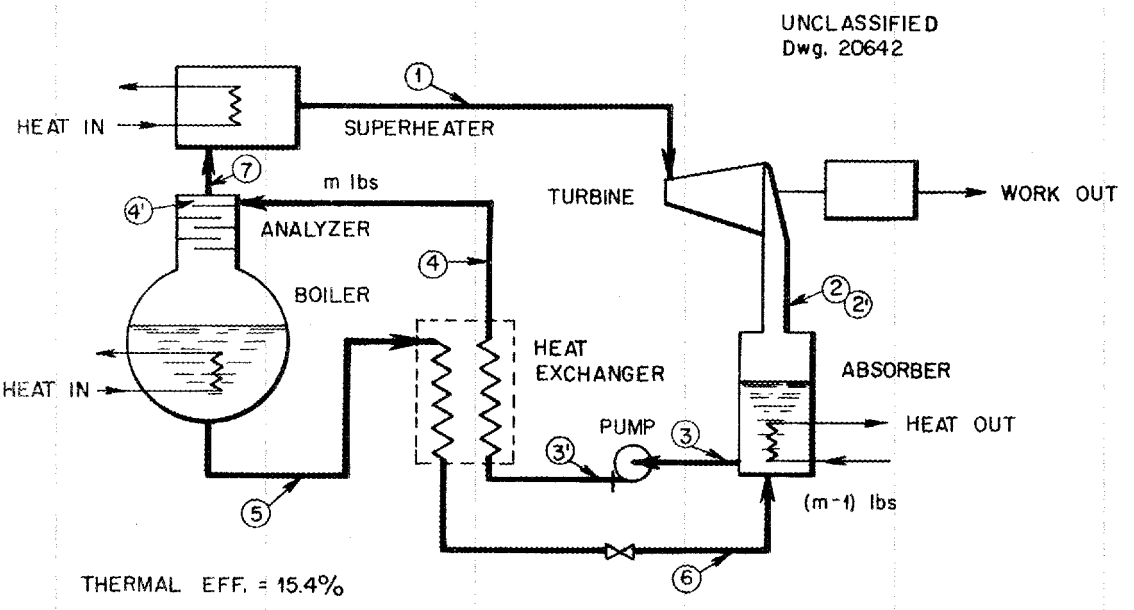

Figure 11. Ammonia-Water Power Cycle (With Superheater, Case V)

Table IV

Properties of State Points in An Ammonia-Water Power Cycle (with superheater)

State Point		t	p	h	w	s	Condition
1		330	248	834	0.748	1.3450	30 °F superheat
2	*	122.5	10	619	0.748	1.3450	wet mixture, x = 88.5%
2'	**	129	10	662	0.748	1.418	wet mixture, x = 93.7%
3		100	10	5.6	0.200	0.1154	sat. liquid
3'		100	248	6.3	0.200	0.1154	subcooled liquid
4		296	248	242	0.200	0.4665	subcooled liquid
4'		300	248	248.7	0.200	0.4743	sat. liquid
5		330	248	300	0.134	0.5228	sat. liquid
6		110	10	35.5	0.134	0.1418	subcooled liquid
7		300	248	799	0.748	1.313	sat. vapor

* Values assume turbine efficiency of 100%
** Values assume turbine efficiency of 80%

Assumed Conditions:

