NASA CONTRACTOR REPORT

NASA CR-1994

PRELIMINARY DESIGN OF AN AUXILIARY POWER UNIT FOR THE SPACE SHUTTLE

Volume II — Component and System Configuration Screening Analysis

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Prepared by
AIRESEARCH MANUFACTURING COMPANY
Los Angeles, Calif.
for Lewis Research Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MAY 1972

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
CR-1994		

4. Title and Subtitle	5. Report Date
PRELIMINARY DESIGN OF AN AUXILIARY POWER UNIT FOR THE SPACE SHUTTLE. VOLUME II - COMPONENT AND SYSTEM CONFIGURATION SCREENING ANALYSIS	May 1972
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
W. L. Burriss and M. L. Hamilton	71-7300-2
	10. Work Unit No.
9. Performing Organization Name and Address	11. Contract or Grant No.
AiResearch Manufacturing Company Los Angeles, California	NAS3-14408
	13. Type of Report and Period Covered
12. Sponsoring Agency Name and Address	Contractor Report
National Aeronautics and Space Administration Washington, D. C. 20546	14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

The auxiliary power unit (APU) for the space shuttle is required to provide hydraulic and electrical power on board the booster and orbiter vehicles. Five systems and their associated components, which utilize hot gas turbines to supply horsepower at gearbox output pads, were studied. Hydrogen-oxygen and storable propellants were considered for the hot gas supply. All APU's were required to be self contained with respect to dissipating internally generated heat. These five systems were evaluated relative to a consistent criteria. The system supplied with high pressure gaseous hydrogen and oxygen was recommended as the best approach. It included a two-stage pressure-compounded partial-admission turbine, a propellant conditioning system with recuperation, a control system, and a gearbox. The gearbox output used was 240 HP. At the close of the study a 400 HP level was considered more appropriate for meeting the prime shuttle vehicle needs, and an in-depth analysis of the system at the 400 HP output level was recommended.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
Power	Electric power	Unclassified - unlimited
Flight control power	APU	
Turbomachinery	Hydraulic power	

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	228	$3.00

* For sale by the National Technical Information Service, Springfield, Virginia 22151

FOREWORD

This report is the second volume of a series that comprises the following:

- Volume I - Summary
- Volume II - Component and System Configuration Screening Analysis
- Volume III - Details of System Analysis, Engineering, and Design for Selected System
- Volume IV - Selected System Supporting Studies
- Volume V - Selected System Cycle Performance Data

This report summarizes the Phase I portion of the program, in which the various component and system concepts were compared and evaluated. Volumes III, IV, and V contain the Phase II work, in which preliminary design of the selected APU system concept was performed.

CONTENTS

SECTION I

INTRODUCTION AND SUMMARY

This report presents the results of the Phase I studies on Contract NAS3-14408, "Space Shuttle APU Study". The Space Shuttle APU will supply electric and hydraulic power for the Space Shuttle Booster and Orbiter vehicles, using a hot gas turbine with a working fluid provided by hydrogen-oxygen combustion or from storable propellants. The following systems were studied and evaluated in establishing a recommended system concept for Phase II of the program:

Low-Pressure Cryogenic H_2-O_2 Supplied System

High-Pressure Cryogenic H_2-O_2 Supplied System

High-Pressure Gaseous H_2-O_2 Supplied System

Dual-Mode Airbreathing/Cryogenic H_2-O_2 System

Monopropellant System

Sections 3 through 7 describe these systems and their performance. In Section 8, these systems are compared and evaluated. The high-pressure gaseous H_2-O_2 supplied system is the recommended approach. The gaseous propellants are to be supplied by another system (the Auxiliary Propulsion System), which has much greater total propellant requirements. It is also recommended that for optimum APU performance, the hydrogen and oxygen be delivered at the lowest temperature and highest pressure possible. These parameters will be determined by APS optimizations which have not been completed at the time of this report. At present, it appears probable that the APS propellant conditioning provisions will deliver propellant conditioning provisions will deliver propellant at pressures in excess of 650 psia and at temperatures in the range from 200 to 400^{o}R.

TURBINE

A two-stage pressure-compounded axial-flow impulse turbine has been selected for the Space Shuttle APU. Two-stage velocity-compounded turbines were considered in detail (Appendix A) and were eliminated on the basis of greater mechanical design problems for aerodynamic designs which would provide performance equivalent to the pressure-compounded turbines.

Figure I-I is an isometric drawing of the turbine gearbox configuration (less alternator and hydraulic pumps). Appendix B describes the turbine mechanical design.

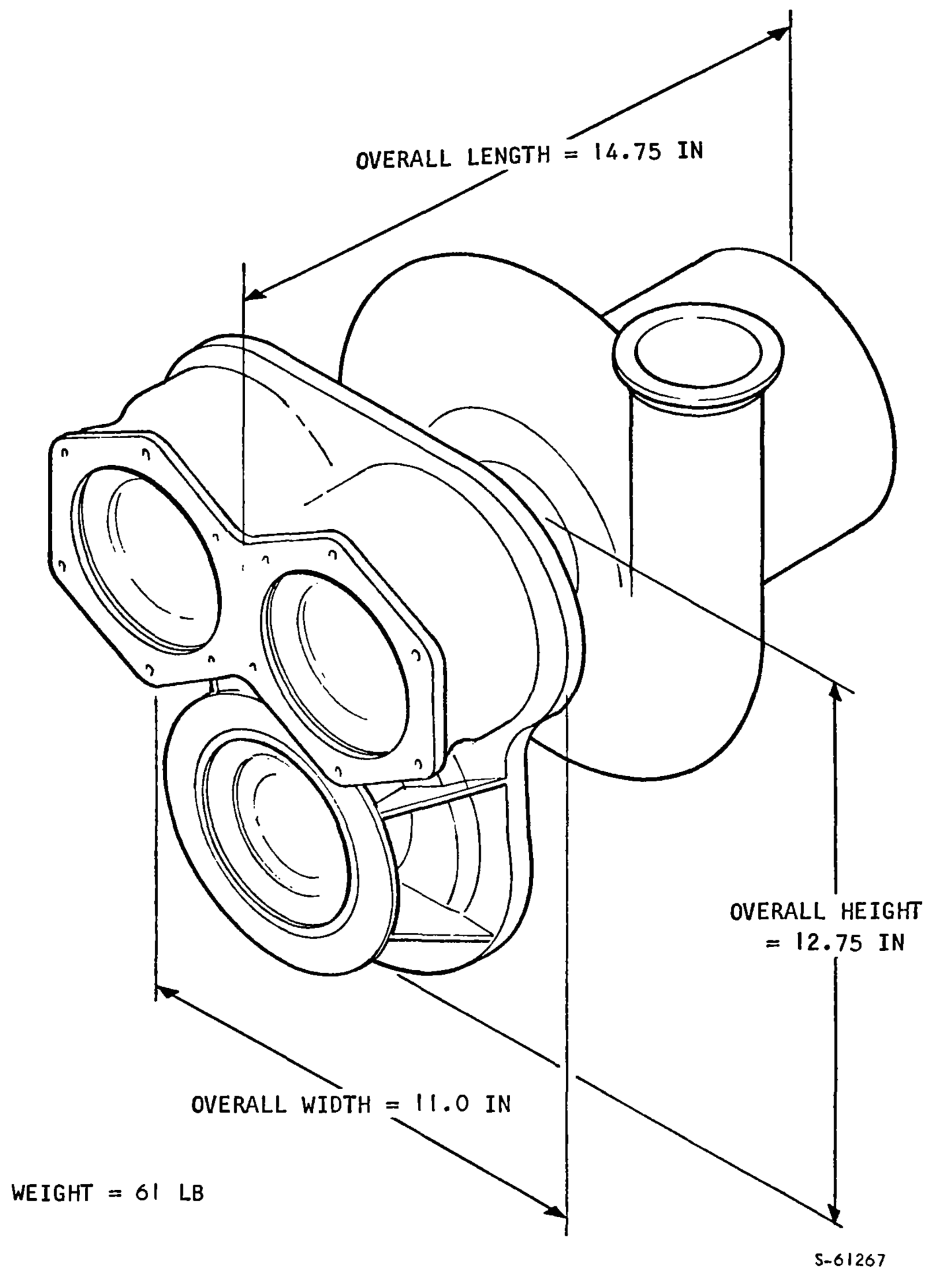


Figure 1-1. Turbine Gearbox Assembly (Less Generator and Hydraulic Pumps)

CONTROLS

Pressure-modulating or pulse-modulating controls can be used for turbine speed control. Table I-I compares the characteristics of these two types of control. The pressure modulating control has been assumed in the studies partially because its steady-state performance is readily predictable. The performance obtainable with pulse modulation control requires transient-state analysis. This will be performed early in Phase II to determine the incentive for this type of control.

TABLE 1-1

CONTROL CONCEPT SELECTION

Factor	Pulse Modulating Type Turbine Speed Control	Pressure Modulating Type Turbine Speed Control
Low-Power-Output Fuel Consumption	Superior for < ±3 percent speed control accuracy	Superior for > ±3 percent speed control accuracy
Performance Predictability	(1) Requires transient analysis (2) Filling and emptying losses uncertain	Highly predictable
Suitability for Variable-Delivery Hydraulic Pump Drive	Requires hydraulic accumulator (WT = 50-100 lb) for stability	Stable without accumulator
Suitability for Alternator Drive	Unsuited for alternator paralleling (required with parallel APU's on A-C bus)	Readily adapted to provisions required for load sharing
Operating Life	(1) Large number of cycles on control valves (2) Continuous Igniter operation required	No design parameters outside state-of-the-art limits for long life service
Turndown Ratio	Required turndown ratio (1.8:1) No problem	Required turndown ratio (10:1) Near state-of-the-are limit for control valves

SECTION 2

DESIGN CONSIDERATIONS

INTRODUCTION

Study Objectives

The primary study objectives are as follows:

(a) Evaluate candidate APU system configurations and select a preferred concept.

(b) Perform preliminary design analyses, engineering, and layouts of the selected APU system concept.

(c) Recommend areas requiring technology development to ensure APU availability for Shuttle program.

This report summarizes the results of the Phase I studies which were primarily concerned with the first task above. Emphasis was given in the Phase I studies to the factors and parameters greatly influencing system weight and system selection. This involved performance of turbine parametric studies (described in Appendix A) to establish the effects of various turbine design parameters and the optimum design approaches. The mechanical design studies (Appendix B) established practical design limits and the validity of the selected design for reliable long-life operation. Other components having major impact on system performance and weight include the cryogenic tankage (Appendix C) and the heat exchangers (Appendix D). The analytical methods used for cycle performance analysis and overall propellant consumption are given in Appendixes E and F. Appendix G shows a combustor design based on previous experience.

Section Contents

This section presents the analytical processes used to establish the performance of the various APU systems. It summarizes the data provided by the cycle performance program described in Appendix E. This program has been the primary evaluation tool used in the study. The section also presents the methods used to provide parametric data showing the relationship between the total APU energy output, the power output, and the system weight. Additionally, the system interfaces relating to satisfactory thermal performance of the hydraulic pump, the generator, and the APU lube system are analyzed to establish the component arrangement best fulfilling the thermal management requirement that all APU heat loads be absorbed within the APU system.

SUMMARY OF PHASE I

The Phase I work can be divided into two general categories, one covering the initial system studies during the first six weeks of the program, and a second covering the subsequent work with the five system concepts selected by NASA, using revised power/altitude profiles.

Initial Systems Studies

The work performed during this period was reviewed at a meeting at NASA-LeRC on October 15, 1970. Table 2-1 summarizes the initial study requirements and the implications to system design which resulted in the candidate system concepts shown in Figure 2-1. At this time, it was concluded that there was considerable incentive for use of recuperated hydrogen-oxygen systems. In addition, because of the relatively long portion of the original mission represented by atmospheric flight, an dual-mode airbreathing/propellant turbine system appeared to be strongly competitive. The other systems were found to be less attractive and competitive only under certain special conditions.

Final Candidate System Studies

The following five system configurations were selected by NASA for study during the last half of Phase I:

1. 600 psia H_2-O_2 system with propellants supplied as liquids at 35 psia
2. Optimum pressure H_2-O_2 system supplied from special supercritical cryogenic tankage.
3. H_2-O_2 system with propellants supplied at 300 psia, 500°R.
4. Dual-mode airbreathing gas turbine and H_2-O_2 turbine with supercritical propellant supply
5. Monopropellant (75% N_2H_4 -24% $N_2H_5NO_3$-1% H_2O) system.

Two-stage (pressure- or velocity-compounded) axial-flow propellant turbines were to be evaluated using pulse- or pressure-modulating speed controls. Consideration was given to the effect of variable O/F ratio in recuperated H_2-O_2 cycles, tankage weight penalties, etc. with an amphasis on parametric presentation of performance indicating sensitivity of system selection to various design requirements.

1. Power Profiles

NASA also supplied revised power-altitude profiles for the booster and orbiter vehicles. Tables 2-2 and 2-3 summarize these profiles. It should be noted that the detailed profiles supplied by NASA, contained in Appendix F, were used in estimating the propellant requirements for the booster and orbiter vehicle missions. As compared with the earlier power profile, the following can be noted:

(a) Booster mission is 177 min long at an average gearbox output power of 46.5 shp, with less than 20 min duration outside the atmosphere.

(b) Orbiter mission is 58.4 min total (with two phases separated by 7 to 30 days inactive storage in orbit) at an average 36 shp at the gearbox, with essentially all operation outside the atmosphere.

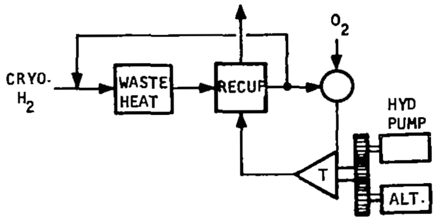
TABLE 2-1

INITIAL DESIGN REQUIREMENTS AND IMPLICATIONS

REQUIREMENT	IMPLICATIONS TO SYSTEM DESIGN
PROPELLANT SUPPLIES TO BE CONSIDERED • LOW-PRESSURE CRYOGENIC (SHARED) TANKAGE • HIGH-PRESSURE CRYOGENIC (SHARED) TANKAGE • LOW-PRESSURE GASEOUS SUPPLY • LOW-PRESSURE CRYOGENIC (SEPARATE) TANKAGE • HIGH-PRESSURE CRYOGENIC (SEPARATE) TANKAGE • STORABLE MONOPROPELLANTS AND BIPROPELLANTS	• IT WILL BE NECESSARY TO CONSIDER A VARIETY OF DIFFERENT CANDIDATE SYSTEM CONCEPTS TO ACCOMMODATE THESE POSSIBLE PROPELLANT SUPPLIES SINCE PROPELLANT SUPPLY SELECTION MUST BE PERFORMED ON A VEHICLE SYSTEM BASIS.
POWER-ALTITUDE PROFILE (see table below)	• THE APU WILL BE REQUIRED TO FUNCTION AT VARYING OUTPUT POWER LEVEL AND DISCHARGE PRESSURE. OFF DESIGN PERFORMANCE WILL BE AN IMPORTANT FACTOR. • 85 PERCENT OF MISSION IS ATMOSPHERIC FLIGHT, MAKING POSSIBLE USE OF AIRBREATHING GAS TURBINE OR RAM AIR HEAT SINK FOR PORTION OF MISSION.
HEAT SINK THE APU IS REQUIRED TO BE SELF-CONTAINED WITH RESPECT TO DISSIPATING INTERNALLY GENERATED HEAT.	• WITH CRYOGENIC SYSTEMS, THE WASTE HEAT CAN BE DISSIPATED IN THE PROPELLANT FLOW; WITH STORABLE PROPELLANTS, A SUPPLEMENTAL HEAT SINK (RAM AIR OR EVAPORANTS) WILL BE REQUIRED.

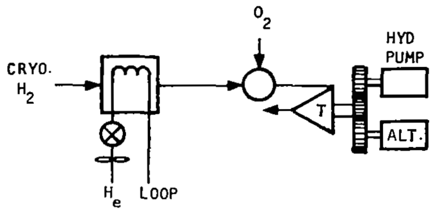
POWER-ALTITUDE PROFILE

POWER STATE	OUTPUT POWER	AMBIENT PRESS., PSIA	DURATION, MIN.
PEAK	100%	14.7	9
PEAK	100%	0	9
MODE	22%	10	144
IDLE	7	0	18



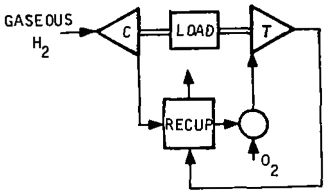
RECUPERATIVE H_2-O_2 SYSTEM

- HIGH-PRESSURE CYCLE
  - DIRECT FEED WITH HIGH-PRESSURE CRYOGENIC TANKAGE
  - WITH PUMPS FOR LOW-PRESSURE CRYOGENIC TANKAGE
- LOW-PRESSURE CYCLE
  - DIRECT FEED FOR LOW-PRESSURE CRYOGENIC TANKAGE



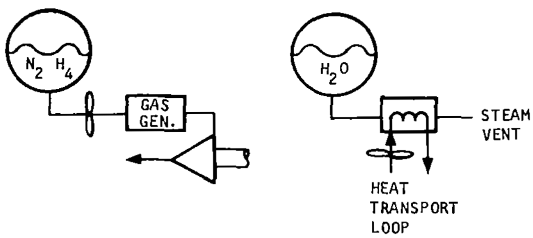
NONRECUPERATIVE H_2-O_2 SYSTEM

- H_e THERMAL TRANSPORT LOOP FOR APU COOLING



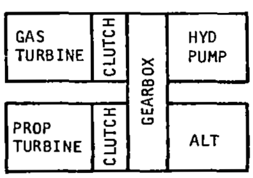
OPEN BRAYTON CYCLE SYSTEM

- DIRECT DRIVE CENTRIFUGAL COMPRESSOR
- GEARBOX DRIVE POSITIVE DISPLACEMENT COMPRESSOR
- SEPARATE TURBOCOMPRESSOR (2-SHAFT ENGINE)



MONOPROPELLANT SYSTEM

- THERMAL TRANSPORT LOOP FOR APU COOLING
  - EXPENDABLE EVAPORANT (H_2O) HEAT SINK
  - RAM AIR HEAT SINK (FOR ATMOSPHERIC FLIGHT)



DUAL-MODE AIRBREATHING/PROPELLANT TURBINE SYSTEM

- PROPELLANT TURBINE
  - MONOPROPELLANT
  - HYDROGEN-OXYGEN
- GAS TURBINE
  - JP FUEL
  - HYDROGEN
- HEAT SINKS
  - EXPENDABLE EVAPORANT (H_2O)
  - RAM AIR
  - FUEL (IF H_2 IS USED)

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Figure 2-1. Initial Candidate System Concepts

TABLE 2-2

BOOSTER VEHICLE MISSION PROFILE SUMMARY

Mission Phase	Turbine Gearbox SHP		Altitude	Duration, min.
	Average	Peak		
Preflight	70	112	S.L.	17
Boost	78	239	S.L. to 250 kft to 10 kft	20
Mode	39	45	10 kft	120
Landing	50	230	10 kft to S.L.	10
Postflight	66	-	S.L.	10
Total APU Operating Time			177 min.	
Average Power Level			46.5 SHP	

TABLE 2-3

ORBITER VEHICLE MISSION PROFILE SUMMARY

Mission Phase	Available Gearbox Output SHP		Altitude	Duration, sec.
	Average	Peak		
Ascent				
Boost	20.0	20	S.L. to 250 kft	250
Orbital injection	44.0	90	250 kft to orbit	250
Descent				
Deorbit	23.9	40	Orbit to 168 kft	1600
Reentry	60.4	100	168 kft to 54 kft	1000
Landing	28.5	200	54 kft to S.L.	400
Total APU Operating Time			58.4 min.	
Average Power Level			36 SHP	

2. APU Power Terminology

Throughout this report, data are displayed in terms of either gross turbine shaft power, net available output power (actual hydraulic and electric power, after accounting for the pump and generator power losses), and gearbox available output power. The different values have been used since data such as specific propellant consumption are usually presented in terms of gross turbine shaft power, whereas NASA has referenced data to available gearbox output power. In the case of a pumped system, where the propellant pumps are driven electrically using power from the generator, not all of the gearbox output power is available for supplying hydraulic and electric power to the vehicle, since some of the power is used for the APU pumps. Thus, the most convenient terminology is net available output power, which accounts for the propellant pumping energy required by the APU and for the power losses in the hydraulic pump and the generator.

3. Hardware Commonality

Based on the mission power profiles, it can be seen that a dual-mode system (one using an air-breathing engine at low altitudes, with either a hydrogen-oxygen or monopropellant APU for high altitudes) is attractive for the booster mission, but all of the orbiter mission must be supplied by a propellant turbine. The split in the orbiter mission will have an impact on the cryogenic storage vessel design. It can be concluded that there is enough similarity in general requirements to permit consideration of hardware commonality between the orbiter and booster vehicle systems.

SYSTEM ANALYSIS

During the last half of Phase I, primary emphasis was placed on determining the performance of the various candidate cycles as a function of their required output power and the ambient pressure. Discussion later in this section gives the logic leading to selection of the cycle configurations for the hydrogen-oxygen systems for which the cycle performance program was prepared. Prediction of the monopropellant system performance can be accomplished by a relatively simple process described in Section 7. Prediction of the dual-mode system performance can be made by combining the hydrogen-oxygen cycle performance program results with those from a similar program for a hydrogen-fueled gas turbine engine. This process is discussed in Section 6.

Cycle Performance Maps

The cycle performance program described in Appendix E generates overall system performance as a function of hydraulic and electric power output, and ambient pressure. It determines pressure losses in the system ducting, temperature levels at each point, flows, heat loads, etc., and performs the necessary iterations to determine turbine discharge pressure, cooling loop recycle flow, O/F ratio for desired turbine inlet temperature, power balance, etc. The program inputs, component performance maps, and output data are given in Appendix E. Typical state point data based on the program output data are shown in later sections of this report.

Vehicle Power/Altitude Analysis to Determine Propellant Consumption

The cycle performance program data can be plotted to establish the APU propellant consumption as a function of the power output and the ambient pressure. These maps, when combined with the specified booster and orbiter power/altitude/time data, can establish the overall propellant consumption for the vehicle mission. A mission power profile program, described in Appendix F, has been generated to accomplish this task. Typical output data, the vehicle mission profiles, and the input data are described in Appendix F. The results of this program allow the performance of various systems to be compared for the specific vehicle profiles of interest.

Primary System Tradeoff Variables

In generating the data for the cycle performance program component maps, it was possible in most cases to establish the optimum component configuration for the required function. Thus, for example, analyses described in Appendix A showed that selection of the pressure-compounded turbine was preferable to selection of a velocity-compounded turbine since the pressure-compounded unit shows slightly superior aerodynamic performance and greatly superior mechanical performance (or, if the velocity-compounded turbine is designed with short blades for equivalent mechanical performance, its aerodynamic performance is further degraded compared to that of the pressure-compounded unit).

1. Turbine Design Point Inlet Pressure/Power Level/ Discharge Pressure

However, in the case of the turbine, it was not possible to establish the optimum turbine design point based only on the turbine performance data. It becomes necessary to consider different turbine design points (both maximum inlet pressure (occurring at sea level full power output) and power output level/discharge pressure combination). Thus, a total of nine turbine designs were evaluated for the hydrogen-oxygen systems. These turbines had maximum inlet pressures ranging from 300 to 1500 psia and design points of either sea level/full power or altitude (10,000 ft)/mode power. Performance maps for each of these turbines are presented in Appendix A.

Evaluation of the two power level/discharge pressure points for the low-pressure liquid supply system described in Section 3 indicated that the altitude/mode power design point turbines show a performance advantage. This is due to the fact that most of the mission is at relatively low output powers at high altitudes. Consequently, the performance of the altitude/mode power turbines only has been evaluated for the other hydrogen-oxygen systems (Sections 4, 5, and 6).

It should be emphasized that evaluation of a single turbine for a single system configuration involves sequential use of four computer programs:

(a) A turbine design program is used to establish turbine geometry.

(b) The turbine geometry is used as an input to the turbine off-design program to establish a turbine performance map (which gives turbine efficiency as a function of power level, discharge pressure, and O/F ratio).

(c) The turbine performance map is then used in the cycle performance program (Appendix E) to establish propellant consumption as a function of output power and ambient pressure.

(d) The APU performance map is used in the mission integration program (Appendix F) to determine the propellant requirements for the booster and orbiter missions.

2. Propellant Inlet Thermodynamic State

A second major system variable is the thermodynamic state at which propellant is assumed to be available to the system. For the low-pressure liquid supply system of Section 3, the propellants are assumed to be available to the cryogenic pumps as saturated liquids at 35 psia -- thus, for that system, the tradeoff is related to the output head that the pumps must provide. However, for the high-pressure integrated tankage system of Section 4, the tradeoff involves the increasing tankage weight penalties at higher pressures vs the improved turbine performance at higher pressures. Finally, for the gaseous feed system described in Section 5, there is no optimum system weight based solely on the APU system performance. For the gas feed system, it is assumed that the propellant will be available to the APU at inlet states having a temperature from 200 to 500^{o}R and a pressure from 300 to 1250 psia. Based on APU system performance alone, the highest enthalpy and pressure head state, the 500^{o}R, 1250-psia inlet condition shows minimum propellant consumption. However, it is also necessary to consider the penalty occurred on-board the vehicle to provide propellant to the APU at this state. Including these conditioning penalties, the optimum propellant inlet condition occurs at low temperatures and reasonably high inlet pressures.

SYSTEM PARAMETRIC DATA

NASA has requested that parametric data be provided to show the APU system weight as a function of total energy output for systems designed for various combinations of peak power and mode power. The power combinations requested by NASA and the design system (base point for scaling) are presented in Table 2-4. The power ratings are in terms of available gearbox output power, not turbine shaft power.

TABLE 2-4

NASA REQUESTED APU SYSTEMS

	Design	NASA Requested APU Systems			
Peak power, hp	225	100	300	500	750
Mode power, hp	40	30	45	60	80
Turndown ratio*	5.63	3.33	6.67	8.33	9.38

*Turndown ratio is defined as the ratio of peak power to mode power.

In arriving at the final system weight vs energy curves for these APU systems, the method illustrated in Figure 2-2 was used. First, the same split between hydraulic and electrical loads was assumed as for the design case which results in the following system requirements:

	Design	NASA Requested APU Systems			
Peak power	225	100	300	500	750
Generator load, kw	20	9	27	45	67
Hydraulic load, gpm at 4000 psi	70	37	110	183	279

Then the systems fixed weights were calculated based on the peak power requirement (see tables in Sections 3 through 7). The propellant required for the design system (225/40 hp) as a function of energy was calculated next. Based on a constant system efficiency, the propellant weight increases linearly with energy required and the stored propellant weight is nearly linear, except for a slight variation in tank load factors. The propellant weight variation with power of the NASA requested APU systems was obtained using the design system mode power SPC variation with turndown ratio. The fixed weight and propellant weight for each NASA requested system was then added to get the final total weight vs energy parametric curves.

Figure 2-3 shows the hydrogen-oxygen SPC variation at peak power and at mode power for the NASA requested APU systems. The SPC is presented as a percentage of the 225/40 hp system SPC. The peak power SPC variation with peak power output was obtained from turbine designs using the design program. The mode power SPC was obtained from the 225/40 hp system performance map taking into account the efficiency and pressure ratio changes with turndown ratio and the scale effect of system size.

Figure 2-4 shows an equivalent curve for 75-24-1 monopropellant systems.

HYDROGEN-OXYGEN CYCLE THERMAL MANAGEMENT

Introduction

The following discussion examines the possible Space Shuttle APU system cycle configurations and establishes the logic leading to selection of the preferred component configuration. The discussion considers APU cycles using hydrogen and oxygen as the cycle energy source; it assumes these fluids are available at cryogenic energy levels (either low-pressure liquid or high-pressure, low-temperature fluids). The following topics are analyzed:

> Cycle Heat Sink -- Hydrogen, oxygen, or expendable evaporant are considered.
>
> Cycle Heat Generation -- Waste heat generated by each system component is analyzed.

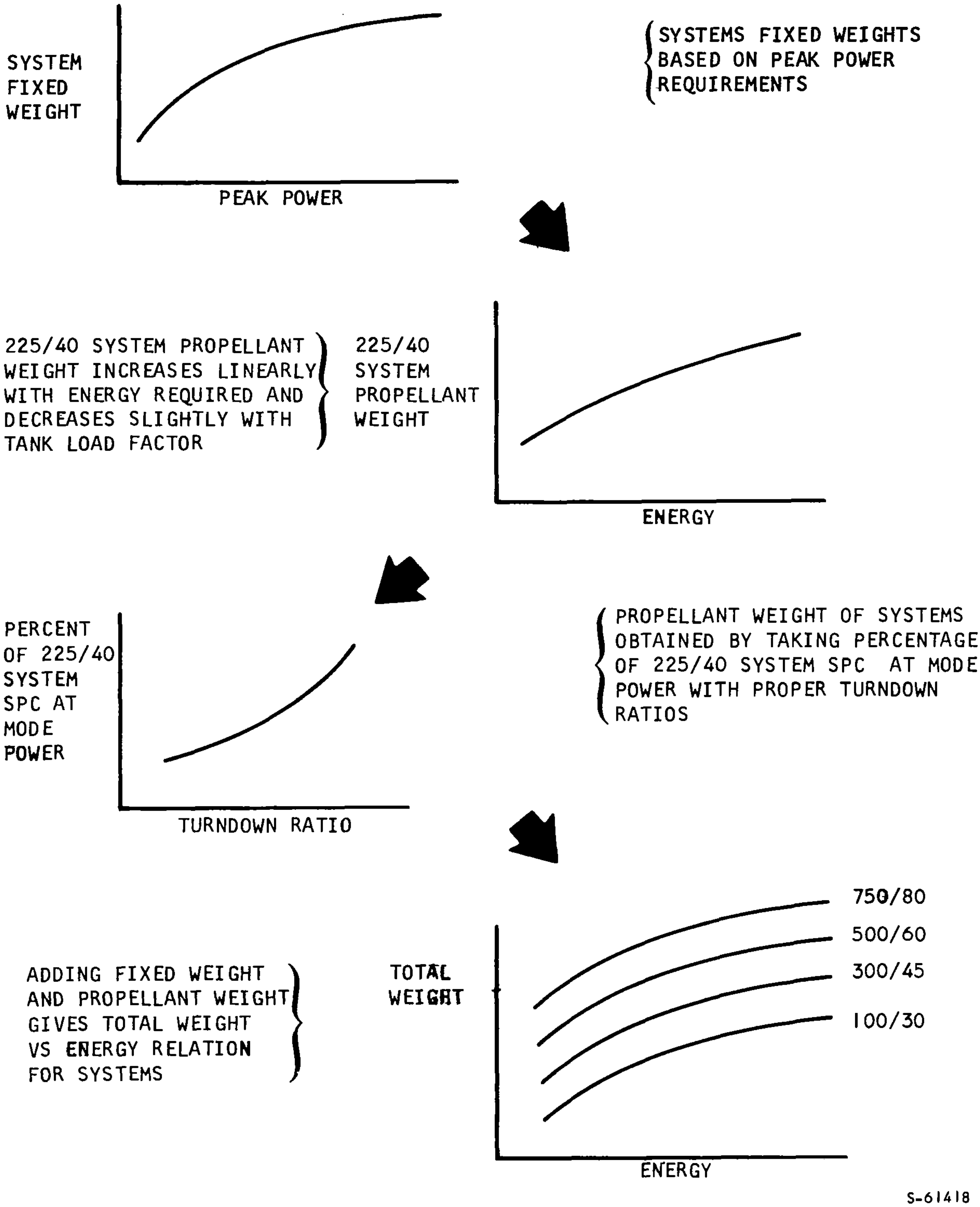

Figure 2-2. Method of Obtaining Total Weight vs Energy Relation for NASA Requested Systems

