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ANALYTICAL INVESTIGATION OF CYCLE CHARACTERISTICS FOR ADVANCED TURBOELECTRIC SPACE POWER SYSTEMS

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SUMMARY

An investigation was made of the relative influence of turbine inlet temperature, radiator temperature, and turbine efficiency on radiator area for Rankine cycles with rubidium, potassium, and sodium as working fluids. It was determined that, whereas turbine inlet temperature and turbine efficiency have gross effects on radiator size, for a given inlet temperature a considerable latitude in the selection of radiator temperature may be accepted with only minor effects on required radiator size.

Also investigated was the influence on turbine efficiency and design of the factors that distinguish alkali-metal vapor turbines from conventional gas turbines. The turbine configuration was determined to be a function of the involved working fluids and rotor blade speed. For a given blade speed, the number of stages required for high turbine efficiency was found to vary directly with turbine specific work output, and therefore to vary in the ratio 5 to 2.5 to 1 for sodium, potassium, and rubidium, respectively. Lower blade speeds than employed in conventional gas turbines may be required to satisfy critical stress considerations resulting from the elevated temperatures involved and the criterion of long-duration reliability. This will increase the number of turbine stages necessary to obtain high turbine efficiency and consequently increase turbine weight.

The question of moisture formation was discussed and a calculation was made to indicate the nature of the aerodynamic losses due to moisture content. Various means of reducing moisture content were considered, including mechanical removal, increased radiator temperature, inefficient expansion, superheat, and reheat. Sample calculations were made in most cases to indicate their comparative effectiveness and resultant penalty in required radiator area.

INTRODUCTION

Currently, there is considerable interest in turboelectric power systems for satellite and space-vehicle use involving long periods of continuous operation. Such systems reject waste heat to a space environment by radiation from a surface that may be in the form of the vehicular skin or a separate distinct radiator. For low power applications of less than 100 kilowatts, the radiator has generally not been a sufficiently large fraction of the total powerplant weight to warrant close optimization. Maximum temperatures for these systems have been less than 1400° F owing to considerations of reliability and the use of existing technology. Furthermore, because of the small sizes involved, component efficiencies are generally low. As a consequence, low-power systems have been characterized by comparatively high radiator specific weights (10 to 30 lb/kw electrical power output). Two examples of such systems are the 3-kilowatt SNAP 2 and the 30-kilowatt SNAP 8 systems currently under development.

As power level is increased and power requirements for electric propulsion are considered, mission analyses indicate that it is necessary that powerplant specific weight be markedly reduced. Since the radiator constitutes a relatively greater percentage of the total weight at high power levels, it is especially important to reduce radiator specific weights. In general, radiator weight will depend not only on the required surface area but also on such specific factors as geometric configuration, meteoroid penetration protection, stresses, type of material, supporting structure, piping, and manifolding. Reliable estimates of radiator weight are therefore difficult to make; however, the major factor of required surface area is readily amenable to analysis.

A number of analyses have been made (e.g., refs. 1 and 2) to investigate the effect of thermodynamic cycle on radiator area. The general conclusions reached from these analyses are that, for reduced radiator area, (1) turbine inlet temperature and turbine efficiency should be as high as practical, (2) the radiator temperature should be approximately 3/4 of the turbine inlet temperature, and (3) liquid-metal working fluids operating in a Rankine cycle appear most attractive. Desirable working fluids generally considered are, in order of increasing temperature level, mercury, rubidium, potassium, and sodium. An example of an advanced nuclear turbogenerating system capable of powering a manned interplanetary vehicle is given in reference 2.

This analysis will investigate further the relative effect of each of the cycle factors on radiator area for the design of advanced power systems. Rubidium, potassium, and sodium were considered as working fluids with maximum temperatures arbitrarily selected as 2100°, 2300°, and 2500° R, respectively, for considering the turbines associated with these fluids. The relative effects of turbine inlet temperature, radiator temperature, turbine efficiency, and working fluid on radiator size were determined.

In view of the importance of turbine efficiency and reliability, further study was made of some of the design characteristics and problems associated with vapor turbines for advanced systems. Some of the turbine factors considered were specific work output, speed-work parameter, number of stages, and moisture formation.

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SYMBOLS

A	area, sq ft
g	gravitational constant, 32.2 ft/sec^2
ΔH	turbine specific work output, Btu/lb
h	enthalpy, Btu/lb
J	mechanical equivalent of heat, 778.2 ft-lb/Btu
m	moisture content, lb of moisture/lb of mixture
n	number of turbine stages
P	power, kw
q_r	heat rejected from cycle, Btu/lb
q_s	heat supplied to cycle, Btu/lb
S	entropy, Btu/(lb)($^{\circ}$R)
T	absolute temperature, $^{\circ}$R
U_m	turbine mean rotor blade speed, ft/sec
w	weight-flow rate, lb/sec
ϵ	emissivity
η	efficiency
λ	turbine speed-work parameter, $U_m^2/gJ\Delta H$
σ	Stefan-Boltzmann constant, 0.173×10^{-8} Btu/(sq ft)(hr)($^{\circ}$R^4)

Subscripts:

aer	aerodynamic performance, excluding moisture
c	cycle

G	generator
id	ideal
R	radiator
ST	stage
T	turbine
vap	vaporization
1	start of heat-addition process, also pump exit
2	end of heat-addition process, also turbine inlet
3	start of heat-rejection process, also turbine exit
4	end of heat-rejection process, also pump inlet

CYCLE ANALYSIS

Fundamental Considerations

The equations necessary to determine the effect of the factors influencing radiator size will be developed herein. The factors considered will be turbine inlet temperature, radiator temperature, turbine efficiency, and working fluid. For simplicity, the Carnot cycle will be used as a model to determine the effect of temperature level and turbine efficiency. This will be shown to be applicable when the properties of rubidium, potassium, and sodium are considered in the Rankine cycle.

Radiator area. - The assumptions used to derive the expression for radiator area are:

(1) The waste heat wq_r is rejected from the working fluid at constant temperature T_3.

(2) The entire outer surface of the radiator is isothermal and equal in magnitude to T_3.

(3) The effect of environmental sink temperature is negligible (ref. 3).

The expression for the rate of heat removal using these assumptions is obtained from the Stefan-Boltzmann equation as

$$wq_r = \frac{A_R \sigma \epsilon T_3^4}{3600} \text{ (Btu/sec)} \tag{1}$$

Substituting the value for σ into equation (1) and solving for radiator area result in

$$A_R = \frac{2.081 \; wq_r \times 10^{12}}{\epsilon T_3^4} \text{ (sq ft)} \tag{2}$$

Equation (2) is valid for all of the cycles considered in this analysis within the limitations of the assumptions that have been made. For practical radiators (e.g., fin and tube arrangement), the total radiator area is given by equation (2) divided by the fin effectiveness of the configuration (ref. 3).

Generator power output. - The power developed by the generator is determined by the net power developed by the cycle and the generator efficiency. In equation form,

$$P_G = \eta_G P_{net} \quad \text{(kw)} \tag{3}$$

The net cyclic power is determined by the thermal power supplied to the cycle and the cycle efficiency:

$$P_{net} = 1.0556 \; \eta_c wq_s \quad \text{(kw)} \tag{4}$$

or

$$P_{net} = 1.0556 \; (wq_s - wq_r) \quad \text{(kw)} \tag{5}$$

Substituting equations (4) and (5) into equation (3) results in an expression for generator power output valid for all power cycles:

$$P_G = \frac{1.0556 \; wq_r \eta_G \eta_c}{(1 - \eta_c)} \quad \text{(kw)} \tag{6}$$

Specific radiator area. - Specific radiator area A_R/P_G is defined as the number of square feet of radiator surface required per kilowatt of electrical power output. The general expression valid for the cycles of interest is determined by dividing equation (2) by equation (6):

$$\frac{A_R}{P_G} = \frac{1.971\times10^{12}}{\epsilon\eta_G T_3^4}\left(\frac{1}{\eta_c} - 1\right) \quad \text{(sq ft/kw)} \tag{7}$$

Because the generator efficiency and surface emissivity are not factors in the thermodynamic cycle, constant values of 0.95 and 0.90 were assigned to η_G and ϵ, respectively, to give

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$$\frac{A_R}{P_G} = \frac{2.306}{(T_3/1000)^4}\left(\frac{1}{\eta_c} - 1\right) \quad \text{(sq ft/kw)} \tag{8}$$

Radiator area is therefore favored by high values of both radiator temperature T_3 and cycle efficiency η_c. However, high values of η_c are characterized by low values of T_3. Therefore, for a given turbine inlet temperature, there is an optimum value of radiator temperature that will minimize radiator area.