1 lb flow to the turbine throttle; m = 9.3 lbs

$w_{f_3} = w_{f_1} = w_{f_7} = w_{f_4}$ $\qquad t_3$ = min. of 100 °F

$t_6 = t_3 + 10°$ $\qquad t_1$ = max. of 330 °F

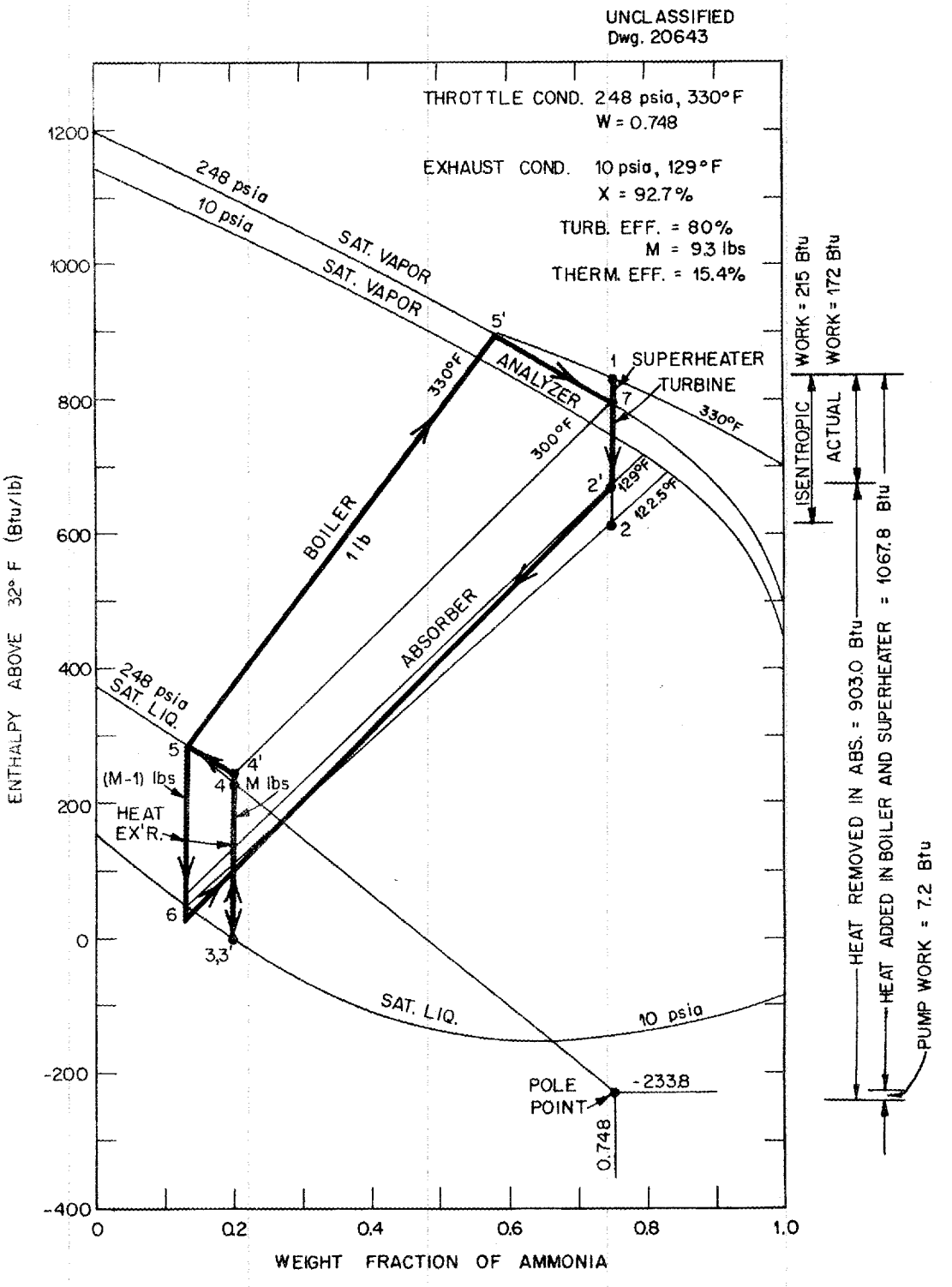

Figure 12. Enthalpy-Concentration Diagram, Ammonia-Water Power Cycle (Case V)

VII. COMPARATIVE STEAM CYCLES

The significance of the efficiencies obtained from the ammonia-water cycle studies is best analyzed by comparison with an equivalent steam cycle operating between the same temperature limits and with the same limiting conditions for the turbine exhaust, turbine efficiencies, etc.

Table VI summarizes the nineteen different steam cycle conditions computed for purposes of comparison. Figure 13 shows a typical saturated steam cycle with two open heaters, and Table V shows the properties at the various state points in the cycle together with an indication at the bottom of the table of the method used in calculating the thermal efficiency of the steam cycle.

Keenan and Key's Thermodynamic Properties of Steam[9] were used as a data source and values taken from the Mollier Diagram were considered sufficiently accurate. The total pump work required was assumed as 0.5 Btu/lb in all cases.

As in the ammonia-water power cycle studies, the limit to turbine expansion was taken as either 100 ^{o}F or 83.5 percent exhaust quality (for 100 percent turbine efficiency) or 88 percent (for 80 percent turbine efficiency), depending on whichever limiting condition is reached first. There are three exceptions, Items 9, 13 and 14, Table VI, which were computed as special cases.

Where feedwater heaters are employed in the steam cycle, simple open-type heaters were assumed. No allowances were made for pressure drops, heat losses, etc., and optimum extraction pressures were approximated by assuming that the temperature rise from condenser hotwell to boiler saturation temperature is equally divided among the heaters and the boiler. This principle is proven

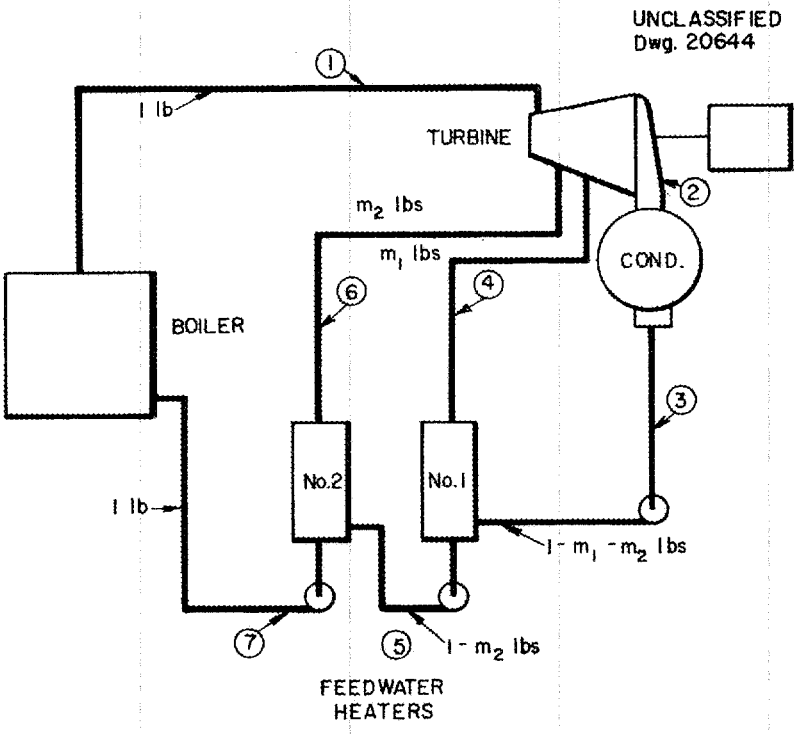

Figure 13. Steam Cycle Study (Case 8, Table VI)

TABLE V

Properties at State Points in Steam Cycle (Fig. 13)

Point	t, °F	psia	h, Btu/lb	s, Btu/lb-°F abs	Condition
1	330	103.1	1188	1.6002	sat. vapor
2	146	3.4	958	1.6002	wet mixture
3	146	3.4	115		sat. liquid
4	207	13.3	1038	1.6002	wet mixture
5	207	13.3	175		sat. liquid
6	268	40.5	1115	1.6002	wet mixture
7	268	40.5	237		sat. liquid

Calculations in Steam Cycle (Fig. 12)

Heater No.1: $m_2 = \frac{1237 - 175}{1115 - 175} = 0.0658$ lbs

Heater No.2: $m_1 = \frac{0.9342\ (175 - 115)}{1038 - 115} = 0.0612$ lbs

Useful work = (1188 - 1115) + 0.9342(1115 - 1038) + 0.873 (1038 - 958)
= 214.5 Btu/lb (less 0.5 pump work) = 214.0 Btu/lb

Heat Added to Cycle = 1188 - 237 = 951 Btu/lb

Net Thermal Eff of Cycle = 214/951 = 22.5%

in most thermodynamics texts, such as Keenan's Thermodynamics[10]. Saturated liquid was assumed to leave each heater. These assumptions are in line with the same assumptions used in the ammonia-water cycle studies.

