9/3-73
LA-5295-MS

INFORMAL REPORT

113
Hh. 230

OK
T

**A Preliminary Study of
Open-Cycle Space Power Systems**

THIS DOCUMENT CONFIRMED AS
UNCLASSIFIED
DIVISION OF CLASSIFICATION
BY IL Cucchiera / pab
DATE 8/1/73

los alamos
scientific laboratory
of the University of California
LOS ALAMOS, NEW MEXICO 87544

S1506

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

UNITED STATES
ATOMIC ENERGY COMMISSION
CONTRACT W-7408-ENG. 38

MASTER

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

In the interest of prompt distribution, this LAMS report was not edited by the Technical Information staff.

Printed in the United States of America. Available from
National Technical Information Service
U. S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22151
Price: Printed Copy ~~\$8.00~~; Microfiche \$0.95

4.00

LA-5295-MS
Informal Report
UC-33

ISSUED: June 1973

A Preliminary Study of Open-Cycle Space Power Systems

by

L. A. Booth
J. C. Hedstrom

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

Fig

A PRELIMINARY STUDY OF OPEN-CYCLE SPACE POWER SYSTEMS

by

L. A. Booth and J. C. Hedstrom

ABSTRACT

Specific masses of earth-orbiting electric power systems in the tens-of-megawatts range with lifetimes of a few hours are estimated. With the low total energy requirement corresponding to short operating times, open-cycle systems, consisting of a disposable working fluid and a heat-engine power-conversion system, were chosen for study. Results indicate that open-cycle system masses are lower than optimistic projections of closed-cycle-system masses for operating times up to $\sim 10^4$ s.

I. INTRODUCTION

Some interest has been expressed recently in the technical feasibility of developing electric power systems to be launched into earth orbit for operation at power levels of the order of tens of megawatts, with lifetimes of the order of thousands of seconds. Reference specifications for such systems are:

Electric power, 1.0 to 100 MW at ≥ 10 kVdc

Operating time, 1800 s

Acceptable system mass, $< 10^5$ kg.

Throughout the duration of the U.S. space power system technology program, the principal design criterion has been that of minimum specific mass, i.e., lowest ratio of total system mass to electric power output. Although system development and capital costs are major considerations, the heretofore high earth surface-to-orbit launch costs ($\sim \$450/\text{kg}$ for a Saturn-V payload) have generally given lighter systems the advantage on a cost-effectiveness basis.

At power levels greater than one kilowatt, heat-engine power conversion systems become the lightest because their conversion efficiencies are higher

than those of direct-conversion systems, e.g., solar-electric or thermoelectric; and the desired system operating lifetimes (from a few months to a few years) dictate that the heat-engine systems be closed-cycle. However, for these closed-cycle heat-engine systems, the aforementioned cost premium for light weight has limited power levels to tens of kilowatts, with the specific mass of the lightest of these developed systems being $\sim 100 \text{ kg/kW}_e$.⁽¹⁾ Optimistic projections of this state-of-the-art technology into the hundreds-of-kilowatts range indicate a minimum specific mass of $\sim 30 \text{ kg/kW}_e$.⁽²⁾ and a recent estimate for a power system based on undeveloped technology has a specific mass of $\sim 5 \text{ kg/kW}_e$ at 10 MW_e.⁽³⁾

Clearly, systems with specific masses of this order of magnitude would exceed the maximum of 10^5 kg at 100 MW_e. However, the specified short lifetime permits the cycle working fluid to become an expendable mass replacing, and substituting for, the relatively heavy heat-rejection and regeneration equipment of the closed-cycle system. Such an open-cycle system would be less complex and its mass would strongly depend on the total electric energy requirement (or operating lifetime). The study of promising open-cycle power systems is the subject of this report.

II. OPEN-CYCLE SYSTEM STUDIES

System Identification

The open-cycle systems for this study consist of a working fluid and a heat engine. The working fluid is the mass of material which, when thermal energy is added, has sufficient enthalpy and entropy to perform work by expansion, and the heat engine is the device in which this expansion work is transformed into electrical energy. Thermal energy may be added to the working fluid either by heat transfer from an external heat source such as a fission reactor, or by the chemical reaction of chemically reactive substances. In the latter case, the products of the chemical reaction become the working fluid.

For minimum specific mass, the enthalpy (or heat capacity) in terms of working-fluid mass should be as high as possible. Because the heat capacity of hydrogen is at least a factor of three higher than that of any other substance, hydrogen is the ideal working fluid, even though it must be stored in space as a cryogenic liquid. With hydrogen as the working fluid, the thermal heat source could be a nuclear rocket propulsion reactor. Because these reactors have the highest power density of any developed fission-energy source, they would provide the lowest specific mass for heating hydrogen.

For good performance (low specific mass) with chemical reaction products as a working fluid, the hydrogen content should be as high as possible, again because of the favorable heat capacity. Chemical reactants may be divided into two classes: (1) those that must be stored in space as cryogenics, and (2) those that may be stored in space at ambient earth surface temperatures and pressures. Cryogenic reactants whose reaction products have good performance characteristics are oxygen (LO_2) and hydrogen (LH_2), whereas space-storable reactants with good performance are monomethylhydrazine (MMH) and dinitrogen tetroxide (N_2O_4).

Two types of heat engines would be practical for this space application: (1) a Faraday type MHD generator, where the change in working-fluid enthalpy is converted directly into electricity; and (2) a turbine-alternator, where the working-fluid

enthalpy is converted into shaft work in the turbine while the shaft work is converted into electricity in the alternator.

Four combinations of these working fluids and heat engines were studied:

Working Fluid	Thermal Energy Source	Heat Engine
LH_2	Nuclear propulsion reactor	MHD generator
LH_2	Nuclear propulsion reactor	Turbine-alternator
LO_2/LH_2	Chemical reaction	Turbine-alternator
N_2O_4/MMH	Chemical reaction	Turbine-alternator

A schematic depicting the major components of these systems is shown in Fig. 1.

