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Review Copy

WRITTEN BY Grimaldi SECTION Compact Powers System GROUP Advanced Technology UNIT Advanced Systems APPROVED BY: (SUPERVISOR) M. G. Coombs	**ATOMICS INTERNATIONAL** *A Division of North American Aviation, Inc.* **TECHNICAL DATA RECORD** PROGRAM Advanced Systems PROJECT Advanced Space Power Plants	TDR NO. 5615 GO 3621 LEDGER ACCT. 7590 SUB-ACCT. 2070 TWR DATE August 29, 1960 PAGE 1 OF 5

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SUBJECT:

Space Handbook Turbines

CONTENTS:

I STATEMENT OF PROBLEM

As part of the space handbook study several fluids, power and temperature level have been investigated to obtain order of magnitude performance and weights of possible nuclear powerplant systems.

II SUMMARY

Due to the lack of time and the general nature of this study three detailed turbine analyses were conducted to determine the general size, type, number of stage and performances of three representative turbines. The balance of the units, 32 inch total, were obtained by ratioing from these representative conditions.

Lacking a physical layout of the units it is impossible to determine stresses or weights with any great degree of accuracy. However, the tip speeds have been kept within acceptable limits and the weights were estimated on the basis of proportions established from past experience. Therefore, the information generated in this study, while not precise, is useful in estimating the general order of performance and weights of space power systems and should not in any way be construed as final or optimum design values.

III DISCUSSION

The turbine power requirements were based on an alternator efficiency of 85% and a pump power requirement of 5% of the gross turbine output. Both of these quantities can vary somewhat with alternator coolant temperature level and with cycle

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operating pressure ratio. However, these variations will have minor effects on the size and performance of the turbine.

A preliminary investigation of the type of turbine showed that a net output of 3000 kw at 1600°F a tandem design, that is, two identical turbines on the same shaft, each handling one-half the total flow, with a shaft speed of 8000 rpm would result in a lighter, more efficient design than a single unit with a shaft speed of 6000 rpm. Both of these speeds are the maximum allowable speeds consistent with reasonable blade tip velocities which are expected to result in reasonable stress levels and erosion damage. Similar weight saving was observed at the other power levels at 1600°F with similar blade tip velocity limitations. While effect of configuration was not investigated at the other temperature levels of interest, similar results are expected. Therefore, all turbines are of the tandem type.

Three specific turbines were analysed as follows:

3000 kw Rubidium 1600°F inlet temperature
3000 kw Potassium 1600°F inlet temperature
3000 kw Sodium 1800°F inlet temperature

The analysis was based on curves of efficiency as a function of specific speed and specific diameter obtained from Sunstrand Turbo Division. Wherever possible the maximum efficiency design was used. However, in the case of the last stage, potassium turbine at 1600°F and 1800°F and the last three stages of the sodium turbines, the maximum efficiency design resulted in excessive tip velocities. Therefore, a small compromise in efficiency was made to obtain a reasonable design.

Once these basic units were designed, the balance of the units were sized and the performance estimated by multipling the efficiencyand diameters by the ratio of the gross or single stage performance and specific diameter as corrected for density effects. Since the sodium turbine at 1800°F involved some performance compromises, this procedure could not be used. Therefore, the ratio were based on the last stage only. No attempt has been made to establish which of these two methods give the best results. However, both methods give reasonable answers. The performance and sizes resulting from this study are given in Tables I and II.

The weights were estimated as follows. From an existing design, the equivalent disk weight was established for the 1600°F condition. The allowable stress varies as the log of the temperature. On the assumption that allowable stress at the 2200°F temperature level was 110% of the value at 1600°F, the effective disk thickness was determined from

$$t = (\bar{t}_{1600} + .12 \text{ hrs. } \frac{T}{2060}) \quad (1)$$

where

$\bar{t}$ = equivalent disk thickness
T = absolute temperature of operation

From past experience it is known that the turbine weight is two to five times the total disk weight depending on the size of the turbine. With this as the starting point, an equation for estimating this factor was determined. The resulting equation for turbine weight is

$$W_T = W_D n \left(1.4 + \frac{1.8}{\bar{D}}\right) \qquad (2)$$

where W_T = Turbine weight

W_D = Average disk weight

n = Number of stages
$\bar{D}$ = Average disk diameter

By the use of equations (1) and (2), the specific weight of the turbines were estimated and presented in figure 1.

Several things can be noted from this figure. As far as the turbine is concerned, there is definitely no advantage in the use of potassium or sodium as a working fluid. The specific weights of the rubidium units decrease with increasing temperature. Potassium and sodium do not show equivalent trends. This inconsistency can be explained in part in that the low temperature units are proportionately smaller due to the tip speed compromise than in the high temperature units where this compromise is not necessary. Another factor that enters into this comparison of trends is that the reduction in diameter for the potassium and sodium systems is more than made up by decrease in strength of the material which requires a thicker, heavier construction at the higher temperatures. No attempt has been made to determine the effect of pressure on casing thickness and therefore, weights. However, the proportions given by equation (2) are based on higher pressure systems than currently considered. Therefore, all of these weights and especially the low temperature end of the curves are conservative in this respect.

From the forgoing discussion, it can be seem that uncertainty of specific weights is quite high. However, since the turbine represents about 10% of the plant weight even an error of 50% in specific weight will not produce a greater error than 5%. Since the possible error in performance or size estimate is no more than 5%, an additional possible error of 5% in system weight is anticipated from the effect of turbine performance on radiator weight.

Therefore, the figures generated in this study are good for the estimate of system weights and performance within the limitations presented above.