TABLE VI

THERMAL EFFICIENCIES OF STEAM POWER CYCLES

Item No.	Boiler Outlet °F	Super-heat °F	Number Fd.Wat. Heaters	Assumed Turbine Eff., %	Turbine Exhaust		Net Thermal Eff. of Cycle %
					Quality	°F	
1	280	0	0	100	83.5	110	21.3
2	280	0	2	100	83.5	110	22.3
3	280	30	0	100	85.8	100	19.9
4	300	0	0	100	83.5	124	21.4
5	300	0	1	80	86.4	100	20.0
6	330	0	0	100	83.5	146	21.4
7	330	0	1	100	83.5	146	22.2
8	330	0	2	100	83.5	146	22.2
9	330	0	0	100	79.4	100	26.5
10	330	30	0	100	83.5	111	22.9
11	330	30	2	100	83.5	111	24.1
12	330	0	0	80	88.0	146	17.1
13	330	0	1	80	85.1	100	21.9
14	330	30	1	80	87.6	100	19.8
15	360	0	0	100	83.5	171	21.2
16	380	0	0	100	83.5	188	21.0
17	380	0	1	100	83.5	188	21.9
18	380	0	2	100	83.5	188	22.3
19	380	30	0	100	83.5	148	22.9

(Total pump work = 0.5 Btu/lb in all cases)

VIII. SECOND LAW ANALYSIS OF AMMONIA-WATER AND COMPARABLE STEAM CYCLES

It was stated previously that the ammonia-water power cycle appeared attractive in that the heat could be added in the boiler at a higher average temperature than in the boiler of a steam system, but the data presented in Tables II and VI indicate that the thermal efficiency of the ammonia-water cycle is less than that of a steam cycle. In order to explain more fully why this situation exists it is necessary to make a thermodynamic study based on a second law analysis. If the overall changes in availability and effectiveness are found for each piece of equipment in the cycle for both the ammonia-water and the steam systems, the thermodynamic effect that each has on the thermal efficiency can be evaluated. The particular processes in the cycle that cause one system to have an advantage over the other is then apparent.

Keenan[11] states, "Availability may be defined as the maximum work which can result from interaction of system and medium when only cyclic changes occur in external things except for the rise of a weight." If minor effects are neglected, available energy, or the availability function, may be defined as, $b = h - T_o s$. For the dead state, where p_o is 14.696 psia and t_o is 75 ^oF, this function is written, $b_o = h_o - T_o s_o$.

Maximum work would occur when all the available energy of the working substance is converted into work, the magnitude of which would be $(b - b_o)$. For any internal process the change in availability per pound of working fluid is, $\Delta b = \Delta h - T_o \Delta_s$. When the change in availability of the external surroundings is considered, the sum of these internal and external changes will be the overall change in availability, and is designated by I. The ideal value for I is zero, but in actual equipment it is always less than zero, reflecting the irreversibilities that are present. Effectiveness, E, for the cycle can be defined as

the ratio of the net useful work of the cycle to the overall loss in availability by the heat source. The effectiveness of a particular piece of equipment in the cycle is defined as the ratio of the overall gain in availability to the overall loss in availability.

Table VII summarizes the results of a study made to determine the changes in availability and effectiveness of each piece of equipment in the ammonia-water and steam power cycles. Since the table is based on a net output of 1 Kwh, the gross turbine outputs reflect the pump work required for the cycle. In the absorber of the ammonia-water system, it may be noted that there is approximately three times the loss in available energy than that which occurs in the condenser of a steam system. This can be explained by the fact that the turbine exhaust in the ammonia-water cycle is at a higher temperature than the minimum temperature for heat rejection. Since the pump efficiency has little effect on the overall thermal efficiency of the cycle, for simplicity it has been assumed 100 percent and therefore, the pumping process has zero change in availability and 100 percent effectiveness. It should be noted, however, that the high circulation rate through the pump of the ammonia-water system makes the pump work approximately 10 times greater than that required in the steam system. This higher circulation rate also affects the heat exchanger in the ammonia-water system in that even though the effectiveness of the exchanger is higher than for the feedwater heaters of a steam system, the overall loss of availability is greater than in the steam system. Also, a comparison of the amount of heat transferred in the ammonia-water exchanger and in the steam system feedwater heaters indicates a considerably larger physical size for the former even though several feedwater heaters are used. In the boiler the effectiveness was greater and the change in availability less for the ammonia-water system than for the steam system in

TABLE VII

Second Law Analysis of Ammonia-Water and Comparable Steam Cycles
1 kwh Net Output

	Q or W Btu		I Btu		E %	
	NH_3-H_2O	Steam	NH_3-H_2O	Steam	NH_3-H_2O	Steam
Case III and Item 5 Table VI						
Turbine	+ 3,540	+ 3,415	- 802	- 800	81.5	81.0
Absorber/condenser	- 18,020	- 13,560	- 1,550	- 594	12.1	22.3
Pump	- 135	- 3.3	0	0	100.0	100.0
Heat exchanger	± 45,000	± 1,490	- 1,186	- 115	88.6	59.4
Boiler-analyzer	∓ 21,450	∓ 17,000	- 419	- 884	94.3	85.4
Cycle			- 3,957	- 2,393	45.9	59.1
Case IV and Item 13 Table VI						
Turbine	+ 3,510	+ 3,420	- 798	- 855	81.6	81.0
Absorber/condenser	- 18,100	- 12,200	- 1,645	- 555	11.2	22.2
Pump	- 96	6.2	0	0	100.0	100.0
Heat exchanger	± 30,500	± 1,630	- 1,143	- 143	83.3	58.2
Boiler	∓ 21,500	∓ 15,610	- 121	- 506	98.4	90.7
Cycle			- 3,707	- 2,059	48.7	63.6
Case V and Item 14 Table VI						
Turbine	+ 3,540	+ 3,415	- 806	- 795	81.6	81.0
Absorber/condenser	- 18,540	- 13,700	- 1,850	- 652	9.1	21.6
Pump	- 134	3.3	0	0	100.0	100.0
Heat Exchanger	± 44,700	± 1,770	- 1,170	- 150	88.6	59.4
Boiler-Anal.-Super.	∓ 22,000	∓ 17,200	- 239	- 816	96.8	86.0
Cycle			- 4,065	- 2,413	45.8	58.6

every case. This is explained by the fact that the heat is added at a higher average temperature. Although the initial premise is correct in that the effectiveness of the boiler in the ammonia-water cycle is greater, the values in Table VII show that the losses in availability in the other components of the cycle offset this gain to the extent that the steam cycle, when considered on an overall basis, has 30 percent more effectiveness and has only about two-thirds the loss in available energy.

It can be concluded that the ammonia-water system has no significant thermodynamic advantage over a steam system as a means of converting heat into useful work.

Sample Calculations - Case III (Second law analysis)

Turbine:

Change in internal availability,

$$\Delta b_i = \Delta h - T_o \Delta s,$$

but, $\Delta h = -W = -172$ Btu/lb,

$$T_o \Delta s = T_o (s_2' - s_1),$$

$$T_o \Delta s = 535(1.385 - 1.313) = 38.5 \text{ Btu/lb},$$

then, $\Delta b_i = -172 - 38.5 = -208.5$ Btu/lb.