Determination of System Masses

By defining a converter efficiency as

$$\eta_c = \Delta H_w / H_o^* \quad (1)$$

where ΔH_w is the change in working-fluid enthalpy that is converted into work in the heat engine, and H_o is the enthalpy of the working fluid at the inlet to the heat engine. The mass of the working fluid is:

$$m_p = \frac{\xi}{\eta_c H_o} \quad (2)$$

where ξ is the total energy output of the heat engine. In terms of specific mass (in kg/kW), Eq. (2) may be written

$$m_{sp} = \frac{m_p}{Q_w} = \frac{\Delta \theta}{\eta_o H_o} \times 1000 \quad (3)$$

Fig. 1. Schematic of open-cycle power systems.

*For nomenclature see pp 16 and 17.

where $\Delta\theta$ is the time interval of the average heat-engine power duration, Q_w .

The effect of η_c on M_{sp} for the time interval of 1800 s is shown in Fig. 2. For the aforementioned working fluids, the enthalpy, defined as $H_o = \frac{\gamma R}{(\gamma-1)} T_o$, is shown in Fig. 3.^(4,5)

Associated with the storage of these fluids is a tankage and equipment mass, expressed by:

$$m_T = m_i + \frac{m_p}{\lambda}, \quad (3a)$$

where m_i is an inert mass and λ is a fluid volume factor. Based on studies for advanced rocket propulsion systems,^(2,6) these factors for the three working-fluid combinations are:

	m_i (kg)	λ
LH ₂	1151	0.882
LO ₂ /LH ₂	100	0.85
N ₂ O ₄ /MMH	65	0.86

For the liquid-hydrogen (LH₂) and liquid-oxygen/liquid-hydrogen (LO₂/LH₂) systems these factors include refrigeration-system masses for zero LH₂ boiloff in earth orbit.

The mass of the nuclear rocket propulsion reactor system including appropriate engine components is expressed by:⁽⁷⁾

Fig. 2. Effect of conversion efficiency (η_c) on working fluid specific mass.

$$m_{ne} = 1935 + 1005 (e^{-F/39340} - 1) + 0.02555 F, \quad (4)$$

where the "equivalent thrust" factor

$$F = \frac{2Q_w}{\sqrt{H_o} \eta_c} - 11120.$$

In addition, a neutron/gamma-ray shield must be included to reduce the internal heating of superconducting wires to $< 10^{-5}$ W/g. The mass of this shield is expressed by:⁽⁸⁾

$$m_{sld} = 500 + 2.36 (10)^{-6} \frac{Q_w}{\eta_c}. \quad (5)$$

For the chemical systems (LO₂/LH₂ and N₂O₄/MMH), the combustion-engine masses are included in the λ -factor for tankage and equipment mass described above.

For a Faraday-type MHD generator with hydrogen as the working fluid [seeded with 10.0 wt% cesium, at inlet conditions of 2700 K and $7.6 (10)^5$ N/m²], relationships for specific mass (based on calculations by L. E. Ring)⁽⁹⁾ and the conversion efficiency, are shown in Fig. 4. The specific mass includes the MHD generator, a superconducting magnet, a liquid-helium refrigeration system, and a 25% contingency for ducting, cabling, and supporting structure.

The development of the η_c function is presented in the appendix. Because the generator output voltage is less than 10 kVdc at power levels $< \sim 35$ MW_e, an inverter-transformer-rectifier system must be

Fig. 3. Stagnation enthalpy of working fluids.

Fig. 4. Specific mass and conversion efficiency for nuclear MHD system.

added to fulfill the load requirements. As shown in Fig. 4, this additional requirement significantly increases the specific mass of the MHD system at the lower power levels. The relationships for the conversion efficiency and specific mass follow:

$$\log_{10} (\eta_c) = -1.2541 + 0.32001 \log_{10} \left(\frac{Q_e}{10^6} \right) - 0.013882 \left(\log_{10} \left(\frac{Q_e}{10^6} \right) \right)^2; \quad (6)$$

at $Q_e > 3.5 (10)^7$ W (without power conditioning),

$$\log_{10} (M_s) = -0.1107 - 0.44481 \log_{10} \left(\frac{Q_e}{10^6} \right) + 0.075338 \left(\log_{10} \left(\frac{Q_e}{10^6} \right) \right)^2; \text{ and} \quad (7)$$

at $Q_e < 3.5 (10)^7$ W (with power conditioning),

$$\log_{10} (M_s) = 0.31387 - 0.16544 \log_{10} \left(\frac{Q_e}{10^6} \right) - 0.027854 \left(\log_{10} \left(\frac{Q_e}{10^6} \right) \right)^2 \quad (8)$$

where M_s is in kg/kW_e.

An approximate mass scaling law was developed for the turbine portion of the turbine-alternator heat engine.⁽¹⁰⁾ Because $Q_w = \dot{m}_p \Delta H_w$, there is an optimum ΔH_w providing a minimum turbine-plus-working fluid mass, i.e., $\dot{m}_p \propto 1/\eta_c$ and $\eta_c \propto \Delta H_w$. The development of the mass scaling law and the optimization procedure are described in the appendix.