Because all three cycles considered have isothermal heat rejection at temperature T_3, inspection of equation (8) reveals that the only difference between analyzing the cycles is the manner in which cycle efficiency η_c is determined for the various cycles.

Carnot Cycle

The Carnot cycle is used as a simple means of showing the effect of temperature level on radiator area. A temperature-entropy diagram (hereinafter referred to as a T-S diagram) for the Carnot cycle is shown in figure 1(a). The cycle efficiency of the Carnot cycle is given by

$$\eta_c = 1 - \frac{T_3}{T_2} \tag{9}$$

Substituting into equation (8) results in the expression for specific radiator area valid for a Carnot cycle:

$$\frac{A_R}{P_G} = \frac{2306}{(T_3/1000)^3}\left(\frac{1}{T_2 - T_3}\right) \quad \text{(sq ft/kw)} \tag{10}$$

Specific radiator area is therefore a function only of turbine inlet and radiator temperature. Differentiating equation (10) and setting the result equal to zero yield a temperature ratio T_3/T_2 of 0.75 for minimum radiator area for the Carnot cycle.

A range of turbine inlet temperatures was selected from 1700° to 2500° R in increments of 200°. For each value selected, a range of radiator temperatures was chosen that bracketed the optimum temperature for each case. Equation (10) was then solved for specific radiator area, and the results are presented in figure 2. Also shown on the figure as a dashed line is the optimum 3/4-temperature-ratio condition referred to previously.

The results of figure 2 clearly indicate the effect of increased turbine inlet temperature in reducing radiator area. For the range of temperatures considered, a 200° increase in turbine inlet temperature represents, on the average, a 30-percent reduction in radiator area at optimum radiator temperature. Figure 2 also shows that a considerable variation from the optimum radiator temperature may be accepted without a gross penalty in radiator area. For example, an approximate 400° spread in radiator temperature bracketing the optimum value represents only a 10-percent penalty in specific radiator area.

It is recognized, of course, that increased temperatures may impose serious design problems. For example, increased maximum temperature may incur serious corrosion or require the use of high-density refractory materials that are difficult to fabricate. Furthermore, the allowable design stress of the turbine rotor blades will be reduced, since the stress-to-rupture strength of rotor blade material is drastically reduced at these temperatures when the system must operate continuously for months or years.

Modified Carnot Cycle

The modified Carnot cycle is used to show the effect of turbine efficiency on radiator area. A T-S diagram of a modified Carnot cycle is shown in figure 1(b). The turbine expansion process is shown as a dashed line because the area under an irreversible process line on a T-S diagram has no meaning.

The cycle efficiency of a modified Carnot cycle with turbine losses is given by

$$\eta_c = \eta_T \left(1 - \frac{T_3}{T_2}\right) \tag{11}$$

and differs from that of a Carnot cycle only by turbine efficiency. Substituting into equation (8) results in the expression for specific radiator area valid for the modified Carnot cycle:

$$\frac{A_R}{P_G} = \frac{2.306}{(T_3/1000)^4}\left[\frac{T_2}{\eta_T(T_2 - T_3)} - 1\right] \quad \text{(sq ft/kw)} \tag{12}$$

Turbine inlet temperatures T_2 of 1900° and 2500° R were selected to study the relative effect of turbine efficiency and radiator temperature. Turbine-efficiency values of 0.20, 0.40, 0.60, 0.80, and 1.00 were selected, and radiator temperatures from 1000° to 2200° R were used. Equation (12) was then solved for specific radiator area, and the results are presented in figure 3.

It is noted from the figure that, for a given turbine inlet temperature, radiator area is more sensitive to turbine efficiency than it is to radiator temperature. A 10-percent difference (percent of local value) in turbine efficiency is equivalent to an approximate 400° spread bracketing the optimum radiator temperature for a turbine inlet temperature of 1900° R, and an approximate 500° spread for a turbine inlet temperature of 2500° R. This 10-percent drop in efficiency represents about a 15-percent increase in radiator area. This results from the fact that a drop in turbine efficiency has a compounding effect on required radiator size. First, more heat per pound of working fluid flowing must be rejected because less work is extracted in the turbine, which results in a larger radiator for the same flow rate (q_r in eq. (2)). Secondly, the flow rate must be increased to meet the same total power output requirement and hence a further increase in required radiator size (w in eq. (2)). When a drop in generator efficiency is considered, the first effect discussed is not present, although the second is. This means that specific radiator area is more sensitive to turbine efficiency than it is to generator efficiency.

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As turbine efficiency drops, there is a small shift in the optimum radiator temperature ratio from 0.75 at $\eta_T = 1.00$ (Carnot cycle) to 0.80 at $\eta_T = 0.20$. Neither the shift in optimum temperature ratio nor the increase in radiator area accompanying a decrease in turbine efficiency affects the wide latitude in radiator temperature that should be permissible.

It is apparent from the results presented that one of the most important requirements of power conversion systems of the type considered herein is that they include turbines that are highly efficient. Some of the factors that influence turbine efficiency are discussed in a later portion of this analysis.

Modified Rankine Cycle

Monatomic sodium, rubidium, and potassium were selected as representative high-temperature liquid metals to determine the effect on radiator size of actual working fluids producing power in a modified Rankine cycle. A sketch of this cycle on a T-S diagram and enthalpy-entropy (h-S) diagram is shown in figure 4. State point 3_{id} represents the ideal turbine exhaust point following isentropic expansion. Also shown on figure 4 is a schematic of the major components required by this cycle along with station location and direction of flow. When the turbine inlet temperature is limited, radiator area will be minimum when saturated vapor enters the turbine at the limiting temperature. This is true because the average temperature of heat addition is maximum, and therefore cycle efficiency for a given radiator temperature is also maximum. Variations in heat-addition processes are considered in a later portion of this analysis.

The assumptions made concerning the modified Rankine cycle are:

(1) Required pump work $(h_1 - h_4)$ is negligible.

(2) All processes with the exception of the turbine expansion are reversible.

The assumption of negligible pump work specifies that the net cycle work output is equal to the turbine work output ΔH, which is determined from turbine efficiency and ideal turbine work as

$$\Delta H = \eta_T(h_2 - h_{3,id}) \quad \text{(Btu/lb)} \tag{13}$$

The heat supplied to the cycle is

$$q_s = h_2 - h_1 \quad \text{(Btu/lb)} \tag{14}$$

and cycle efficiency is found by dividing equation (13) by equation (14) to give

$$\eta_c = \frac{\eta_T(h_2 - h_{3,id})}{(h_2 - h_1)} \tag{15}$$

Substituting equation (15) into equation (8) then results in an expression for specific radiator area valid for a modified Rankine cycle:

$$\frac{A_R}{P_G} = \frac{2.306}{(T_3/1000)^4}\left[\frac{(h_2 - h_1)}{\eta_T(h_2 - h_{3,id})} - 1\right] \quad \text{(sq ft/kw)} \tag{16}$$

It is noted from figure 4 that the selected modified Rankine cycle model (1-2-3-4), with saturated vapor at the turbine inlet, differs from the modified Carnot cycle only by the shaded triangular area as shown. Therefore, the results of the modified Rankine cycle would be expected to approximate the results of the modified Carnot cycle if the ratio of the triangular area to total area in figure 4 is small. Again, using turbine inlet temperatures of 1900° and 2500° R, the same values of turbine efficiency (0.20, 0.40, 0.60, 0.80, and 1.00) were selected to show this comparison and determine the effect of working fluid. Sodium, rubidium, and potassium properties were taken from references 4 to 6, and equation (16) was solved for specific radiator area. The results are presented in figures 5(a) and (b) for turbine inlet temperatures of 1900° and 2500° R, respectively. Calculated points for Na, Rb, and K are shown as circles, squares, and diamonds, respectively. Also shown for comparison are the corresponding specific-area variations obtained for the modified Carnot cycle presented in figure 3.

For the range of conditions covered in figure 5, the specific-area variations for the three alkali vapors considered are about the same. These variations are also not too different from those given by the modified Carnot cycle. At minimum radiator area, the Rankine cycle values are up to about 7 percent greater than the Carnot values at 1900° R and up to 10 percent greater at 2500° R. These results indicate, as might be seen from their T-S diagrams, that the ratio of triangular area to total area in figure 4 is relatively small for the fluids considered. Small values of this ratio are characterized by low liquid specific heats and high latent heats of vaporization. For simplicity in preliminary cycle calculations, therefore, it may be sufficient to use the modified Carnot cycle for determining the effects of turbine efficiency, turbine inlet temperature, and radiator temperature on specific radiator area, providing the T-S diagrams of the working fluid considered are similar to those of the three vapors in figure 5.