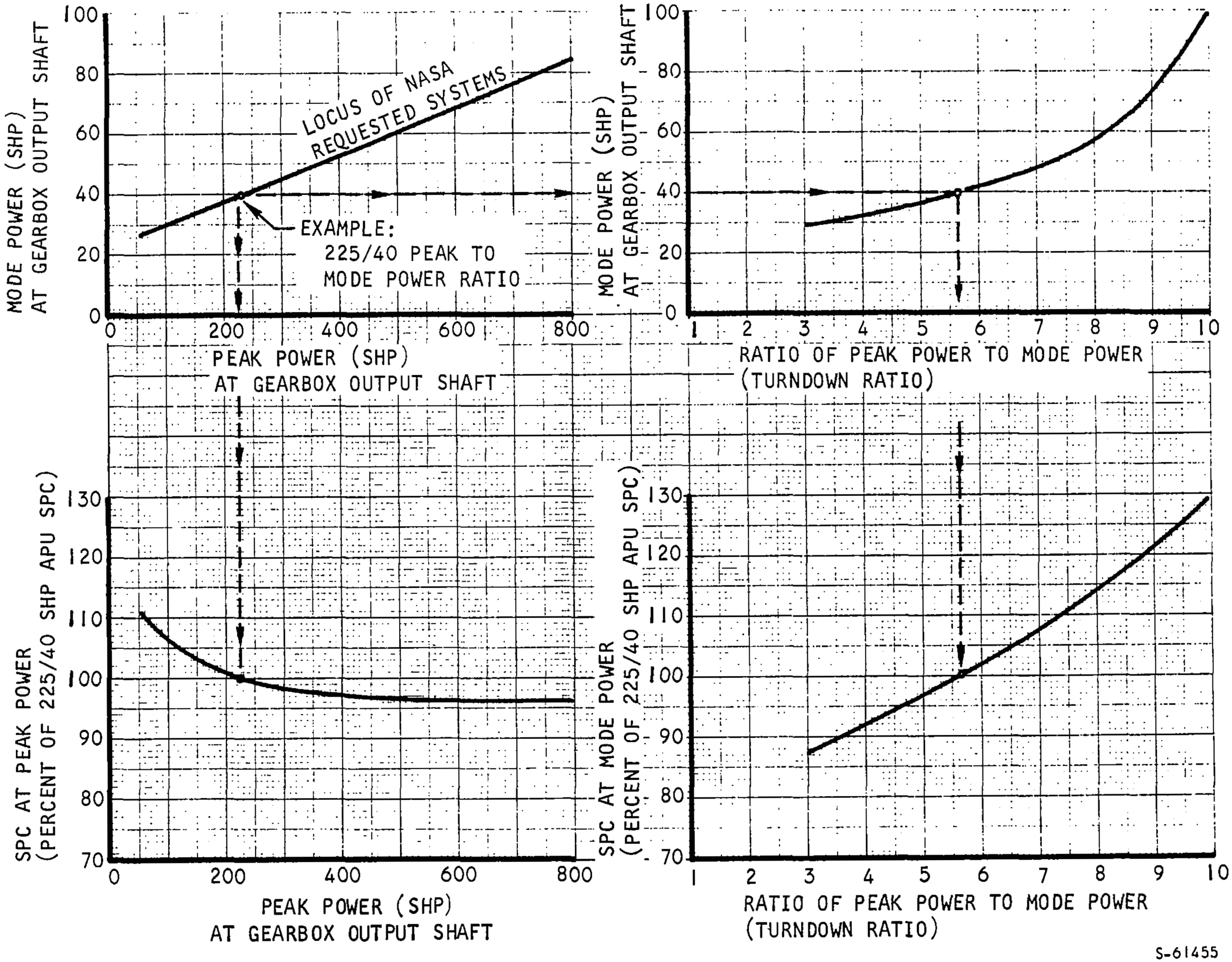

Figure 2-3. Hydrogen-Oxygen System SPC Variations at Peak Power and Mode Power for NASA Requested Systems

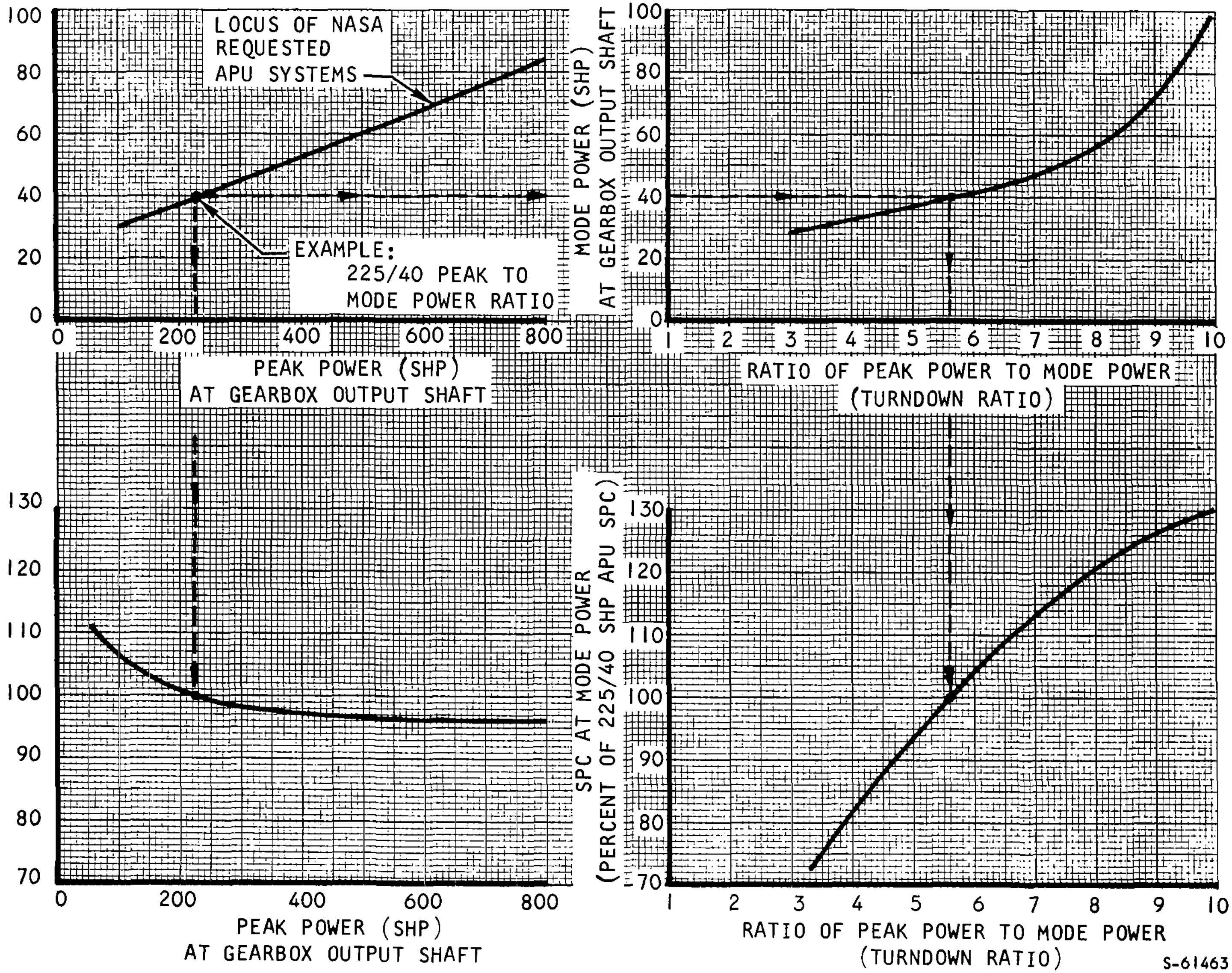

Figure 2-4. Monopropellant System SPC Variation at Peak Power and Mode Power for NASA Requested Systems

Heat Transfer Limitations - Limits on temperatures necessary for proper component operation are analyzed.

Cycle Configuration -- *Unrecuperated and recuperated cycles are* examined, a recuperated cycle is shown to be necessary for proper component cooling.

Performance Considerations -- Effect of recuperation on system performance is analyzed.

The analysis indicates that the component arrangements and thermal management concepts for the three hydrogen-oxygen systems of Sections 3 through 5 are nearly identical.

Requirements

The APU is required to provide both hydraulic and electric power at various output power levels and at various time periods throughout the booster and orbiter missions. The unit must have a restart capability since it may be shut down for prolonged intervals between uses. It also must be self-contained, relying on itself for component cooling. Thus, the APU must either use the hydrogen as a heat sink (the oxygen has insufficient heat capacity) or it must carry an expendable heat sink such as water.

Methods of Component Cooling

Studies to date indicate that at full load about 20 percent of the maximum turbine shaft power output will appear as heat losses in the various system components, such as the generator, hydraulic pump, gearbox, turbine, combustor, lube pump, and cryogenic fluid pumps (if used). Thus, for an APU developing 250 hp at the turbine output shaft, about 50 hp will appear as heat in the APU components. This heat load excludes any heat returned to the APU by the hydraulic fluid---under certain conditions, almost all of the hydraulic pump power output can be converted to heat that must be dissipated. At part loads the heat losses become far larger than the useful output power. At zero useful output power there is about 20 hp of heat generated in a 250-shp APU.

Therefore, for a 3-hr APU duty cycle a minimum of about 60 hp-hr of component heat must be dissipated. The two most likely methods of dissipating this heat are to use the cycle hydrogen flow as a heat sink (the oxygen has insufficient heat capacity) or to use an expendable evaporant such as water. Boiling water would absorb about 1100 Btu per lb water, requiring about 139 lb water per APU. Alternatively, adding the component heat to the cycle hydrogen flow would reduce the amount of oxygen required to raise the hydrogen temperature to the desired turbine inlet temperature. The effect of heat addition is therefore to change the O/F ratio at which the cycle is operated, while maintaining the total hydrogen flow almost constant. For a turbine inlet temperature of 2260^0R, the APU would require an O/F ratio of 1.30 if the hydrogen and oxygen were input to the combustor at an enthalpy corresponding to that of liquids. Adding component heat to the hydrogen would lower the cycle O/F ratio to about 1.22 assuming that 20 percent of the turbine work appears as waste heat. This O/F reduction is equivalent to a saving of about 24 lb of oxygen for an APU outputting a total energy of 300 hp-hr.

Consequently, because of its substantial weight advantage to the system, the concept of using hydrogen as the cycle heat sink is selected.

Component Operating Temperatures

The first step in establishing a cycle configuration is to investigate the heat sink temperatures required by the various cycle components. Figure 2-5 summarizes these limitations.

I. Maximum Acceptable Heat Sink Temperatures

Both the hydraulic oil and the lube oil (used to absorb gearbox heat losses) must be maintained at temperatures below about 300°F maximum and 450°F maximum at certain local points. Thus, these fluids must be cooled to levels below 300°F if they are to be capable of absorbing the component waste heat. In the case of the lube oil, there is a tradeoff between the oil flow rate and the temperature to which the oil must be cooled since the heat to be absorbed is independent of the oil flow rate.

For the hydraulic fluid, the fluid flow rate is fixed by the pump capabilities so that the heat sink temperatures can be determined uniquely. Assuming a 4000 psi hydraulic system in which all the hydraulic power (85 percent drive efficiency) appears as waste heat, the fluid must be cooled to about 265°F maximum if its return temperature to the pump is to be maintained below 300°F (additional heat input by vehicle structure might lower the maximum **cooled-fluid temperature somewhat). For the parameter studies in this report only 15 percent of the hydraulic output power is assumed to appear as waste heat**

An additional component requiring a relatively low heat sink temperature is the generator. Typically, aircraft generators are either oil or air cooled with a maximum heat rejection temperature of about 250°F. Thus, a logical arrangement of the hydraulic and oil heat loads is to place them after the generator heat load since the generator has a lower maximum heat rejection temperature.

2. Minimum Acceptable Heat Sink Temperatures

The minimum acceptable heat sink temperatures for both the hydraulic fluid and the lube oil are about 0°F since much lower temperatures cause an excessive increase in the fluid viscosity, making pumping increasingly difficult.

Although the other cycle heat loads, such as the turbine and combustor, do not have a definite minimum heat rejection temperature, large temperature differences between the coolant and the component are likely to cause thermal stress problems during the transients occuring on startup/shutdown and sudden load changes. Therefore, these components are ideally placed at a point in the cycle flow path where their inlet coolant temperature does not change suddenly. Such thermal buffering can be obtained by placing these loads after the hydraulic and lube oil heat exchangers which, because of the large quantities of fluid, have a stabilizing influence on the inlet hydrogen temperature.

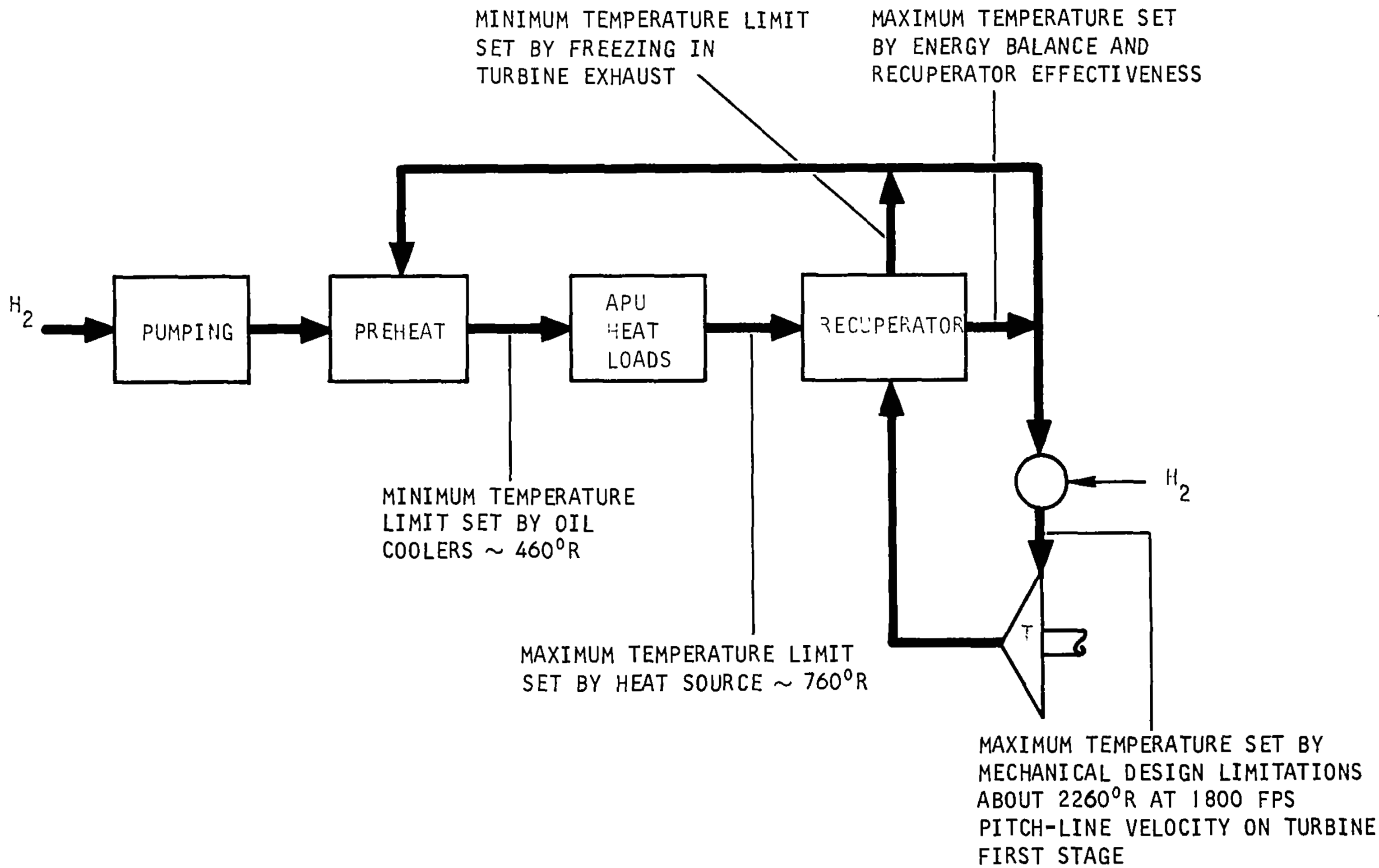

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Figure 2-5. H_2 - O_2 Cycle Component Temperature Limitations

Previous applicable combustor designs indicate that with a reasonable insulation thickness, the exposed insulation wall temperature can be maintained within safe limits and the energy loss from the combustor surface is small. External combustor cooling, thus, is not necessary.

3. Resulting Arrangement of Cycle Heat Loads

Based on the foregoing, the resulting arrangement of the cycle heat loads would be as follows:

Generator

Hydraulic fluid heat exchanger

Lube oil heat exchanger

Turbine cooling

Figure 2-6 shows the flow path for the hydrogen assuming that a high pressure is obtained by pumping low-pressure liquid.

Effect of Additional Heat Input

For cycle analysis studies, the heat loads shown in Figure 2-6 may be considered as a single large heat load. Thus, it is possible to determine the effect on cycle performance as a function of the magnitude of the heat input to the hydrogen flow prior to reaction in the combustor. It is additionally necessary to consider the temperatures at which such heat is available since a temperature differential is essential to heat transfer.

Figure 2-7 shows a simplified schematic of the APU cycle in which heat is added to the hydrogen flow. The resulting effect of such heat addition on the specific propellant consumption (SPC) at full output power is shown in Figure 2-8. The data indicate that optimum cycle performance is obtained when all of the component waste heat (including propellant pumping power in cycles using pumps) and all of the available energy in the turbine exhaust are used to preheat the propellant flow to the combustor. The data of Figure 2-8 show approximately a 20 percent reduction in SPC with full waste heat utilization.

1. Limitation on Amount of Recuperation

Figure 2-8 assumes that the turbine exhaust is cooled to 700^{0}R. Colder exhaust temperatures would cause condensation of the exhaust gas water; condensation in turn might lead to freezing which could block the turbine exhaust duct. If a large amount of waste heat is available upstream of the recuperator, the recuperator discharge temperature will be higher than 700^{0}R and full cycle recuperation will not be possible. However, data presented in Section 3 inidcate that the amount of recuperation is approximately the same at all power conditions, although at low power conditions the full heat recovery is not obtained -- there is considerably more waste heat available at low power conditions.

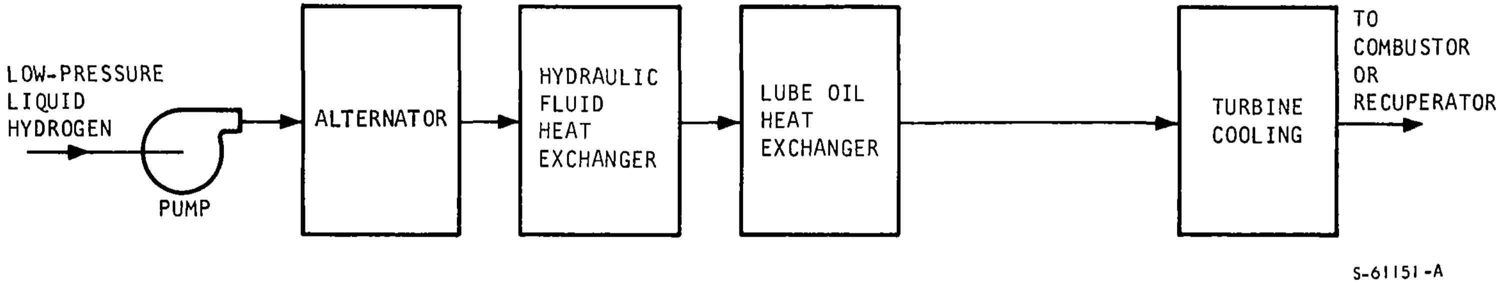

Figure 2-6. APU System Heat Load Arrangement

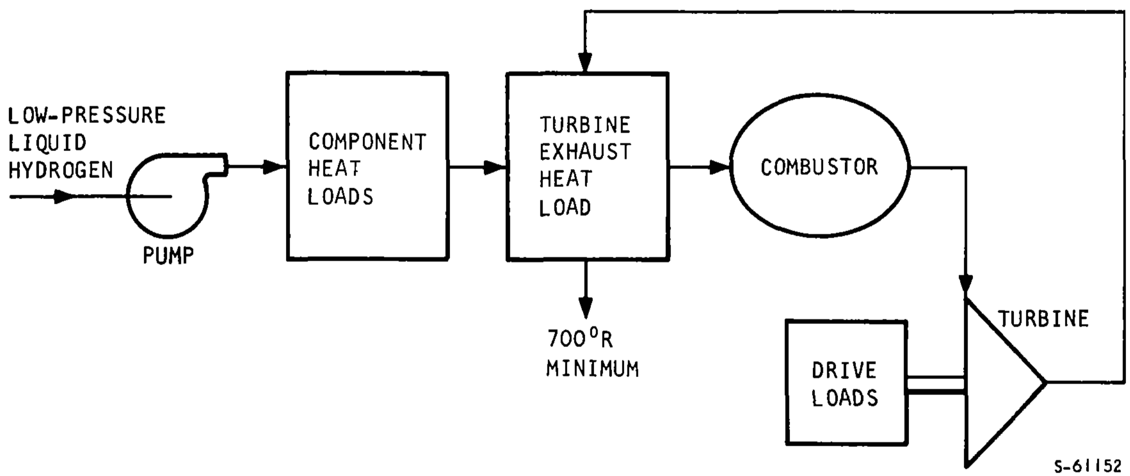

Figure 2-7. Simplified APU Cycle Schematic

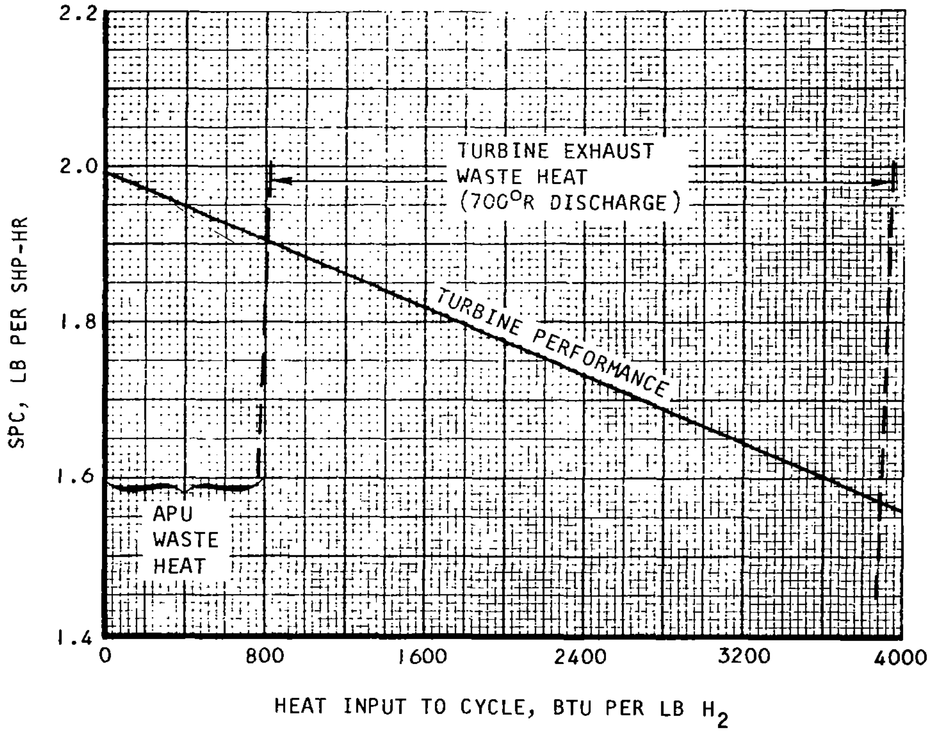

Figure 2-8. SPC vs Heat Added to Cycle Hydrogen Flow

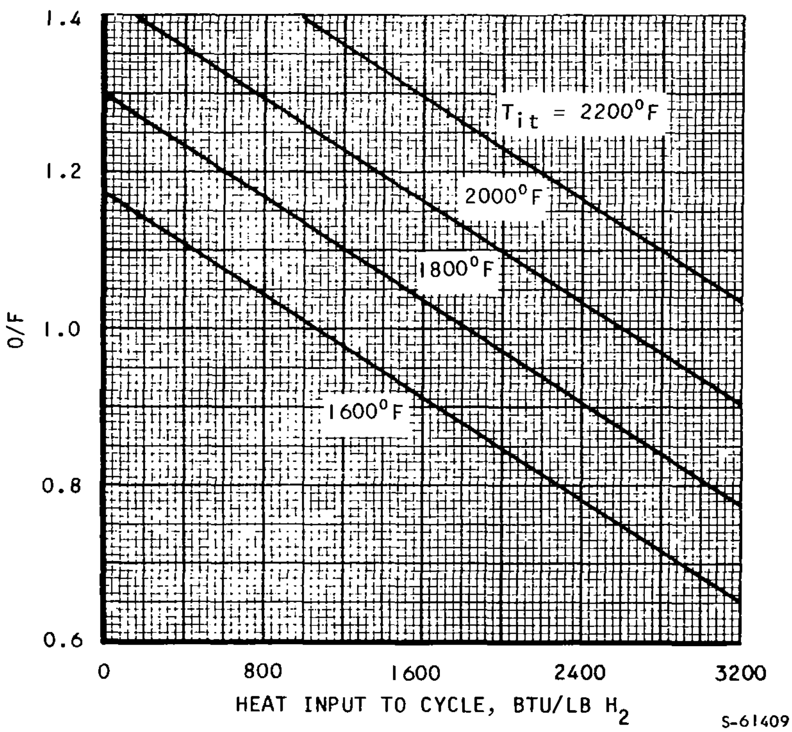

Figure 2-9. Cycle O/F Ratio vs Heat Added to Hydrogen Flow

2. Heat Addition vs O/F Ratio Relationship

Figure 2-9 shows the relationship between the O/F ratio and the heat input to the cycle. It indicates that increased heat inputs reduce the O/F ratio; however, the hydrogen flow remains almost constant. This is illustrated by the data of Figures 2-10 and 2-11 which are indicative of turbine part-load performance. Figure 2-10 shows the turbine SPC vs O/F ratio -- these data have been used to generate Figure 2-11 which shows the hydrogen and oxygen mass flows as a function of the cycle O/F ratio at this part-power point. As with the full-power point, these part-power data show that there is a strong incentive to reduce the system SPC by maximizing the recovery of turbine exhaust gas heat. It should be noted that an additional component, a hydrogen recuperator, is required in order to transfer heat from the turbine exhaust gas to the cycle hydrogen flow. However, this component weighs only about 16 lb, and as discussed below, will be essential to the cycle for reasons other than SPC optimization.

Control of Component Operating Temperatures

Although the earlier discussion has established the preferred arrangement of the component heat loads, it remains to check that the heat capacity of the hydrogen flow is compatible with the various component heat loads. Preliminary studies indicate the component heat loads at full power output are approximately as follows:

Generator = 179 Btu/min (at 10 hp output)

Hydraulic pump = 1360 Btu/min

Hydraulic system = 1270 Btu/min (heat added to fluid by hydraulic loads)

APU lube system = 567 Btu/min (lube pump and gearbox losses)

Turbine cooling = 771 Btu/min

Also, at sea level full power, the hydrogen flow into the combustor is about 4.12 lb/min. Assuming the component arrangement shown in Figure 2-6, then the hydrogen temperature at the inlet to the hydraulic fluid heat exchanger would be 64°R. This temperature is well below the minimum desirable temperature of 460°R at the hydraulic fluid heat exchanger. Alternatively, rearranging the components so that the turbine heat load is placed in front of the hydraulic heat exchanger, the hydrogen temperature is still only 118°R. Therefore, to meet the temperature limitations imposed by the hydraulic fluid and the lube oil, it is necessary to modify the system of Figure 2-7 into several possible arrangements:

Recycling a portion of the hot hydrogen flow to raise the temperature (Figure 2-12)

Preheating the cold hydrogen with part of the turbine exhaust heat load (Figure 2-13).

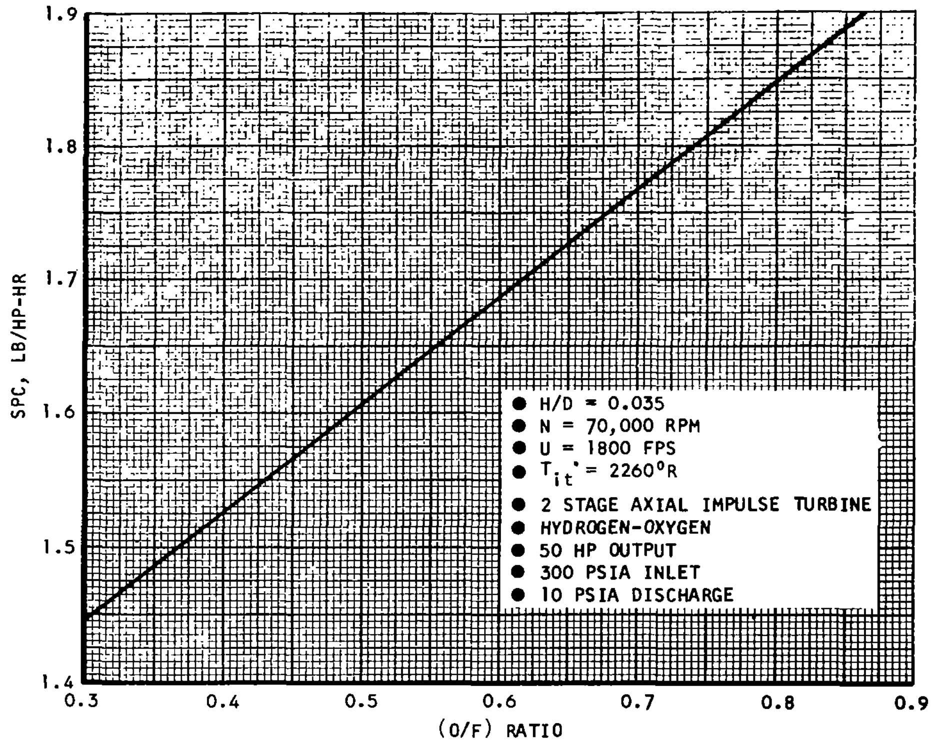

Figure 2-10. SPC vs (O/F) Ratio.

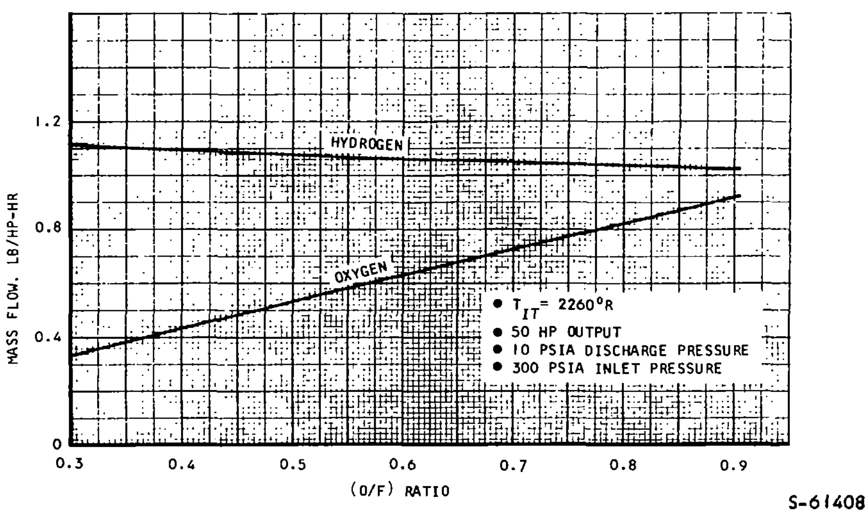

Figure 2-11. Hydrogen and Oxygen Flows vs (O/F) Ratio

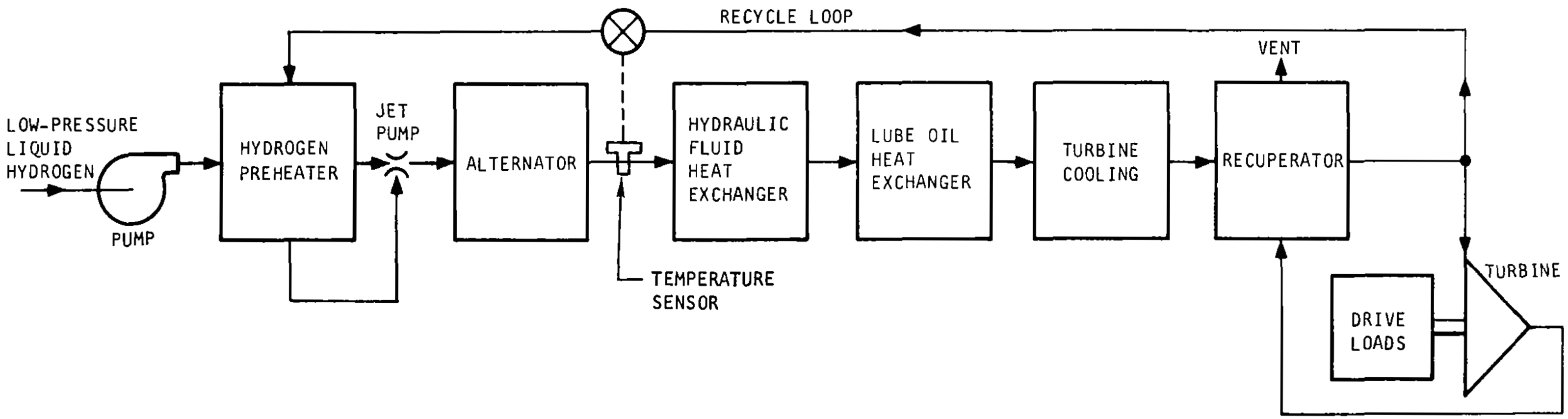

Figure 2-12. Recycle Loop for Component Temperature Control

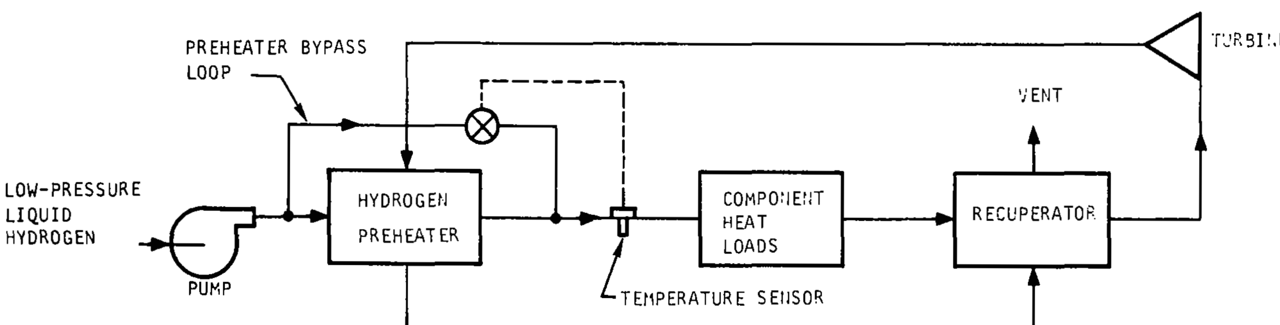

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Figure 2-13. Recuperative Preheater System for Component Temperature Control

Preheating the cold hydrogen with a regenerative recuperator system (Figure 2-14)

Using an intermediate heat transfer loop for component cooling (Figure 2-15)

The complexity and weight of the intermediate transfer loop system make it noncompetitive with the other systems. Table 2-5 compares the other three systems.