The working-fluid properties for the LH₂, LO₂/LH₂, and N₂O₄/MMH turbine systems, derived from thermochemical⁽¹¹⁾ and rocket engine performance data⁽⁴⁾ are presented below:

Turbine System	1st Stage Inlet Temp., K	Gas Constant (R), J/kg·K	Heat Capacity Ratio (γ)	Mixture Mass Ratio
LH ₂	1350	4157	1.4	-
LO ₂ /LH ₂	1350	1569	1.3	3.35
N ₂ O ₄ /MMH	1350	569	1.25	0.7

For the alternator portion of the turbine-alternator heat engine, a mass scaling relationship was developed (see appendix) using as a reference a 4160-V, 5000-kVA three-phase ac machine with a superconducting rotor developed by Westinghouse Electric Corp.⁽¹²⁾ This relationship is represented by:

$$\log_{10} (M_a) = -0.25806 - 0.21120 \log_{10} \left(\frac{Q_e}{10^6} \right) - 0.014869 \left(\log_{10} \left(\frac{Q_e}{10^6} \right) \right)^2 \quad (9)$$

and includes the alternator with a minimum output voltage of 10 kV, a liquid-helium refrigeration system, and a rectifier for conversion to dc voltage.

Based on the foregoing mass relationships, total system masses were determined for electrical power levels in the range of 1.0 to 100 MW at voltages ≥ 10 kVdc. The following power-conversion efficiencies were used throughout the systems.

System	Thermal Power	Heat Engine Power		Load Power
		Q_w	Q_e	
MHD (> 35 MW)	Q_w/η_c	Q_e	Q_e	Q_e
(35 MW)	Q_w/η_c	Q_e	$Q_e/0.95$	Q_e
Turbine-Alternator	Q_w/η_c	$Q_e/0.98$	$Q_e/0.99$	Q_e

The system specific masses for a time interval of 1800 s are shown in Fig. 5, and system mass breakdowns for load power levels of 1.0, 10.0, and 100 MW are presented in Table I.

At operating times other than 1800 s, system masses are shown in Figs. 6, 7, and 8 for power levels of 1.0, 10.0, and 100 MW, respectively.

TABLE I
PERFORMANCE AND SPECIFIC MASS BREAKDOWN
OF OPEN-CYCLE POWER SYSTEMS

System Identification ^a	I			II			III			IV		
Load Power, MW _e	1.00	10.0	100	1.00	10.0	100	1.00	10.0	100	1.00	10.0	100
Heat engine output, MW _e	1.05	10.5	100	1.03	10.3	103	1.03	10.3	103	1.03	10.3	103
Thermal power, MW	18.6	91.8	467	3.19	31.9	240	3.07	30.7	232	2.48	24.8	248
Conversion efficiency (η_c)	0.0566	0.1144	0.2140	0.3230	0.3230	0.4297	0.3351	0.3351	0.4446	0.4155	0.4155	0.4155
Specific mass breakdown, kg/kW ^b												
Thermal system												
Working fluid	0.8635	0.4273	0.2285	0.2213	0.2213	0.1726	0.3865	0.3865	0.3111	0.7505	0.7505	0.7505
Storage tank ^c	0.1155	0.0572	0.0306	0.0296	0.0296	0.0231	0.0682	0.0682	0.0549	0.1222	0.1222	0.1222
Inert	1.151	0.1151	0.0115	1.151	0.1151	0.0115	0.100	0.0100	0.0010	0.0650	0.0065	0.00065
Nuclear engine	1.951	0.1967	0.0351	1.971	0.1935	0.0280	—	—	—	—	—	—
Nuclear shield	0.5428	0.0712	0.0157	0.5073	0.0575	0.0104	—	—	—	—	—	—
Heat engine												
MHD generator and equipment	0.7625	0.3261	0.2000	—	—	—	—	—	—	—	—	—
Turbine	—	—	—	0.3442	0.4439	0.1451	0.1518	0.1958	0.0881	0.1973	0.2544	0.0624
Alternator and equipment	—	—	—	0.5208	0.2972	0.1515	0.5208	0.2972	0.1515	0.5208	0.2972	0.1515
Power conditioning	1.28	0.979	—	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030	0.030
Total power system	6.67	2.17	0.521	4.78	1.39	0.572	1.26	0.988	0.637	1.69	1.46	1.12

^aI - LH₂/Nuclear/MHD; II - LH₂/Nuclear/Turbine-Alternator; III - LO₂/LH₂/Turbine-Alternator; IV - H₂O₂/MHD/Turbine-Alternator.

^bBased on load power at voltage ≥ 10 kVdc.

^cIncludes chemical reaction engine for Systems III and IV.

Fig. 5. Specific mass of open-cycle power systems for operating time of 1800 s at various power levels.

Fig. 6. Specific mass of 1.0 MW_e open-cycle power systems for various operating times.

Comparison with Other Power Systems

At power levels of up to ~ 10 MW_e, two other types of prospective systems could have masses less than 10^5 kg: (1) thermionic systems and (2) helium-cooled, closed-cycle nuclear MHD systems.

Estimates of thermionic power-system masses are shown in Figs. 6 and 7 for 1.0 and 10.0 MW_e,

respectively. These estimates may apply for maximum lifetimes of some tens of hours. Systems with longer lifetimes would be heavier by a factor of approximately four. A mass estimate for a 100-MW system was not made, but the specific mass would be about the same as that at the 10-MW level.

Also shown in Fig. 7, is the helium-cooled, closed-cycle nuclear MHD system described by Seikel

Fig. 7. Specific mass of 10 MW_e open-cycle power systems at various operating times.

and Nichols⁽³⁾. The mass of a closed-cycle nuclear MHD system, shown in Fig. 8, was estimated for a power level of 100 MW_e.⁽¹⁴⁾ The 100-MW system is based on an extrapolation of the 100-MW system with a modified nuclear-rocket reactor as the thermal energy source.

III. DISCUSSION OF RESULTS

The mass estimates for all power systems discussed in this study, including the thermionic and closed-cycle nuclear MHD systems, are based to some degree on an extrapolation of current technology, i.e., none of these systems could be assembled from "off-the-shelf" components. Therefore, one must be careful in drawing conclusions from the results presented in Table I and in Figs. 5 through 8. The following comments are offered on this basis for the mass estimates.