Change in external availability,

$$\Delta b_e = W = 172 \text{ Btu/lb}$$

Effectiveness is then,

$$E = \frac{\Delta b_e}{\Delta b_i} = \frac{172}{-208.5} = 81.5\ \%$$

Overall change in availability is then,

$$I = \Delta b_i + \Delta b_e = -208.5 + 172 = -38.5 \text{ Btu/lb}.$$

Absorber:

Change in internal availability,

$$\Delta b_i = \Delta H - T_o \Delta S,$$

but, $\Delta H = Q_{out} = -868$ Btu,

and, $T_o \Delta S = T_o \left[(m - 1)(s_3 - s_6) + (s_3 - s_2') \right],$

$$T_o \Delta S = 535 \left[8.3(0.115 - 0.142) + (0.115 - 1.385) \right],$$

$$T_o \Delta S = -795,$$

then, $\Delta b_i = -868 + 795 = -72.9$ Btu

Change in external availability, assuming cooling water in at 75 °F, and out at 85 °F,

$$\Delta b_e = \Delta H - T_o \Delta S,$$

again, $\Delta H = Q = 868$ Btu,

and, $T_o \Delta S = m T_o (s_{out} - s_{in})$,

where $m = \frac{\Delta H}{\Delta h}$ and Δh = 10 Btu/lb cooling water,

m = 86.8 lbs,

then, $\Delta b_e = 86.8(535)(0.1024 - 0.0839)$,

Δb_e = 8.8 Btu

As this energy is not generally recovered it should not be credited to the cycle.

Effectiveness is then,

$$E = \frac{\Delta b_e}{\Delta b_i} ,$$

$$E = \frac{8.8}{-72.9} = 12.1\ \%$$

Overall change in availability is,

$$I = \Delta b_i ,$$

$$I = -72.9 \text{ Btu}$$

Heat Exchanger:

Change in availability of "m" stream,

$$\Delta b_m = m\left[(h_4 - h_3) - T_o(s_4 - s_3)\right] ,$$

where m is 9.3 lbs as in sample calculation of Case II.

then, $\Delta b_m = 9.3\left[(242 - 6) - 535(0.467 - 0.115)\right]$,

Δb_m = 439 Btu

Change in availability of "(m - 1)" stream,

$$\Delta b_{m-1} = (m - 1)\left[(h_6 - h_5) - T_o(s_6 - s_5)\right] ,$$

$$\Delta b_{m-1} = 8.3\left[(35 - 300) - 535(0.142 - 9.523)\right] ,$$

Δb_{m-1} = -496 Btu.

Effectiveness,

$$E = \frac{\Delta b_m}{\Delta b_{m-1}} = \frac{439}{-496} ,$$

$$E = 88.6\ \%$$

Overall change in availability,

$$I = \Delta b_m + \Delta b_{m-1},$$

$$I = 439 - 496 = -57 \text{ Btu}$$

Boiler and Analyzer:

Change in internal availability,

$$\Delta b_i = \Delta H - T_o \Delta S,$$

where, $\Delta H = Q_{in} = +1032.8$ Btu,

and, $T_o \Delta S = T_o \left[(m - 1)(s_5 - s_4) + (s_1 - s_4) \right]$,

$$T_o \Delta S = 535 \left[(8.3)(0.523 - 0.467) + (1313 - 0.467) \right],$$

$$T_o \Delta S = 701 \text{ Btu},$$

then, $\Delta b_i = 1032.8 - 701 = 332$ Btu

Change in external availability, (assuming heating steam saturated at 350°F),

$$\Delta b_e = -Q_{in} - mT_o(s_{out} - s_{in}),$$

where, $m = \frac{-Q_{in}}{\Delta h(\text{steam})} = \frac{1032.8}{870.8}$,

$$m = 1.18 \text{ lb. steam/lb vapor to turbine},$$

then, $mT_o \Delta s = 1.18(535)(1.0754) = 680$ Btu,

$$\Delta b_e = -1032.8 + 680 = -353 \text{ Btu}$$

Effectiveness,

$$E = \frac{\Delta b_i}{\Delta b_e} = \frac{332}{-353},$$

$$E = 94\ \%$$

Overall change in availability,

$$I = \Delta b_i + \Delta b_e = 332 - 353,$$

$$I = -21 \text{ Btu}$$

Cycle Effectiveness,

$$E = W/\Delta b_e(\text{boiler}) = 164.8/-353,$$

$$E = 46.4\ ^{o}/_{o}$$

In order to correlate the above results with the quantities reported in Table VII, it is necessary to multiply them by 20.8. This conversion factor is the required mass flow rate to the turbine necessary to obtain a net output of 1 kwh.

IX. RECOMMENDATIONS FOR FURTHER STUDY

The writers believe that the ammonia-water power cycle shows so little promise that further study of this working fluid is not warranted.

A two-substance working fluid wherein water is the more volatile component, such as a lithium chloride-water combination, possibly could have better thermal efficiencies than that of a comparable conventional steam cycle.

The properties of enthalpy and entropy are essential when making a heat-power cycle study. If little-known substances are used as working fluids a great deal of preliminary effort will be required to determine sufficient accurate property data.