This comparison is based on full power output and placing the turbine heat load in front of the hydraulic heat exchanger. It assumes the hydraulic and the lube oil heat exchanger weights are same for all concepts. The recycling system with recuperator and the regenerative recuperator system have equal weights. A more detailed comparison including off design performance, transient conditions, and control methods will be required for final selection of the prefered concept. However, the recycle system with recuperator (Figure 2-12) is selected as the base line system because it is known to work for off-design conditions while the performance of the other concept (Figure 2-14) is unknown at present.

Conclusions

It can be concluded that the optimum APU cycle is one using hydrogen as the heat sink for the component heat loads. The arrangement of heat loads as shown in Figure 2-12 meets the temperature limitations imposed by the hydraulic fluid and the lube oil. Such a cycle must be recuperated in order to meet these component temperature requirements if the propellant is supplied to the APU at temperatures below $400^{0}R$. Additionally, this recuperation provides a substantial improvement in cycle performance, primarily through reduction of the oxygen required by the APU.

The cycle component arrangement is applicable to all three of the hydrogen-oxygen systems considered during the second half of Phase I. Only when the incoming propellant is supplied at a temperature exceeding $400^{0}R$ is it possible to operate the cycle without recuperation, while still maintaining acceptable component operating temperatures over the entire operating regime. This conclusion is explained in detail in Section 5.

However, when the propellant is supplied at a high temperature, then at part-power operation at low ambient pressures, there is insufficient heat capacity available in the hydrogen flow to adequately cool the components. Consequently, at these high propellant inlet temperatures, it becomes necessary to supplement the hydrogen cooling with water boiling. This concept is described in Section 5.

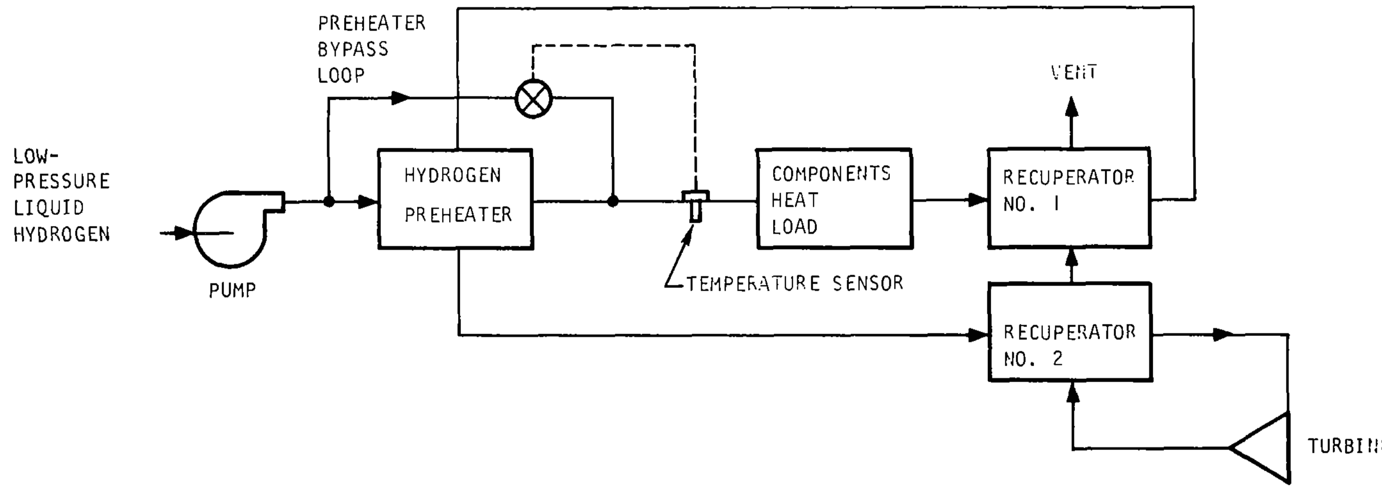

Figure 2-14. Regenerative Recuperator System for Component Temperature Control

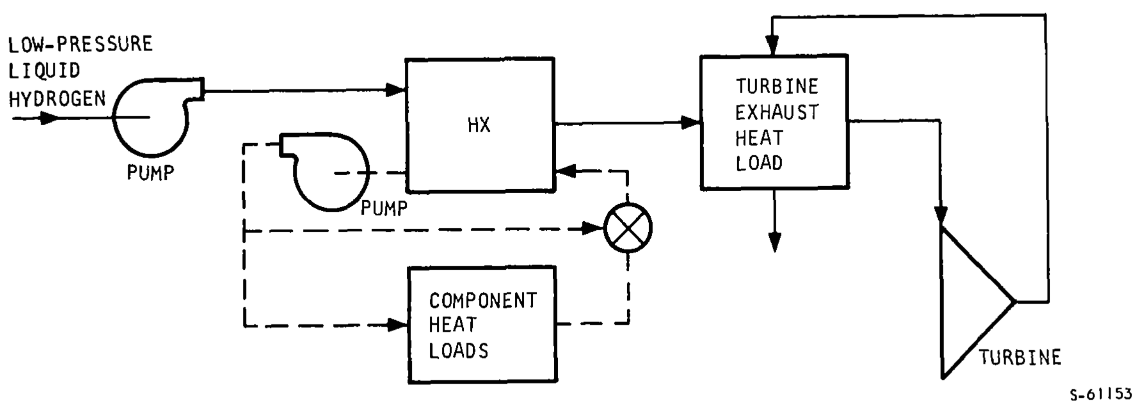

Figure 2-15. Intermediate Heat Transfer Loop for Component Temperature Control

TABLE 2-5

COMPARISON OF RECYCLE SYSTEM, RECUPERATIVE PREHEATER SYSTEM AND REGENERATIVE RECUPERATOR SYSTEM

	Recycle System with Recuperator	Recuperative Preheater System	Regenerative Recuperator System
Estimated weights for weight comparison (recuperator, H_2 preheater, and required extra components)	23.3 lb	45.0 lb	24.0 lb
Advantage	• Low system weight • Known off-design performance	• Simple system • No ejector, no recycle loop needed • Easy control	• Simple system • No ejector, no recycle loop needed • Easy control • Low system weight
Disadvantage	• Ejector needed for recycle loop • Larger piping needed for recycle loop • Higher chance of condensing problem in recuperator	• Relatively higher in weight • Off-design performance unknown	• Off-design performance unknown
Comments	• Good design • Known off-design performance • Selected as the baseline system	• Simple system • System weight too high, no further consideration is recommended	• Simple System • Possible weight reduction can be qchieved by further improvement in recuperator and preheater design • Capability of off-design performance is unknown, more study is needed for the system

SECTION 3

LOW-PRESSURE CRYOGENIC H_2-O_2 SUPPLIED SYSTEM WITH PROPELLANT PUMPS

INTRODUCTION

H_2 and O_2 are supplied to this system as low-pressure (35 psia) cryogens. The APU system incorporates pumping provisions to deliver the propellants at a higher pressure level for efficient cycle performance. Since the propellants enter the system at a low temperature, propellant thermal conditioning can be provided by a recuperative cycle which dissipates APU waste heat and provides high cycle efficiency.

General Description

In the system schematic shown in Figure 3-1, the propellant pumps are driven by electric motors. This approach was selected over other approaches (gearbox drive or hydraulic motor drive) for simplicity in the startup procedure and for flexibility of installation (the APU may be located a considerable distance from the propellant tankage). After passing through the pump and the delivery line to the APU, the hydrogen is preheated by hot recycle hydrogen gas. For optimum jet pump performance the temperatures of the incoming and recycle hydrogen flows should be approximately equal; hence, it is necessary to use a hydrogen preheater to assure temperature equilibration. The preheated and recycled hydrogen flows are mixed to provide a suitable flow and temperature level for cooling of the generator, hydraulic fluid, gearbox lubricant, and turbine housing. After passing through the recuperator, the hydrogen flow is split, a portion flowing to the combustor, the remainder being recirculated in the thermal loop by the jet pump. The oxygen flow is preheated by the hydrogen before passing into the combustor which produces the hot high-pressure working fluid for the turbine. The turbine exhaust gas passes through the recuperator before being dumped overboard.

DESIGN POINT SELECTION

The primary variables establishing the design point for this system are

Turbine inlet pressure

Turbine design point (power level and discharge pressure)

Selection of the optimum turbine inlet pressure is dependent on the cycle thermal balance. Optimum cycle performance is obtained where all of the component waste heat (including propellant pumping power) and all of the available energy in the turbine exhaust are used to preheat the propellant flow to the combustor. To prevent water condensation, which might in turn lead to freezing, the turbine exhaust recuperator must have a discharge temperature of about 700°R minimum. However, when there is a large amount of waste heat available upstream of the recuperator, the discharge temperature will be higher than 700°R, and full cycle recuperation will not be possible. Thus, the turbine inlet pressure selection involves trading-off the improvement in turbine performance with increasing pressure against the possible degradation

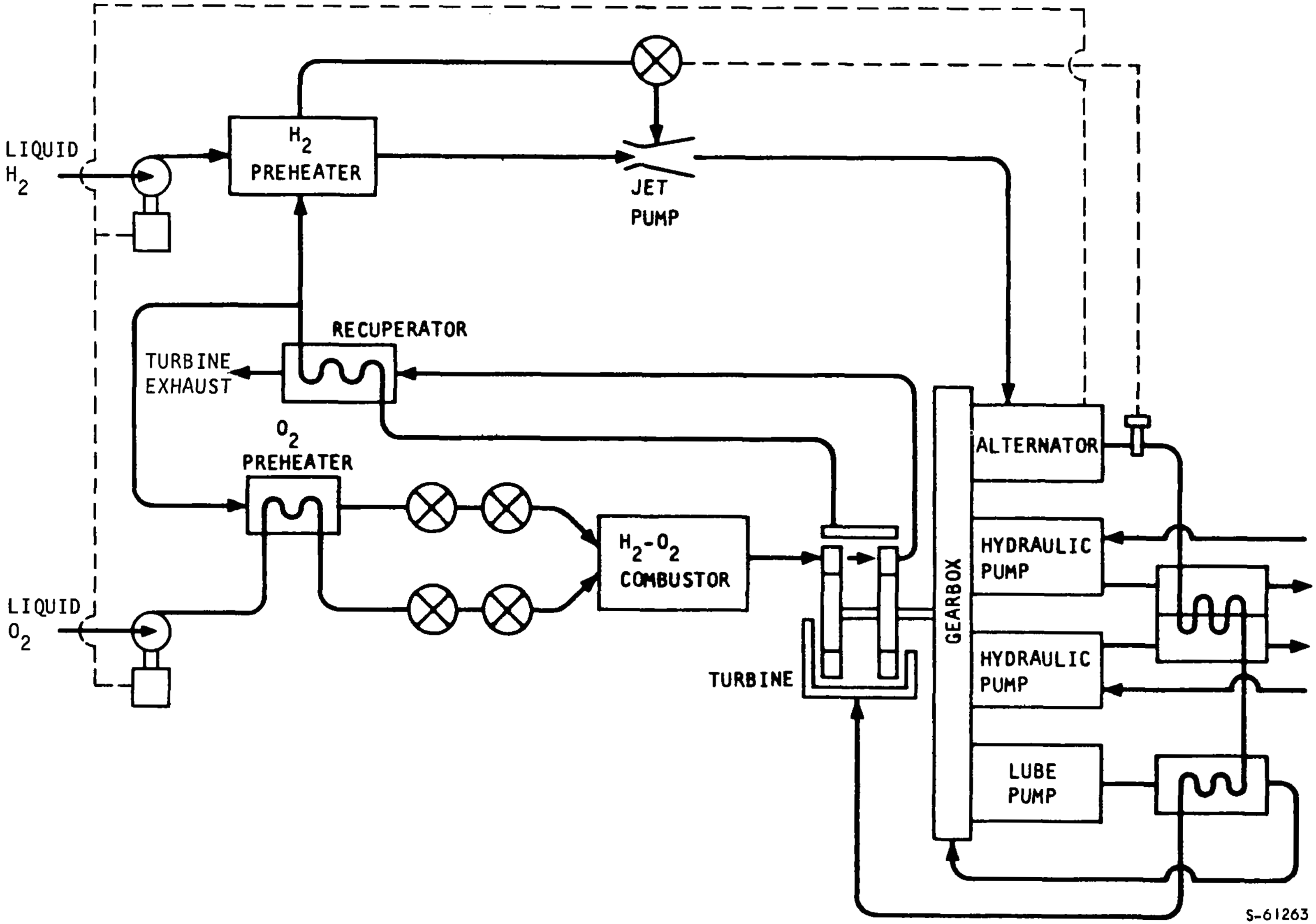

Figure 3-1. Low Pressure Cryogenic H_2 - O_2 Supplied System

in propellant conditioning performance due to generation of higher waste heat loads (as a result of increased propellant pumping power) with increasing pressure.

The turbine design point selection involves comparing system performance using different turbine performance maps representative of different design conditions. For this system, a total of nine turbine designs were evaluated. These turbines were designed at either sea level, full power, or at altitude, mode power (which represents the bulk of the system output energy). Their inlet pressures span the range from 300 to 1500 psia.

Figures 3-2 through 3-10 show the APU system propellant consumption in terms of the net output power and the ambient pressure for each of the nine turbines considered. The step increase in propellant consumption is due to pressurization of the second hydraulic pump for loads above about 95 hp. Using these data, the mission propellant requirement program described in Appendix F was used to establish the total propellant required for the booster and orbiter missions. These data are shown in Table 3-1 and Figures 3-11 and 3-12. From these data, the following conclusions can be drawn:

> For the booster mission, there is incentive to design the turbine for altitude mode-power condition. For the orbiter mission, this design point also shows a slight advantage.
>
> Minimum system weight, including pump weight and ducting weight variations with pressure, is obtained at a design pressure of 600 psia.

Figures 3-13 through 3-23 show typical performance data for this system when operated with a turbine designed at altitude mode power with a maximum sea level, full power inlet pressure of 600 psia.

Specific Propellant Consumption (SPC)

Two specific propellant parameters are of interest, one reflecting net hydraulic and electrical power output, the other indicating gross turbine shaft power input to the APU gearbox. Figures 3-13 and 3-14 show the net and gross SPC's as a function of net and gross output power. At sea level ambient pressure, the SPC follows the expected decrease with increasing output power level. However, at low ambient pressures, the gross SPC remains relatively constant, experiencing a slight decrease with power level at low output. This characteristic is explained by analysis of the turbine pressure ratio and the discharge pressure at various ambient pressures. Figure 3-15 shows a plot of the pressure ratio vs net output power. The data indicate that at low ambient pressures the turbine pressure ratio is almost constant. However, at low ambient pressures, the turbine discharge pressure changes by a factor of almost nine as the net output power is increased from zero to full power as shown in Figure 3-16. The resulting effect is to provide an almost constant turbine efficiency at low ambient pressures; this is shown in Figure 3-17. Consequently, since the turbine efficiency is almost constant (in fact, it increases slightly with decreasing power), the gross SPC at zero ambient pressure is also almost constant. In contrast, at sea level ambient pressure, the turbine discharge pressure varies by only 2.6 psia from zero to full power; therefore, power modulation requires varying the pressure ratio which causes a decrease in efficiency as power is reduced.

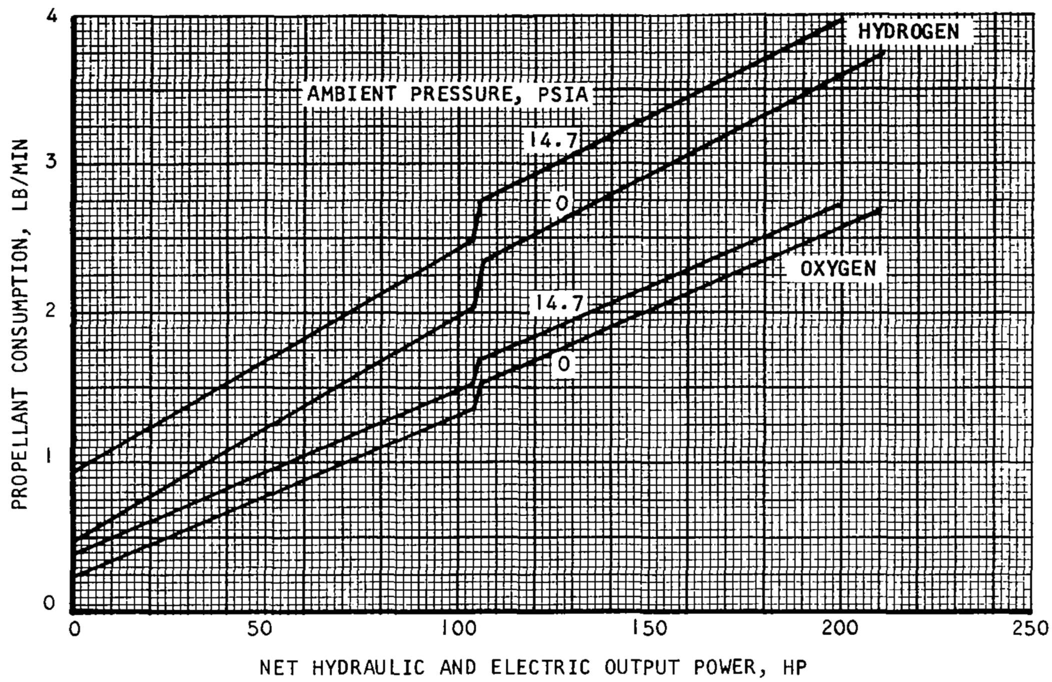

Figure 3-2. APU Performance Map; Turbine Design Point = Sea Level, Full Power, 1500 psia Maximum Pressure.

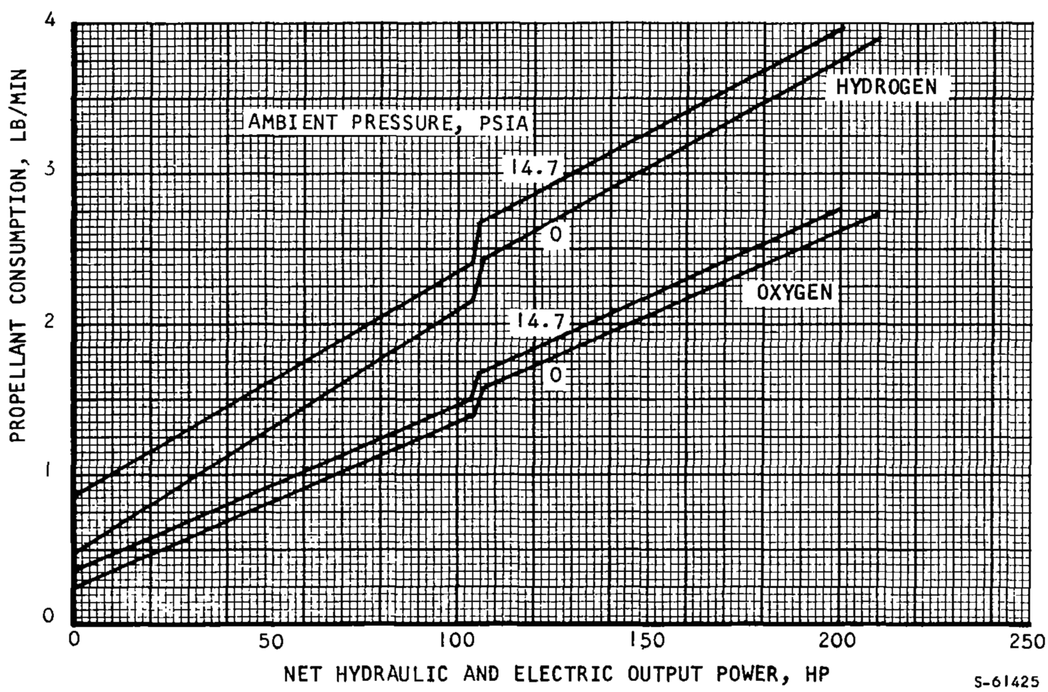

Figure 3-3. APU Performance Map; Turbine Design Point = Altitude, Mode Power, 1500 psia Maximum Pressure.

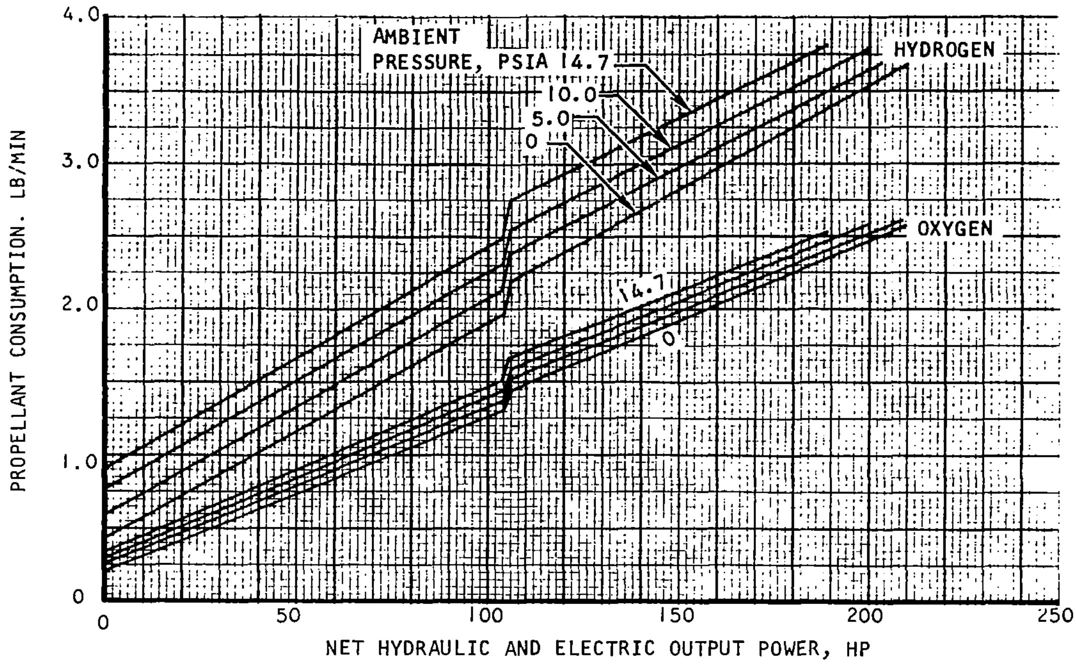

Figure 3-4. APU Performance Map; Turbine Design Point = Sea Level, Full Power, 1200 psia Maximum Inlet Pressure.

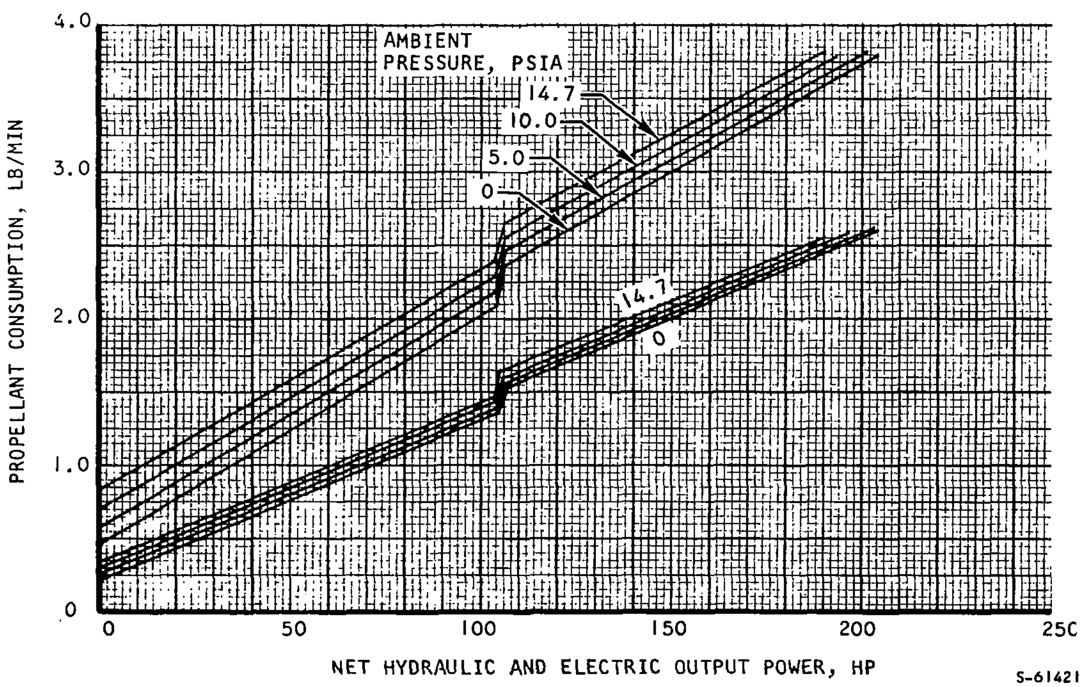

Figure 3-5. APU Performance Map; Turbine Design Point = Altitude, Mode Power, 1200 psia Maximum Pressure.

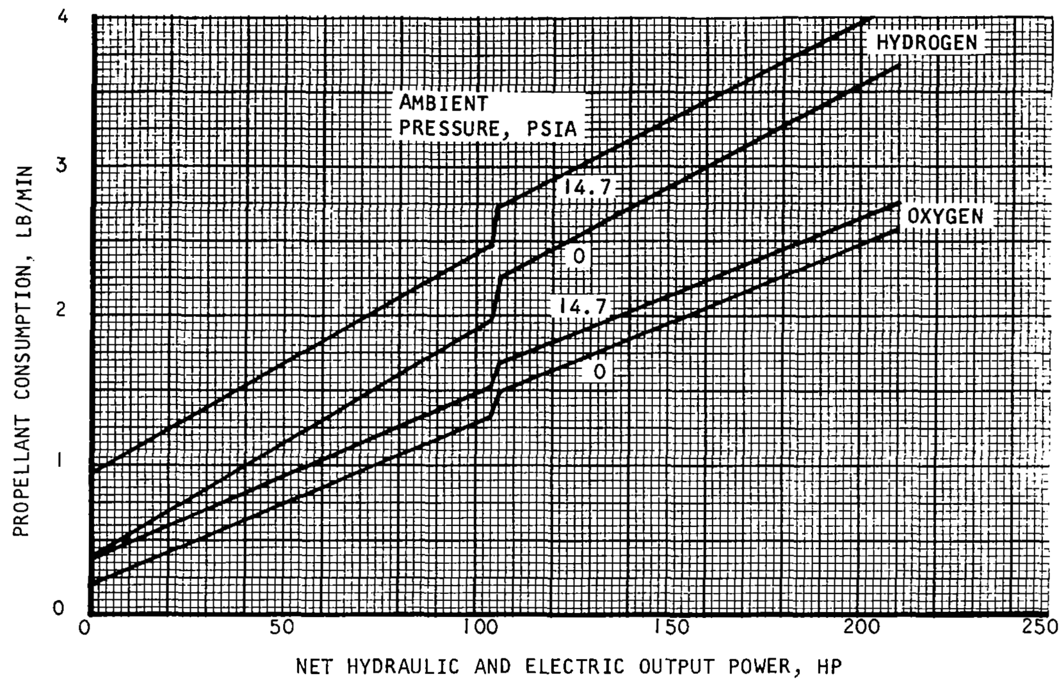

Figure 3-6. APU Performance Map; Turbine Design Point = Sea Level, Full Power, 900 psia Maximum Pressure.

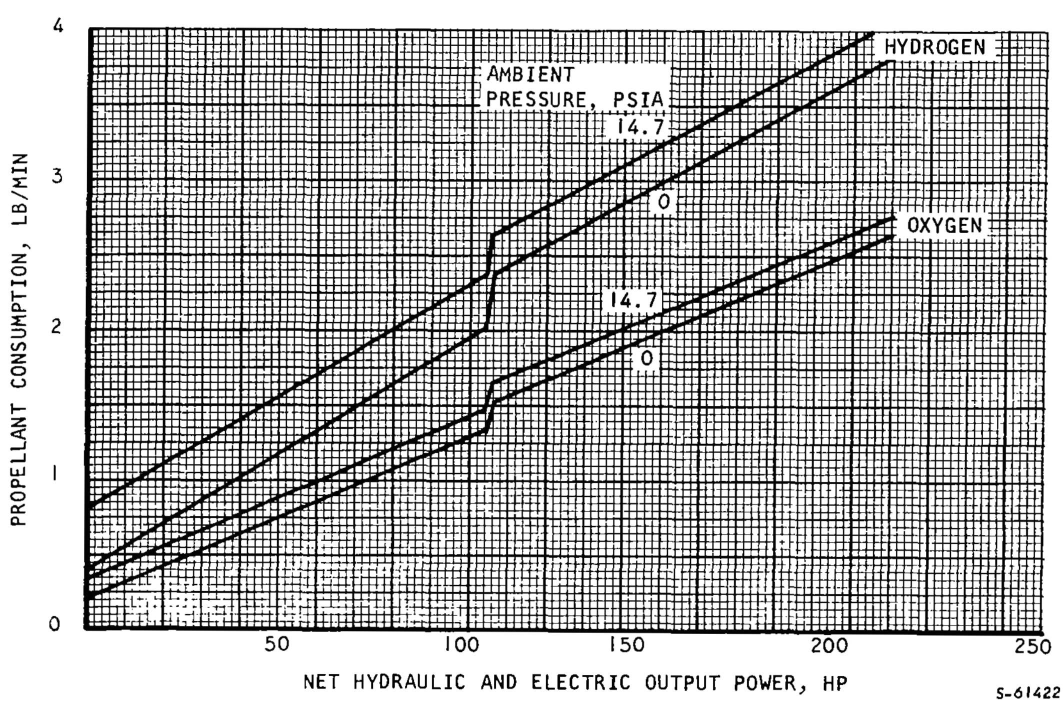

Figure 3-7. APU Performance Map; Turbine Design Point = Altitude, Mode Power, 900 psia Maximum Pressure

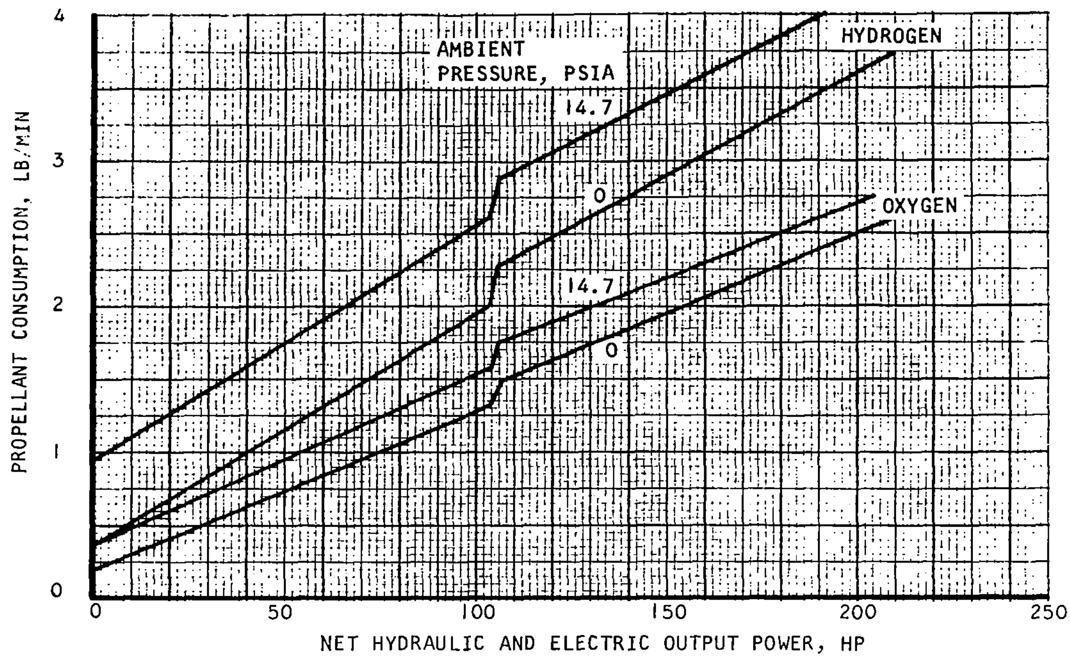

Figure 3-8. APU Performance Map; Turbine Design Point = Sea Level, Full Power, 600 psia Maximum Pressure.

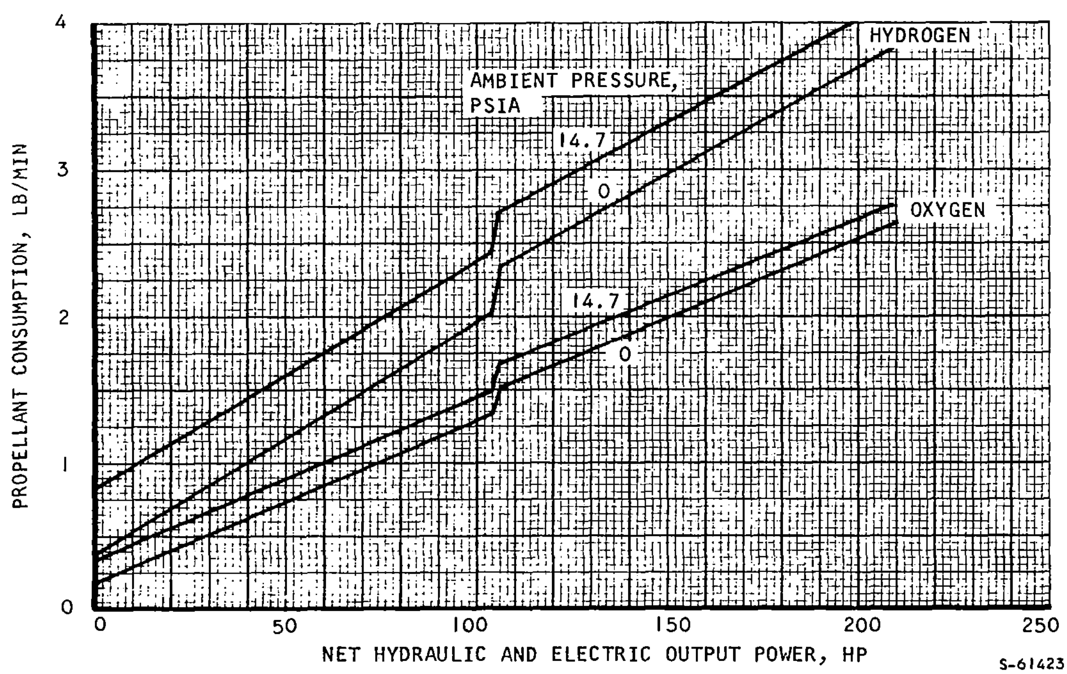

Figure 3-9. APU Performance Map; Turbine Design Point = Altitude, Mode Power, 600 psia Maximum Pressure.