Mass estimates that are based on preliminary engineering designs developed in the U.S. advanced space propulsion program, which may be accurate to within ten percent, were derived for the following:

- The nuclear energy system including the reactor and nuclear engine components.
- Working-fluid storage systems including tanks, insulation, and support structure.
- Chemical energy systems including chemical-engine components.

The mass estimates of the following components are based on either experimental equipment or an extrapolation of size within a factor of ten from

Fig. 8. Specific mass of 100 MW_e open-cycle power systems at various operating times.

that of existing equipment or preliminary designs. These estimates may be accurate within ~ 50%.

- Turbines up to ~ 10 MW shaft power.
- Alternator (with superconducting rotor) up to ~ 10 MW_e output.

The mass estimates of the following components are based on either mass scaling laws derived from physical principles or on extrapolation of size greater than a factor of ten from that of existing equipment or preliminary designs. These estimates may be accurate within a factor of two.

- Turbines at > 10 MW shaft power
- Nuclear reactor shielding
- Superconducting alternators at > 10 MW_e output
- MHD generators and superconducting magnets
- Cryogenic refrigeration equipment
- Thermionic reactor systems
- Closed-cycle nuclear MHD systems
- Power conditioning equipment
- Contingencies associated with, e.g., integrated system support structure, ducting, control systems, and cabling.

With the above-stated reservations, the following conclusions may be drawn upon comparison of the results in Table I and Figs. 5 through 8.

Low Power Levels (up to ~ 10 MW_e)

1. At power levels less than ~ 5 MW_e and lifetimes up to a few thousand seconds, the chemical/turbine-alternator systems are clearly lighter than

the LH_2 /nuclear systems. This is ascribed primarily to the fact that the size of a nuclear reactor becomes criticality-limited at power levels less than $\sim 10 \text{ MW}_e$, as described by Eq. (4), and that its mass therefore is constant at this low power. Between the two chemical systems, the LO_2/LH_2 system is the lightest because of the higher hydrogen content in the working fluid.

2. A severe mass penalty is imposed on the LH_2 /nuclear/MHD system by the addition of power conditioning to raise the voltage to 10 kV, effectively weighing this system out of competition with the others. This mass penalty would also exist for MHD systems operating with chemical reaction products as the working fluid.

3. For operating times longer than $\sim 2 (10)^4 \text{ s}$, the thermionic power system would be lighter than the open-cycle systems. Although no clear choice between the thermionic and the closed-cycle nuclear MHD system is indicated, the former would probably be less expensive.

High Power Levels (10 to 100 MW_e)

1. In this power range the choice of the lightest system is even less clear, particularly at more than $\sim 35 \text{ MW}$ where power conditioning is not needed for the MHD system. At the lower end of this power range, the LO_2/LH_2 /turbine-alternator system may be preferable because of lower capital cost, but at some power less than 100 MW the launch

cost of an apparently heavier chemical system may be predominating and a nuclear system could become economically competitive.

2. If a nuclear system is cheaper, a choice between MHD and turbine-alternator is not conclusive, except, of course, where the MHD system is penalized with power-conditioning ($< 35 \text{ MW}_e$).

3. As in the low-power-level range, the open-cycle systems are lighter than the closed-cycle system for lifetimes up to $\sim 2 (10)^4 \text{ s}$. Also, the total mass of an open-cycle system would probably exceed 10^5 kg at a lifetime somewhat less than 10^4 s .

IV. CONCLUSIONS

The following conclusions may be drawn from the observations discussed above:

- Up to operating times of $\sim 10^4 \text{ s}$, the open-cycle systems are lighter than the thermionic or closed-cycle systems over the power range of 1.0 to 100 MW_e .

- At power levels of the order of 1.0 MW_e , the LO_2/LH_2 /turbine-alternator system appears to be a clear choice.

- At power levels of the order of 100 MW_e , the choice between nuclear-MHD, nuclear-turbine-alternator, and LO_2/LH_2 -turbine-alternator is not clear.

APPENDIX

DEVELOPMENT OF MASS SCALING LAWS

I. MHD GENERATOR SYSTEM

Basic Principles of MHD Generator Design

For one-dimensional flow of a perfect gas through a Faraday-type MHD duct, the conservation equations of energy, momentum, and mass, the equation of state, and Ohm's law are, respectively:

$$\bullet \rho u C_p \frac{dT}{dx} = jE \quad (\text{A1})$$

$$\bullet \frac{dP}{dx} = jB \quad (\text{A2})$$

$$\bullet \rho u A = \text{const} \quad (\text{A3})$$

$$\bullet \rho = P/RT \quad (\text{A4})$$

$$\bullet j = \kappa (uB - E). \quad (\text{A5})$$

By defining a turbine efficiency as

$$\eta_t = \frac{E}{uB} \quad (A6)$$

Equations A4 and A5 may be substituted in Eq. A1 to obtain

$$\frac{dx}{dT} = \frac{C_p P}{R \eta_t (1 - \eta_t) \kappa u B^2 T} \quad (A7)$$

Assuming that B^2/P is constant and the Mach number, defined as $N_M = u/\sqrt{\gamma RT}$, is constant throughout the duct length, then Eq. A7 may be integrated with $\kappa = f(T)$ to determine a duct length for a given temperature ratio.

By assuming that the practical maximum length-to-diameter ratio $(L/D) \approx 12$ (based on current MHD generator technology) and that the volume within the duct is approximated by

$$\frac{A_1 L}{3} \left(2 + \frac{A_2}{A_1} \right), \quad (A8)$$

the mass flow rate can be calculated from Eq. A3 by

$$\dot{m}_p = \frac{P_1 N_M A_1}{RT_1} \sqrt{\gamma RT_1} \quad (A9)$$

where the subscript 1 refers to the inlet conditions at the duct, and the subscript 2 refers to the duct exit conditions.