X. APPENDIX

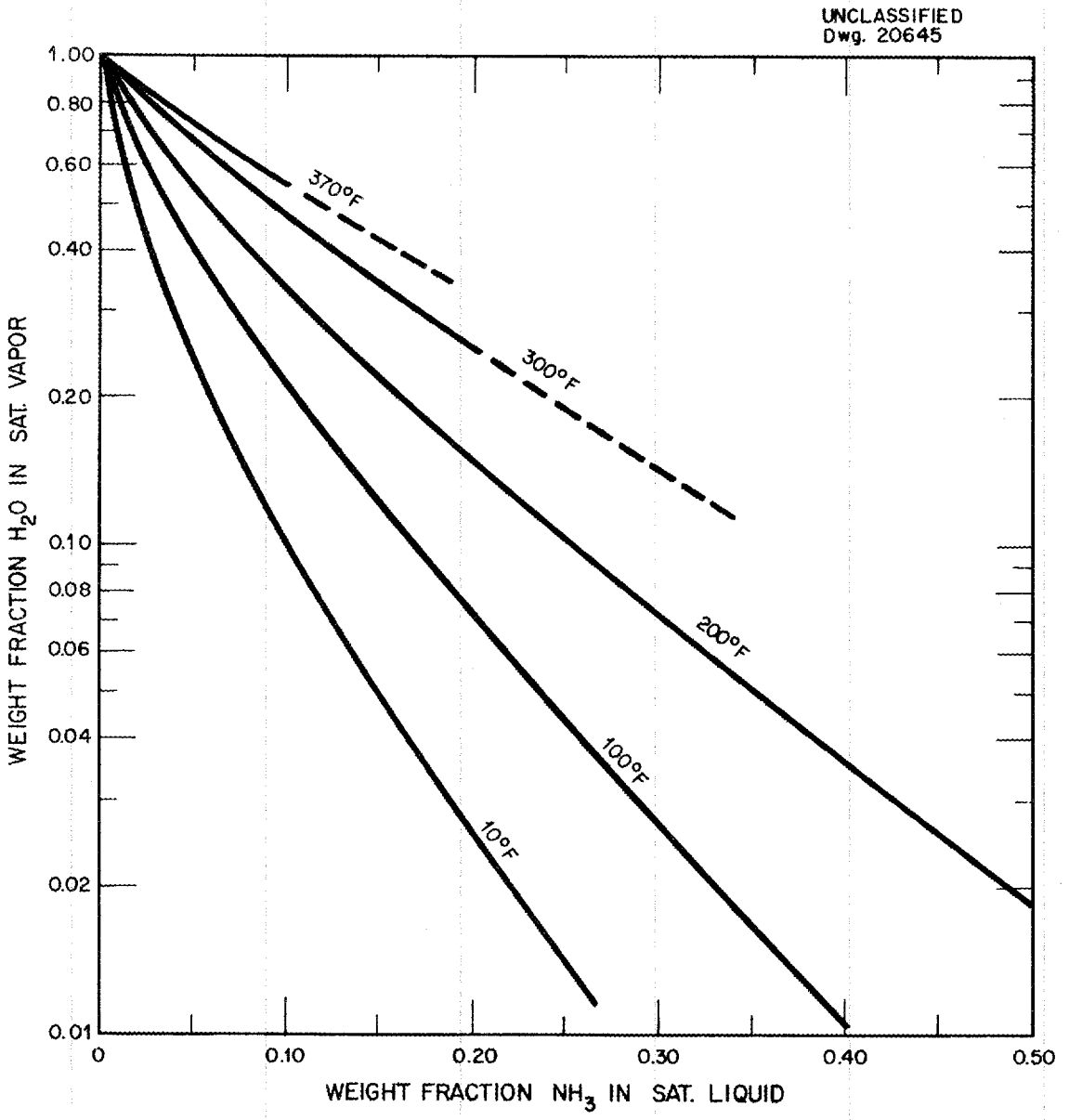

Figure 14. Concentration NH_3 in Saturated Vapor vs. Concentration in Saturated Liquid at Various Temperatures

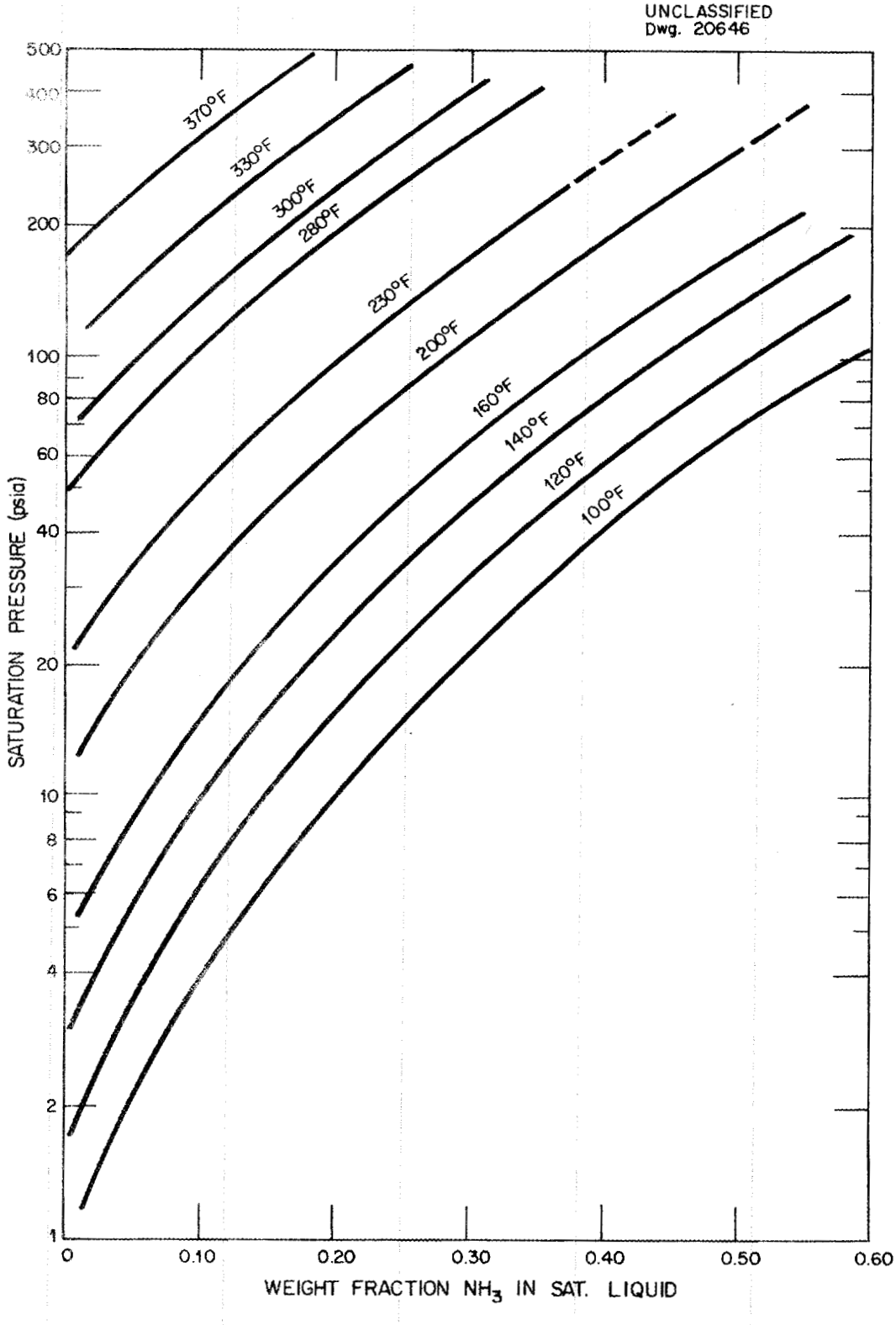

Figure 15. Saturated Pressure of NH_3-H_2O Solutions at Various Temperatures and Concentrations