TURBINE CONSIDERATIONS

The requirements to be met by turbines used in space turbogenerating systems are clear; they must have a high degree of reliability and they must be highly efficient. There are three factors that influence reliability and efficiency that distinguish this type of turbine from the usual gas turbine; namely, the elevated temperatures involved, the requirement of prolonged continuous operation, and the use of a condensing alkali metal as the turbine driving fluid. The remainder of the analysis will therefore consider some of the effects of these factors on turbine design and performance.

Aerodynamic Factors Affecting Turbine Efficiency

Speed-work parameter, λ. - In gas-turbine practice, turbine efficiency has been analytically related to the design requirements imposed on a turbine (ref. 7). The design variables selected for this correlation are turbine specific work output ΔH and mean rotor blade speed U_m. These variables are expressed in a dimensionless form and defined as the speed-work parameter λ:

$$\lambda \equiv \frac{U_m^2}{gJ\Delta H} \qquad (17)$$

Similarly, for multistage turbines, stage speed-work parameter is defined in terms of stage specific work output ΔH_{ST} as

$$\lambda_{ST} \equiv \frac{U_m^2}{gJ\Delta H_{ST}} \tag{18}$$

When the work is equally split between stages,

$$\Delta H_{ST} = \frac{\Delta H}{n} \tag{19}$$

where n is the number of turbine stages. Substitution into equation (18) results in

$$\lambda_{ST} = \frac{nU_m^2}{gJ\Delta H} \tag{20}$$

A detailed investigation of the relation between efficiency and speed-work parameter has been made (ref. 7), and the results correlated with experimental data from a large number of turbines. It was determined from these results that reasonably high efficiencies can be maintained for individual stage speed-work parameters λ_{ST} as low as 0.5. In this analysis, it is assumed that similar efficiency characteristics will be obtained for vapor turbines, and the value $\lambda_{ST} = 0.5$ can be used as a suitable criterion for high efficiency.

Turbine work as a function of working fluid. - Referring to figure 4, the net cycle work of the Rankine cycle (turbine work for this analysis) is equal to the area enclosed by path $1\text{-}2\text{-}3_{id}\text{-}4$. This area could also be represented as an equivalent rectangle equal to $(T_2 - T_3)$ times ΔS. For fluids having normal bell-shaped T-S diagrams such as figure 4, ΔS is nearly equal to the change in entropy in vaporization of the liquid. Thus, for comparable ΔT, turbine work outputs for different fluids are directly proportional to their particular change in entropy of vaporization. Further, because for vaporization,

$$\Delta S_{vap} = \frac{\Delta h_{vap}}{T}$$

work is proportional to the latent heat of vaporization, Δh_{vap}, at the vaporization temperature. A generalization can therefore be made: Fluids with high latent heats of vaporization will have higher-specific-work turbines characterized by lower flows and more stages than fluids with small latent heats.

Stage number, n. - To conserve turbine weight, it is desirable for a given application using a specific working fluid to have a turbine that has maximum efficiency with as few stages as practical. Substituting the aforementioned value of $\lambda_{ST} = 0.5$ into equation (20) and solving for stage number result in

$$n = \frac{0.5\ gJ\Delta H}{U_m^2} \tag{21}$$

It is noted from equation (21) that stage number is reduced by increasing the mean blade speed U_m. In view of this, current gas turbines E-486
tend to operate at blade speeds limited by stress considerations, in the neighborhood of 1000 feet per second. However, when the operating temperatures are pushed to extreme limits, as they are for liquid-metal turbines for advanced space applications, the allowable design stresses are drastically reduced, especially from a reliability standpoint for long-time continuous use. Consequently, turbines in this class will undoubtedly be designed to operate at lower blade speeds and have more stages than conventional turbines that have comparable specific-work outputs but lower operating temperatures.

To show the differences in stage number as a function of working fluid, equation (21) was solved for a range of mean blade speeds for the fluids monatomic sodium, potassium, and rubidium. The selected blade speed and turbine inlet temperatures for the fluids involved are presented in the following table. For simplicity, the turbine temperature ratio of 3/4 was used throughout:

Fluid	Na	K	Rb
Mean blade speed, ft/sec	400-1000	300-1000	300-1000
Turbine inlet temperature, $^{\circ}R$	2500-2000	2300-1800	2100-1600

The values of turbine work were determined by assuming a turbine efficiency of 0.80 and using the Mollier charts and tables derived from references 4 to 6. The resulting relation of stage number and blade speed for the three working fluids considered is shown as figure 6.

The effect on stage number of being forced to lower blade speeds as a result of extreme temperature and reliability requirements is shown in the figure. For example, reducing the mean blade speed of a sodium turbine from 800 feet per second to 600 feet per second would require an increase in stage number from 8 to 14 stages. The effect becomes greater as blade speed is reduced further.

The turbine specific-work outputs for Rb, K, and Na are seen from figure 6 to be in the approximate proportions of 1, 2.5, and 5, respectively. As discussed previously, this results from the fact that their latent heats of vaporization have approximately the same proportions.

Consequently, for the same blade speeds, the ratio of required turbine stages is also the same. Sodium, for example, would require approximately twice as many stages as potassium and five times as many as rubidium. However, for equal power requirements, the required system weight flow and hence fluid inventory weight (which may be quite large) would naturally be in the reverse ratio. This leads to the conclusion that it would be improper to select any fluid as a more favorable turbine driving fluid without a more detailed investigation of turbine characteristics and inventory weight.

The spread in specific-work output for each of the fluids involved is not very large when it is noted from figure 6 that a range of 500° R in turbine inlet temperature was taken for each fluid. The decrease in turbine temperature difference (T_2 - T_3) caused by decreasing the turbine inlet temperature and using the same radiator temperature ratio of 3/4 is compensated for by the increased ΔS of vaporization at lower temperatures to make their product substantially the same. This trend can be noted from figure 4.

Moisture Formation

As previously discussed, it is desirable to operate alkali metal systems with saturated or near-saturated vapor entering the turbine. This means that a significant amount of moisture droplets may form within the turbine, which will adversely affect both mechanical reliability and aerodynamic performance. There are three questions to answer concerning moisture: (1) Do moisture droplets form during the turbine expansion process? (2) If moisture does form, will the turbine mechanically operate for a long continuous time, or will blade erosion destroy its reliability? (3) If it does form and can be tolerated mechanically, what penalty in turbine performance might occur? None of these questions can be answered at the present time with certainty.

If the phenomenon of supersaturation occurs and no moisture forms during the expansion process, a small loss in turbine work output will result from a decrease in available energy due to the expansion process. This would probably be accepted in view of the serious implications that moisture formation poses, but to assume that supersaturation will occur does not appear sound. Therefore, this analysis assumes that moisture formation does occur and that the fluid maintains an equilibrium expansion through the turbine; that is, the properties of the fluid are equal to those indicated by the state point on a T-S or h-S diagram.

The quantitative determination of the permissible moisture allowed to form within a turbine represents a research area that would require a considerable effort. However, if it is found that moisture does form and that the turbine cannot tolerate it, there are two apparent solutions: Either steps must be taken to reduce or prevent moisture formation, or the moisture must be collected as fast as it forms before it can do any damage. The first solution can be accomplished by using such cycle controls as a very inefficient turbine, superheat, reheat, or higher than optimum radiator temperatures. These will be discussed later. The second solution would utilize a mechanical collection scheme, returning the liquid to the cycle. Partial moisture removal has been done with mercury vapor turbines and may represent a satisfactory solution.

If moisture does form and can be tolerated by the turbine mechanically, the assignment of a penalty to the performance of the turbine can only be assumed. The validity of trying to extrapolate the moisture problems of known fluids to the alkali metals is highly questionable. However, a loss assumption can be made and the effect of moisture calculated, realizing that only qualitative observations have any significance. In view of this, only the results for one fluid, monatomic sodium, are presented. The results for potassium and rubidium were found to be similar.