Therefore, for given inlet stagnation conditions, the duct length and corresponding mass flow rate can be calculated for an assumed temperature ratio, knowing the value $\kappa = f(T)$ for the given working fluid.

The power level, Q_w , is then

$$Q_e = \dot{m}_p C_p T_1 (1 - T_2/T_1). \quad (A10)$$

Calculations

With inlet stagnation conditions of

$$T_0 = 2700 \text{ K}$$

$$P_0 = 7.60 (10)^5 \text{ N/m}^2$$

$$\kappa_0 = 25 \text{ A/V}\cdot\text{m for 10 wt\% cesium}$$

$$H_0 = 4.1 (10)^7 \text{ J/kg for 10 wt\% cesium}$$

and the following assumptions,

$$\eta_t = 0.7 \text{ (const)}$$

$$\gamma = 1.25 \text{ (const) for 10 wt\% cesium}$$

$$N_M = 0.5 \text{ (const)}$$

$$R = 3743 \text{ J/kg}\cdot\text{K for 10 wt\% cesium}$$

$$B_1 = 5.0 \text{ T,}$$

various MHD generator parameters were calculated⁽⁹⁾ for temperature ratios of 0.95, 0.90, 0.85, and 0.80, as summarized in Table A-I.

TABLE A-I

CALCULATED MHD GENERATOR PARAMETERS

T_2/T_1	0.95	0.90	0.85	0.80
L (m)	1.07	2.61	4.87	8.26
$\dot{m}$ (kg/s)	0.55	2.69	7.46	16.62
A_2/A_1	1.41	2.03	2.98	4.47
A_1 (m ²)	0.00473	0.0231	0.0641	0.1429
Q_e (MW)	1.41	13.57	56.62	168.3
η_c	0.062	0.123	0.185	0.247
M_B (kg/kW _e)	0.665	0.305	0.24	0.19
Voltage (kV)	3.02	7.30	13.4	22.4

The specific masses include the generator, the superconducting magnet, and the liquid-helium refrigeration system. The generator masses are based on a representative, existing system and are proportional to the calculated duct volumes. Superconducting magnet and refrigeration system masses were taken from Stekly and Thome⁽¹⁵⁾ and from Stekly, et al.⁽¹⁶⁾

Equations (6) and (7) in the text of this report were derived by a curve fit to the calculated data in Table A-I with an extra 25% in mass added for ducting between the reactor and the generator and for other contingencies.

Power Conditioning

At generator output voltages less than 10 kV (at power levels less than ~ 35 MW), an inverter-rectifier system must be added to meet the load requirements. Estimates of inverter-rectifier system masses are as follows:⁽¹³⁾⁽¹⁷⁾

	Power Level, MW _e		
	1.0	10	100
Voltage ratio	4/1	2/1	2/1
Specific mass, kg/kW _e			
Transformer	0.38	0.24	0.16
Inverter/rectifier	0.29	0.25	0.15
Filters and control system	0.07	0.05	0.03
Cabling and hardware	0.55	0.44	0.30
Total specific mass, kg/kW _e	1.29	0.98	0.64

Equation (8) in the text of this report was derived from the sum of these masses and the generator system masses in Table A-I.

II. TURBINE-ALTERNATOR SYSTEM

Basic Principles of Turbine Design⁽¹⁰⁾

The turbine design is based on a four-stage set of blading with a constant mean blade radius (r_m) as shown in Fig. A-1.

If we define the nozzle gas velocity through the first stage as $u_n = \sqrt{2\Delta H_n}$, where ΔH_n is the enthalpy change through the first stage, then the

impinging (normal to blade motion) gas velocity at the first stage is

$$u_1 = u_n \cos \alpha = \sqrt{2 C_p T_1 \left[1 - (P_2/P_1)^{(\gamma-1)/\gamma} \right]} \cos \alpha \quad (A11)$$

where T_1 is the inlet gas temperature to the first stage, and P_2/P_1 is the stage exit-to-inlet pressure ratio. From the blade geometry in Fig. A-1, the flow area at the first stage is

$$A = 2\pi \left(\frac{\Delta r}{r_m} \right) r_m^2 (1-b) \quad (A12)$$

where b is a blockage factor represented by the frontal area of the blades. The gas density at the exit of the first stage is

$$\rho = \frac{P_1}{RT_1} \left(\frac{P_2}{P_1} \right)^{1/\gamma} \quad (A13)$$

The mass flow rate, expressed by the conservation equation, becomes the product of Eqs. A11, A12, and A13, or

$$\dot{m}_p = \frac{P_1}{RT_1} \left(\frac{P_2}{P_1} \right)^{1/\gamma} 2\pi r_m^2 \left(\frac{\Delta r}{r_m} \right) (1-b) \cos \alpha \sqrt{2 C_p T_1 \left[1 - (P_2/P_1)^{(\gamma-1)/\gamma} \right]} \quad (A14)$$

The enthalpy change through the four stages of the turbine set is

$$\Delta H_w = C_p (T_{ex} - T_1) \quad (A15)$$

where T_{ex} is the exit temperature and T_1 is the inlet temperature, and the temperature ratio across the four stages is

$$\frac{T_{ex}}{T_1} = 1 - \eta_t \left[1 - \left[(P_2/P_1)^4 \right]^{\frac{\gamma-1}{\gamma}} \right] \quad (A16)$$

where η_t is the turbine efficiency (per set of four stages), defined as $\Delta H(\text{actual})/\Delta H(\text{isentropic})$.