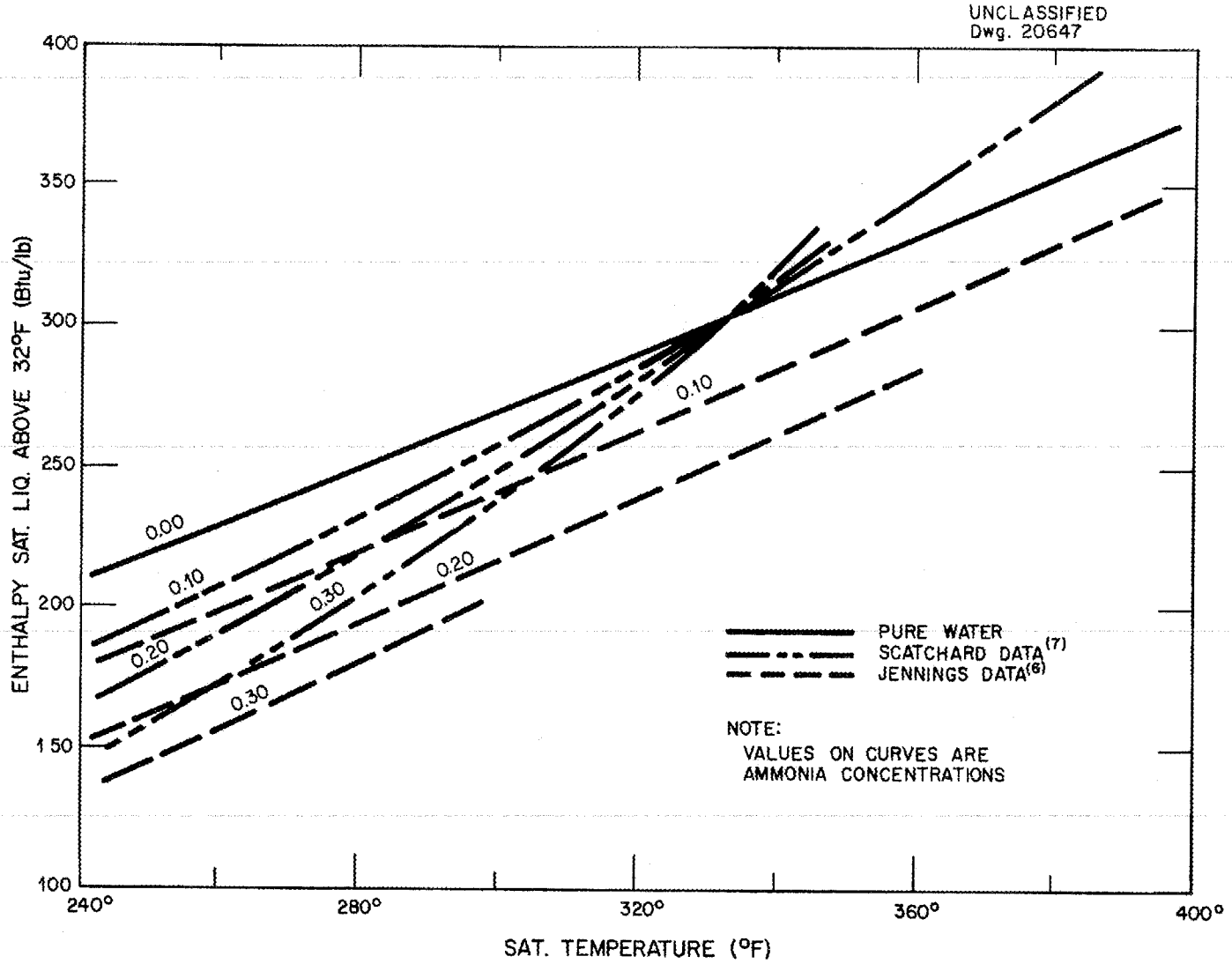

Figure 16. Enthalpy Saturated Liquid vs. Temperature and Concentration

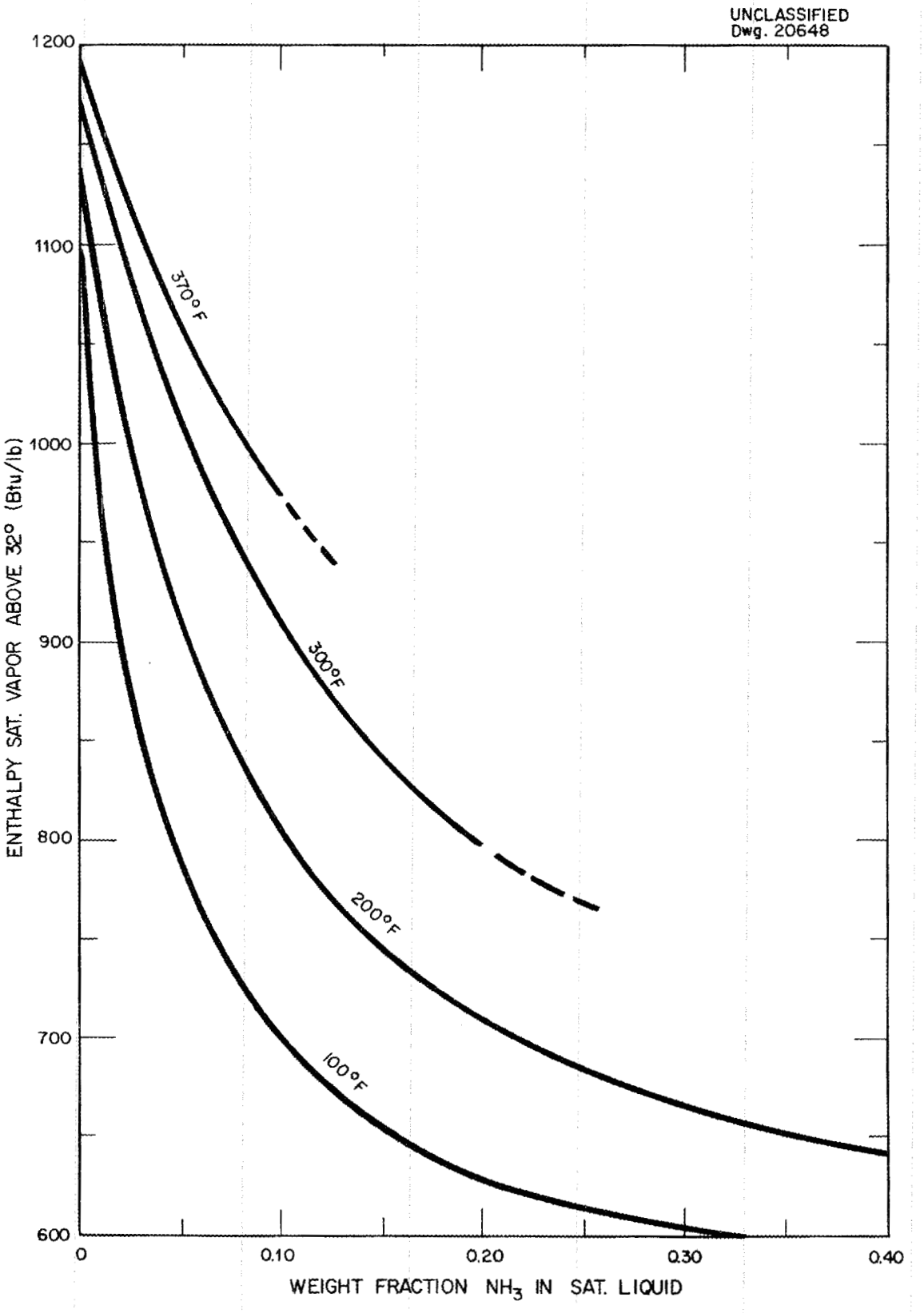

Figure 17. Enthalpy Saturated Vapor Scatchard Data(7)