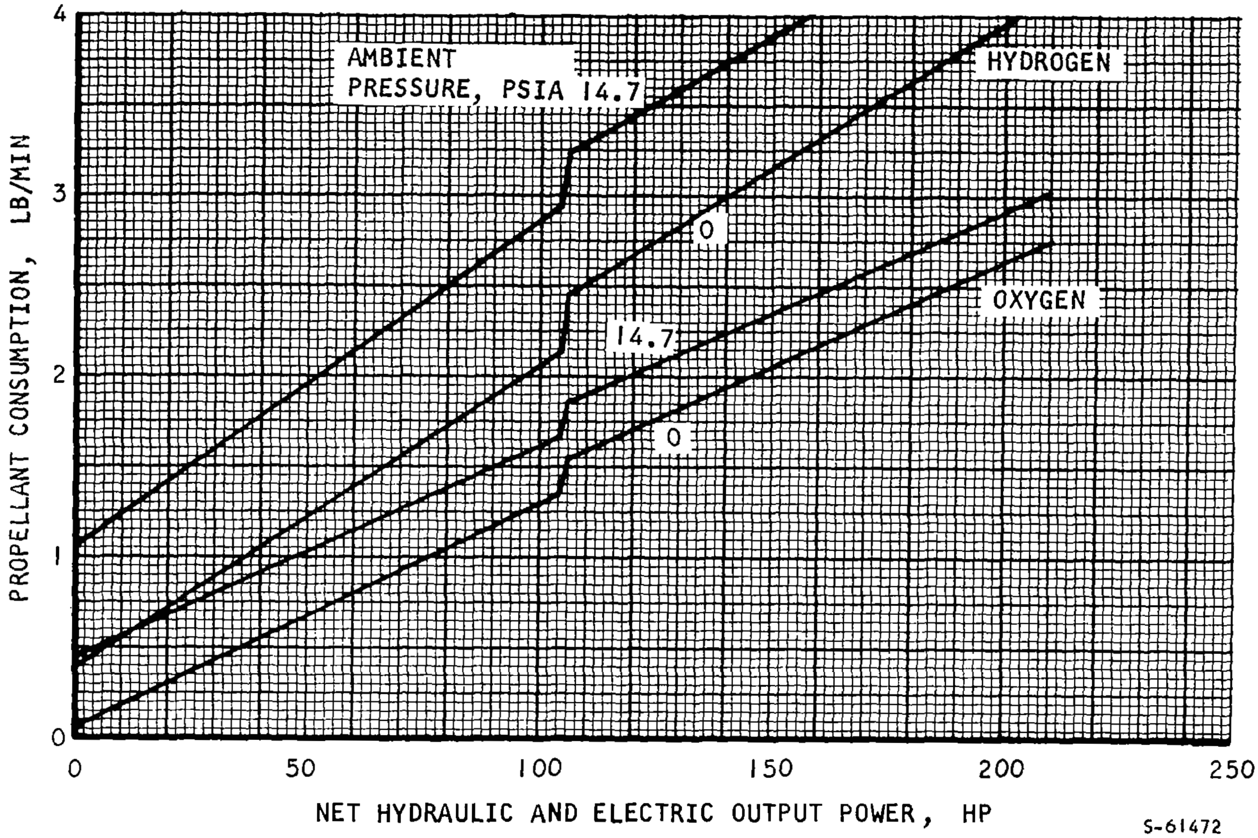

Figure 3-10. APU Performance Map; Turbine Design Point = Altitude Mode Power 300 psi Maximum Pressure

TABLE 3-1

MISSION PROPELLANT REQUIRED

Booster Propellant, lb		P_{inlet} Maximum, psia	Orbiter Propellant, lb	
Sea Level	Altitude		Sea Level	Altitude
--	358.5	300	-	91.3
307.4	285.8	600	82.6	80.2
303.6	282.2	900	83.5	81.2
303.9	290.1	1200	85.6	86.8
312.3	299.6	1500	90.4	89.9

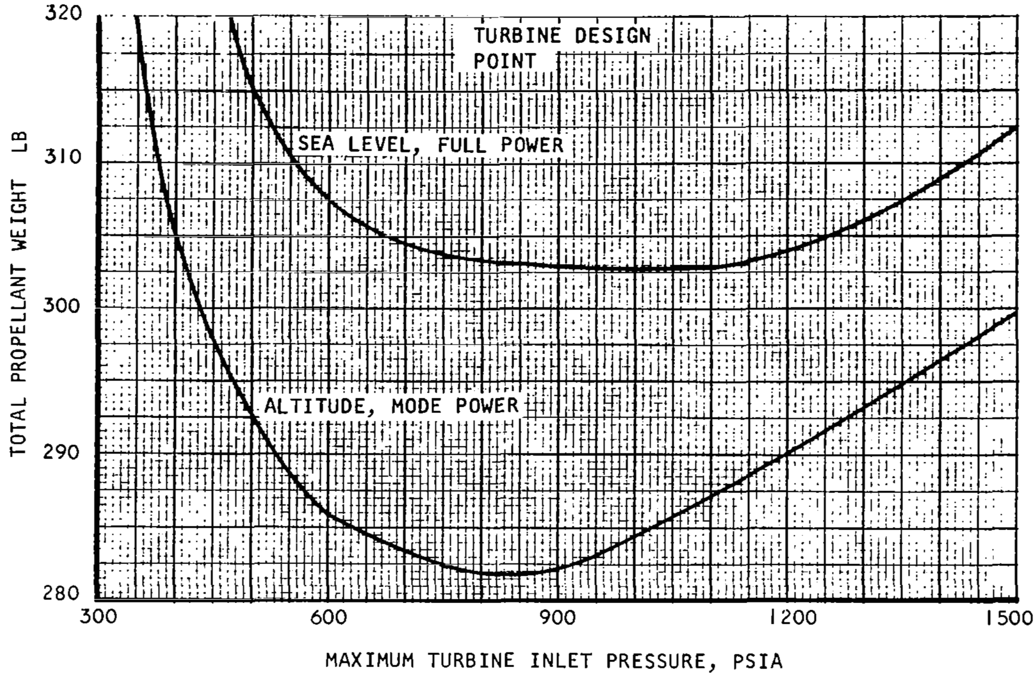

Figure 3-11. APU Propellant for Booster Mission.

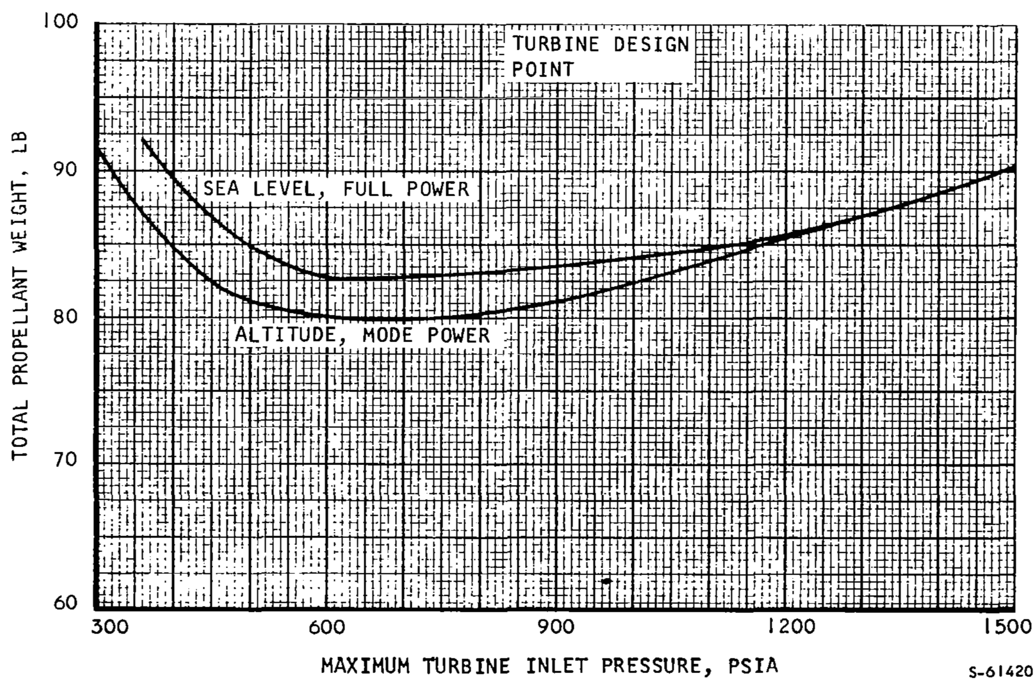

Figure 3-12. APU Propellant for Orbiter Mission.

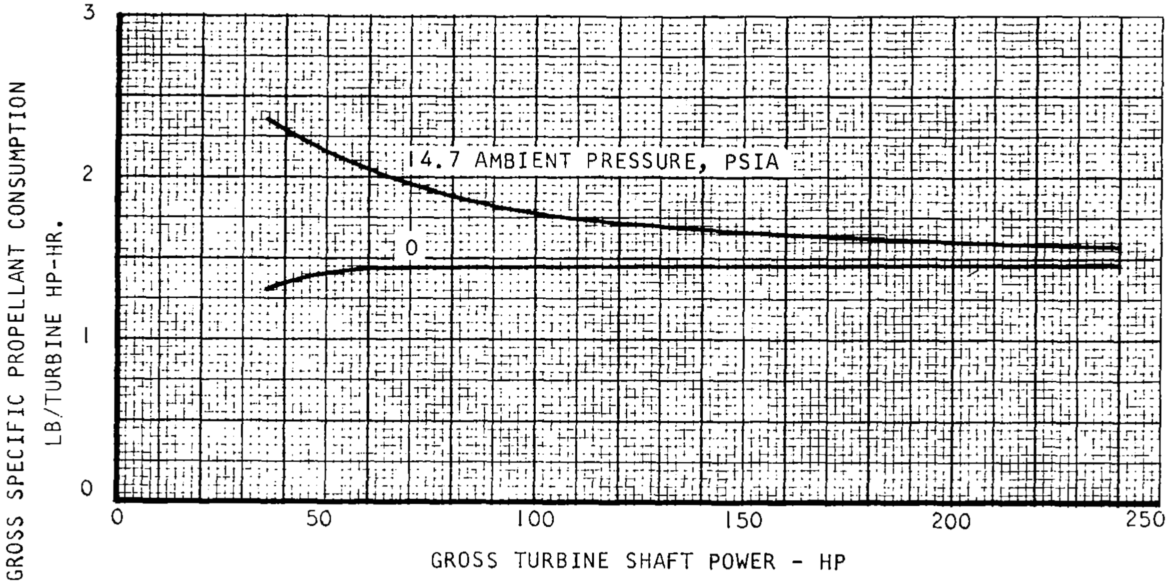

Figure 3-13 Specific Propellant Consumption vs Gross Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

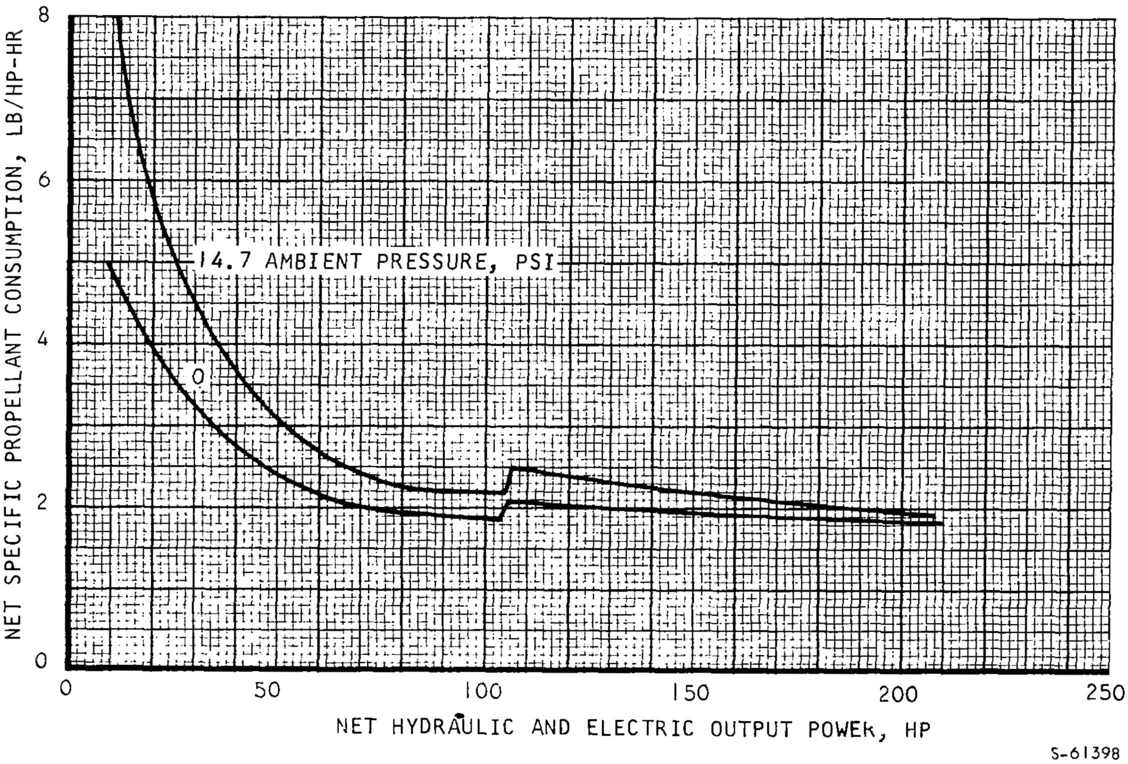

Figure 3-14. Propellant Consumption vs Net Output Power, Hp Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

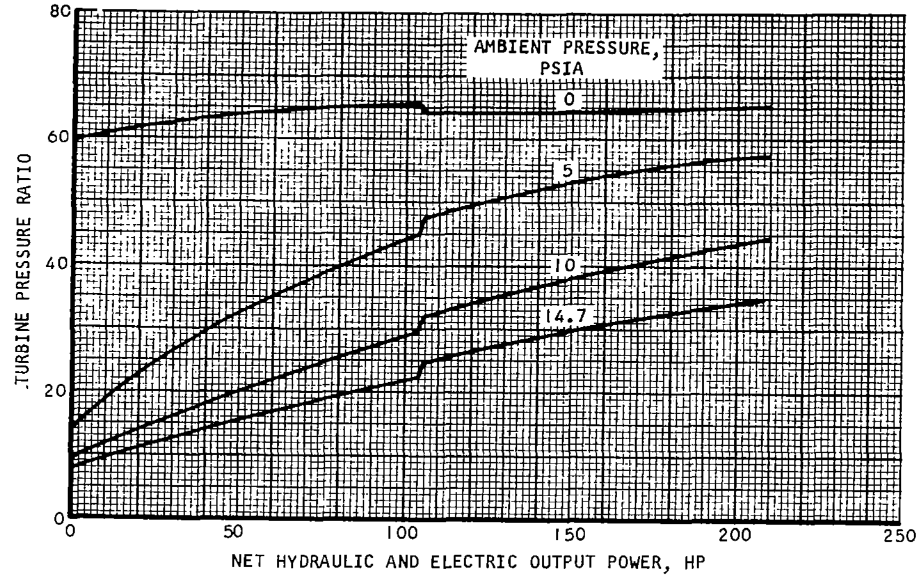

Figure 3-15. Turbine Pressure Ratio vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

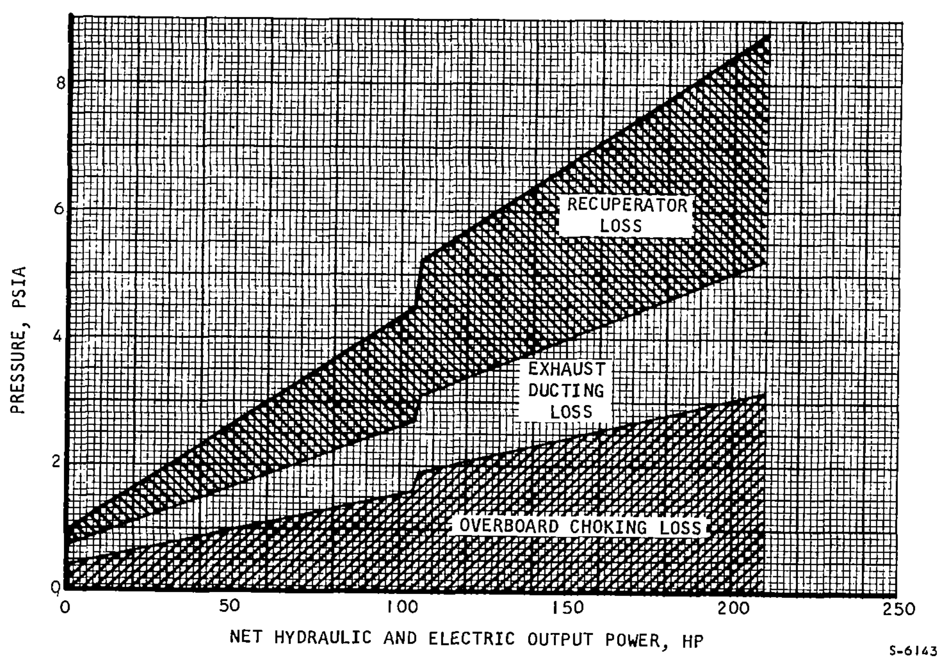

Figure 3-16. Turbine Discharge Pressure at Zero Ambient Pressure; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

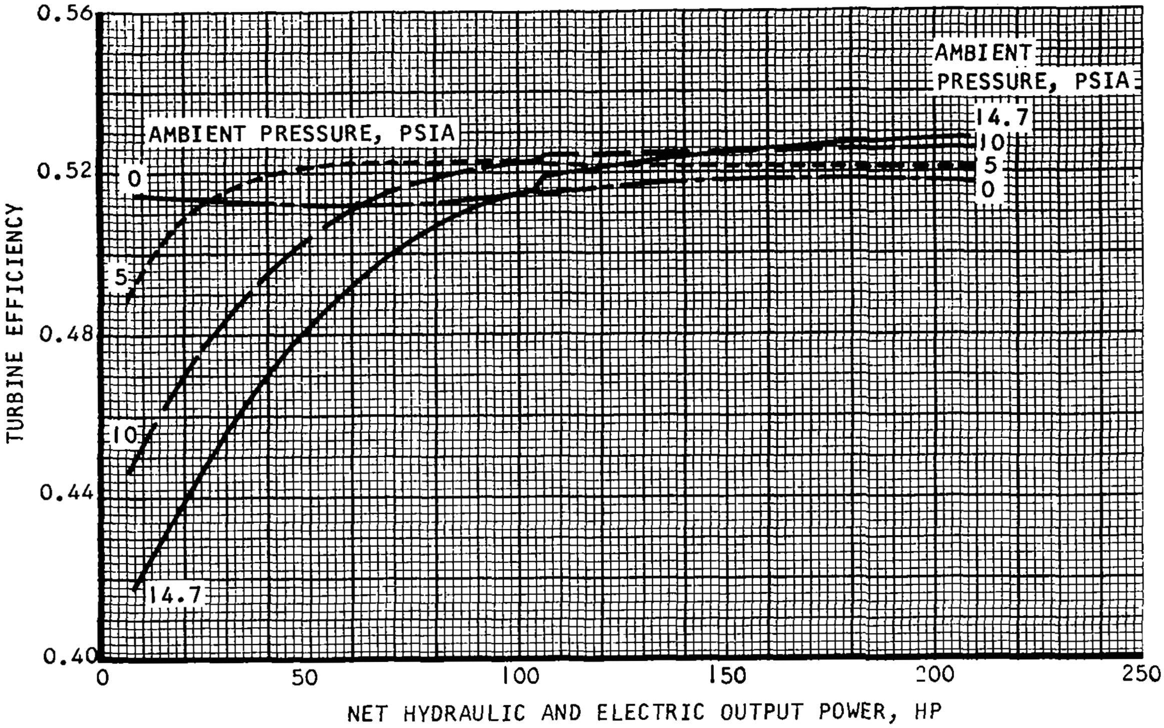

Figure 3-17. Turbine Efficiency vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

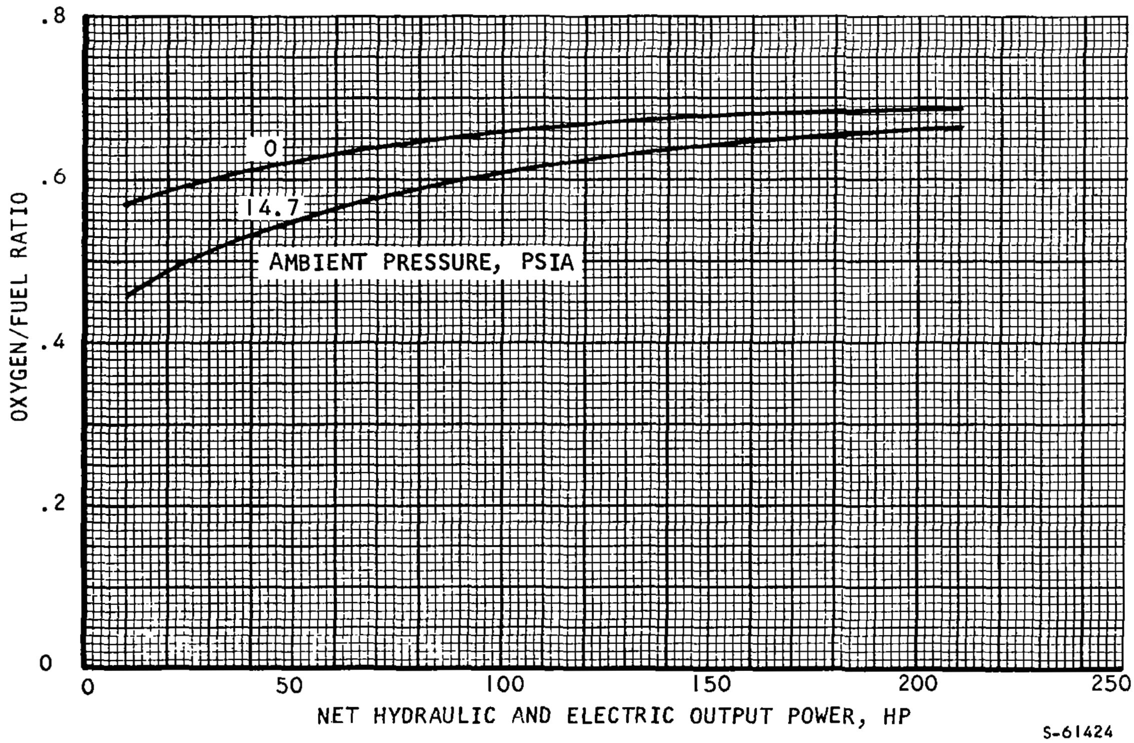

Figure 3-18. O/F vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

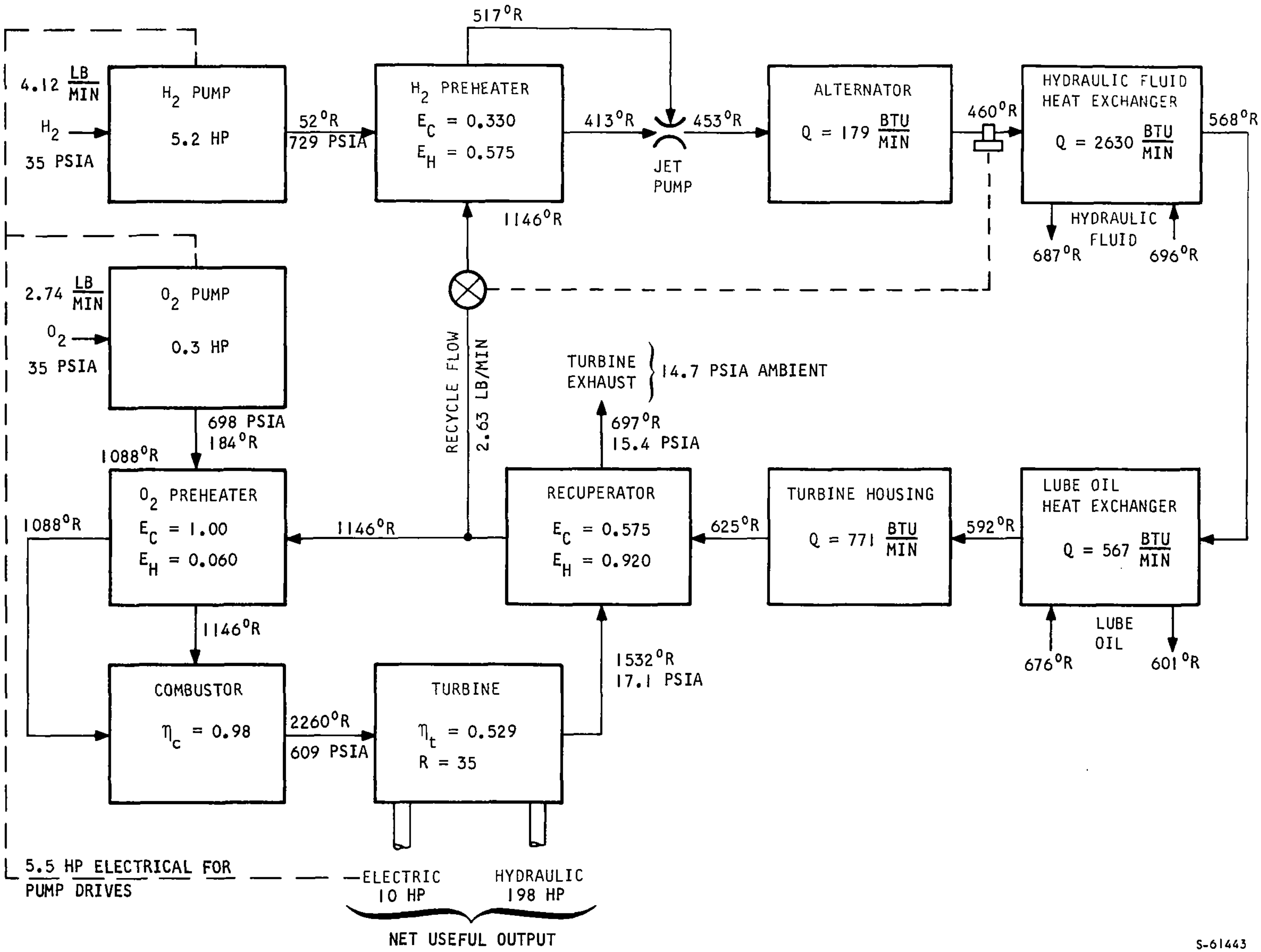

Figure 3-19. Sea Level, Full Power Cycle State Points - Low-Pressure Cryogenic Supplied System.

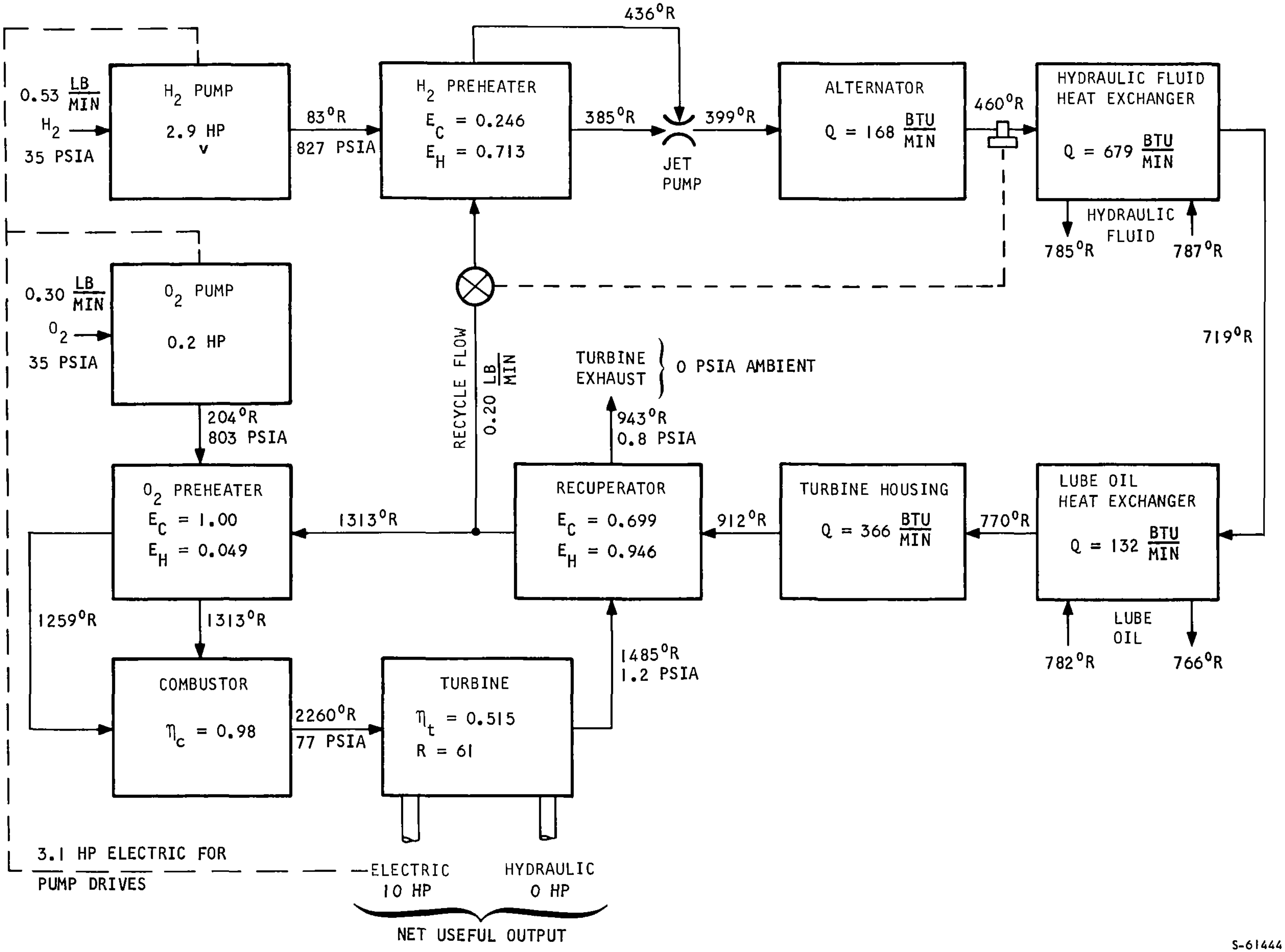

Figure 3-20. Space, Low Power Cycle State Points - Low-Pressure Cryogenic Supplied System.

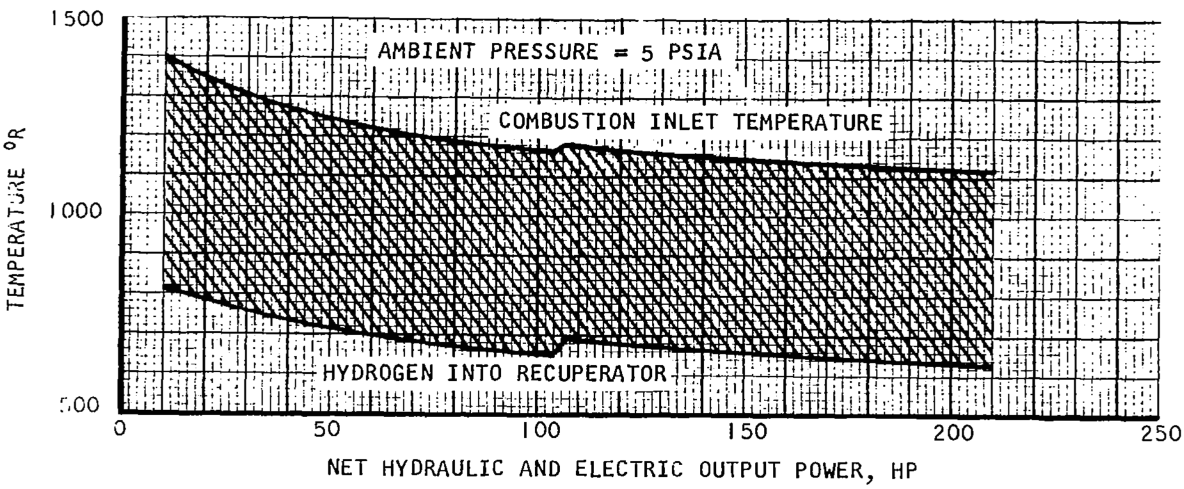

Figure 3-21. Temperature vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

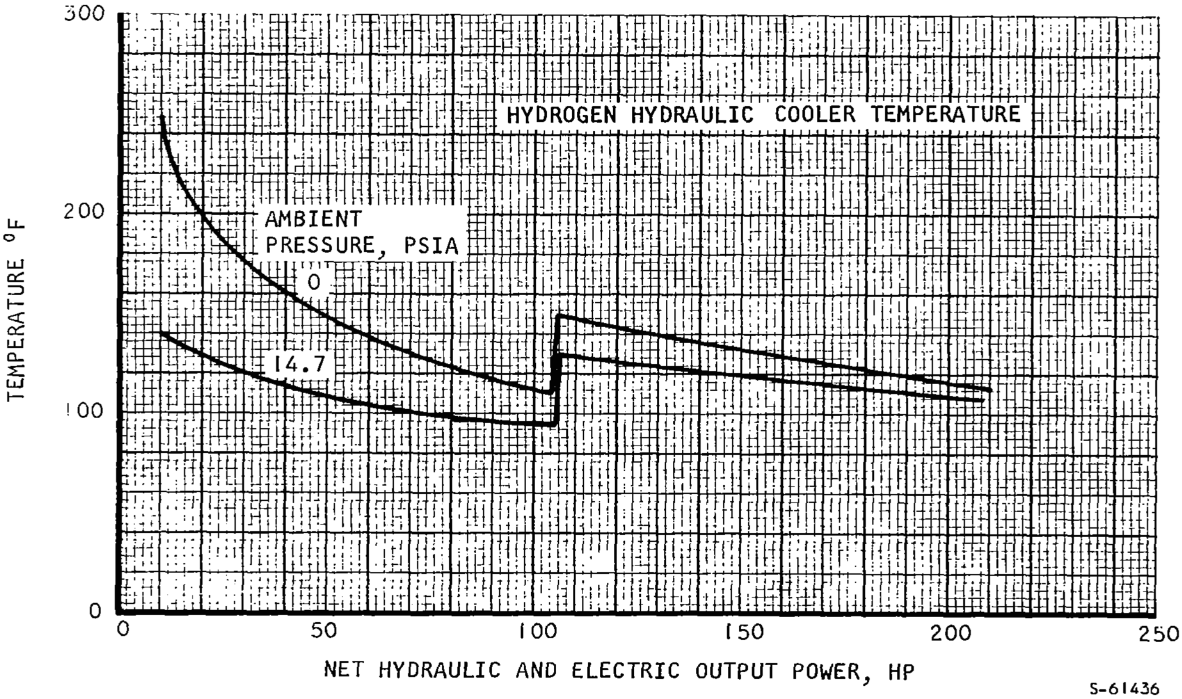

Figure 3-22. Hydraulic Temperature vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure.