Then, the mass of housing to completely enclose the turbine set of four stages can be approximated by:

Fig. A-1. Design basis for four-stage turbine:
(a) velocity diagram; (b) blade geometry.

$$10 \left[2 + \left(\frac{P_2}{P_1} \right)^4 \right] \frac{P_1}{\sigma_m} \rho_m r_m^3 \quad (A17)$$

and the mass of the rotor can be approximated by:

$$3(2+b) \rho_m r_m^3, \quad (A18)$$

where σ_m is the stress and ρ_m the density of the structural material. Assuming a doubling of the housing mass to account for the gearing between turbine spools and other contingencies, and assuming that $\sigma_m = 10 P_1$ and $\rho_m = 8000 \text{ kg/m}^3$, the total mass of the four-stage turbine becomes

$$m_{\text{turb}} = 8000 \left[2 \left[2 + \left(\frac{P_2}{P_1} \right)^4 \right] + 3(2+b) \right] r_m^3. \quad (A19)$$

Turbine Design Constraints

Two constraints on stresses in the turbine rotor may be used to determine the stage pressure ratio and inlet pressure. The maximum disk tensile stress is determined by

$$\sigma_t = K \rho_m u_b^2 \quad (A20)$$

where u_b is the blade velocity and, for a hydrogen turbine wheel design, $K = 0.474$.⁽¹⁸⁾ By defining an optimum blade-to-nozzle velocity ratio of 0.5⁽¹⁰⁾ and by combining Eqs. A11 and A20, we obtain

$$P_2/P_1 = \left[1 - \frac{2\sigma_t}{K \rho_m C_p T_1} \right]^{\gamma/(\gamma-1)} \quad (A21)$$

Then assuming a maximum σ_t of $6.07 (10)^8 \text{ N/m}^2$ (88000 psi) with a ρ_m of 8000 kg/m^3 , the ratio P_2/P_1 (per stage) for any set of four stages is

$$P_2/P_1 = \left[1 - \frac{3.2 (10)^5}{C_p T_1} \right]^{\gamma/(\gamma-1)}. \quad (A22)$$

From Eq. A14 it is seen that $r_m \propto 1/\sqrt{P_1}$; therefore, the turbine mass from Eq. (A19) is proportional to $(1/\sqrt{P_1})^3$ so that, for minimum turbine mass, the inlet pressure should be maximized. The limiting value of P_1 can then be determined by the minimum r_m , which is determined by the allowable torsional stress:

$$\sigma_t = \frac{2}{\pi} \frac{Q_w r_m}{u_b r_s^3}, \quad (A23)$$

where r_s is the shaft radius. Assuming that the ratio of r_s to r_m remains constant as depicted in Fig. A-1, the mean blade radius of a turbine set of four stages can then be determined for a given shaft power level with the following parameters estimated from turbine design experience⁽¹⁰⁾:

$$\begin{aligned} \eta_t &= 0.85 \\ \Delta r/r_m &= 1/5 \\ b &= 1/5 \\ \alpha &= 1.22 \text{ rad.} \end{aligned}$$

For blade velocities, determined by Eqs. A20 and A21, at the first-stage inlet for shaft power levels ranging from 1.0 to 100 MW, the following relationship provides a reasonable although non-optimum torsional stress (Eq. A23) from the resulting r_m as a function of P_1 (Eq. A14)

$$P_1 = 6895 (90.91 + 9.091 (10)^{-6} Q_w). \quad (A24)$$

Turbine Working Fluid Mass Optimization

From Eqs. (1) and (3) in the main body of the report, the M_{sp} is inversely proportional to ΔH_w ; and from the previous development of turbine mass scaling, the mass of the turbine is proportional to ΔH_w . Therefore, an optimum ΔH_w exists for a minimum sum of working fluid plus turbine mass. A near-optimum condition was determined by the following procedure:

1. For a given shaft power level and inlet working-fluid temperature to the turbine system, the mass flow rates are calculated for combinations of 1, 2, 3, etc. turbine sets (each with four stages) by the following sequence:

- stage pressure ratio by Eq. A22;
- temperature ratio by Eq. A16;
- enthalpy change by Eq. A15;
- total enthalpy change (ΔH_{wt}) by the sum of enthalpy changes for each successive combination of turbine sets;
- mass flow rate by $\dot{m} = Q_w/\Delta H_{wt}$, where the inlet temperature to the next set is the exit temperature from the previous set.

2. Having determined the mass flow rates for the combinations of turbine sets, the mean blade radii for each set are calculated by Eq. A14, with the inlet pressure to the first set given by Eq. A24 and the inlet pressure to each successive set given by

$$P_1 = P_{1-1} (P_2/P_1)^4 \quad (A25)$$

where P_{1-1} is the inlet pressure to the previous set.

3. The masses of each of the combinations, calculated by Eq. A19, are added to the working-fluid mass, $m_p = \dot{m}_p \Delta\theta$, and the minimum sum of m_p and turbine combination mass becomes the "optimum".

Basic Principles of Alternator Design

The voltage generated in a stator winding of a synchronous alternator is given by:

$$\epsilon = NlB \times v, \quad (A26)$$

where B = magnetic field strength (Wb/m²)

N = number of turns/winding

v = velocity of B (m/s)

l = length of conductor (m).