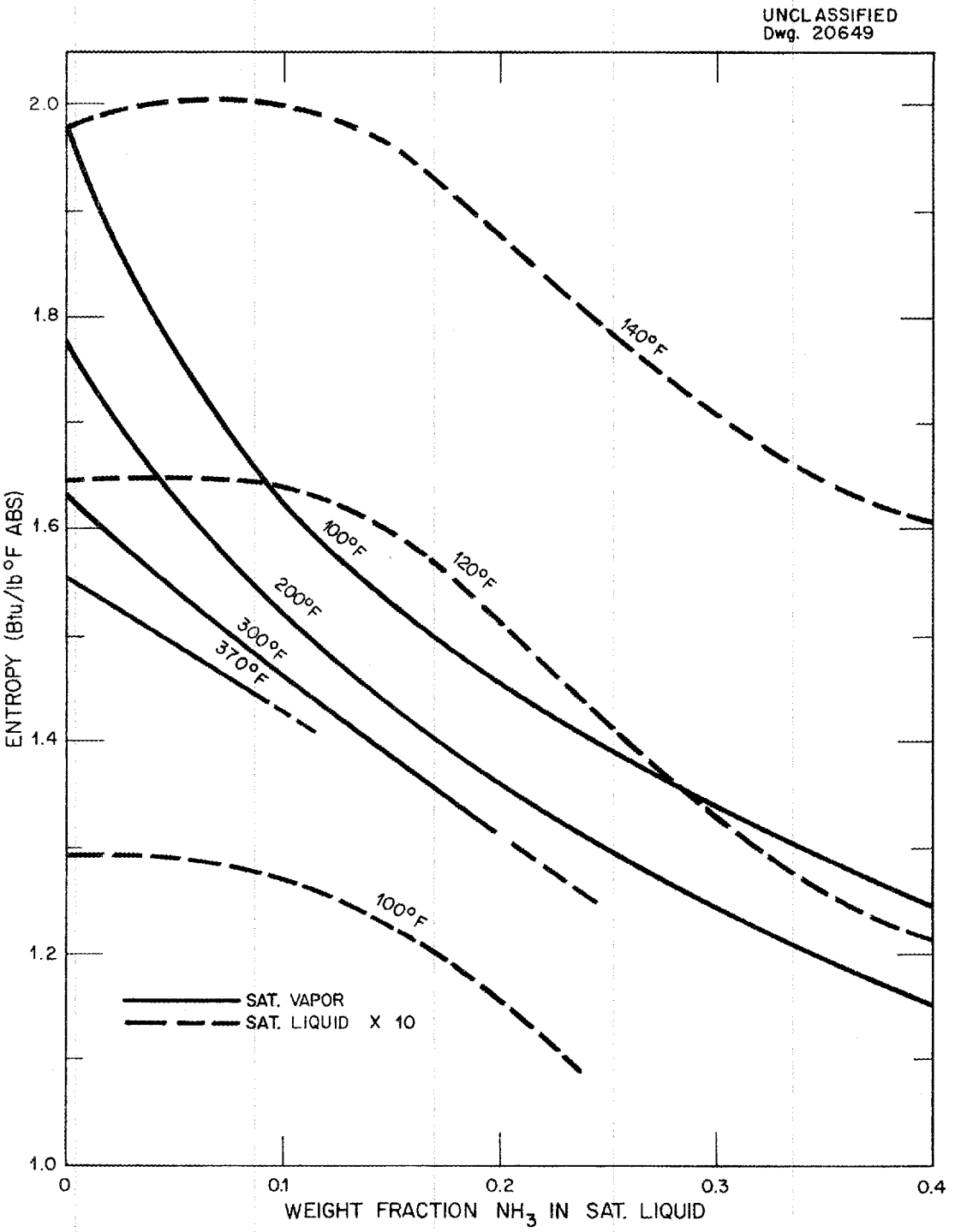

Figure 18. Entropy Saturated Vapor and Liquid

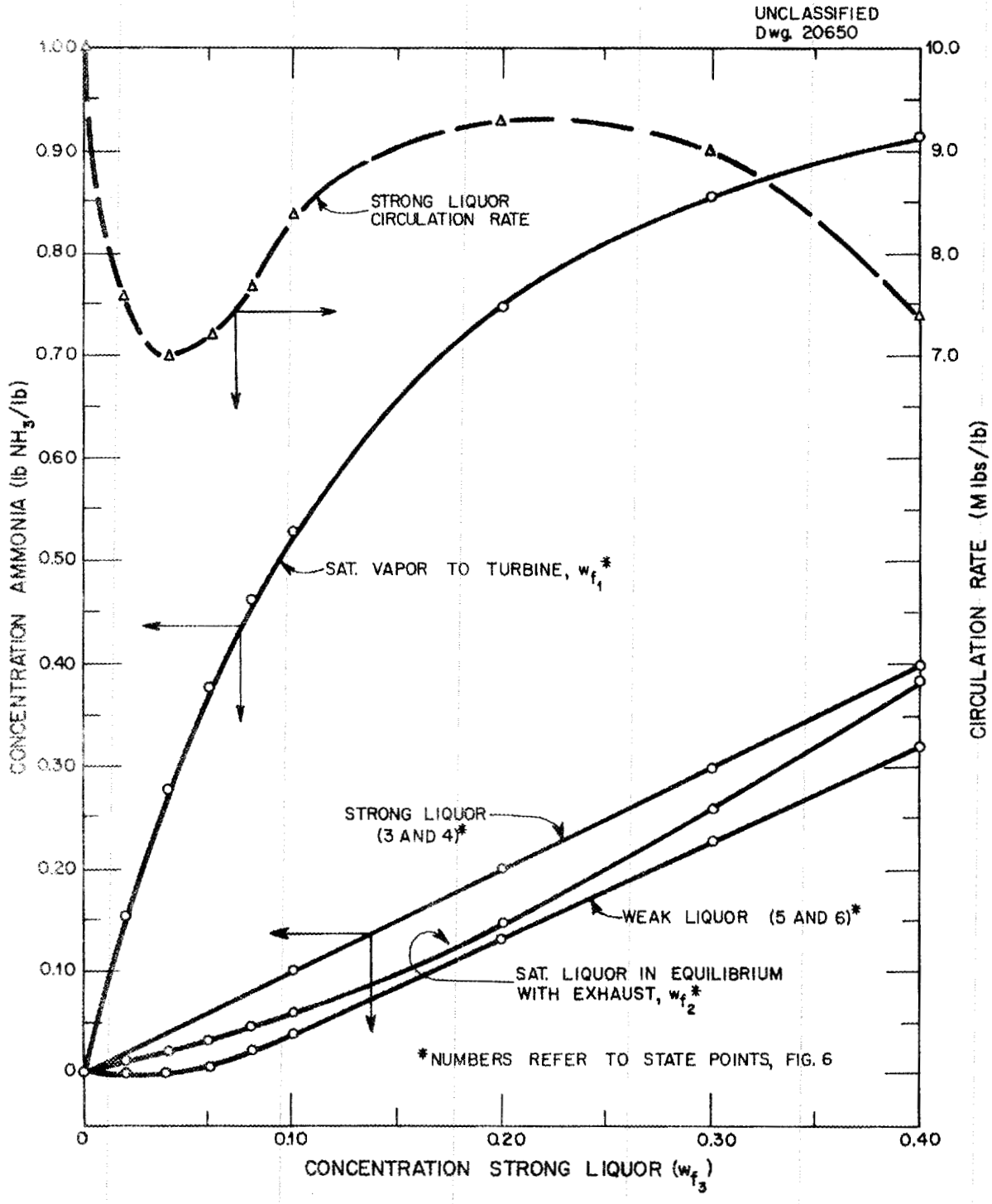

Figure 19. Ammonia Concentrations at State Points Case II, Items 4-12, Table IV

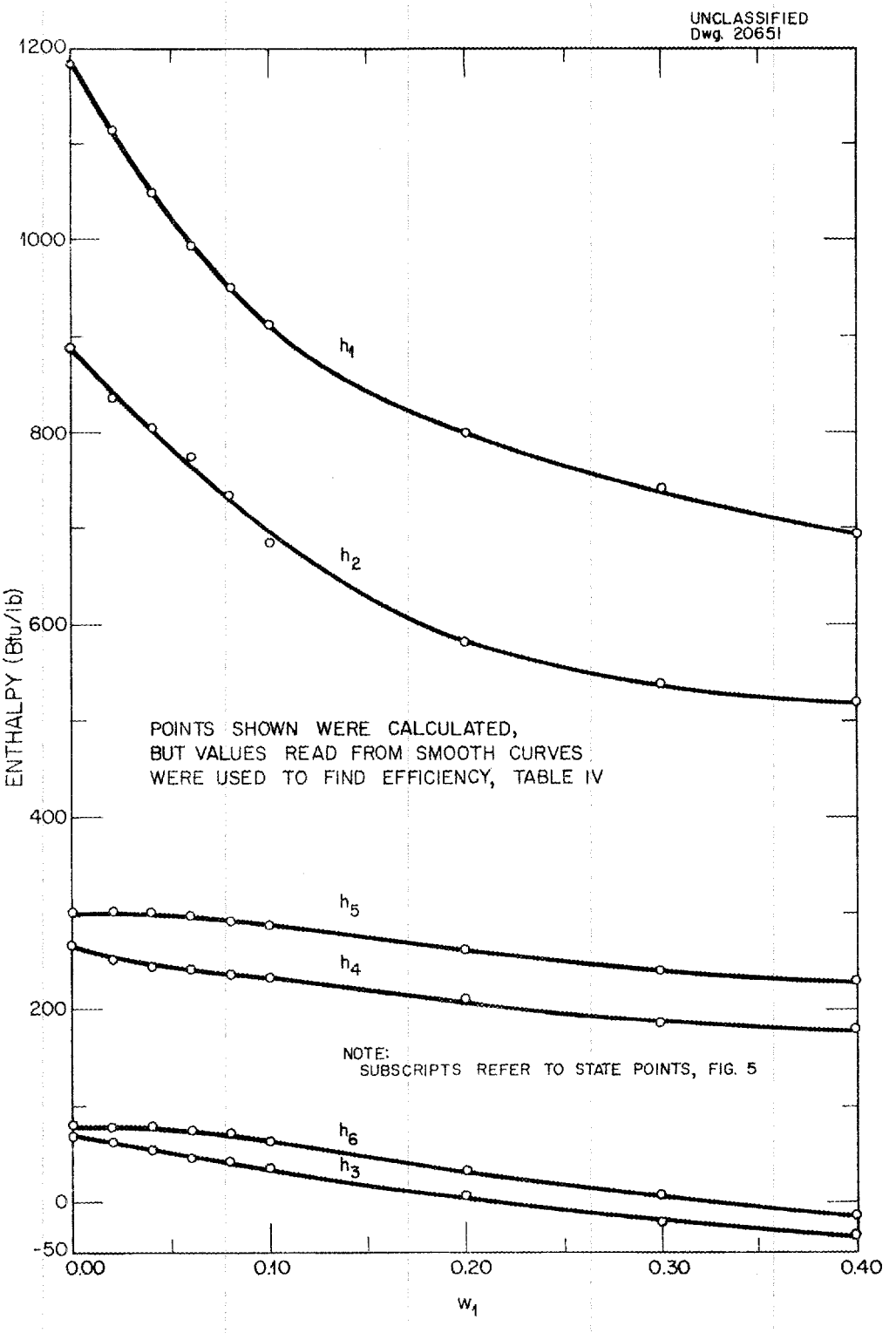

Figure 20. Ammonia-Water Power Cycles Enthalpies vs. w_1, Case II, Items 4-12, Table IV

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11. Keenan, ibid, p. 290

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