TABLE I

Efficiency - %

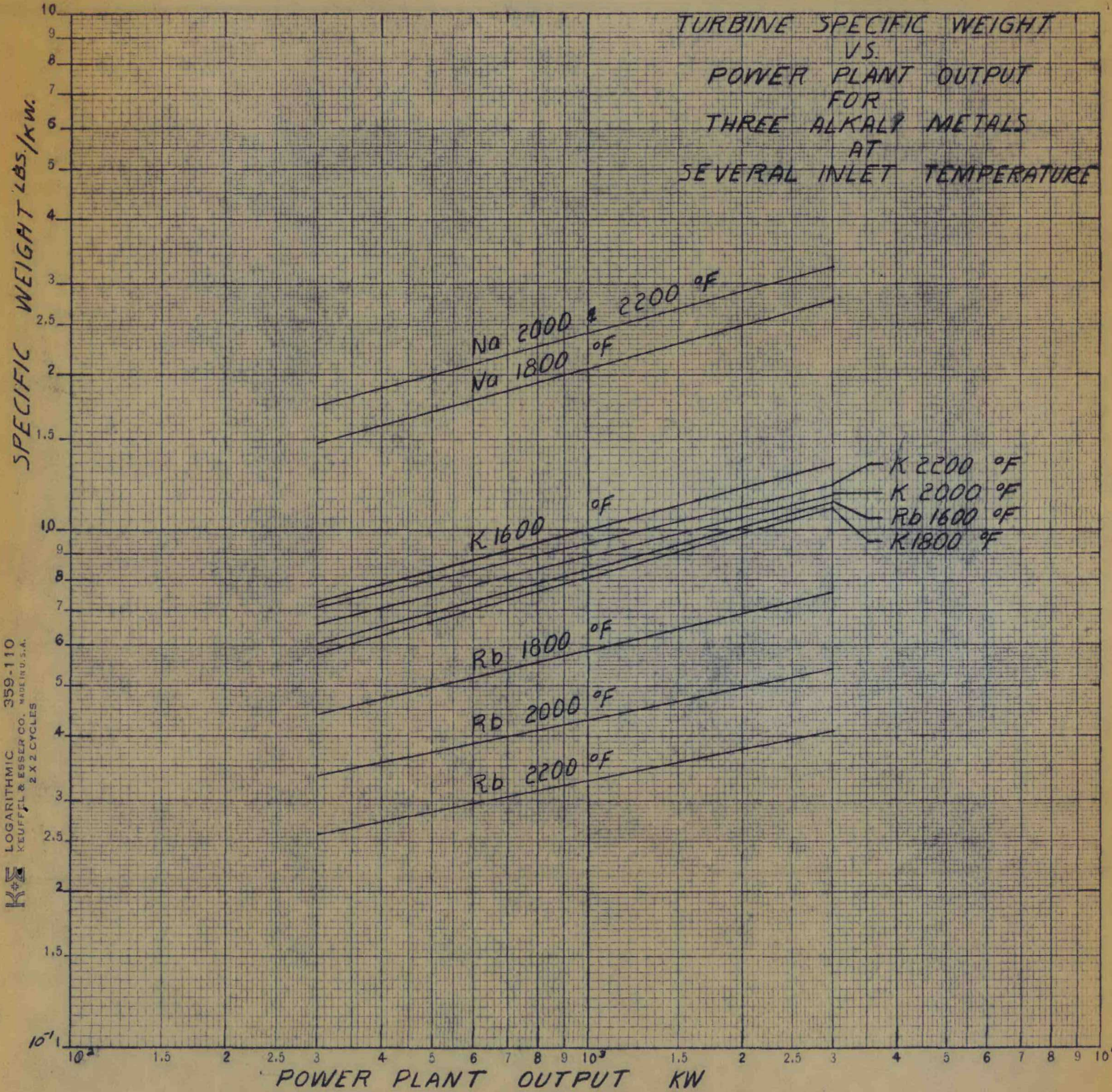
Temperature	1600°F	1800°F	2000°F	2200°F
Rubidium	73	71	68	65
Potassium	75	72	70	67
Sodium	--	68	76	76

TABLE II

Rubidium

	1600°F		1800°F		2000°F		2200°F	
	Length*	Max. Dia.*	Length	Max. Dia.	Length	Max. Dia.	Length	Max. Dia.
3000 kw	2.4	3.2	2.2	2.5	2.2	2.2	2.2	1.7
1000 kw	1.6	2.0	2.0	1.60	1.8	1.4	1.0	1.3
300 kw	1.0	.95	1.2	.76	1.0	.66	1.0	.53
Potassium								
3000 kw	2.8	2.9	2.6	2.4	2.8	2.7	3.0	2.7
1000 kw	2.4	2.4	2.2	2.1	2.2	2.0	2.3	1.9
300 kw	1.4	1.0	1.2	.8	1.3	.9	1.4	.9
Sodium								
300 kw	--	--	3.6	3.1	3.7	2.8	4.0	2.7
1000 kw	--	--	2.8	2.2	2.7	1.9	2.9	1.9
300 kw	--	--	1.4	1.00	1.5	1.00	1.6	.9

* feet


TURBINE SPECIFIC WEIGHT
VS.
POWER PLANT OUTPUT
FOR
THREE ALKALI METALS
AT
SEVERAL INLET TEMPERATURE
SPECIFIC WEIGHT LBS./KW.
POWER PLANT OUTPUT KW
Na 2000 & 2200 °F
Na 1800 °F
K 1600 °F
K 2200 °F
K 2000 °F
Rb 1600 °F
K 1800 °F
Rb 1800 °F
Rb 2000 °F
Rb 2200 °F
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