Because $v = \omega r$, where ω is the speed of rotation, and assuming that l/r is a constant, the voltage for any alternator becomes

$$\epsilon = NB\omega r^2 \quad (A27)$$

where r becomes a characteristic dimension. For convenience we will assume that r in this case is the rotor radius. Assuming that the fractional power loss in the stator is constant, then

$$i^2 R_w \propto Q_e = \epsilon i \quad (A28)$$

where R_w is the resistance of the winding. Assuming that

$$R_w \propto Nl/r_w^2 \quad (A29)$$

and

$$N \propto r/r_w, \quad (A30)$$

then

$$N \propto \frac{\epsilon^{2/3} r^{1/3}}{Q_e^{1/3}} \quad (A31)$$

Substituting Eq. A31 in A27 and assuming that B is constant, the voltage scaling law is

$$\epsilon \propto \frac{\omega^3 r^7}{Q_e} \quad (A32)$$

Because the mass of the alternator is $\propto r^3$, the specific-mass scaling law is then

$$M_s \propto \frac{\epsilon}{\omega^{9/7} Q_e^{4/7}} \quad (A33)$$

Using an experimental Westinghouse Corp. machine with a superconducting rotor⁽¹²⁾ as a reference, the proportionality constant is

$$\frac{(4500 \text{ kg})(60 \text{ Hz})^{9/7} [5(10)^6 \text{ W}]^{4/7}}{(5000 \text{ KW})(4160 \text{ V})^{3/7}} = 3.29 (10)^4$$

and the specific-mass scaling law becomes

$$M_s = 3.29 (10)^4 \frac{\epsilon^{3/7}}{\omega^{9/7} Q_e^{4/7}} \quad (A34)$$

where M_s = specific mass, kg/kW_e

ϵ = generated voltage, V

f = ac frequency, Hz = $\omega/2\pi$

Q_e = generated power, W.

Values of M_s with f as constant parameter are shown in Figs. A-2, A-3, and A-4 for power levels of 1.0, 10, and 100 MW, respectively.

Alternator Design Constraints

The constraints on tensile and torsional stresses for the turbine rotor also apply to the alternator rotor. Because the speed of the alternator rotor will be matched as closely as possible to the rotor speed of the last turbine set, these constraints will apply to the alternator rotor radius. The tensile stress is then

$$\sigma_t \propto \omega^2 r^2 \quad (A35)$$

and the torsional stress is

$$\sigma_T \propto \frac{Q_w}{\omega^3} \quad (A36)$$

Again, scaling from the Westinghouse machine, the ratios of stresses are

Fig. A-2. Specific mass of 1.0-MW_e alternator scaled from Westinghouse reference machine⁽¹²⁾.

$$(\sigma/\sigma_R)_t = 3.19(10)^{-6} f^{8/7} [\epsilon Q_w]^{2/7} \quad (A37)$$

$$\text{and } (\sigma/\sigma_R)_t = 1.641(10)^{-3} \frac{Q_w^{4/7} f^{2/7}}{\epsilon^{3/7}} \quad (A38)$$

These relationships are plotted in Figs. A-5 and A-6, respectively. Because the stress levels in the Westinghouse reference machine are very low, of the order of 10^7 N/m^2 (1500 psi) for σ_t and $< 10^6 \text{ N/m}^2$ (150 psi) for σ_c , arbitrary limits of $(\sigma/\sigma_R)_t = 8$ and $(\sigma/\sigma_R)_t = 5$ were established for the alternator design in this study.

Fig. A-3. Specific mass of 10-MW_e alternator scaled from Westinghouse reference machine⁽¹²⁾.

Turbine-Alternator Matching

With a combination of the relationships developed above, the working fluid-turbine-alternator system mass could be optimized for any power level, but the complexity of such a calculation is beyond the scope of this study. Therefore, a simpler approach was taken by approximately matching the alternator speed to the last-stage turbine speed, then determining an alternator size (Figs. A-2, A-3, and A-4) for a minimum output voltage of 10 kV within the stress constraints of Figs. A-5 and A-6.

Fig. A-4. Specific mass of 100-MW_e alternator scaled from Westinghouse reference machine. (12)

Fig. A-5. Rotor tensile stress related to Westinghouse reference machine. (12)

Fig. A-6. Rotor torsion stress related to Westinghouse reference machine. (12)

The turbine conditions for this alternator sizing at power levels of 1.0, 10.0, and 100 MW_e are given in Table A-II. The stress-level constraints at these conditions are at the 100-MW_e power level, where the tensile stress (Fig. A-5) requires a maximum rotational speed of ~ 140 rev/s. The alternator speeds arbitrarily chosen to "match" the turbine speeds are 400, 220, 140 rev/s (Hz) for power levels of 1.0, 10.0, and 100 MW_e, respectively. The basic alternator specific masses from Figs. A-2, A-3, and A-4 for these power levels are then 0.287, 0.166, and 0.080 kg/kW_e, respectively.

However, we must add the masses of a liquid-helium refrigeration system to cool the superconducting rotor, a final gear box to match the turbine and alternator speeds, and a power-conditioning system to convert the output voltage from ac to dc. These additional masses are given in Table A-III. A curve fit to these data points is represented by Eq. (9) in the main body of this report.

TABLE A-II
CALCULATED TURBINE PARAMETERS

	Turbine Parameters for Various Working Fluids		
	<u>LH₂</u>	<u>LO₂/LH₂</u>	<u>N₂O₄/MMH</u>
<u>Load Power = 1.0 MW_e</u>			
No. of turbine sets	7	4	3
Mass flow rate, kg/s	0.1394	0.2526	0.4848
Turbine mass, kg	344	152	197
Last stage rotor speed, rev/s	554	561	436
<u>Load Power = 10.0 MW_e</u>			
No. of turbine sets	7	4	3
Mass flow rate, kg/s	1.394	2.526	4.848
Turbine mass, kg	4439	1958	2544
Last stage rotor speed, rev/s	236	239	186
<u>Load Power = 100 MW_e</u>			
No. of turbine sets	9	5	3
Mass flow rate, kg/s	10.87	20.33	48.48
Turbine mass, kg	14512	8812	6238
Last stage rotor speed, rev/s	152	138	138