Assumed moisture penalty on aerodynamic performance. - The assumed moisture penalty for this analysis follows reasoning similar to that used in steam practice. Reference 8 indicates that, for each 1 percent of moisture in a steam-turbine stage, there will be a reduction in stage power of about 1.15 percent. The reference attributes this to the braking effect on the rotor blades caused by the moisture particles leaving the nozzle at a velocity approximately 1/10 that of the main body of dry steam. The same type of loss will occur in liquid-metal turbines, the question being whether or not the magnitude will be the same. In view of the many unknowns concerned, this loss for each 1-percent moisture is arbitrarily increased from 1.15 to 1.5 percent for this analysis.

The assumed moisture loss can be presented in the form of a working expression for turbine efficiency. From the basic assumption,

$$\Delta H = (1 - 1.5\ m_3)\Delta H_{aer} \tag{22}$$

where ΔH_{aer} is the work output that would result if there were no losses due to the moisture. The corresponding turbine efficiency is then

$$\eta_T = (1 - 1.5\ m_3)\eta_{aer} \tag{23}$$

To show the effect of moisture, an aerodynamic turbine efficiency excluding moisture loss, η_{aer}, of 0.80 was assumed for sodium at a turbine inlet temperature of 2500° R. Equation (23) was then combined with equation (16) to solve for radiator area for the same range of conditions used for sodium in the CYCLE ANALYSIS for a turbine efficiency of 0.80. The resulting effects of moisture on specific radiator area requirements resulting from the reduction of turbine efficiency are shown in figure 7. The lower curve is the same as the values for sodium of figure 5(b) for a turbine efficiency of 0.80. The difference between the lower and upper curve is therefore the penalty caused by the assumed additional loss due to moisture. Lines of constant moisture are also shown on the figure.

As noted from figure 7, there is an 18-percent increase in minimum radiator requirement due to the effect of moisture. Further, the penalty becomes larger as radiator temperature is decreased, resulting in an increase of 37.6 percent at 1600° R. This naturally results from the increase in moisture at lower radiator temperatures, which range from 5 to 14 percent as the radiator temperature decreases from 2200° to 1600° R.

It must be reiterated that the discussed moisture penalty on turbine performance is only as valid as is the assumed loss of 1.5 percent. However, the calculations show that the penalty in radiator area due to the moisture loss effect may be significant.

<u>Means of reducing moisture</u>. - The most serious effect of moisture would be felt in system reliability. If a turbine can be made to operate reliably with moisture present, the simplest and probably the best solution would be to let it form and accept the penalty in turbine efficiency, since any means taken to eliminate or largely reduce moisture add either complexities to the system or increased weight, or both. However, if it does become necessary to reduce or eliminate moisture, the aforementioned means can be considered:

(1) The partial removal of condensed liquid by mechanical means appears to be an attractive solution if the effective collecting points can be determined and a simple return system provided.

(2) The use of an inefficient turbine expansion process along the saturated vapor line would eliminate moisture at the expense of a great penalty in turbine efficiency and hence radiator area. For example, saturated sodium vapor at 2500° R entering the turbine and expanding in this manner to a radiator temperature of 1800° R results in a turbine efficiency of only 27 percent and a required radiator size of about 3 square feet per kilowatt. As can be seen from figure 5(b), this would result in more than a 300-percent increase in area. The use of an inefficient expansion as a means of moisture control therefore appears prohibitive.

(3) The amount of moisture formation may be reduced by increasing the cycle radiator temperature to a higher than optimum value. Referring

to the upper curve of figure 7 that includes the assumed moisture penalty, it is noted that moisture can be reduced from about 0.09 to 0.06 when the radiator temperature is increased from 2000° to 2180° R. The penalty in specific radiator area accompanying this increase in radiator temperature is seen to be about 9 percent. Also, the effect on other cycle components and weights due to a decrease in cycle efficiency must be considered. However, this method of moisture control adds no complexities or mechanical modifications to the system and appears attractive if the solution of the moisture problem involves only a reduction in moisture of a few points.

(4) Figure 8(a) shows the use of varying amounts of superheat as a means of moisture control. In each case the latent heat of vaporization is added at a temperature below the 2500° R specified for the assumed cycle. The working fluid is subsequently superheated to this desired temperature level. This tends to decrease the cycle efficiency by lowering the average temperature of heat addition, but at the same time it increases the turbine efficiency by lowering the moisture loss effect at the turbine exit. This moisture control factor is shown in the figure by the movement of the expansion point toward the saturated vapor line as the amount of superheat is increased.

The effect of varying amounts of superheat on specific radiator area is shown in figure 9 for monatomic sodium. The calculations were made for η_{aer} of 0.80 and included the moisture loss penalty in the wet region as previously discussed. It is seen from the curves that a general increase in required radiator area is caused by superheating. This indicates that, although turbine efficiency is improved somewhat by a reduction of the moisture loss, the effect of this reduction on radiator area is small compared to the increase in radiator area caused by the decrease in cycle efficiency resulting from the lower average temperature of heat addition. Further, the range of superheating considered does not substantially reduce moisture for a fixed radiator temperature. Therefore, the use of superheat alone as a means of moisture control does not appear attractive, because comparatively large penalties in radiator area are required for only small reductions in moisture.

(5) If it becomes necessary to reduce moisture to a very low value, or completely eliminate moisture, it may be desirable to consider a reheat cycle. Figure 8(b) shows the use of reheating where superheated vapor enters the turbine and is subsequently reheated following some degree of expansion. The initial superheat is the same as in the case of pure superheating. After the superheated vapor enters the turbine it is allowed to expand to, or close to, the saturated vapor state. The vapor is then reheated into the superheat region and reexpanded. The moisture reduction or elimination after the final expansion is such that the turbine exhaust point can fall slightly in the wet vapor region, on the saturated vapor line, or slightly in the superheat region.

The use of reheat as compared with saturated or superheated vapor at the turbine inlet may or may not increase the average temperature of heat addition, which may or may not increase cycle efficiency. However, even if η_c decreases slightly because of this effect, the amount of decrease will be small compared with the corresponding decrease caused by superheating alone. In addition, since the moisture in the turbine exhaust is considerably reduced or possibly eliminated, the moisture loss (eq. (23)) will not be significant. However, it is recognized that the mechanical arrangement of ducting, mixing chambers, and/or heat exchangers required for the reheat schedule could introduce significant losses in the turbine and system. The results presented herein are therefore optimistic, and a prior knowledge of the total losses associated with a reheat arrangement would be needed in order to evaluate accurately the effect of reheat.

In this analysis, it was assumed that in the first reheat the vapor attains a temperature of 2500° R, and 50° R less in each subsequent reheat. For illustrative purposes, the number of reheats was arbitrarily limited to three, which would correspond to interstage reheating of a four-stage turbine. The calculations for specific radiator area were again made for an assumed η_{aer} of 0.80 and included the moisture loss penalty in the wet vapor region. Figure 10 presents the results of these calculations and shows the theoretical advantage of using reheat as compared with superheat alone (fig. 9), in that moisture can be largely reduced or eliminated without penalizing radiator area. The improvement is primarily a result of higher average temperatures of heat addition.

SUMMARY OF RESULTS

The analysis considered advanced Rankine-cycle turboelectric power systems. The relative effects of turbine inlet temperature, radiator temperature, turbine efficiency, and working fluid on radiator area requirements were determined for the alkali metals, rubidium, potassium, and sodium operating in a Rankine cycle. Because of the importance of high turbine efficiency, the factors peculiar to the associated turbines were considered and their influence on turbine design procedures noted. The results of the investigation may be summarized as follows:

1. The temperature level of heat addition to the cycle has a gross effect on required radiator size and should be as high as practical. For any turbine inlet temperature selected, minimum radiator area will occur when the radiator temperature is approximately 3/4 of turbine inlet temperature. The resulting minimum area is, however, sensitive to the selected level of turbine inlet temperature. For example, a 200° increase in this temperature represents, on the average, a 30-percent reduction in minimum radiator requirements.