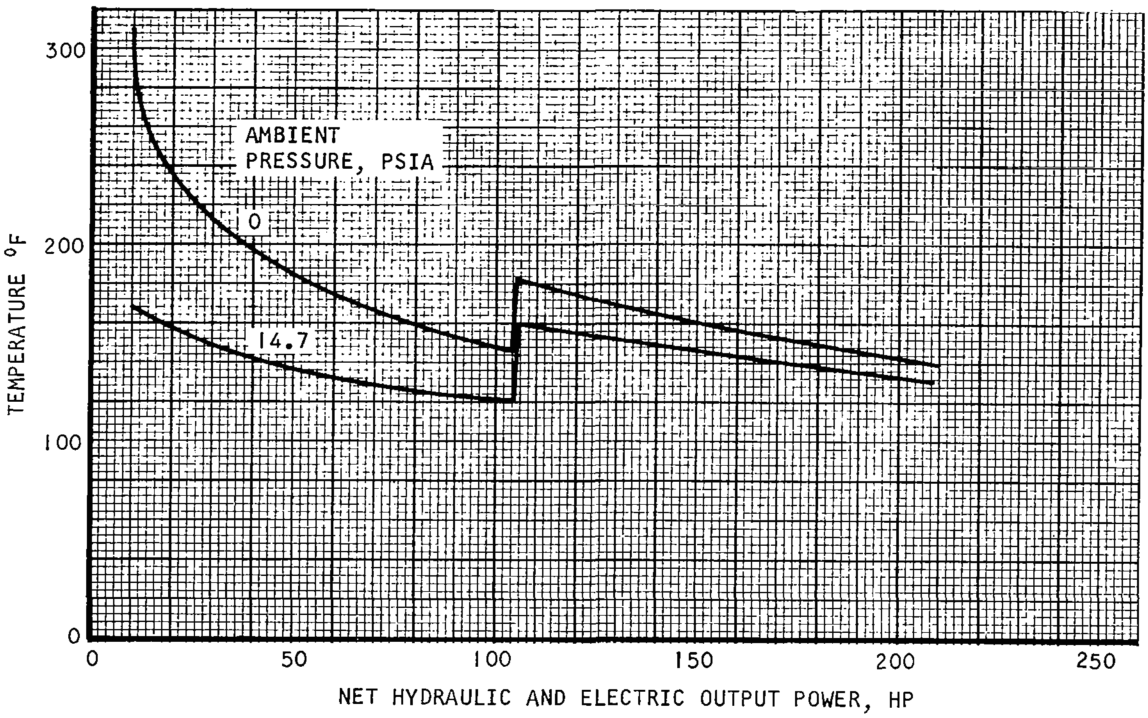

Figure 3-23. Lube Oil Temperature vs Net Output Power; Pumped System: Altitude Design Point, 600 psia Maximum Pressure

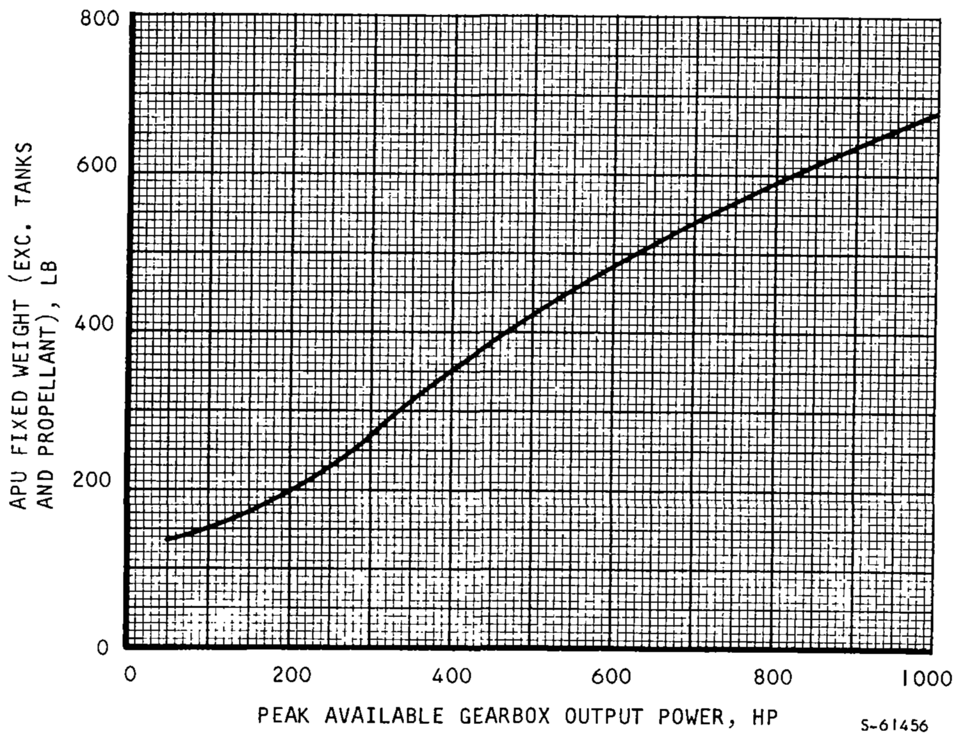

Figure 3-24. APU Fixed Weight; Low-Pressure Cryogenic Liquid Supplied System

O/F Ratio

The O/F ratio shown in Figure 3-18 has the expected decreased with decreasing power level, which reflects an increasing waste heat input. Many of the system losses are not dependent upon power output and, as a consequence, a relatively large amount of waste heat is available at part load. As indicated in Section 2, the effect of this waste heat is to reduce the O/F ratio.

Cycle State Points

Figures 3-19 and 3-20 show the cycle state points for the two operating extremes, sea level full power and space zero hydraulic and 10 hp electric power. The first condition sizes many of the components since the maximum heat loads and flows will be obtained here. The space idle-power condition will be critical with respect to thermal design, since the heat load remains relatively constant and the hydrogen flow is drastically reduced. The present system shows satisfactory performance at both conditions. The data assume full hydraulic fluid flow regardless of power output--thus the fluid temperature rise is quite low.

Cycle Recuperation

Figure 3-21 shows the combustor inlet temperature and the hydrogen temperature into the recuperator. The difference between these two represents cycle recuperation from the turbine exhaust. It will be noted that this difference remains relatively constant with output power.

Hydraulic Fluid Temperature

Figure 3-22 shows the hydraulic fluid temperature as a function of output power for sea level and zero ambient pressure. From the cycle state point diagram, it appears that the peak hydraulic temperature (obtained at minimum load and zero ambient pressure) can be reduced by at least 50^{0}F by sizing the hydraulic oil cooler for a higher effectiveness. The benefit resulting from this reduced temperature must be evaluated against the inreased heat exchanger weight.

Lube Oil Temperature

Lube oil temperature control is essential for proper functioning and reliable operation of the APU. Figure 3-23 shows the predicted lube oil temperature as a function of output power. These temperatures are satisfactory for the intended usage.

WEIGHT

In accordance with a NASA request for parametric data on systems sized for different combinations of peak and mode power, Table 3-2 shows the fixed weight of the APU system components, excluding the propellant tankage and the propellant. The system fixed weight is plotted in Figure 3-24. The data are based on the selected 600 psia maximum turbine inlet pressure. The inflection in the weight curve with power is due to the variation in the weight and speed (hence gearbox torque) of available hydraulic pumps.

TABLE 3-2

APU FIXED WEIGHT LOW PRESSURE CRYOGENIC LIQUID SUPPLY SYSTEM

Component		Peak Power Required: 100 hp	225 hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps		14.0	25.0	44.0	103.0	145.0
Generator		11.2	24.0	34.0	61.5	90.2
Turbine (with containment)		44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)		2.1	3.1	3.6	4.6	5.6
Control Logic Devices		6.0	6.0	6.0	6.0	6.0
Control Valves		4.5	5.0	5.5	6.0	7.0
Pressure Regulators		2.0	2.0	2.0	2.5	3.0
Heat Exchangers	Recuperator	10.2	16.5	19.5	26.8	34.2
	H_2 Preheater	2.6	3.8	4.4	5.7	7.0
	O_2 Preheater	0.5	0.7	0.8	1.0	1.3
	Hydraulic Oil Cooler	2.2	4.4	5.0	6.2	7.4
	Lube Oil Cooler	1.0	1.4	1.8	2.9	4.3
Reactant Pumps	H_2 Pump	8.1	9.9	11.3	13.1	14.9
	O_2 Pump	2.2	2.7	3.1	3.6	4.1
Gear Box with Lube Pump		5.7	17.4	26.6	58.9	92.5
Lube Oil in Sump		7.7	7.7	7.7	7.7	7.7
Instrumentation		1.0	1.0	1.0	1.0	1.0
Ducting		14.9	21.6	24.8	31.7	38.7
Subtotal		139.9	196.2	245.1	386.2	513.9
10 Percent for Vehicle Support Structure		14.0	19.6	24.5	38.6	51.4
Total Fixed Weight		153.9	215.8	269.6	424.8	565.3

The fixed and variable weights of the booster APU system for 225 hp output at the gearbox are tabulated in Table 3-3 for the various possible combinations of tank types (vacuum-jacketed hard shell, and soft shell and storage concepts (separate tankage for each APU, shared tankage for three APU's, shared tankage for four APU's, and shared tankage with the APS--assumes 10,000 lb each of hydrogen and oxygen in tanks).

The fixed and variable weight of the orbiter APU system are shown similarly in Table 3-4.

Finally, using the scaling criteria described in Section 2, it is possible to determine the APU system weight as a function of power level and total energy output. Such curves are shown in Figures 3-25 and 3-26. They assume separate hard shell tanks for each APU on the orbiter, and separate soft shell tanks for each APU on the booster. Thus, these tankage concepts represent the extremes in the possible tank weights. The booster soft shell tanks are the lightest of all tanks considered at any given deliverable contents quantity. The orbiter hard shell tanks are the heaviest.

TABLE 3-3

BOOSTER 225 HP APU SYSTEM WEIGHT - LOW PRESSURE CRYOGENIC LIQUID SUPPLY SYSTEM
(SINGLE APU)

Tank Type	Hard Shell				Soft Shell			
Storage Concept	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Shared With APS	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Shared With APS
Fixed weight, lb	215.8	215.8	215.8	215.8	215.8	215.8	215.8	215.8
Hydrogen weight, lb	187.7	187.7	187.7	187.7	187.7	187.7	187.7	187.7
Hydrogen tank weight, lb	163.2	137.0	135.0	112.5	84.5	65.7	62.0	28.2
Oxygen weight, lb	98.1	98.1	98.1	98.1	98.1	98.1	98.1	98.1
Oxygen tank weight, lb	24.5	17.7	15.7	14.7	17.7	13.7	11.8	6.9
Total system weight, lb	689.3	656.3	652.3	628.8	603.8	581.0	575.4	536.7

TABLE 3-4

ORBITER 225 HP APU SYSTEM WEIGHT - LOW PRESSURE CRYOGENIC LIQUID SUPPLY SYSTEM
(SINGLE APU)

Tank Type	Hard Shell				Soft Shell			
Storage Concept	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Shared with APS	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Shared with APS
Fixed weight, lb	215.8	215.8	215.8	215.8	215.8	215.8	215.8	215.8
Hydrogen weight, lb	51.8	51.8	51.8	51.8	51.8	51.8	51.8	51.8
Hydrogen tank weight, lb	68.4	47.8	45.8	32.1	44.0	26.9	25.4	10.4
Cryogenic weight, lb	28.4	28.4	28.4	28.4	28.4	28.4	28.4	28.4
Oxygen tank weight, lb	52.5	11.4	7.0	4.5	48.3	9.9	6.8	3.5
Total system weight, lb	416.9	355.2	348.8	332.6	388.3	332.8	328.2	309.9

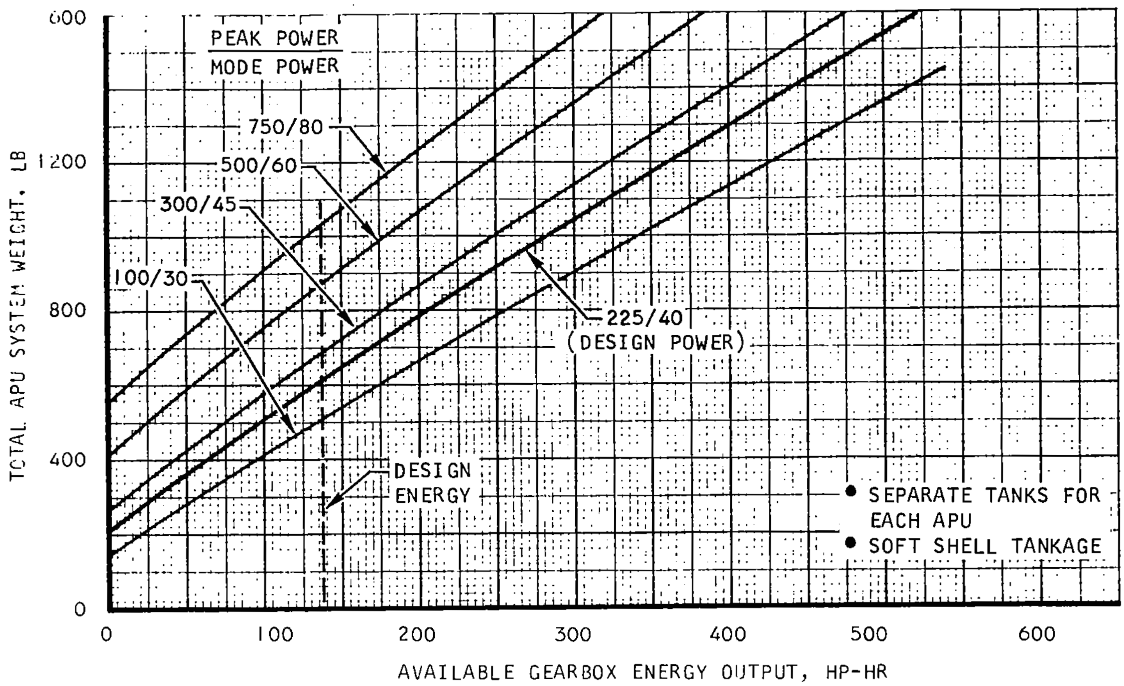

Figure 3-25. Booster Vehicle APU System Weight; Low-Pressure Cryogenic Liquid Supplied System.

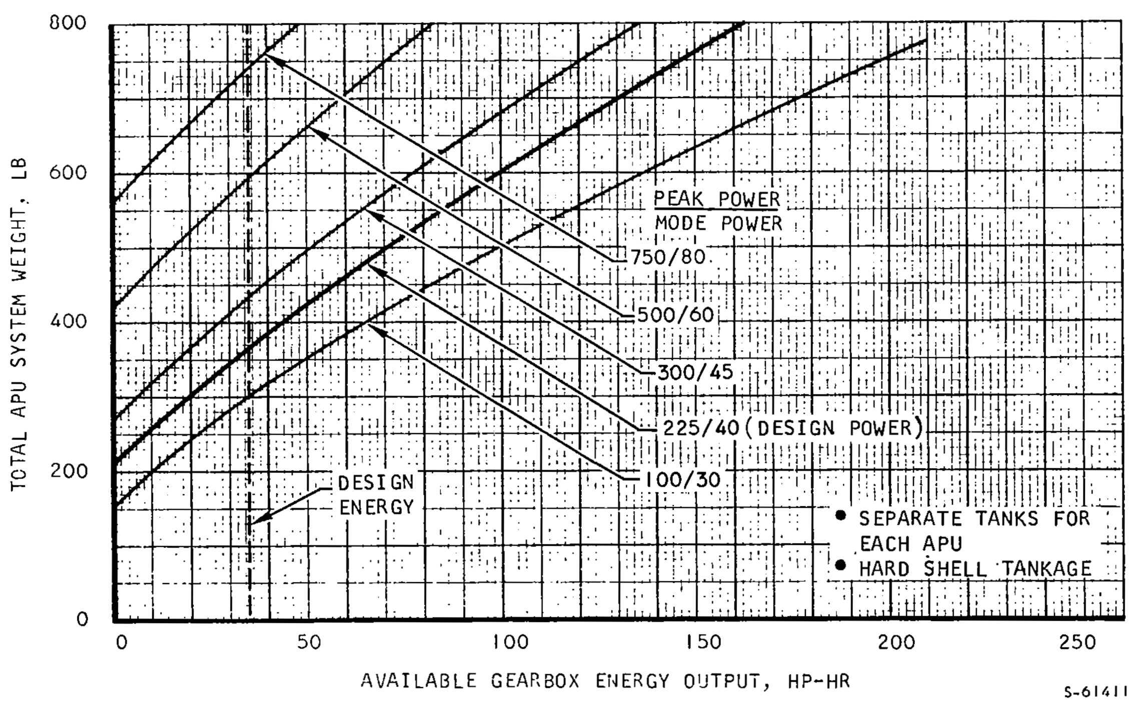

Figure 3-26. Orbiter Vehicle APU System Weight; Low-Pressure Cryogenic Liquid Supplied System.

SECTION 4

INTEGRAL HIGH-PRESSURE CRYOGENIC SUPPLIED SYSTEM

GENERAL DESCRIPTION

In this system, the propellants are supplied from integral high-pressure cryogenic tanks. Tank operating pressure is optimized for minimum system total weight considering the effect of propellant supply pressure on tankage weight and turbine performance. Since the cryogenic tanks are a part of the APU system, the system is required to incorporate provisions for maintenance of the cryogenic tank pressure at the proper level. This will involve adding heat to the contents of the tank for (1) initial pressure buildup to the operating level and (2) maintenance of the operating pressure with propellant withdrawl. This can be accomplished by one or more of the following means:

Internal electrical heaters with electrical power modulated by a pressure switch.

Return heated cryogen to tank by means of a recirculating fan.

Pass heated cryogen through an internal heat exchanger with flow modulated by pressure sensing control.

In the system schematic shown in Figure 4-1, the last method is used. This was selected for simplicity, reliability, and minimum performance penalties to the system.

High pressure hydrogen from the cryogenic tank is preheated by hot hydrogen which is heated by the turbine exhaust in a recuperative heat exchanger. A portion of this preheated hydrogen is passed through the internal heat exchanger to maintain tank pressure at the desired value. The preheated hydrogen next flows through a jet pump where it is augmented by recirculated hydrogen flow in the thermal loop. The thermal loop involves cooling of the generator, hydraulic fluid, gearbox lubricant, and turbine housing. After passing through the recuperator, the hydrogen flow is split, a portion flowing to the combustor and the remainder being recirculated through the thermal loop. The combustor hydrogen flow is used to preheat the oxygen. As before, a portion of the preheated oxygen is passed through the cryogenic oxygen tank to maintain its pressure at the desired value. The gaseous hydrogen and oxygen pass through control valves which maintain the proper flows and pressures to burn in the combustor which supplies a hot high-pressure working fluid to the turbine. After expansion in the turbine, the turbine exhaust gas flows through the recuperator before being dumped overboard.

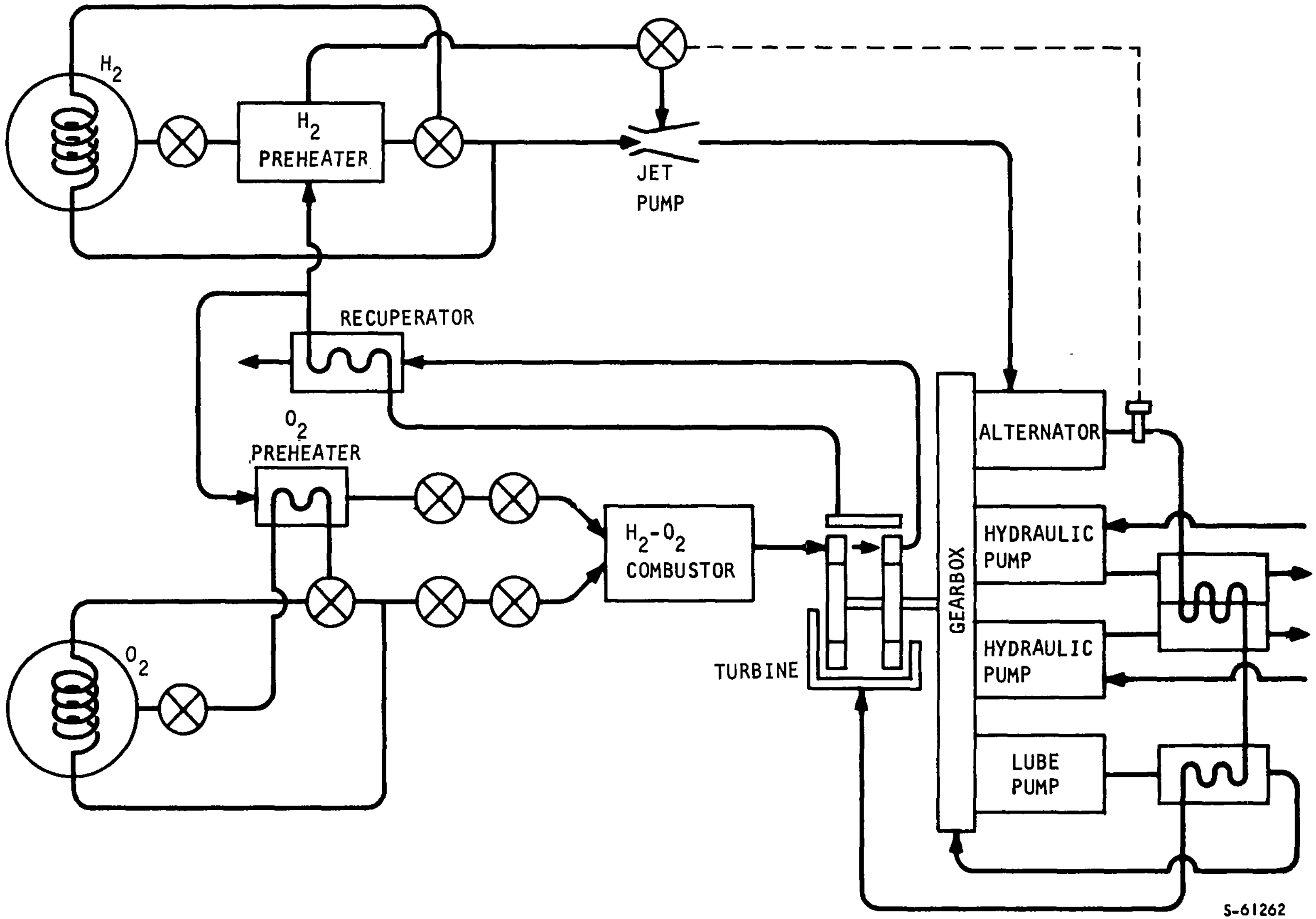

Figure 4-1. Integral High-Pressure Cryogenic Supplied System

CRYOGENIC STORAGE INTEGRATION

As indicated previously, the APU system is required to supply heat input to the cryogenic storage tanks to maintain constant pressure with fluid withdrawal. Figures 4-2 and 4-3 show the required input to maintain constant pressure as a function of cryogen density (which will decrease with usage in a constant volume supercritical storage tank). It will be noted that the heat input per unit mass of fluid withdrawn will vary with density. Part or all of the required heat input could be provided by heat leak into the tank. However, this will be an inefficient design because of the wastage of cryogen by venting during standby or part-load operation. In general, because of the mission and operational requirements, it will be necessary to design a well-insulated tank with very low heat leak.

SYSTEM SUPPLY PRESSURE TRADEOFF

As indicated previously, turbine performance tends to increase with increasing supply pressure. With this system, where the turbine is supplied directly from the cryogenic storage tanks, tank weight will tend to increase with increasing pressure. An optimization study was performed to determine the optimum storage pressure for this system.

As described in Section 2 of this report, the cycle performance program was used to establish propellant consumption maps as a function of storage pressure for the cycle described previously and shown in Figure 4-1. Figures 4-4, 4-5, and 4-6 are the APU performance maps for cryogenic supply pressures of 300, 650, and 950 psia. These maps were used in the integration program to determine the propellant requirements shown in Table 4-1 for the booster and orbiter vehicle mission profiles. Then, the tankage weight penalties (Appendix C) were applied to establish the weight tradeoffs with storage pressure shown in Figures 4-7 and 4-8 for the booster and orbiter missions, respectively. An optimum near 600 psia is obtained for both the orbiter and booster missions.

SYSTEM PERFORMANCE

Propellant consumption is given in Figure 4-5 as a function of output power and ambient pressure. Figure 4-9 is a typical cycle state-point diagram. The primary difference between this and the low-pressure cryogenic supplied system described in the previous section involves elimination of the propellant pumping power penalty and addition of negative heat loads representing thermal requirements for pressure maintenance with fluid withdrawal. Cycle performance parameters are uneventful and generally lower temperatures are obtained than the low-pressure system because of the reduction in parasitic losses (for propellant pumping) and additional heat sink capacity (for cryogenic tank pressurization).

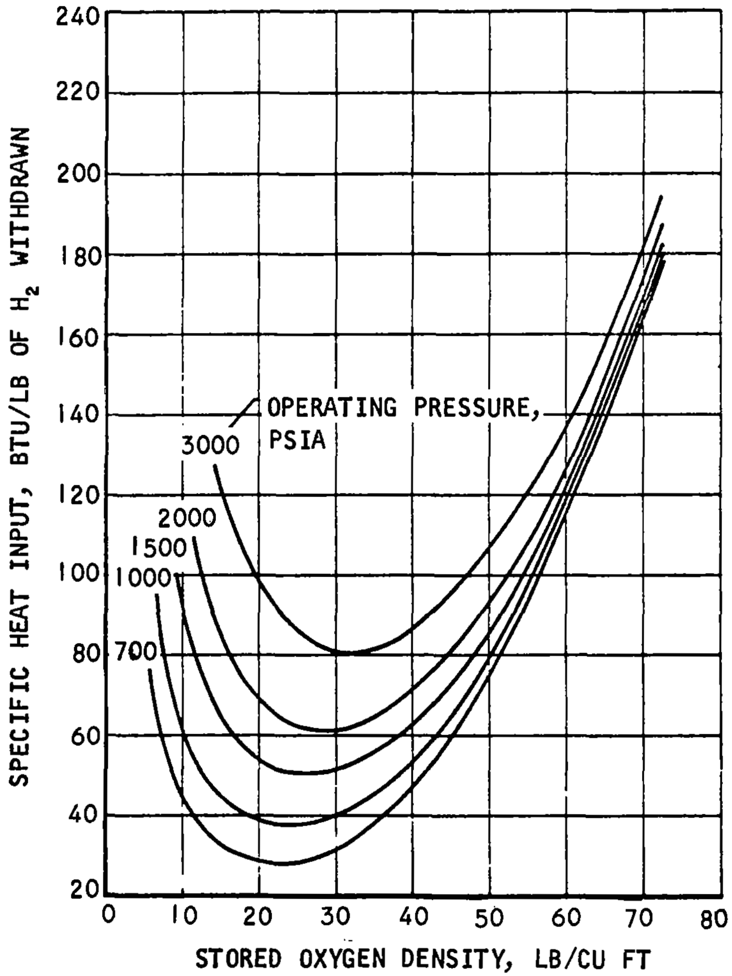

Figure 4-2. Specific Heat Input Required to Maintain Stored Oxygen at a Constant Supercritical Pressure During Flow

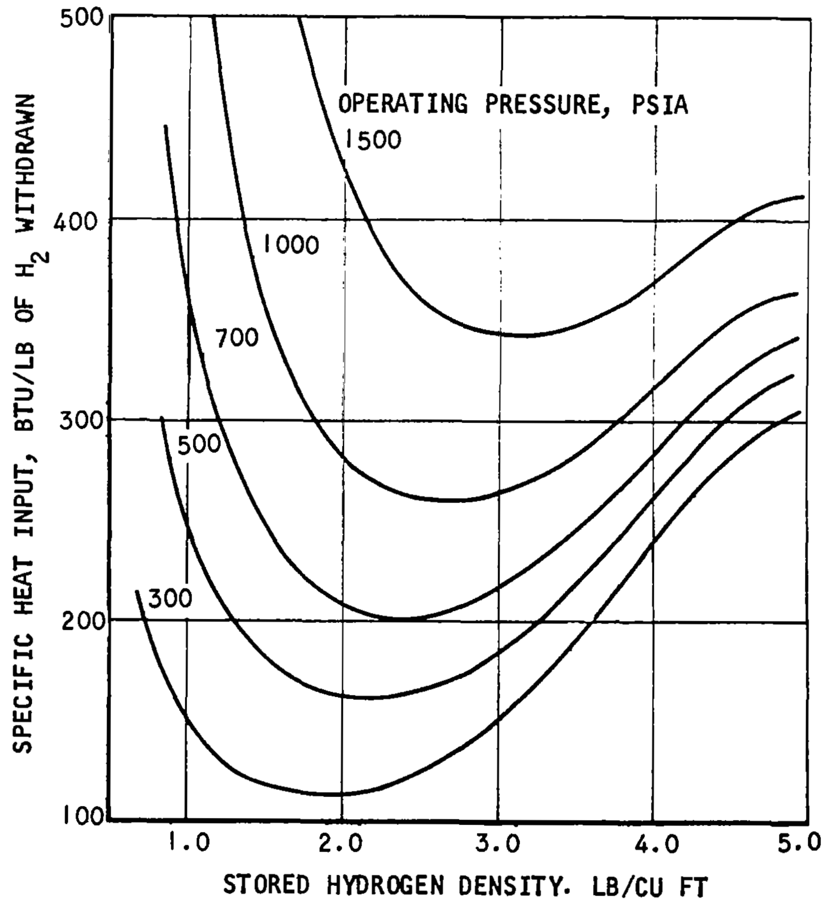

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Figure 4-3. Specific Heat Input Required to Maintain Stored Hydrogen at a Constant Supercritical Pressure During Flow

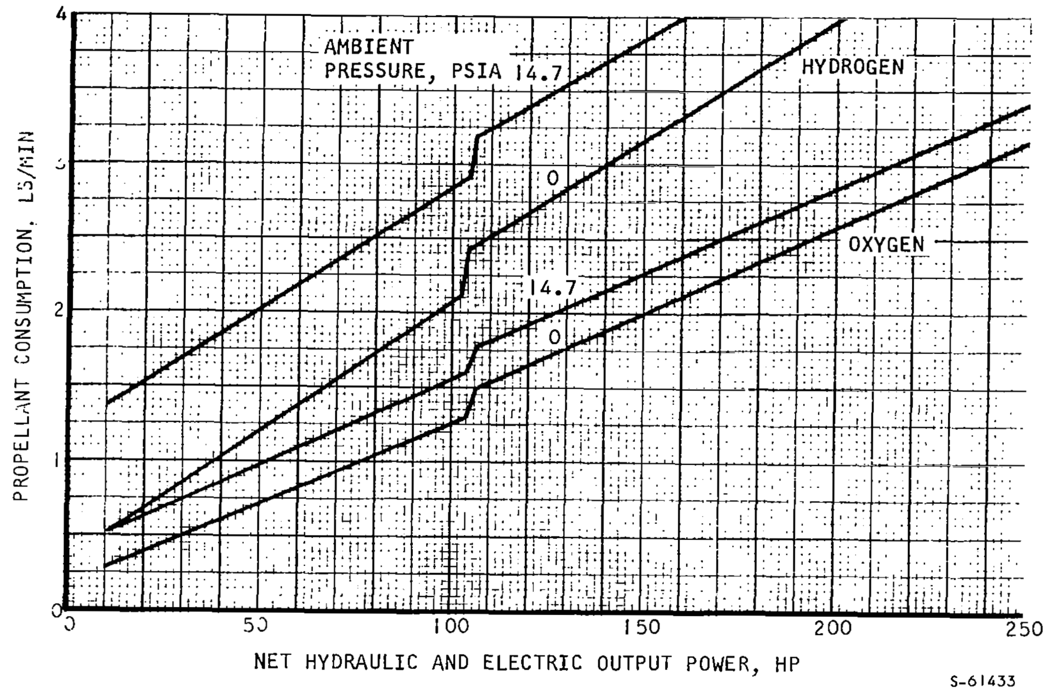

Figure 4-4. APU Performance Map; 300 psia Cryogenic Tanks, Turbine Design Point = Sea Level, Full Power