TABLE A-III
ALTERNATOR SYSTEM SPECIFIC MASS BREAKDOWN

<u>Power Level, MW_e</u>	<u>Alternator</u>	<u>25% Contingency</u>	<u>Gear Box(18)</u>	<u>Refrig. System(15)</u>	<u>ac-dc Rectifier</u>	<u>Total</u>
1.0	0.287	0.072	0.128	0.025	0.03	0.548
10.0	0.166	0.044	0.071	0.015	0.03	0.326
100	0.080	0.020	0.042	0.009	0.03	0.181

NOMENCLATURE

A	area	m^2
b	blockage factor, defined by Equation (A12)	--
B	magnetic field strength	Wb/m^2
C_p	heat capacity at constant pressure	$J/kg \cdot K$
E	MHD duct electric field	V/m
e	electromotive force	V
f	rotation frequency	Hz
F	equivalent thrust factor, defined by Equation (4)	N
H	enthalpy	J/kg
H_o	stagnation enthalpy	J/kg
ΔH_n	enthalpy change in turbine 1st stage	J/kg
ΔH_w	enthalpy change in heat engine	J/kg
i	current	A
j	MHD duct transverse current	A/m^2
l	length defined by Equation (A26)	m
L	MHD duct length	m
m	mass	kg
m_i	inert mass	kg
m_{ne}	nuclear engine mass	kg
m_p	working fluid mass	kg
m_{sld}	nuclear shield mass	kg
m_T	tankage mass	kg
m_{turb}	turbine mass	kg
$\dot{m}_p$	working fluid flow rate	kg/s
M_s	specific mass	kg/kW_e
M_{sp}	working fluid specific mass	kg/kW
N	number of turns per winding in alternator stator	--
N_M	Mach number	--
P	pressure	Pa
P_i	turbine inlet pressure	Pa
Q_e	heat engine electrical power output	W
Q_l	load power	W
Q_t	thermal power	W
Q_w	heat engine power output	W
r	radius	m
r_m	turbine mean blade radius	m
r_s	turbine shaft radius	m
r_w	alternator stator winding wire radius	m
R	gas constant	$J/kg \cdot K$
R_w	alternator stator winding resistance	V/A
T	temperature	K
T_{ex}	turbine exit temperature	K
T_i	turbine inlet temperature	K
T_o	stagnation temperature	K

u	velocity	m/s
u_b	turbine blade velocity	m/s
u_i	turbine impinging gas velocity	m/s
u_n	turbine nozzle gas velocity	m/s
v	magnetic field velocity vector	m/s
x	axial distance through MHD duct	m
α	turbine nozzle angle	rad
γ	heat capacity ratio	--
η_c	conversion efficiency, defined by Equation (1)	--
η_t	turbine efficiency	--
ξ	total energy	J
κ	electrical conductivity	A/V·m
λ	tankage factor, defined by Equation (3)	--
ρ	density	kg/m ³
ρ_m	structural material density	kg/m ³
σ	stress	Pa
σ_m	structural material stress	Pa
σ_R	reference stress	Pa
σ_t	tensile stress	Pa
σ_τ	torsional stress	Pa
$\Delta\theta$	operating time	s
ω	rotation velocity	rad/s

REFERENCES

1. J. E. Boretz, Principal Investigator, "Dual Mode Nuclear Rocket Applications Study", TRW Systems Group, NASA/MFSC, Contract NAS8-28119, Final Report issued 30 Sept., 1972.
 2. Jet Propulsion Laboratory contributions to NASA/AEC Advanced Propulsion Comparison Study (1972).
 3. G. R. Seikel and L. D. Nichols, "The Potential of Nuclear MHD Electric Power Systems", AIAA Paper 71-638, presented at AIAA/SAE 7th Propulsion Joint Specialist Conference, Salt Lake City, Utah (June 1971).
 4. "Performance and Properties of Liquid Propellants", Aerojet-General Corporation (1961).
 5. O. A. Farmer, "Tables of Para-Hydrogen Data in Engineering Units from 36° to 5000°R at Pressures to 5000 psia", USAEC Report LA-3669, Los Alamos Scientific Laboratory (1967).
 6. "Nuclear Stage System Definition Study", McDonnell Douglas Astronautics Co., NASA/MFSC Contract NAS8-27951 (1972).
 7. Based on unpublished work by J. D. Balcomb, Los Alamos Scientific Laboratory (September 1972).
 8. Based on unpublished work by R. E. Malenfant, Los Alamos Scientific Laboratory (February 1973).
 9. Based on unpublished work by L. E. Ring, Arnold Engineering Development Center (February 1973).
 10. Based on unpublished work by D. L. Hanson, Los Alamos Scientific Laboratory (February 1973).
 11. D. R. Stull and H. Prophet, et. al., "JANAF Thermochemical Tables", Second Ed., NSRDS-NBS37, U.S. Dept. of Commerce (June 1971).
 12. E. F. McCann II and C. J. Mole, "Superconducting Electric Propulsion Systems for Advanced Ship Concepts", AIAA Paper 72-590, presented at AIAA/SNAME/USN Advanced Marine Vehicles Meeting, Annapolis, MD (July 1972).
 13. Based on unpublished work by T. G. Frank, Los Alamos Scientific Laboratory (February 1973).
 14. Based on unpublished work by H. J. Newman, Los Alamos Scientific Laboratory (April 1972).
 15. Z.J.J. Stekly and R. J. Thome, "Lightweight Superconducting MHD Magnets", Magnetic Corporation of America, Technical Report AFAPL-TR-72-32, Volume II, for Air Force Aero Propulsion Laboratory (June 1972).
 16. Z.J.J. Stekly, et. al., "Superconducting Saddle Magnet Design Considerations", presented at 12th Symposium Engineering Aspects of Magnetohydrodynamics, Argonne National Laboratory (March 1972).
 17. R. E. Stapleton, Los Alamos Scientific Laboratory, private communication (February 1973).
 18. D. L. Hanson and S. W. Moore, Los Alamos Scientific Laboratory, private communication (February 1973).
-