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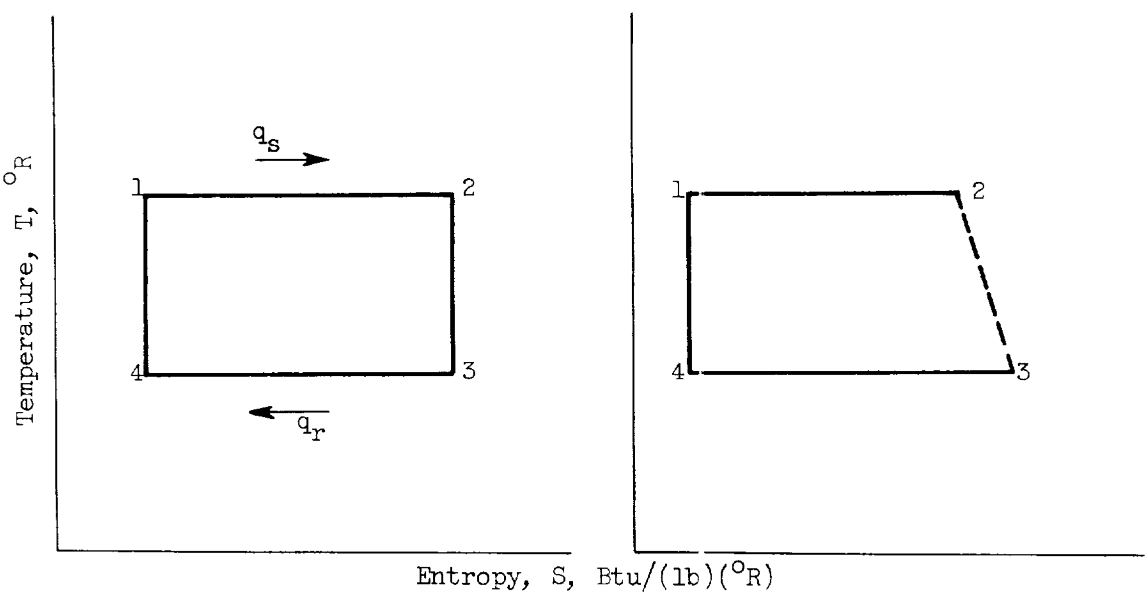


(a) Carnot cycle.

(b) Modified Carnot cycle.

Figure 1. - Carnot and modifiei Carnot cycles.

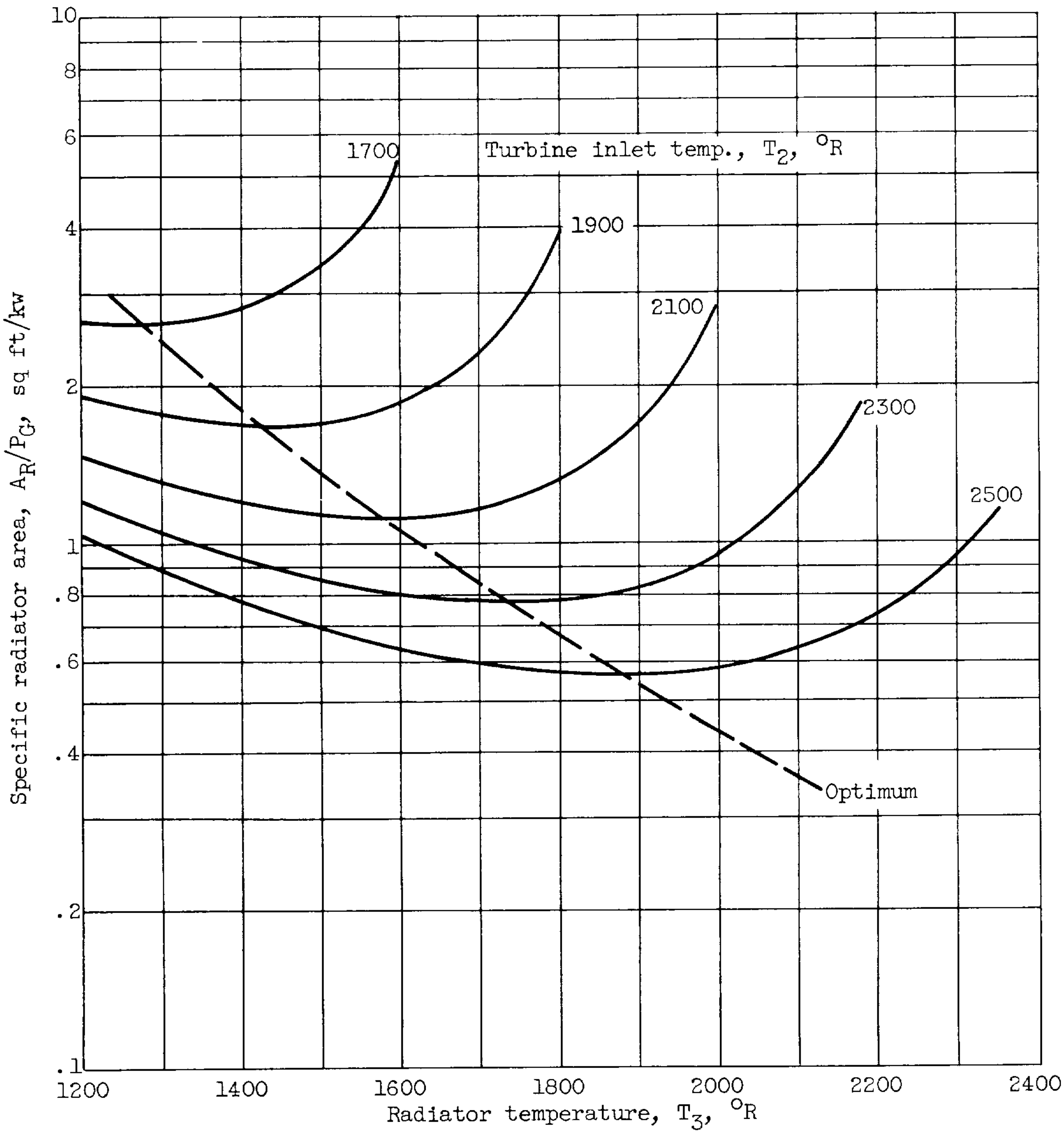
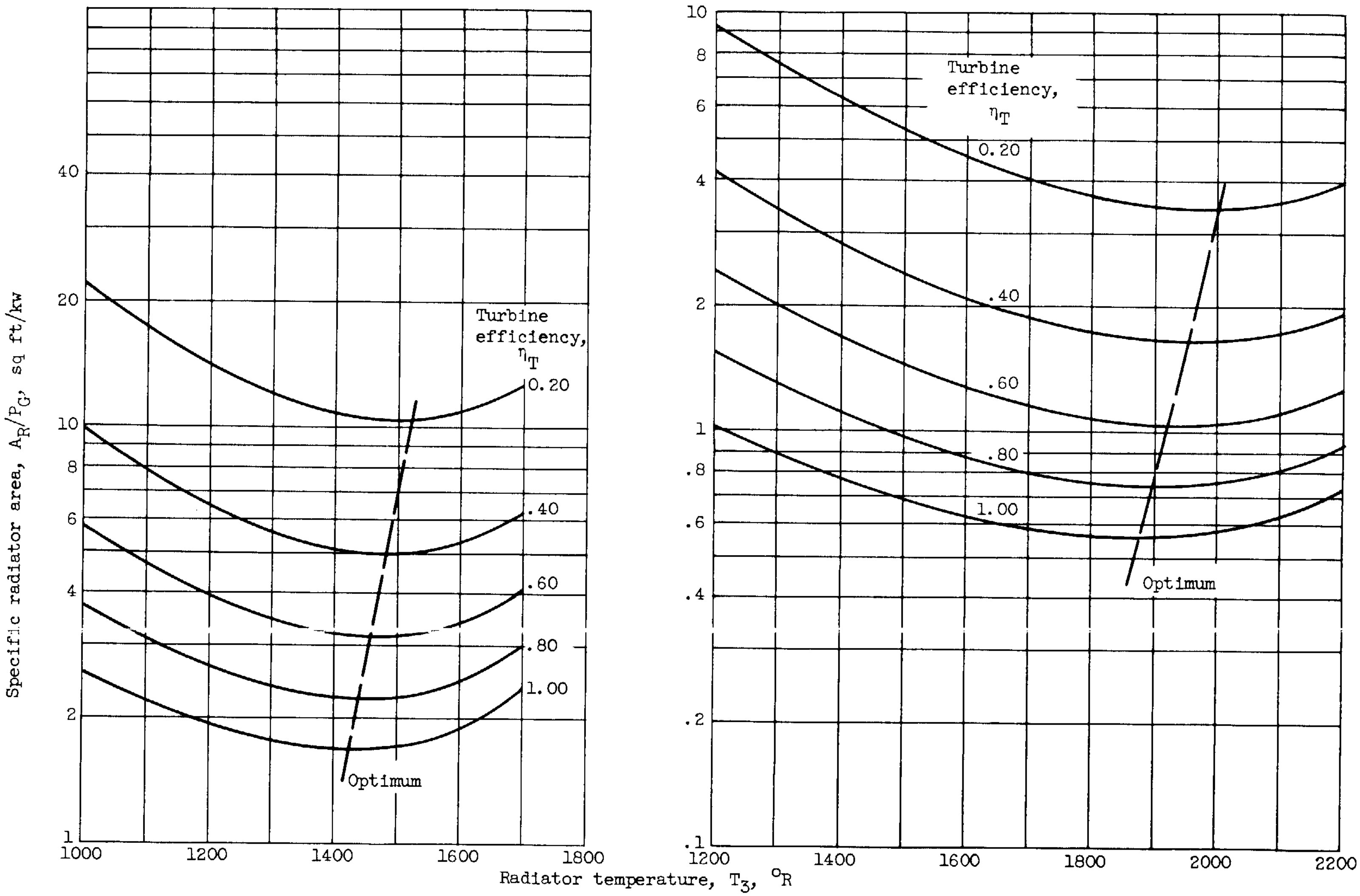


Figure 2. - Effect of turbine inlet and radiator temperatures on radiator area, Carnot cycle.