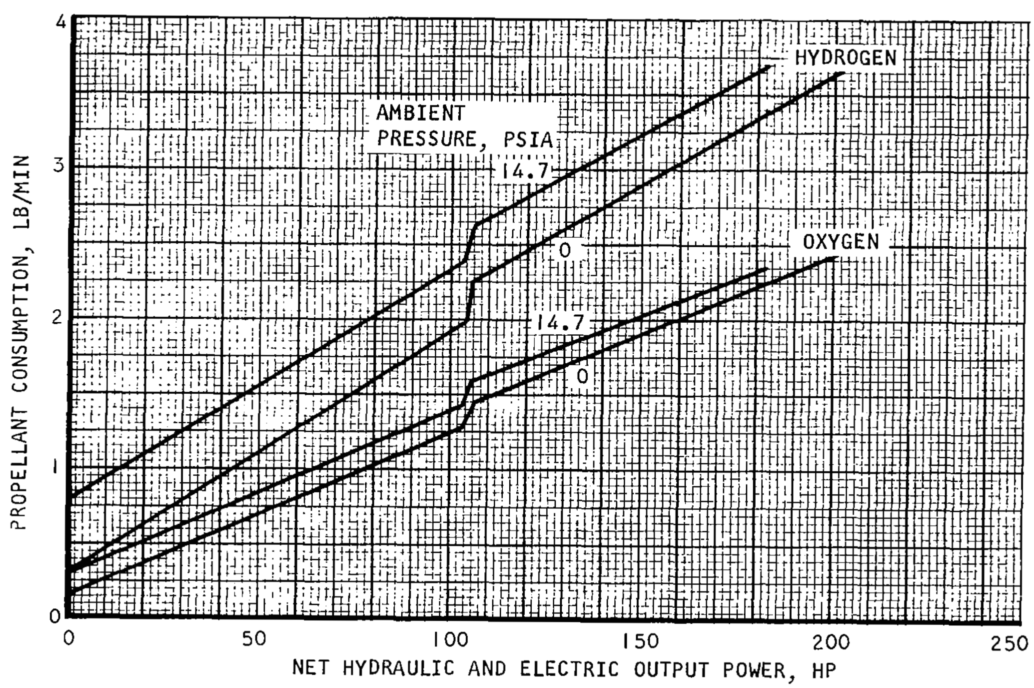

Figure 4-5. APU Performance Map; 650 psia Cryogenic Tanks, Turbine Design Point = Altitude, Mode Power

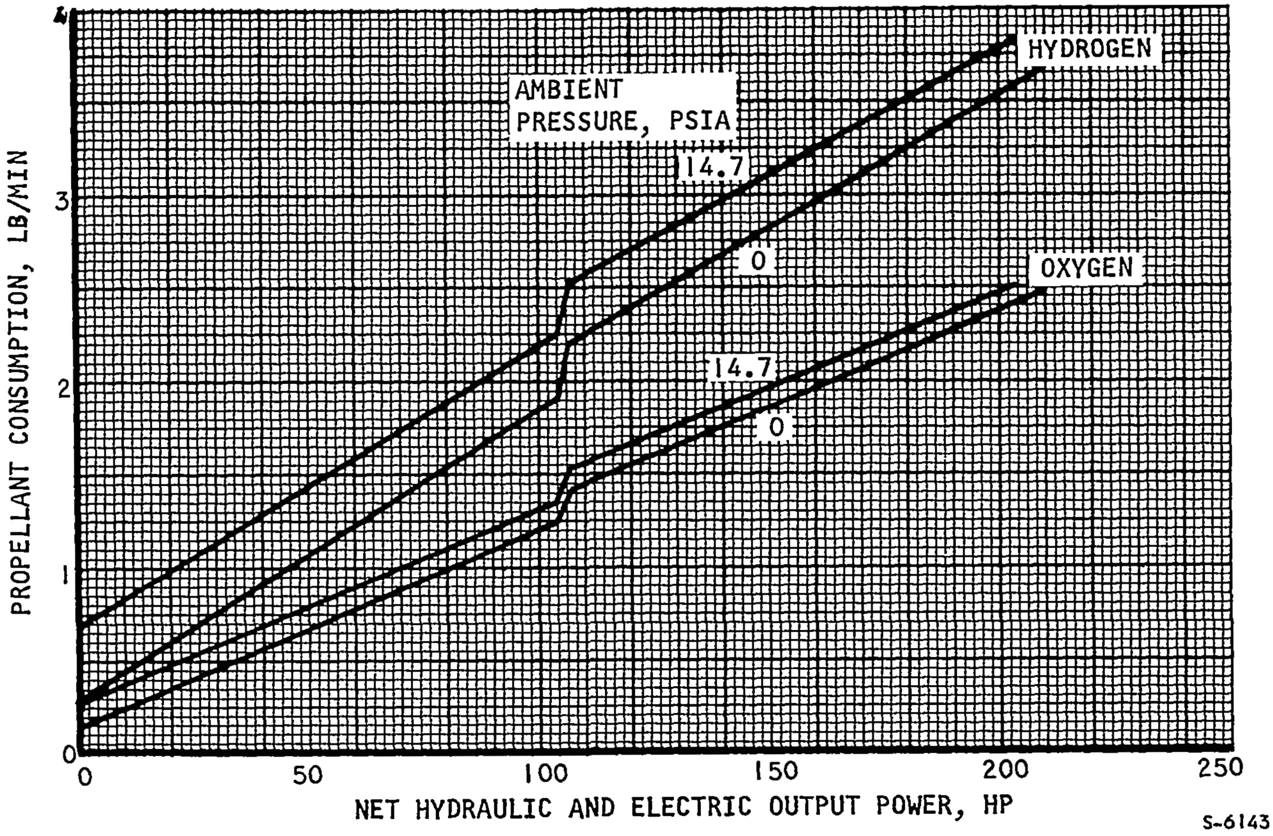

Figure 4-6. APU Performance Map; 950 psia Cryogenic Tanks, Turbine Design Point = Altitude, Mode Power

TABLE 4-I

MISSION PROPELLANT REQUIREMENTS

Cryogenic Storage Pressure, psia	Propellant Requirements, lb Booster Mission		Orbiter Mission	
	H_2	O_2	H_2	O_2
300	238.0	109.9	58.8	29.1
650	176.1	90.4	47.6	25.6
950	167.3	87.0	46.3	25.2

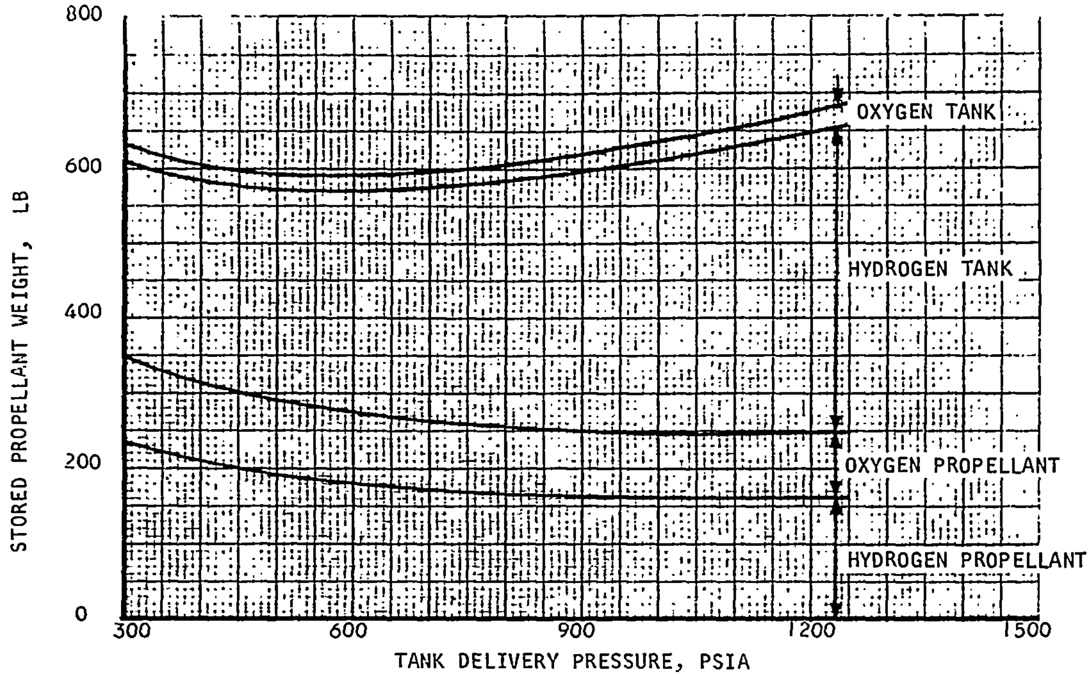

Figure 4-7. APU Stored Propellant Weight vs Delivery Pressure for Booster Vehicle

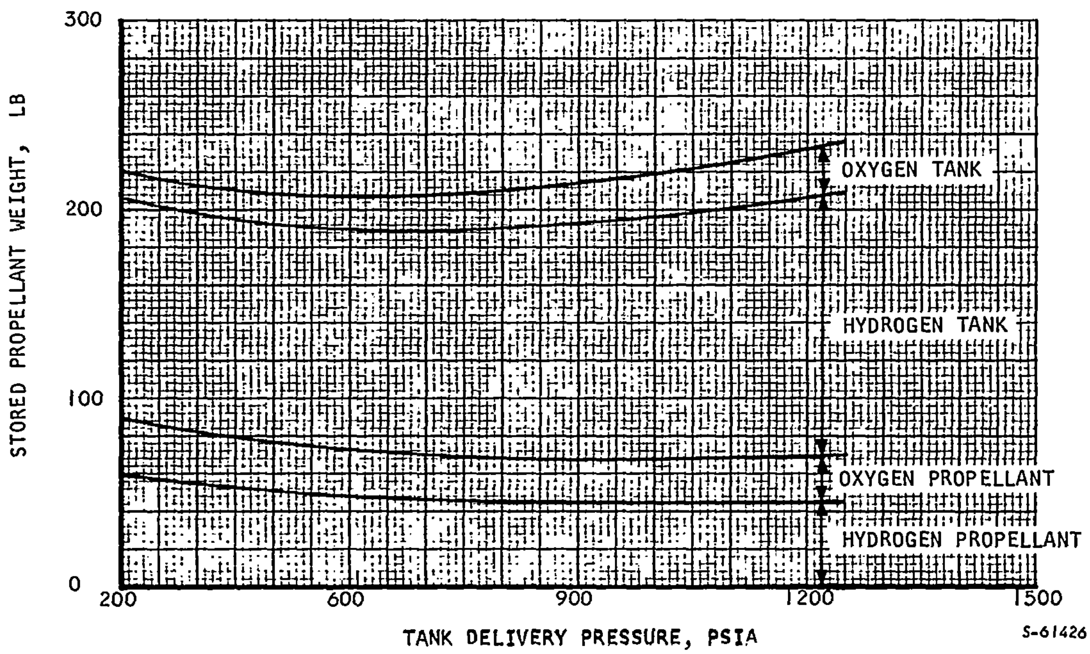

Figure 4-8. APU Stored Propellant Weight vs Delivery Pressure for Orbiter Mission

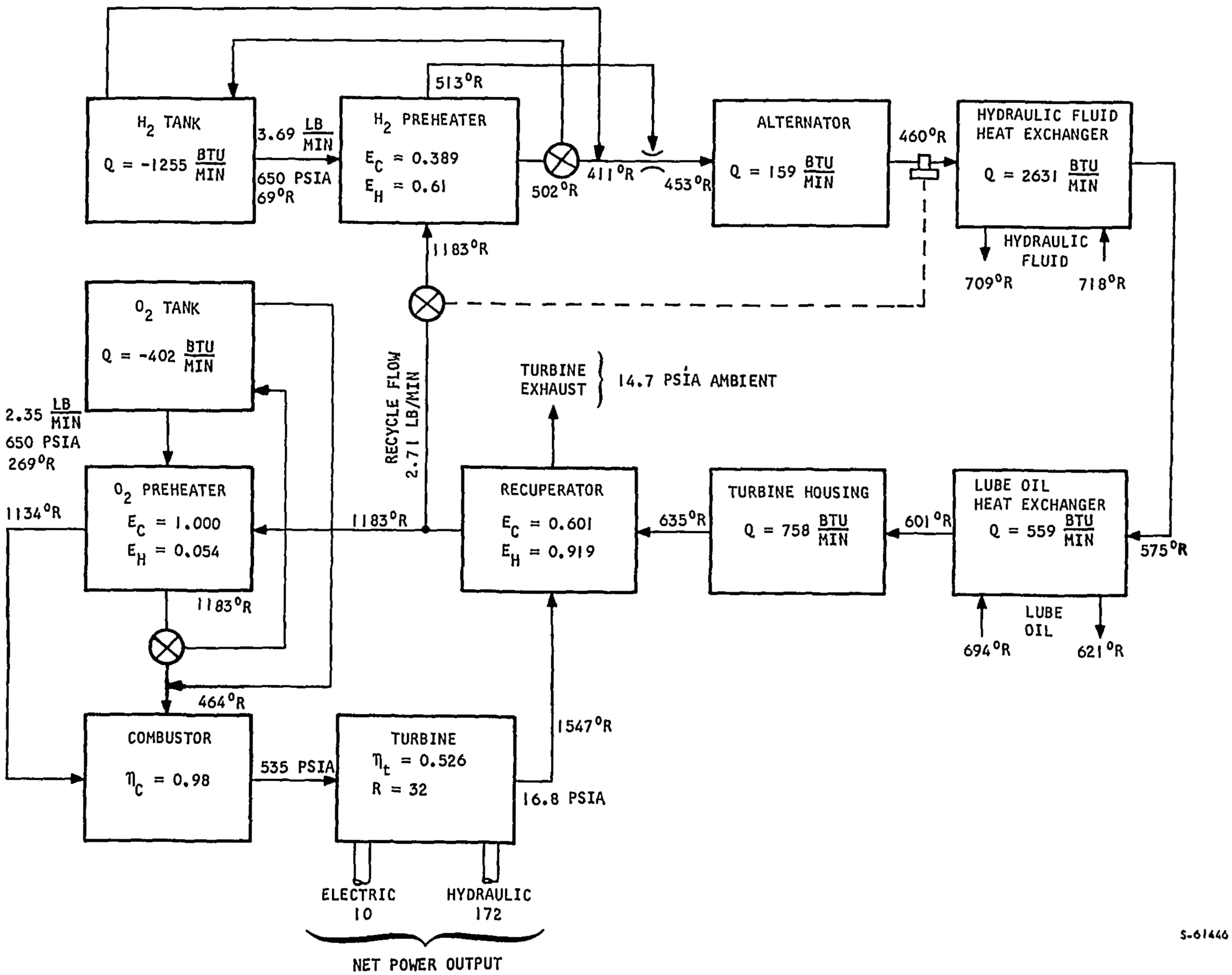

Figure 4-9. Sea Level, Full Power Cycle State Points - Integral High Pressure Cryogenic Supplied System

WEIGHT

Table 4-2 summarizes the APU fixed weight as a function of the peak output power. Figure 4-10 shows a plot of fixed weight vs output power. The inflection in the curve is due to the variation in available pump weights and speeds.

The fixed and variable weight of the booster APU system for 225 hp output at the gearbox are tabulated in Table 4-3 for the various combinations of tank types (vacuum-jacketed hard shell, and soft shell) and storage concepts (separate tankage for each APU, shared tankage for 3 APU's and separate tankage for 4 APU's).

The fixed and variable weights of the orbiter APU system are shown similarly in Table 4-4.

Finally, using the scaling criteria described in Section 2, it is possible to determine the APU system weight as a function of power level and total energy output. Such curves are shown in Figures 4-11 and 4-12. They assume separate hard shell tanks for each APU on the orbiter, and separate soft shell tanks for each APU on the booster. Thus, these tankage concepts represent the extremes in the possible tank weights. The booster soft shell tanks are the lightest of all tanks considered at any given deliverable contents quantity. The orbiter hard shell tanks are the heaviest.

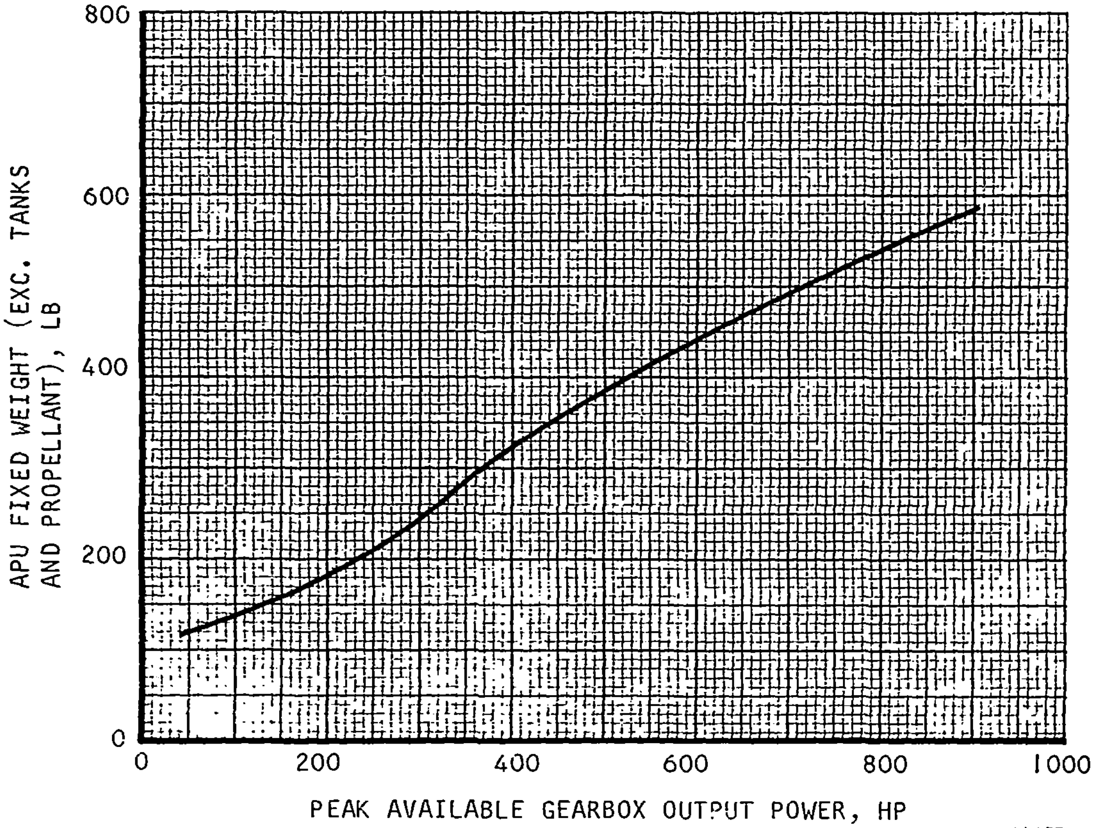

Figure 4-10. APU Fixed Weight; Integral High Pressure Cryogenic System

TABLE 4-2

APU FIXED WEIGHT INTEGRAL HIGH PRESSURE CRYOGENIC

Component \ Peak Power Required		100 hp	225-hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps		14.0	25.0	44.0	103.0	145.0
Generator		9.0	20.0	27.0	45.0	67.0
Turbine (with containment)		44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)		2.1	3.1	3.6	4.6	5.6
Control Logic Devices		6.0	6.0	6.0	6.0	6.0
Control Valves		4.5	5.0	5.5	6.0	7.0
Pressure Regulators		2.0	2.0	2.0	2.5	3.0
Heat Exchangers	Recuperator	10.2	16.5	19.5	26.8	34.2
	H_2 Preheater	2.6	3.8	4.4	5.7	7.0
	O_2 Preheater	0.5	0.7	0.8	1.0	1.3
	Hydraulic Oil Cooler	2.2	4.4	5.0	6.2	7.4
	Lube Oil Cooler	1.0	1.4	1.8	2.9	4.3
Reactant Pumps	H_2 Pump	-	-	-	-	-
	O_2 Pump	-	-	-	-	-
Gear Box With Lube Pump		5.7	17.4	26.6	58.9	92.5
Lube Oil in Sump		7.7	7.7	7.7	7.7	7.7
Instrumentation		1.0	1.0	1.0	1.0	1.0
Ducting		14.9	21.6	24.8	31.7	38.7
Subtotal		127.4	179.6	223.7	353.0	471.7
10 Percent for Vehicle Support Structure		12.7	18.0	22.4	35.3	47.2
Total Fixed Weight		140.1	197.6	246.1	388.3	518.9

TABLE 4-3

BOOSTER 225 HP APU SYSTEM WEIGHT - INTEGRAL HIGH-PRESSURE CRYOGENIC SUPPLIED SYSTEM (SINGLE APU)

Tank Type	Hard Shell			Soft Shell		
Storage Concept	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Separate for Each APU	Common for 3 APU's	Common for 4 APU's
Fixed weight, lb	197.6	197.6	197.6	197.6	197.6	197.6
Hydrogen weight, lb	176.1	176.1	176.1	176.1	176.1	176.1
Hydrogen tank weight, lb	289.0	259.0	250.0	195.5	171.0	169.0
Oxygen weight, lb	90.4	90.4	90.4	90.4	90.4	90.4
Oxygen tank weight, lb	26.2	20.8	19.9	24.4	19.0	18.1
Total system weight, lb	779.3	743.9	734.0	684.0	654.1	651.2

TABLE 4-4

ORBITER 225 HP APU SYSTEM WEIGHT - INTEGRAL HIGH-PRESSURE CRYOGENIC SUPPLIED SYSTEM (SINGLE APU)

Tank Type	Hard Shell			Soft Shell		
Storage Concept	Separate for Each APU	Common for 3 APU's	Common for 4 APU's	Separate for Each APU	Common for 3 APU's	Common for 4 APU's
Fixed weight, lb	197.6	197.6	197.6	197.6	197.6	197.6
Hydrogen weight, lb	47.6	47.6	47.6	47.6	47.6	47.6
Hydrogen tank weight, lb	144.0	105.7	102.0	121.0	88.5	79.5
Oxygen weight, lb	25.6	25.6	25.6	25.6	25.6	25.6
Oxygen tank weight, lb	43.5	9.5	7.7	33.3	8.5	6.9
Total system weight, lb	458.3	386.0	380.5	425.1	367.8	357.2

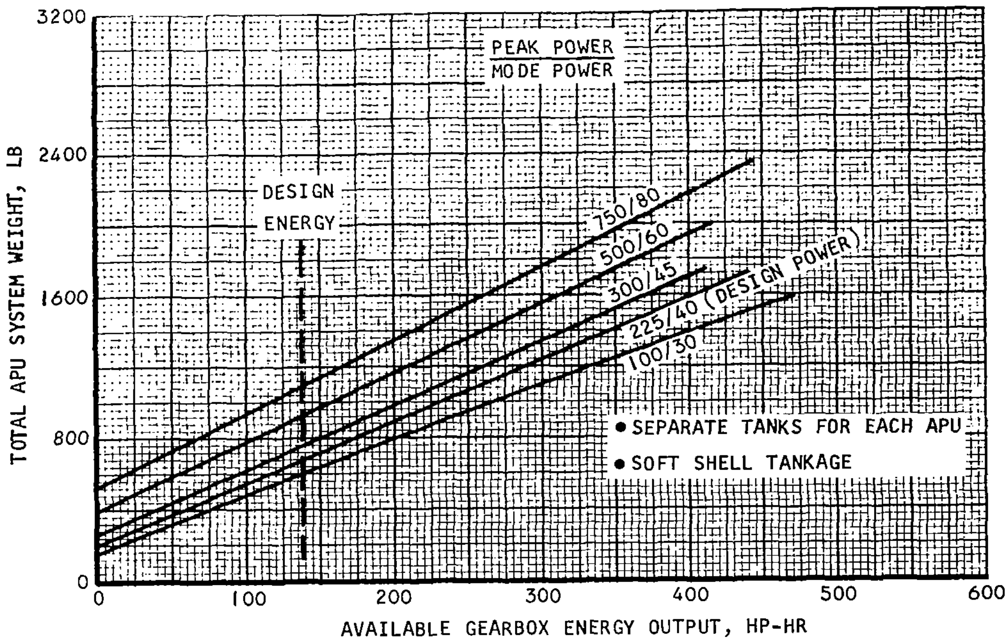

Figure 4-11. Booster Vehicle APU System Weight; Integral High Pressure Cryogenic Supplied Sys+em

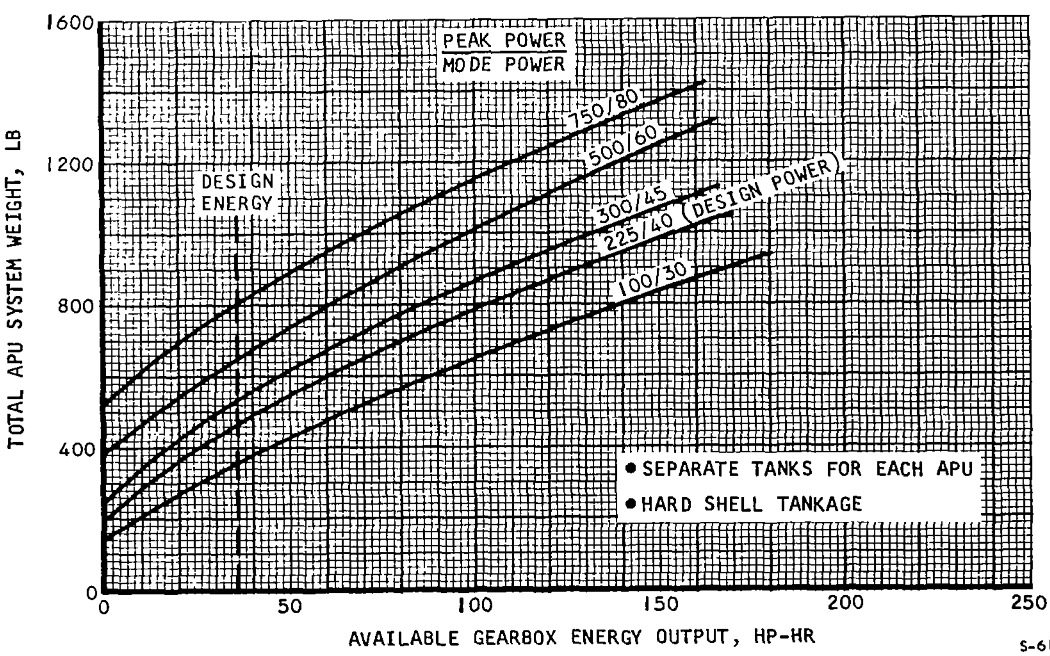

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Figure 4-12. Orbiter Vehicle APU System Weight; Integral High Pressure Cryogenic Supplied System

SECTION 5

GASEOUS H_2-O_2 SUPPLIED SYSTEM OPERATED AT SUPPLY PRESSURE

INTRODUCTION

In this system, the propellants are obtained as high-pressure gases from another sytem. System configuration is somewhat sensitive to the propellant gas inlet temperature. For example, with high propellant inlet temperatures on the order of 500°R, no propellant conditioning is necessary, but the thermal capacity in the propellant flow may not be sufficient under some conditions to meet the system heat sink requirement and a supplemental heat sink (water boiler) will be needed. On the other hand, with low propellant inlet temperatures (200° to 300°R), the heat sink capacity in the propellant flow will be sufficient to reject the internal heat load at acceptable temperature levels. However, to avoid low temperature thermal problems in some components (particularly the oil coolers), it will be necessary to incorporate thermal conditioning provisions similar to those used in the systems described previously in this report. The system schematic given in Figure 5-1 incorporates provisions to meet the entire temperature range. If the inlet temperature range can be restricted, one or the other of the two systems shown in Figures 5-2 and 5-3 will suffice.

High Inlet Temperature Cycle

In this cycle (shown in Figure 5-2) cooling hydrogen flow is recycled through a water boiler (for supplemental cooling and cooling flow augmentation) by a jet pump. The recycle loop flow control senses heat exchanger discharge temperature and regulates recycle flow to maintain temperature at the proper level for cooling of the lubricant and hydraulic fluid. Where supplemental cooling is not required, the recycle loop is shut.

Low Inlet Temperature Cycle

In this cycle, shown schematically in Figure 5-3, hot gas flow from the recuperator is recycled by the jet pump to provide a suitable inlet temperature (on the order of 460°R) to the hydraulic fluid heat exchanger.

Intermediate Inlet Temperature

At intermediate propellant inlet temperature levels (on the order of 400°R), supplemental cooling will be required during some conditions and propellant heating will be required during other conditions. The system shown in Figure 5-1 has provisions to accommodate both requirements and may be needed for intermediate propellant inlet temperature levels, as well as being applicable to conditions where the propellant temperature may vary over a very wide range during the course of a mission.