(a) Turbine inlet temperature, 1900° R.

(b) Turbine inlet temperature, 2500° R.

Figure 3. - Effect of turbine efficiency and radiator temperature on radiator area, modified Carnot cycle.

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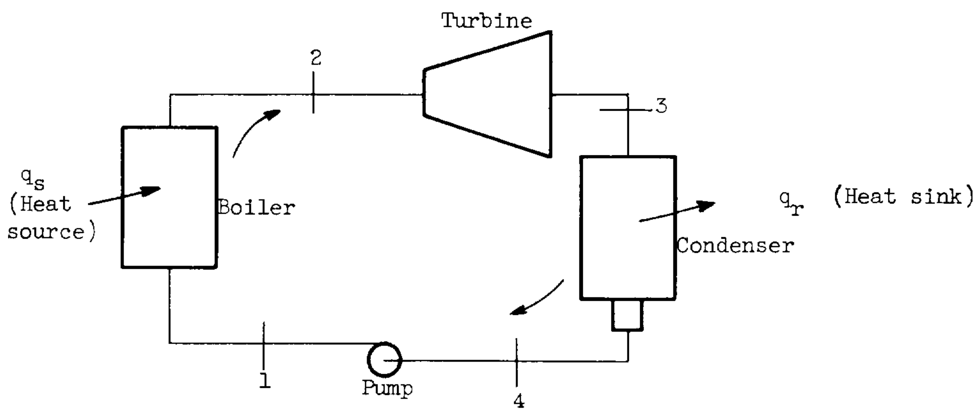


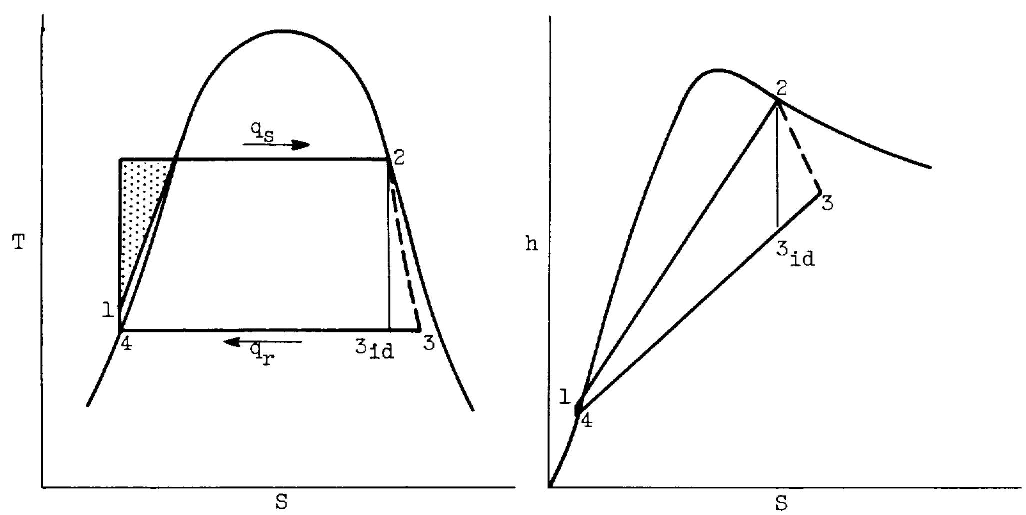
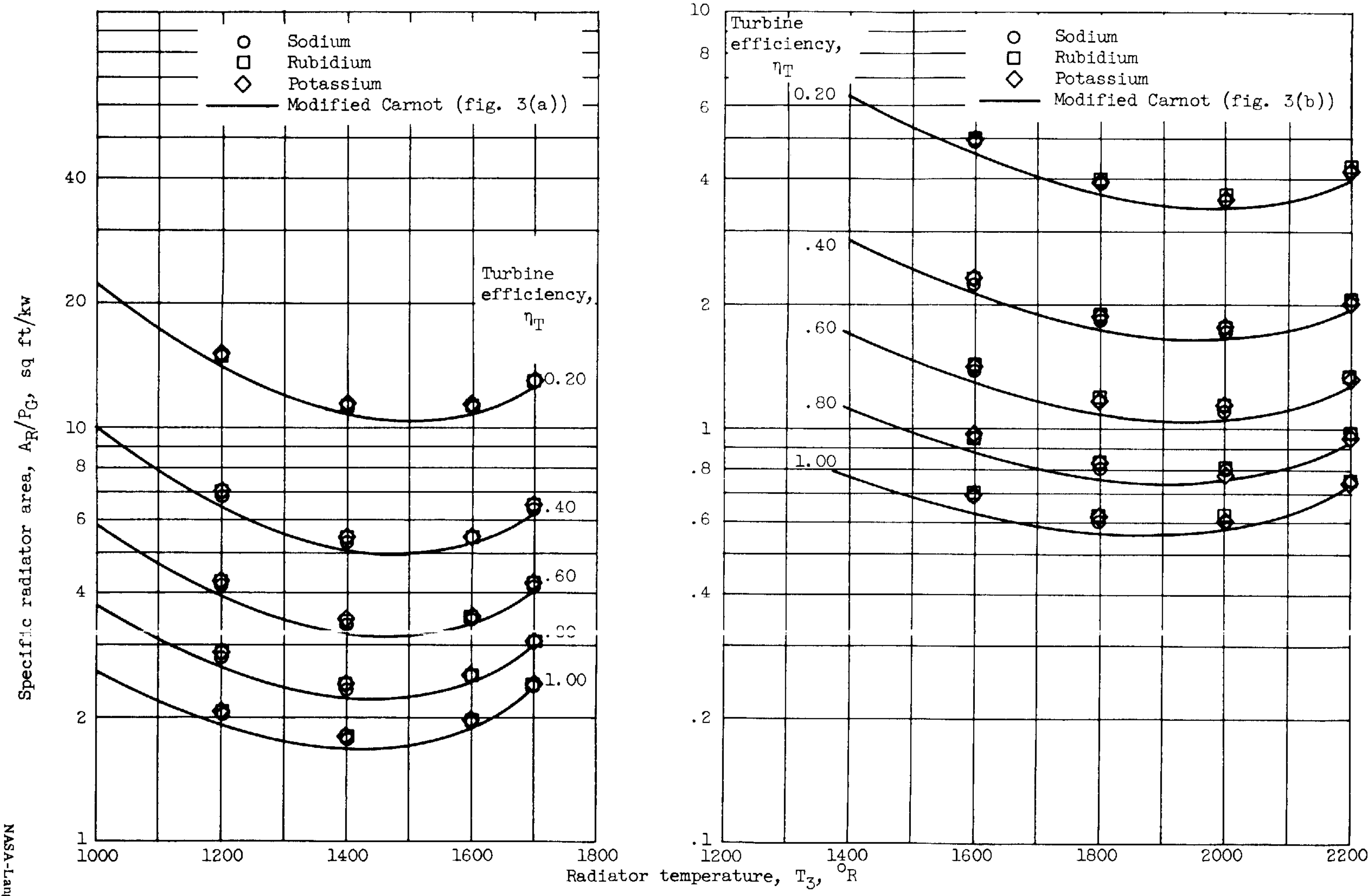


Figure 4. - Modified Rankine cycle, saturated vapor at turbine inlet.



(a) Turbine inlet temperature, 1900° R.

(b) Turbine inlet temperature, 2500° R.

Figure 5. - Effect of working fluids sodium, rubidium, and potassium on radiator area, modified Rankine cycle.

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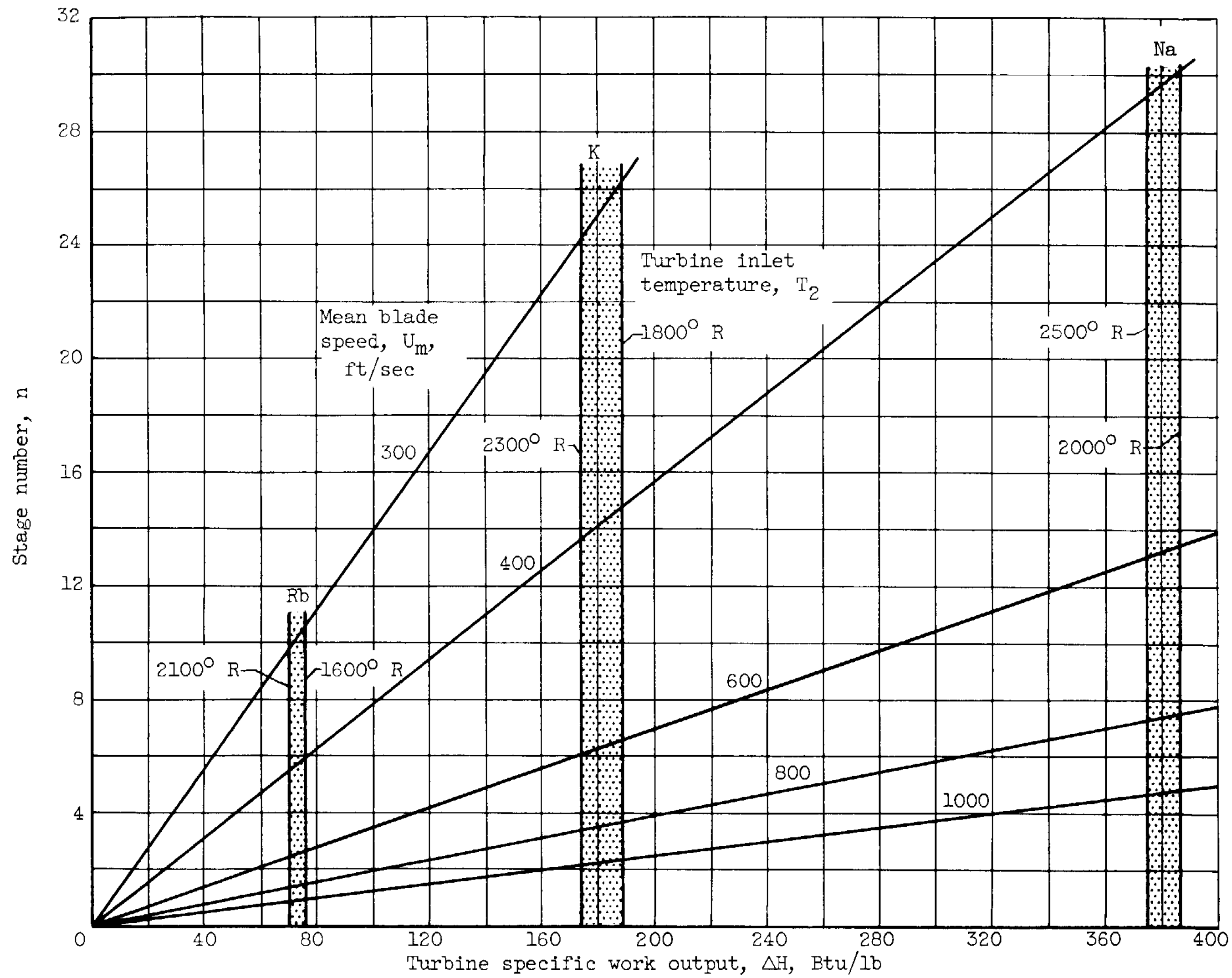

Figure 6. - Turbine stage number as function of blade speed and working fluid. Temperature ratio, T_3/T_2, 3/4; speed-work parameter, λ_{ST}, 0.5; turbine efficiency, η_T, 0.80.

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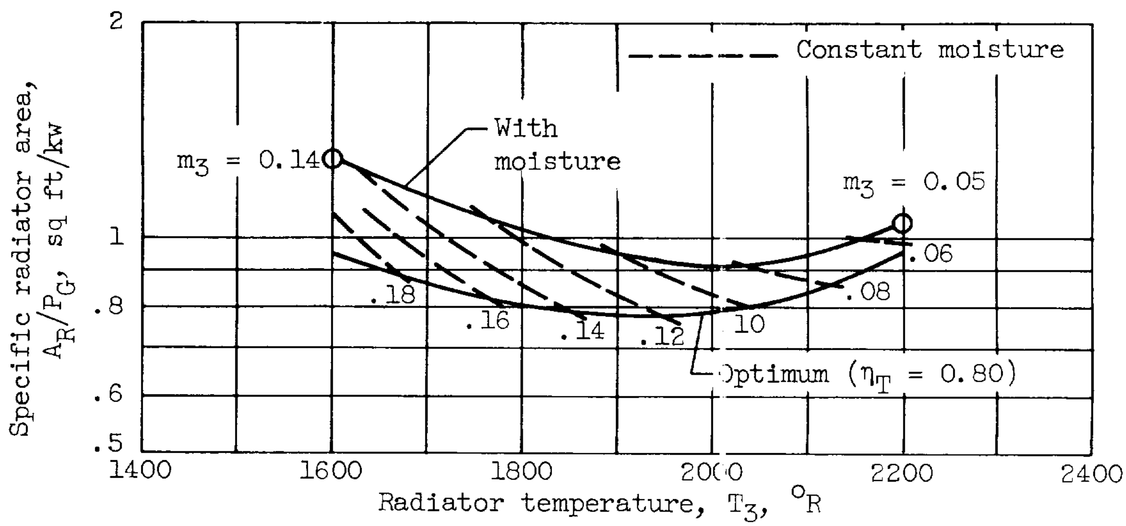

Figure 7. - Effect of moisture on radiator area, sodium. Turbine inlet temperature (saturated vapor), 2500° R.

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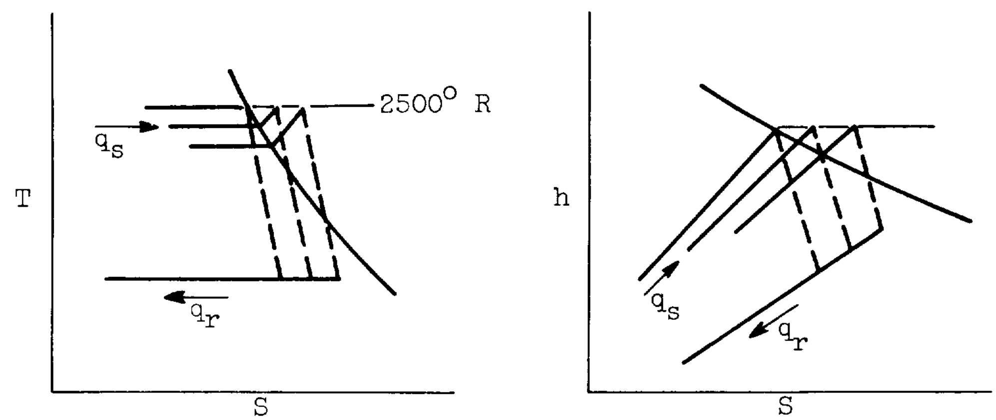
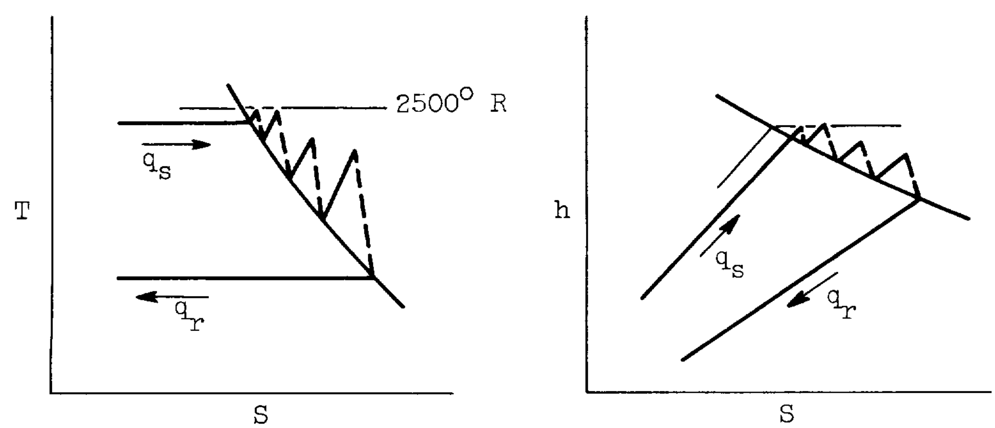


(a) Superheating.