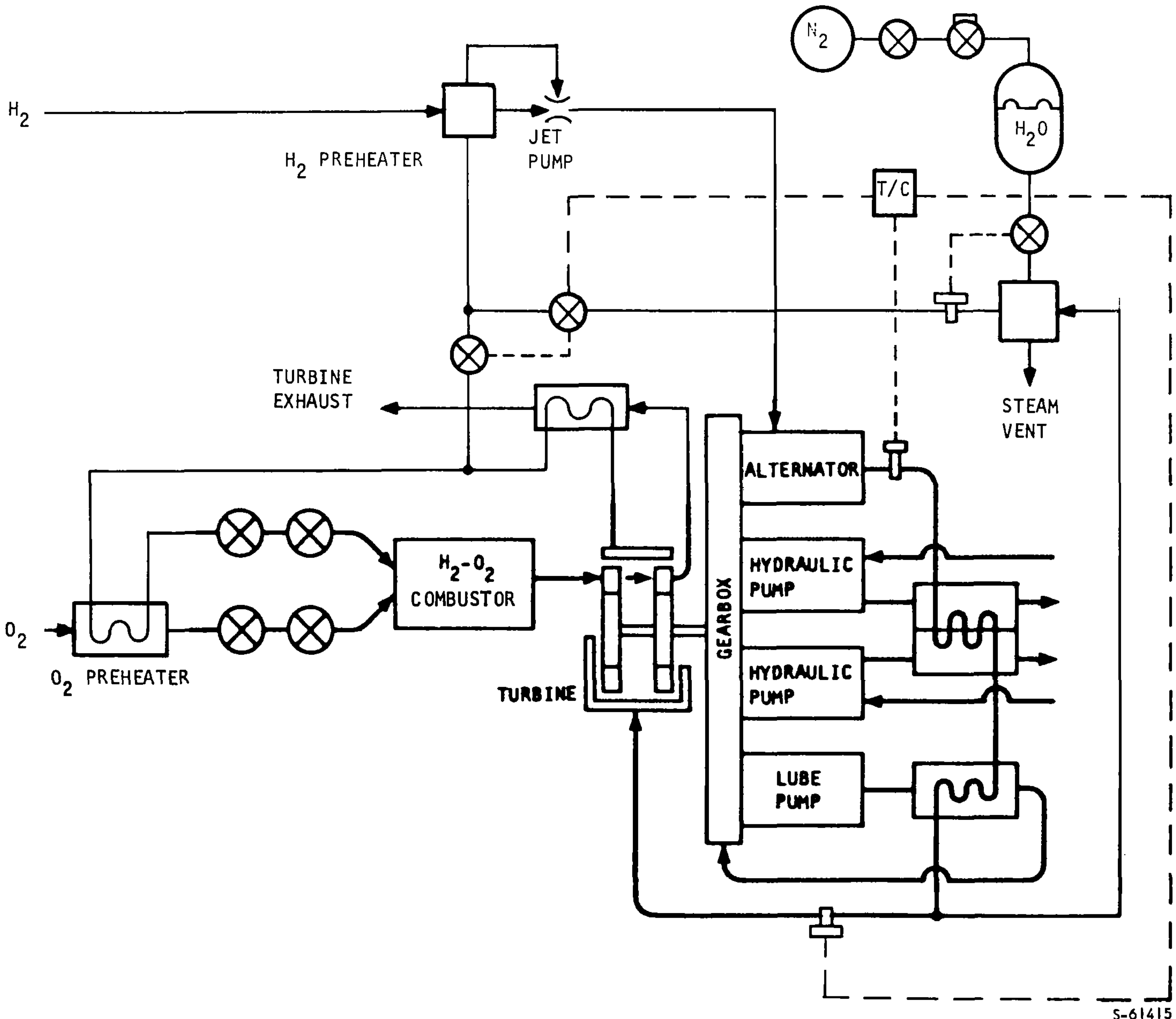

Figure 5-1. High-Pressure Gaseous H_2-O_2 Supplied System

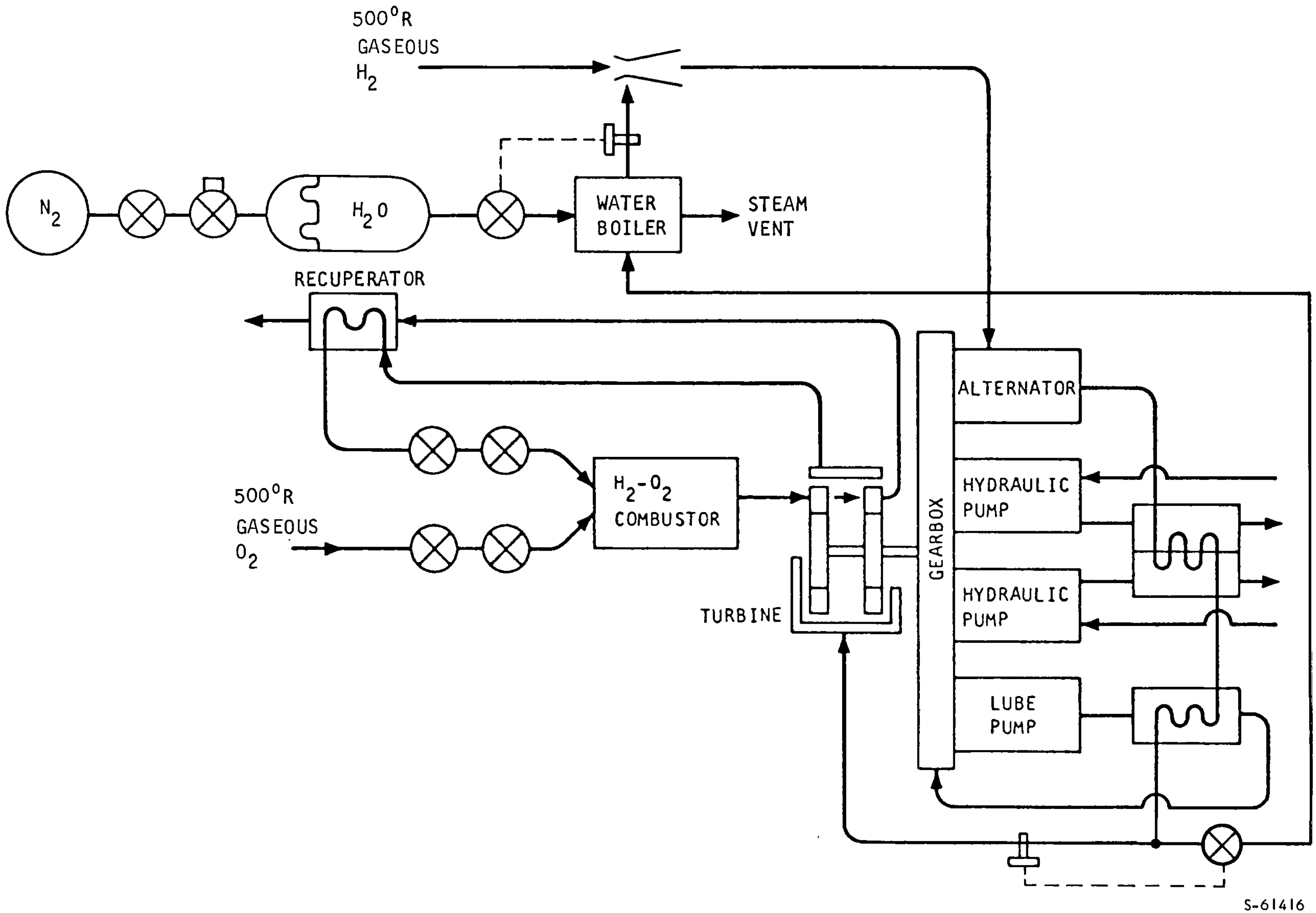

Figure 5-2. High-Pressure Gaseous H_2-O_2 Supplied System/High Inlet Temperature Cycle

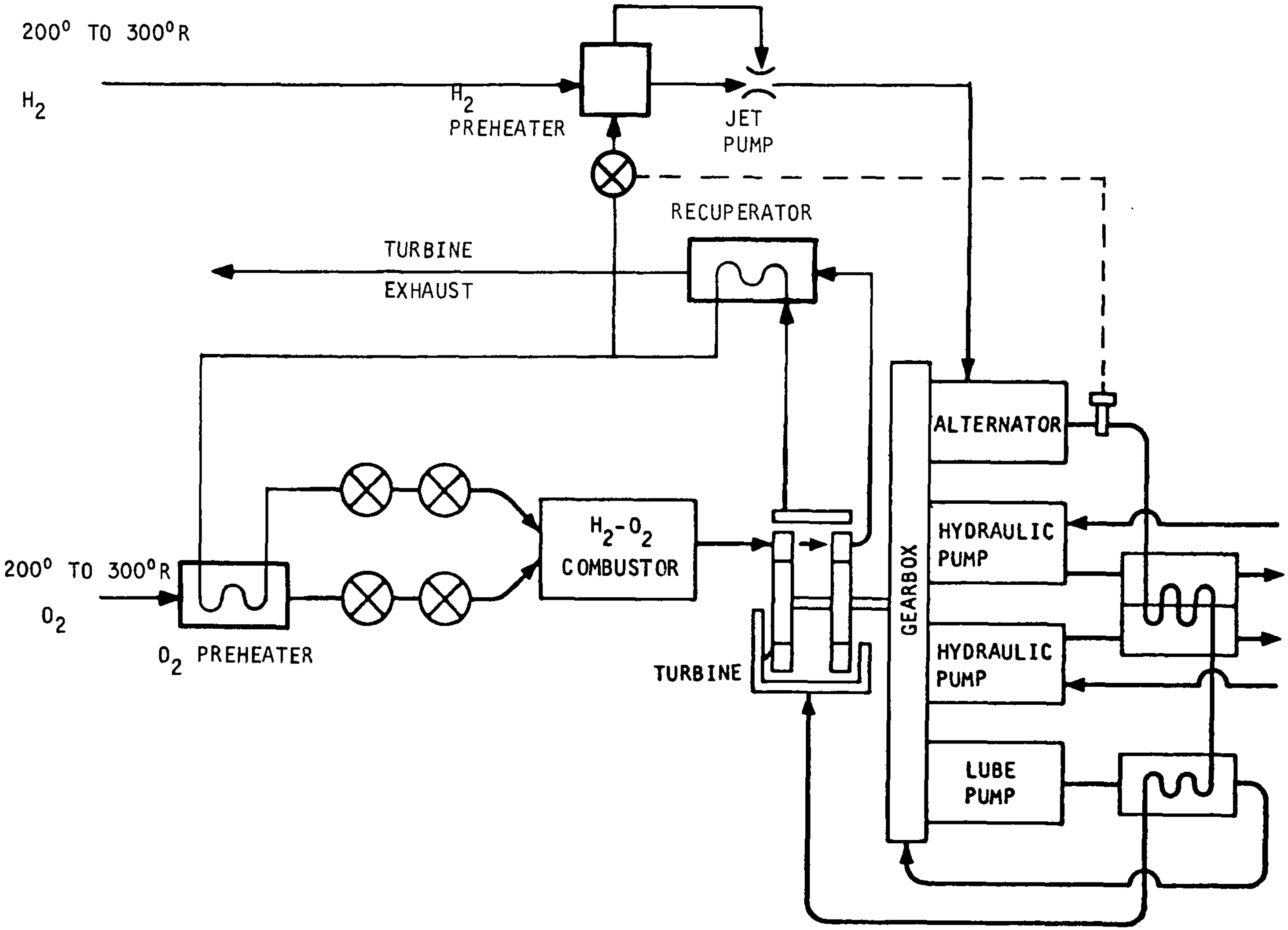

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Figure 5-3. High-Pressure Gaseous H_2-O_2 Supplied System/Low Inlet Temperature Cycle

DESIGN TRADEOFFS

To investigate the effect of gaseous delivery supply temperature and pressure, APU performance maps for inlet temperatures of 200°, 300°, 400°, and 500°R and inlet pressures of 300, 650, 950, and 1250 psia were prepared.

200°R Inlet Temperature

Figures 5-4 through 5-7 show the performance maps for various inlet pressures. The system schematic given in Figure 5-3 is applicable to this inlet temperature. That is, propellant flow has sufficient thermal capacity to provide the entire heat sink for the system and no supplemental cooling is required.

300°R Inlet Temperature

Figures 5-8 through 5-11 give the system performance maps. No supplemental heat sink is required and the system schematic given in Figure 5-3 is applicable to this inlet temperature.

400°R Inlet Temperature

As shown in the performance maps given in Figures 5-12 through 5-15, supplemental cooling is needed under some conditions. Under other conditions, it will be necessary to thermally condition the propellant flow for it to provide the proper temperature level for cooling. As a consequence of these two requirements, the system schematic shown in Figure 5-1 is applicable to this inlet temperature.

500°R Inlet Temperature

The performance maps for a 500°R inlet temperature are given in Figures 5-16 through 5-19. In this case, no thermal conditioning of the propellant flow is necessary and supplemental cooling is needed under some conditions. Therefore, the system schematic given in Figure 5-3 is applicable to this inlet temperature.

Nonrecuperative Cycle

As the propellant feed temperature increases, a point will be ultimately reached where regeneration is not justified. To explore this condition, additional 500°R cases were considered using a nonrecuperative cycle. Figure 5-20 is a performance map for the 300 psia 500°R inlet case. As with recuperative cycles operating at the same inlet conditions, supplemental cooling is needed only at low ambient pressure and low power level.

APU Expendable Requirements

The cycle performance maps described previously were fed into the mission integration program to determine the expendable requirements. Tables 5-1 and 5-2 show the APU expendable requirements (hydrogen, oxygen, and water) as a

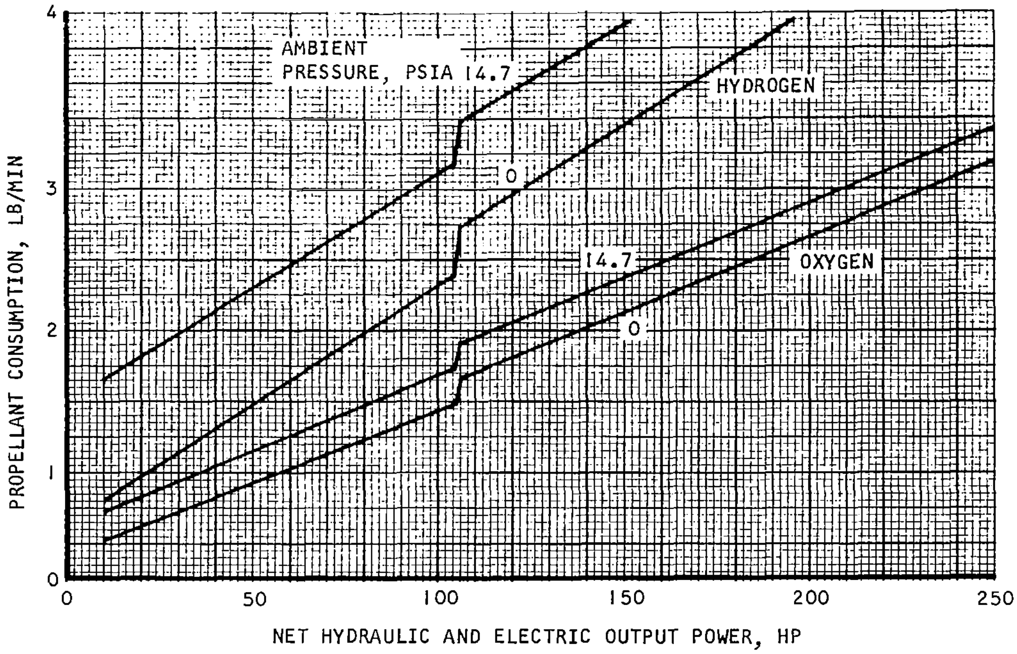


Figure 5-4. APU Performance Map; 200°R, 300 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

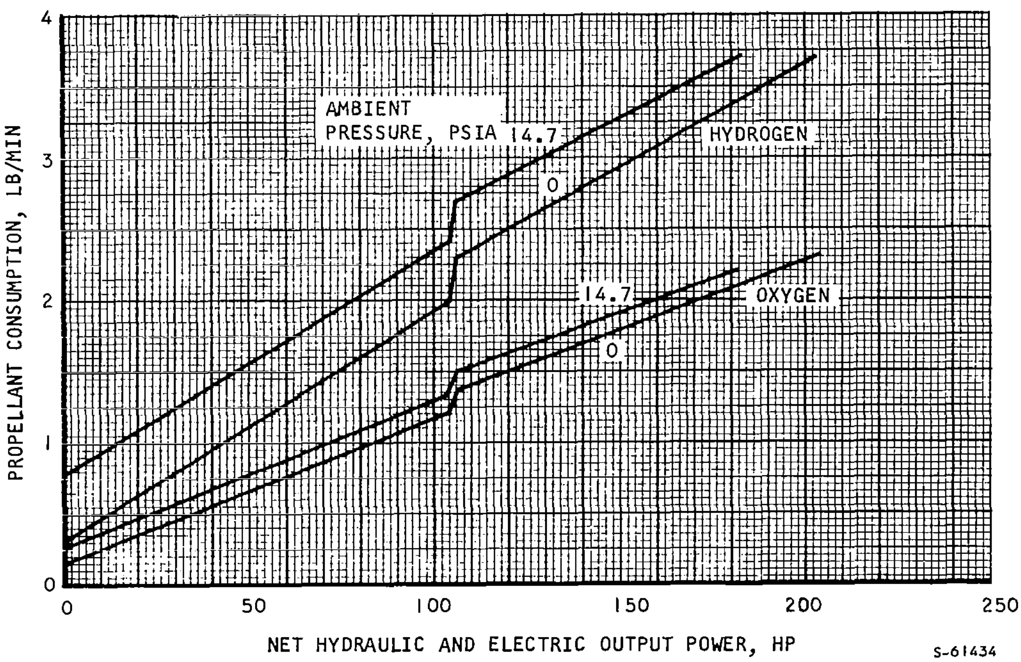


Figure 5-5. APU Performance Map; 200°R, 650 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

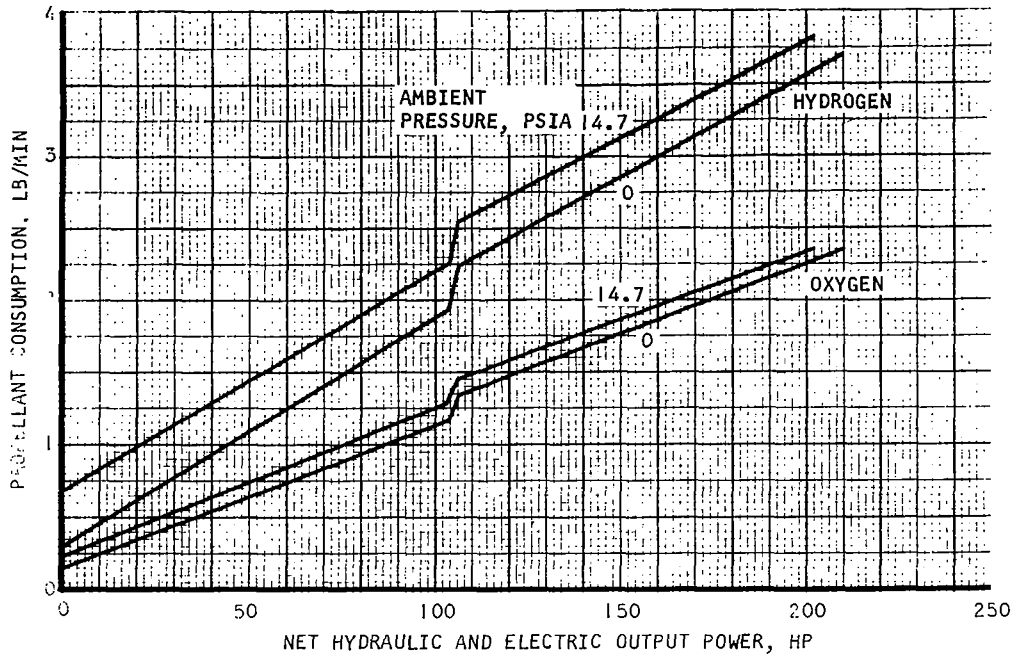


Figure 5-6. APU Performance Map; 200°R, 950 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

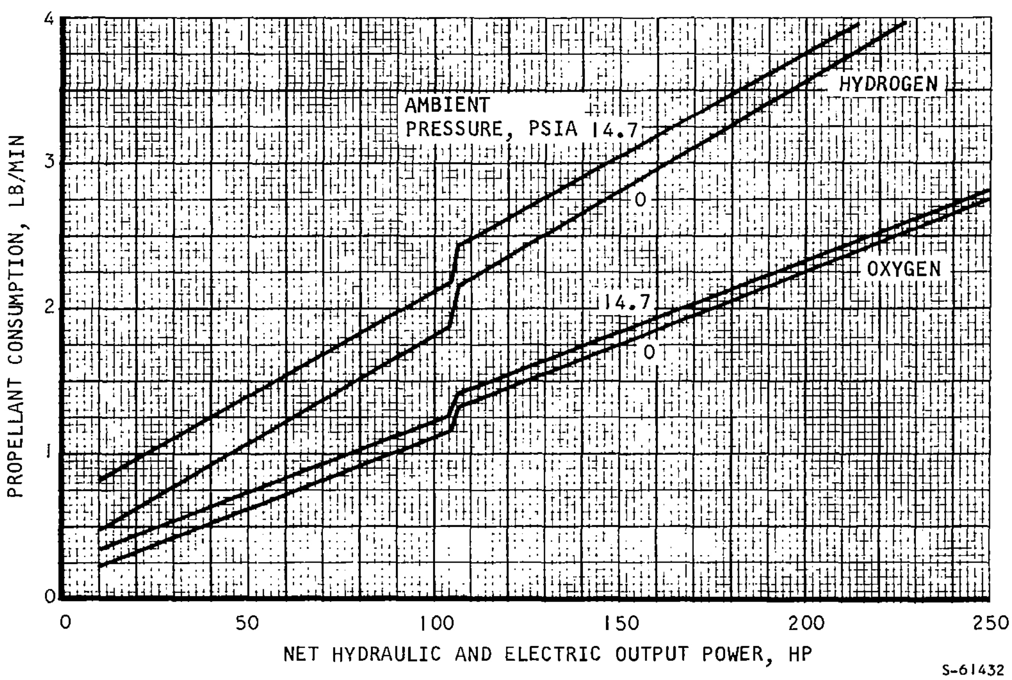


Figure 5-7. APU Performance Map; 200°R, 1250 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

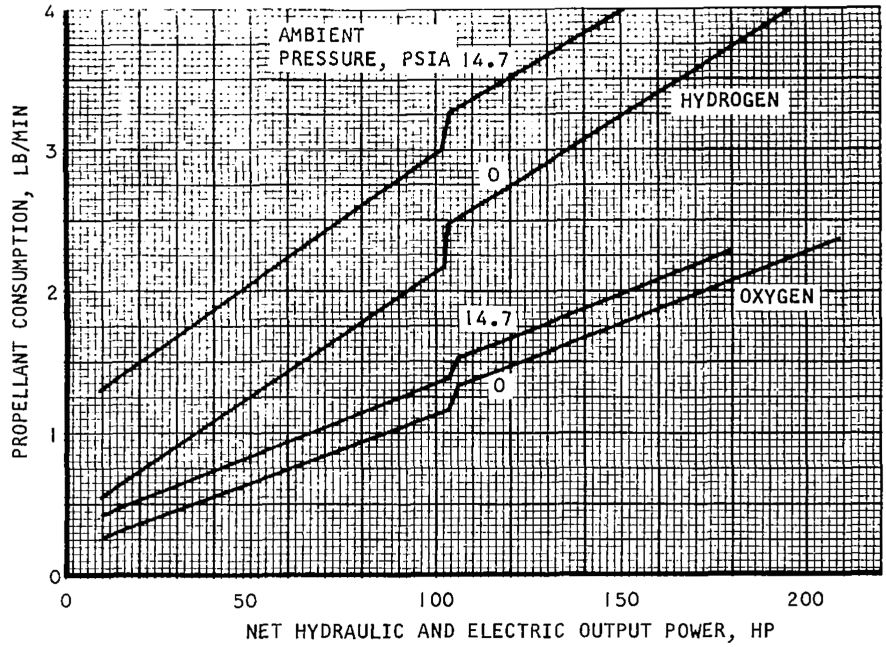


Figure 5-8. APU Performance Map; 300^0R, 300 psia Gas Feed, Turbine Design Point = Altitude Mode Power

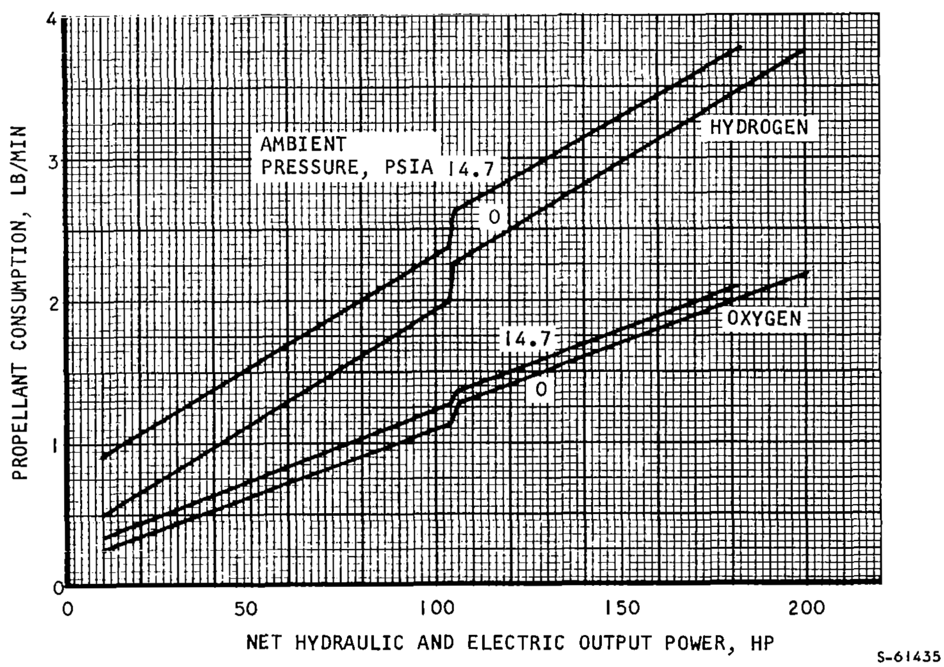


Figure 5-9. APU Performance Map; 300^0R, 650 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

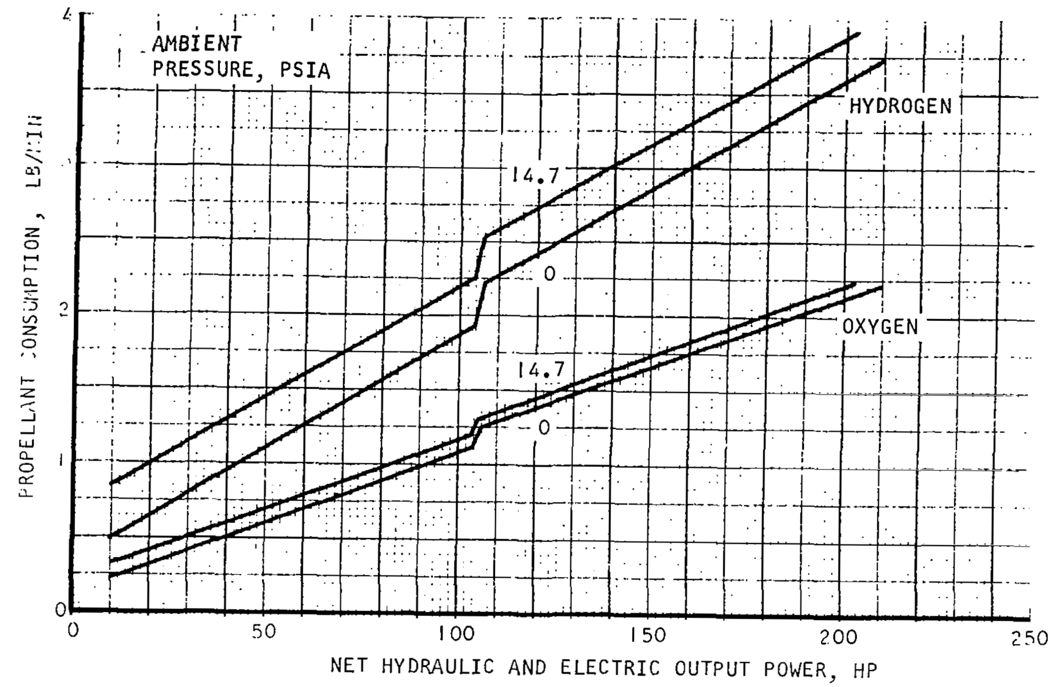


Figure 5-10. APU Performance Map, 300°R, 950 psi Gas Feed, Turbine Design Point = Altitude, Mode Power

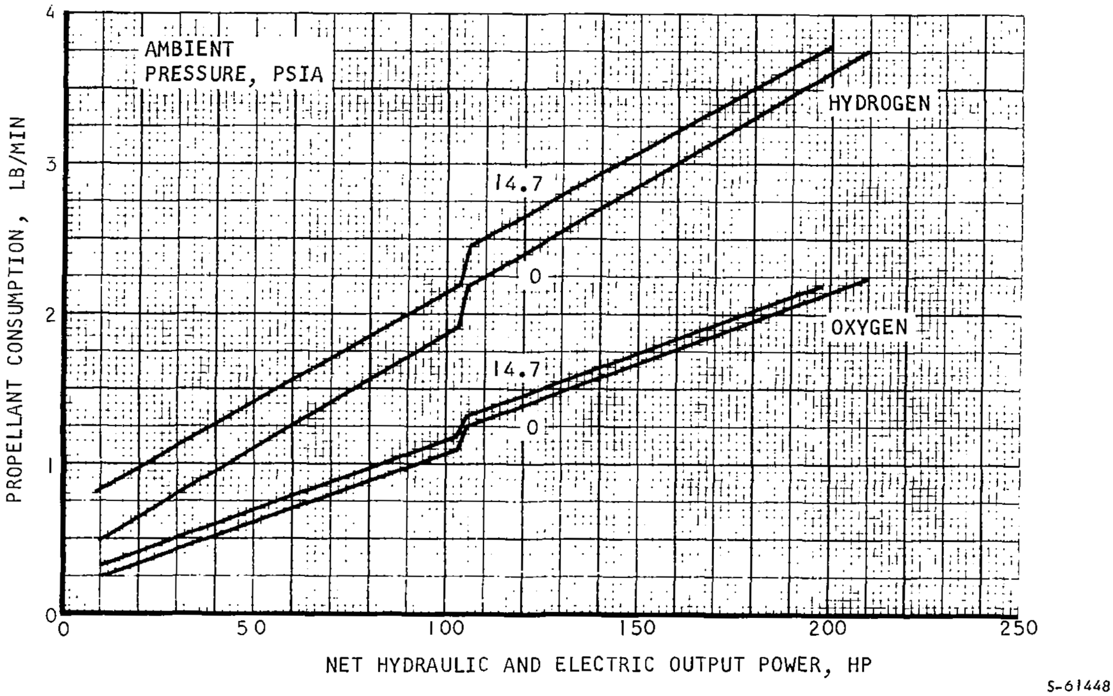


Figure 5-11. APU Performance Map, 300°R, 250 psi Gas Feed, Turbine Design Point = Altitude, Mode Power

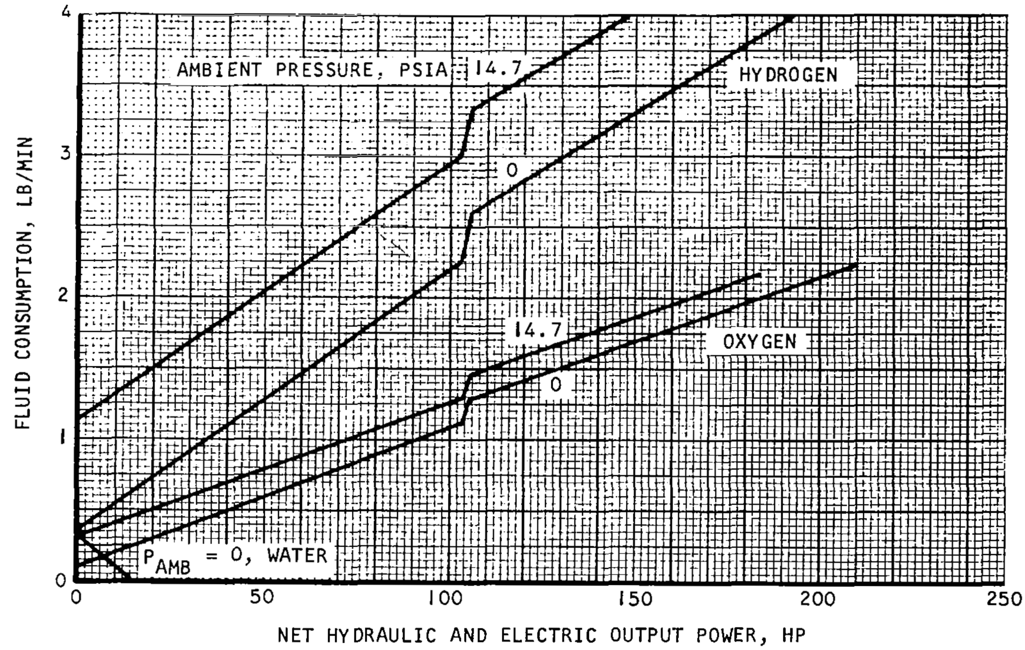


Figure 5-12. APU Performance Map; 400°R, 300 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

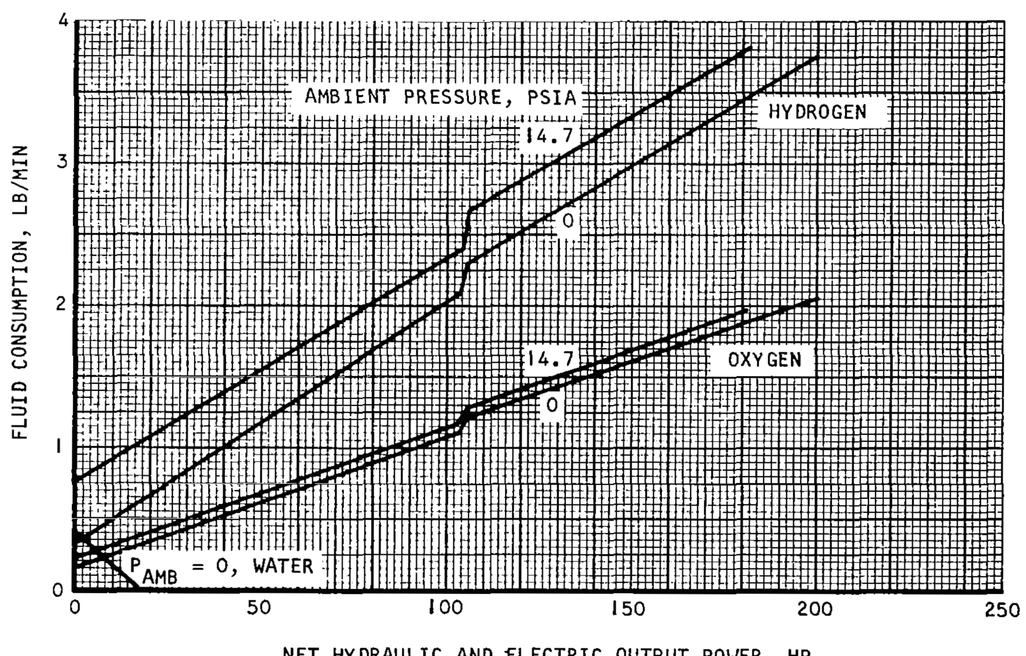


Figure 5-13. APU Performance Map; 400°R, 650 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

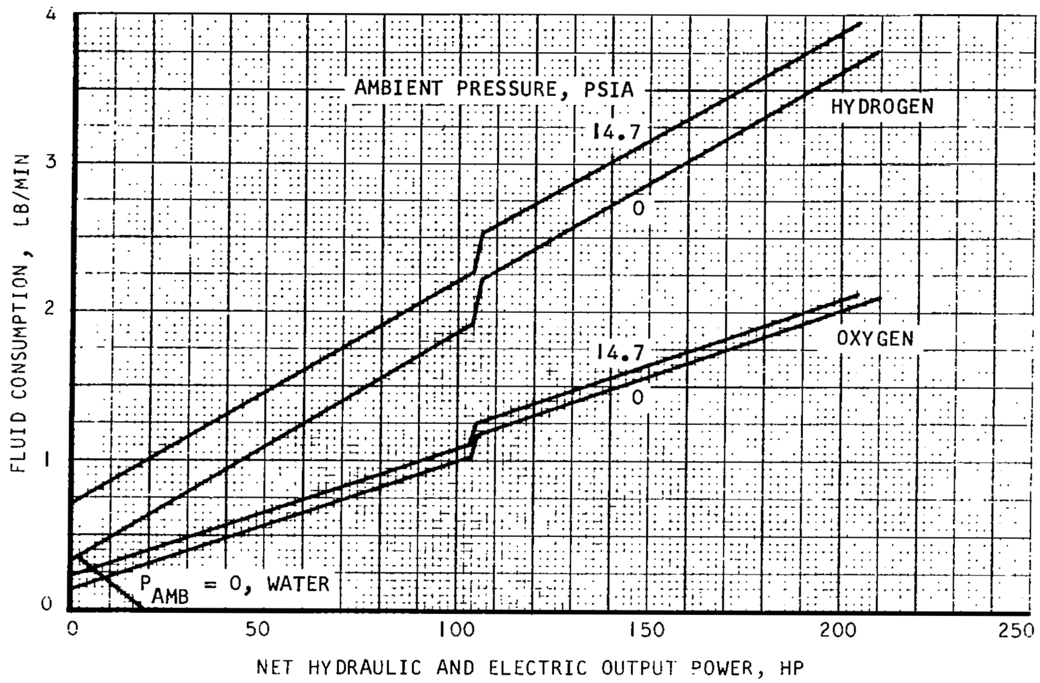


Figure 5-14. APU Performance Map; 400°R, 950 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