(b) Reheating.

Figure 8. - Moisture control by variations in heat addition.

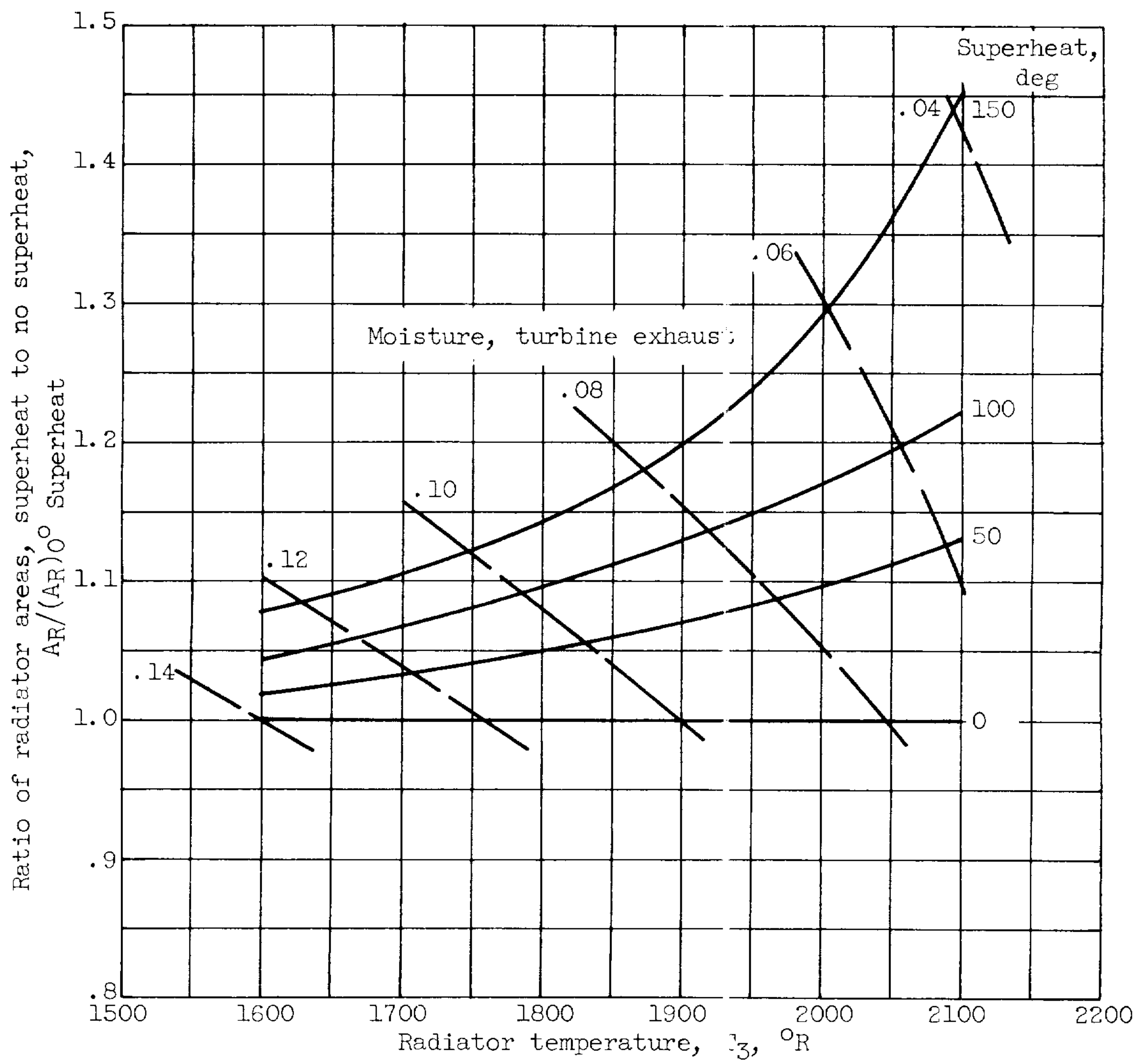


Figure 9. - Effect of superheat on radiator area and moisture content, monatomic sodium. Turbine inlet temperature, 2500° R.

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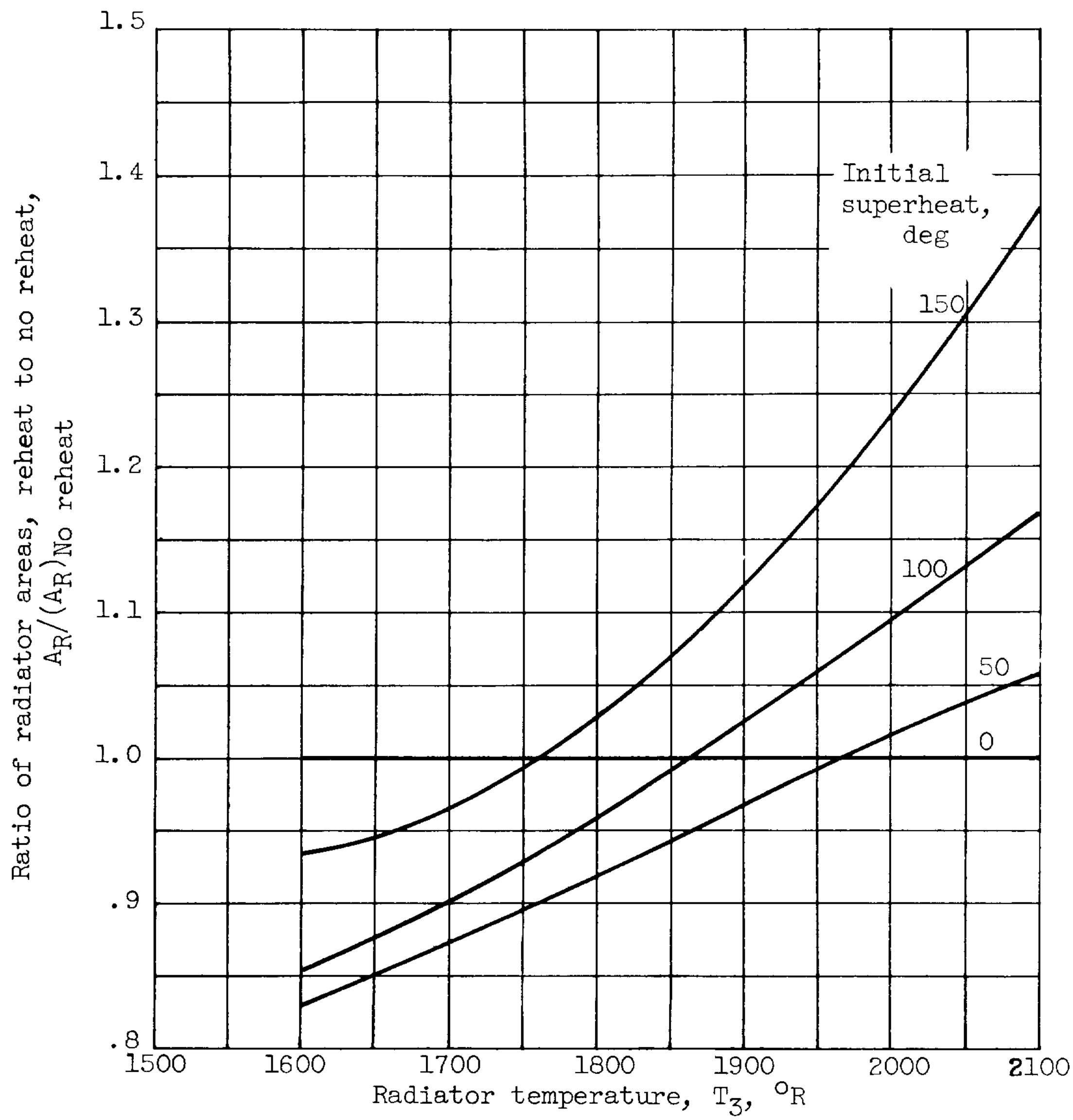

Figure 10. - Effect of reheat on radiator area, monatomic sodium. Turbine inlet temperature, 2500^{o} R; maximum moisture, < 0.04.