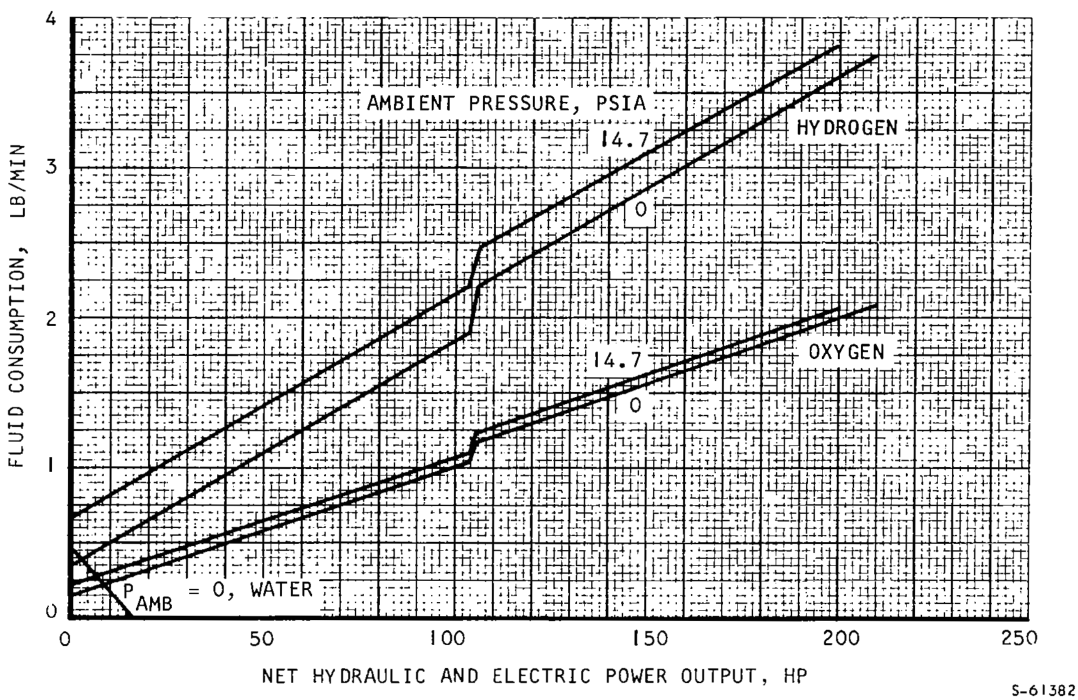


Figure 5-15. APU Performance Map; 400°R, 1250 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

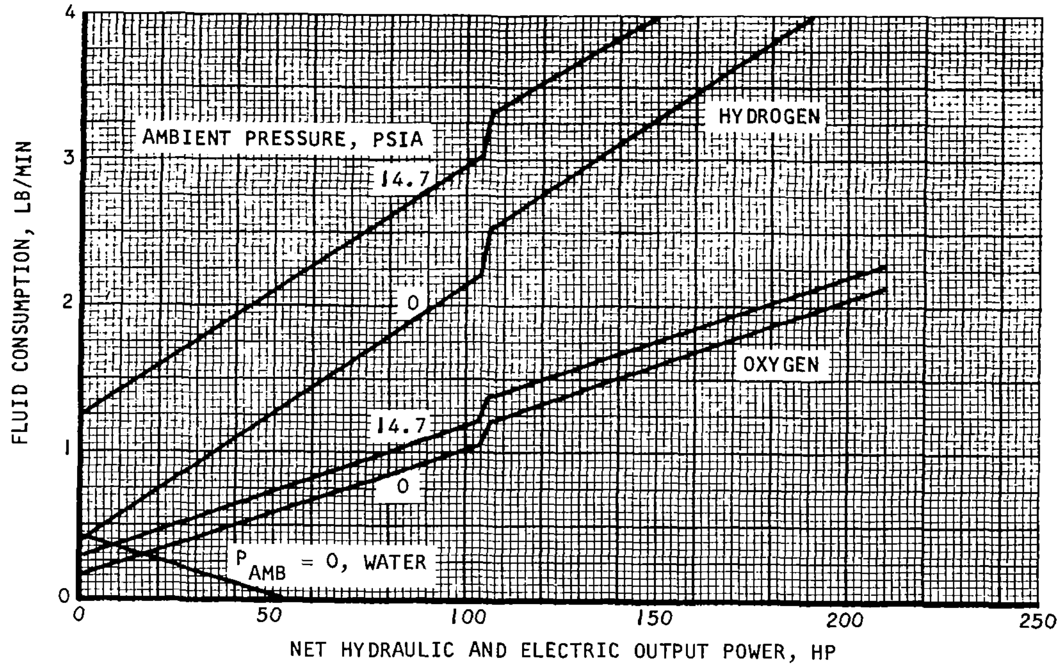


Figure 5-16. APU Performance Map; 500°R, 300 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

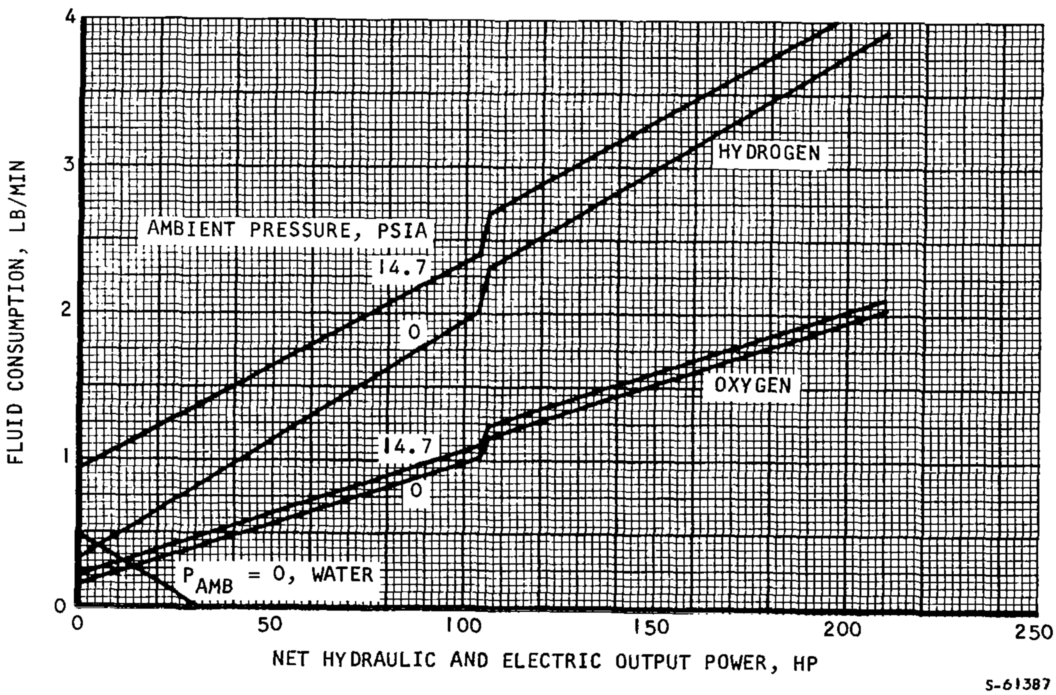


Figure 5-17. APU Performance Map; 500°R, 650 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

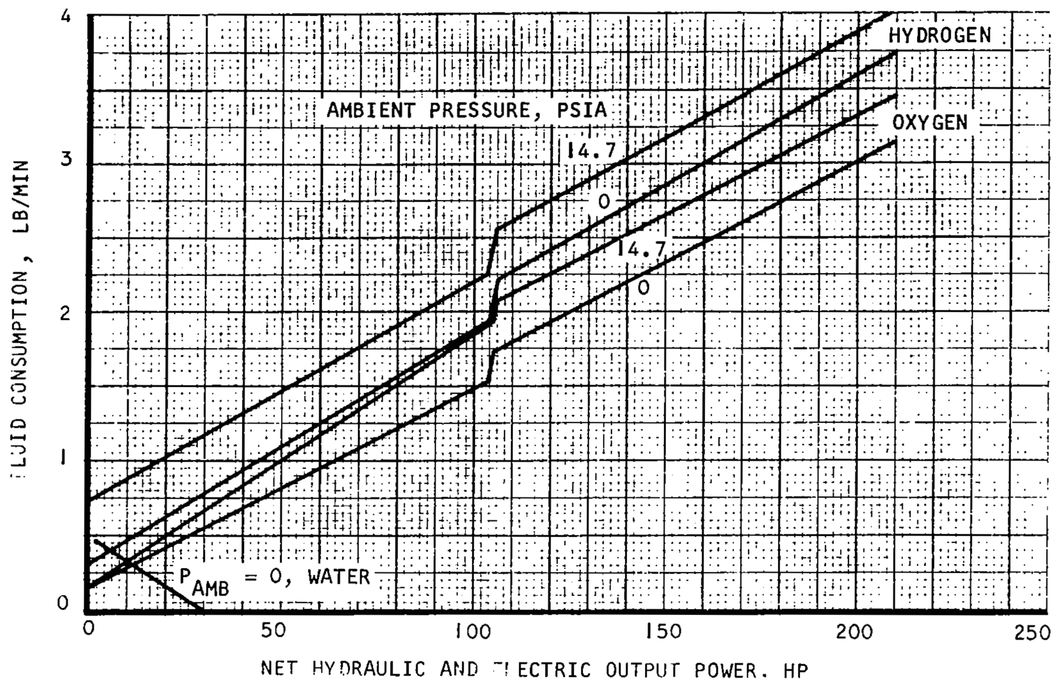


Figure 5-18. APU Performance Map; 500°R, 950 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

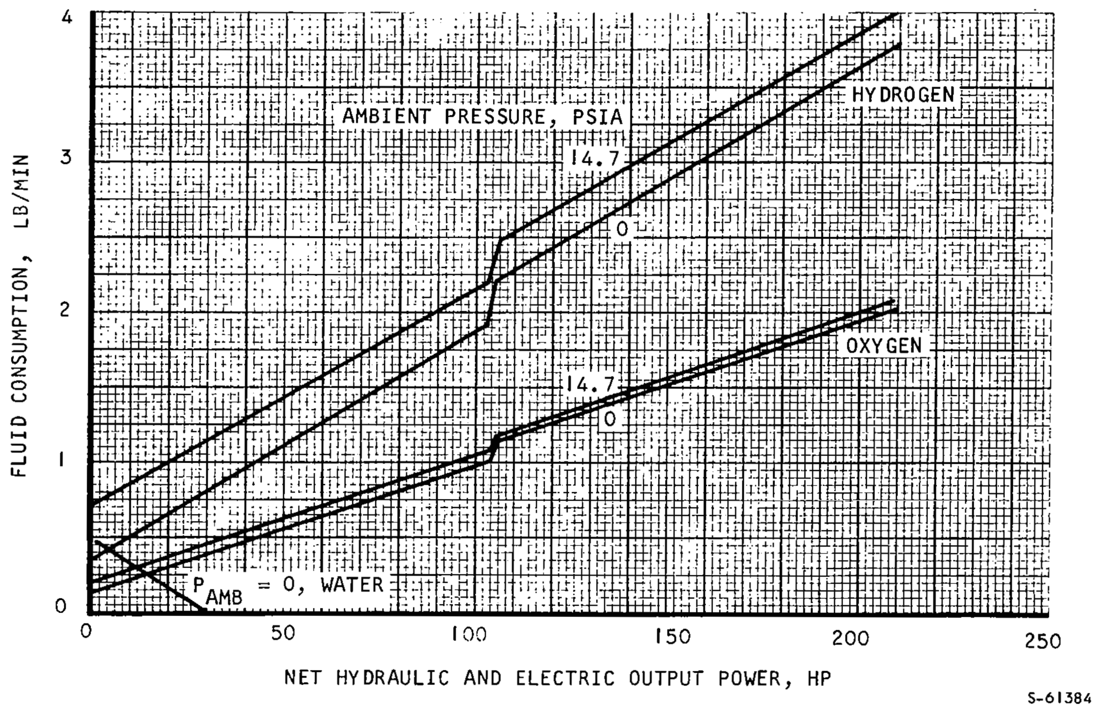


Figure 5-19. APU Performance Map; 500°R, 1250 psia Gas Feed, Turbine Design Point = Altitude, Mode Power

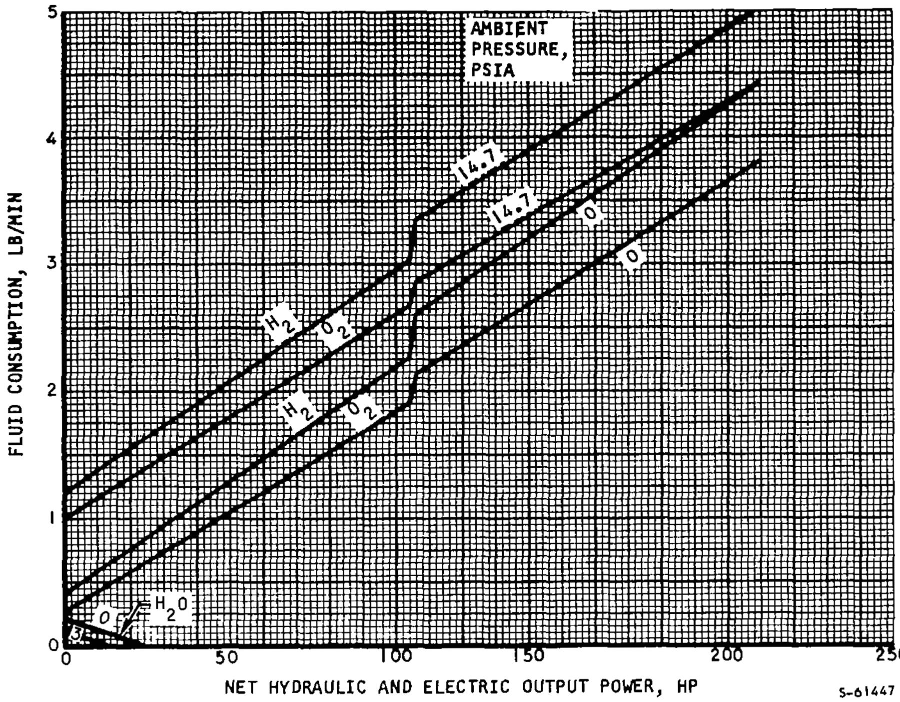


Figure 5-20. APU Performance Map; 500^0R, 300 psia Gas Fixed, Nonrecuperated; Altitude, Mode Power Design Point

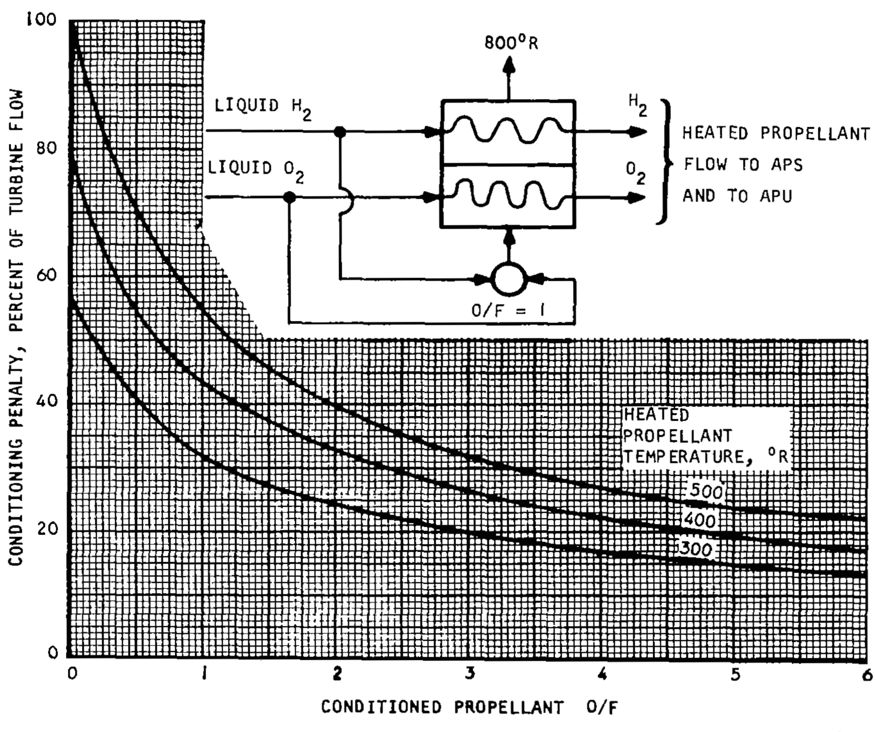


Figure 5-21. Propellant Conditioning Penalty

TABLE 5-1

EXPENDABLE REQUIREMENTS FOR BOOSTER MISSION

Propellant Inlet Temperature	Expendables Required (lb) for Inlet Pressure and Temperature											
	300 psia			650 psia			950 psia			1250 psia		
	H_2	O_2	H_2O	H_2	O_2	H_2O	H_2	O_2	H_2O	H_2	O_2	H_2O
Recuperative cycle												
200°R	242.5	100.4	-	179.4	85.0	-	169.3	82.0	-	163.8	80.2	-
300°R	233.8	91.8	-	178.0	78.5	-	169.1	76.8	-	164.4	75.7	-
400°R	236.7	82.7	0	179.5	74.7	0.016	169.7	73.0	0.249	164.7	71.6	0.42
500°R	249.2	84.1	1.4	184.7	72.6	2.5	174.1	70.7	2.7	169.7	69.7	3.0
Nonrecuperative Cycle												
500°R	245.6	205.0	0.937	-	-	-	-	-	-	-	-	-

TABLE 5-2

EXPENDABLE REQUIREMENTS FOR ORBITER MISSION

Propellant Inlet Temperature	Expendables Required (lb) for Inlet Pressure and Temperature											
	300 psia			650 psia			950 psia			1250 psia		
	H_2	O_2	H_2O	H_2	O_2	H_2O	H_2	O_2	H_2O	H_2	O_2	H_2O
Recuperative Cycle												
200°R	60.0	27.4	-	48.5	24.8	-	47.1	24.3	-	46.3	23.6	-
300°R	59.1	25.4	-	48.8	23.0	-	47.5	22.7	-	46.4	22.6	-
400°R	60.2	21.5	5.7	49.8	22.5	7.4	47.6	22.1	7.5	46.47	21.89	9.5
500°R	61.9	24.0	9.5	50.2	22.0	12.5	48.3	21.7	13.0	47.2	21.6	13.5
Nonrecuperative Cycle												
500°R	63.1	49.2	8.529	-	-	-	-	-	-	-	-	-

function of the propellant inlet conditions for the booster and orbiter missions. It is necessary to consider the penalties to the vehicle for propellant conditioning to properly evaluate overall performance.

SYSTEM INTEGRATION

The NASA guidelines for the gaseous propellant supplied system specify inlet conditions to the cycle of 300 psia and 500°R. Presumably, the propellants would be provided by the auxiliary propulsion system (APS). However, it should be emphasized that the propellant supply and conditioning functions provided by the APS are accomplished at a weight penalty to that system, and as a consequence, this should be taken into account in evaluation of this system relative to others which have integral propellant conditioning provisions.

Propellant Pumping

Propellant pumping can be efficiently performed by the APS turbopump. Presumably, the APU could be supplied from the APS propellant accumulators at relatively low cost to that system because of the relatively low APU propellant flow requirements. It appears that it will not be necessary to penalize the APS (by using larger accumulators or higher capacity pumps) for the propellant pumping function.

Propellant Thermal Conditioning

Thermal conditioning of the APU propellant flows may impose penalties on the APS supply system equivalent to the APU propellant weight. The penalties to the APS for thermal conditioning depend upon the conditioned propellant O/F ratio and temperature, as shown in Figure 5-21. For the low O/F ratios required by the APU, relatively high conditioning penalties are obtained. For example, at an O/F = 0.45 and a 500°R conditioned temperature, the conditioning penalty is represented by 73.5 percent of the APU propellant flow (not including the APS tankage penalties for storage of these additional propellants). Of course, the penalties to the APS can be reduced by reducing the conditioned propellant temperature. (There is substantial incentive in APS design optimization for use of minimum conditioned temperature.)

Heating of Propellant in Feedlines

Since the APU may be installed some distance from the propellant source in the APS, there is a question of heat leak in the propellant feedlines, and its effect on APU design. Figure 5-22 shows the results of a preliminary analysis of feedline heat leak. For the range of ducting run applicable here, it appears that the heat leak in the feedlines will be low and can be reduced to negligible levels by use of appropriate thermal insulation.

Propellant Supply

The APS will be penalized by the direct propellant requirements (given in Tables 5-1 and 5-2) and by propellant required for thermal conditioning the APU propellant flow in the APS. Tables 5-3 and 5-4 give the total hydrogen

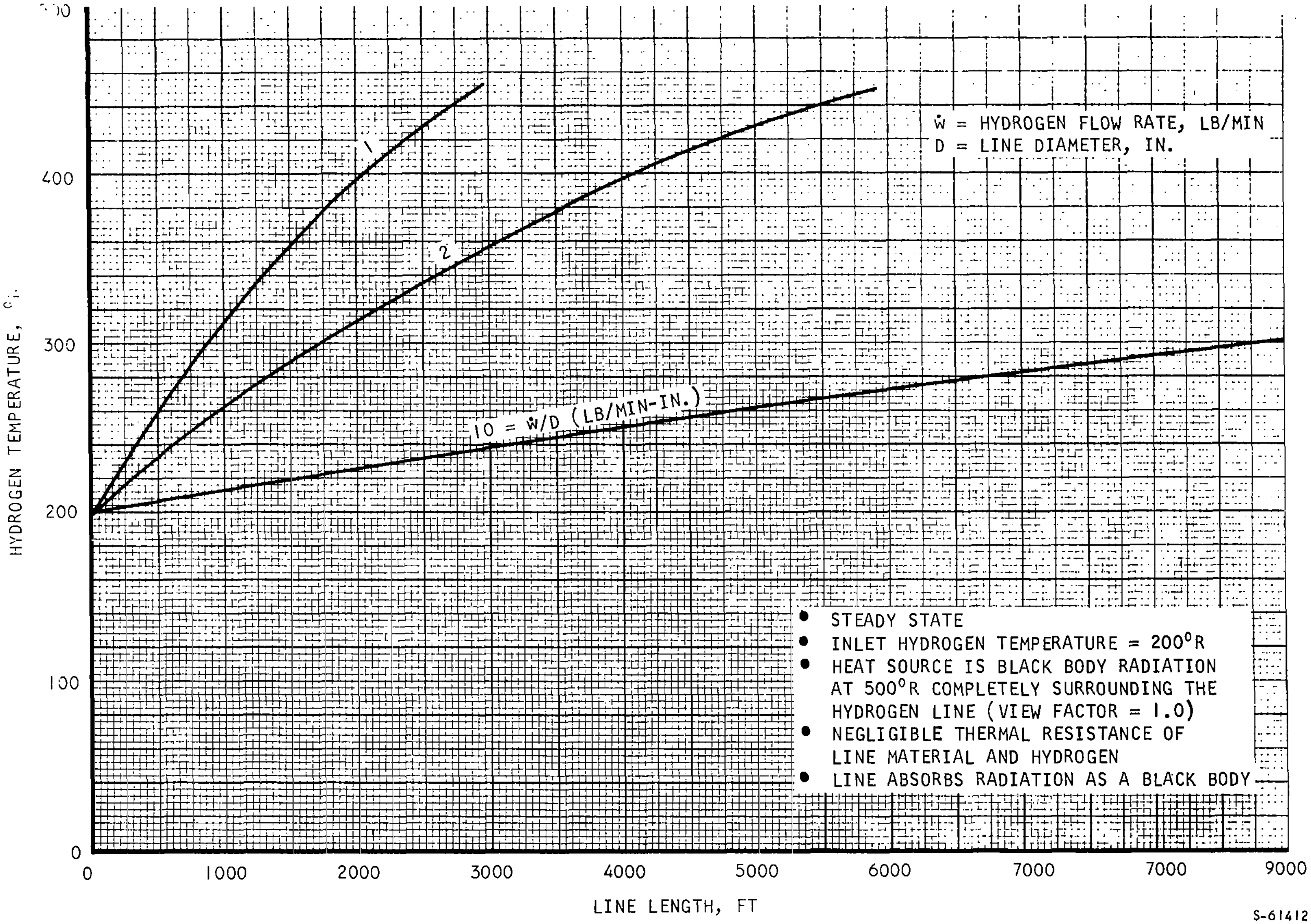


Figure 5-22. Steady-State Hydrogen Temperature as a Function of Line Length, Diameter and Flow Rate When Subjected to Radiation Heat Source

TABLE 5-3

TOTAL PROPELLANT AND CONDITIONING REQUIREMENTS FOR BOOSTER MISSION

Propellant Inlet Temperature	Expendables Required (lb) for Inlet Pressure and Temperature							
	300 psia		650 psia		950 psia		1250 psia	
	H_2	O_2	H_2	O_2	H_2	O_2	H_2	O_2
Recuperative								
200°R	292.7	154.2	217.1	122.7	204.4	117.1	198.1	114.5
300°R	305.3	163.3	231.8	132.3	220.7	128.4	214.9	126.2
400°R	333.7	183.9	253.0	148.2	238.5	141.8	232.1	139.0
500°R	380.2	215.1	280.7	168.6	264.6	161.2	259.0	159.0
Nonrecuperated								
500°R	378.6	338.0	-	-	-	-	-	-

TABLE 5-4

TOTAL PROPELLANT AND CONDITIONING REQUIREMENTS FOR ORBITER MISSION

Propellant Inlet Temperature	Expandables Required (lb) for Inlet Pressure and Temperature							
	300 psia		650 psia		950 psia		1250 psia	
	H_2	O_2	H_2	O_2	H_2	O_2	H_2	O_2
Recuperative								
200°R	72.4	39.8	59.0	35.3	56.9	34.1	55.9	33.2
300°R	77.3	43.6	64.3	38.5	61.9	37.1	60.7	36.9
400°R	84.9	45.8	70.3	43.0	67.3	41.8	65.7	41.1
500°R	94.7	56.8	76.7	48.5	73.8	47.2	72.2	46.6
Nonrecuperated								
500°R	102.2	88.3	-	-	-	-	-	-

and oxygen required from the APS propellant supply as a result of the APU propellant flow.

Figures 5-23 and 5-24 show that there is considerable incentive for operation at maximum pressure and minimum inlet temperature conditions. It should be emphasized that if this function is integrated into the APS propellant conditioning, the requirements for the APS will determine the inlet propellant conditions.

SYSTEM PERFORMANCE

As indicated previously, cycle configuration is somewhat dependent upon the propellant inlet temperature. Three cycle modifications have been identified according to whether (1) recycling of warm gas is needed for propellant conditioning, (2) recycling of cooled gas and a supplemental heat sink are required for system thermal control, or (3) both functions are needed in the system at different times.

Low Inlet Temperature Cycle

Figures 5-25 and 5-26 are typical cycle state point diagrams for a low inlet temperature (200°R) cycle at sea level- high-power-output and space low-power-output conditions. As with previous systems, the temperature of the hydraulic fluid and lubricant increase with decreasing power output and ambient pressure. The temperature levels shown for the hydraulic fluid and lubricant at the space low-power-altitude condition can be reduced by increasing heat exchanger effectivenesses (at a penalty in system fixed weight) or by various cycle modifications (such as relocation of the recirculation loop from downstream of the recuperator to downstream of the turbine housing heat exchanger or the lube oil heat exchanger). These modifications have small effect on cycle performance.

High Inlet Temperature Cycle

Figures 5-27 and 5-28 are typical cycle state point diagrams for a high inlet temperature (500°R) cycle at sea-level high power-output and space low-power-output conditions. At the high power condition, the propellant flow provides an adequate heat sink. At low-power low-ambient-pressure conditions, supplemental cooling is provided by a water boiler. In this case, the water boiler recirculation loop flow is modulated to maintain the hydrogen temperature at the exit of the lubricating oil heat exchanger at 850°R.

Intermediate Inlet Temperature Cycle

Figures 5-29 and 5-30 show typical cycle state-point diagrams for an intermediate inlet temperature (400°R) system. At the high-power-output sea-level condition, warm gas is recycled to bring the hydraulic fluid inlet hydrogen temperature to 460°R. At the low-power-output zero-pressure condition, gas cooled by the water boiler is recycled to maintain the lubricating oil hydrogen outlet temperature below 850°R.

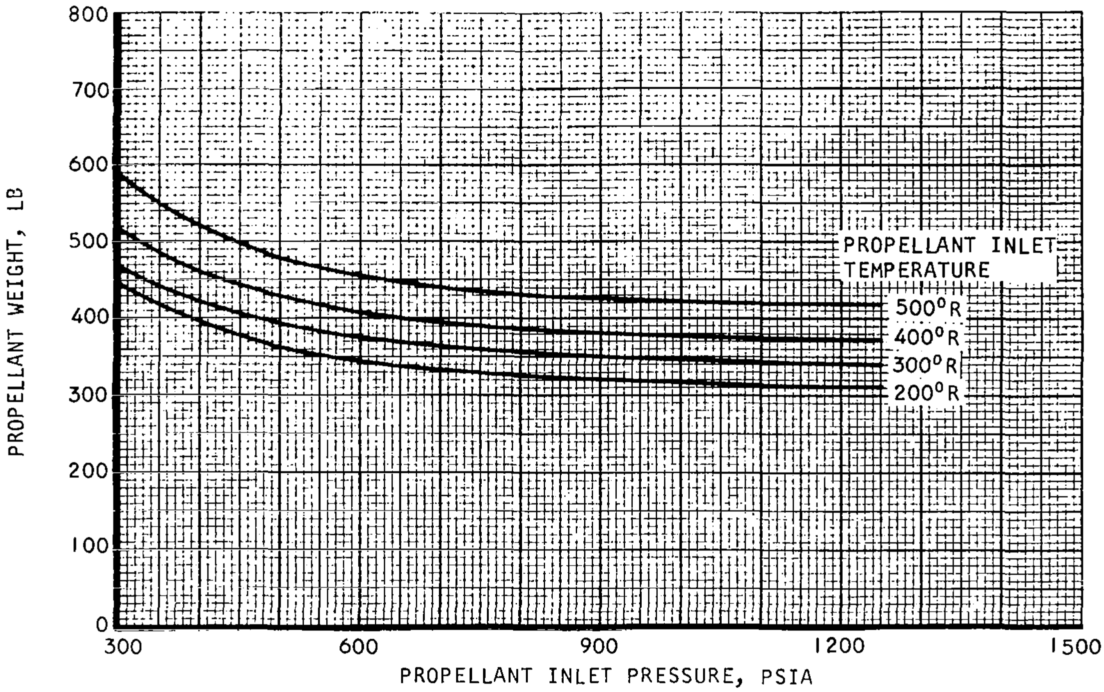


Figure 5-23. Booster Vehicle Total Propellant Requirements

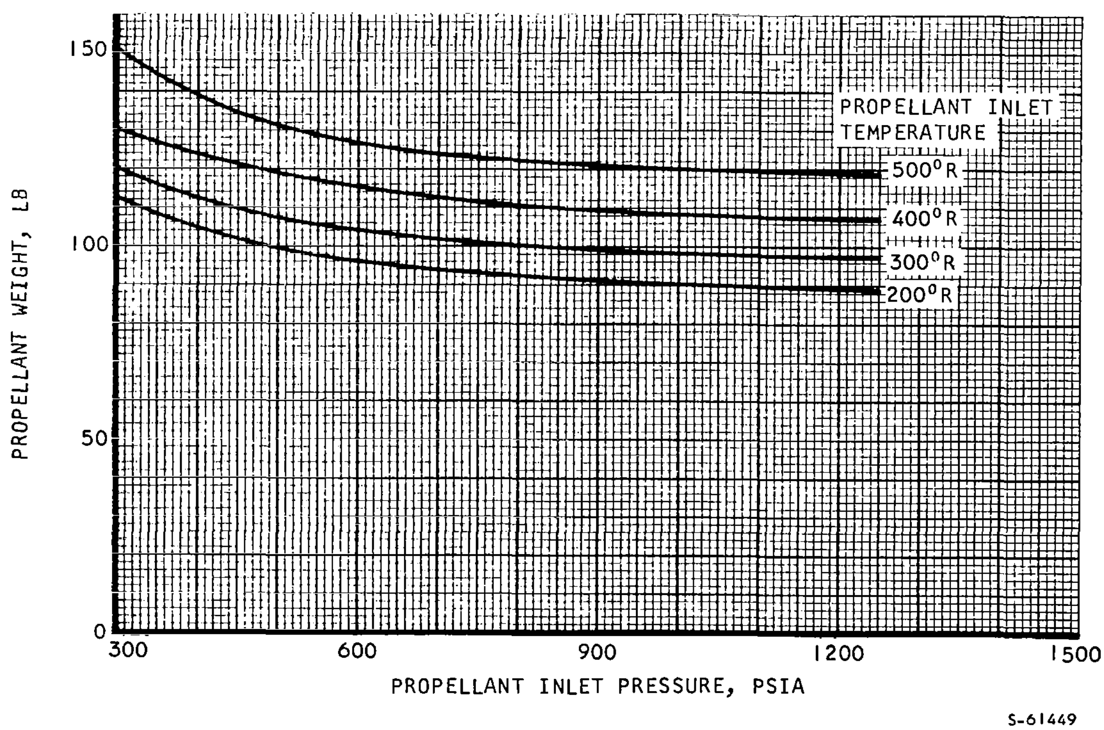


Figure 5-24. Orbiter Vehicle Total Propellant Requirements

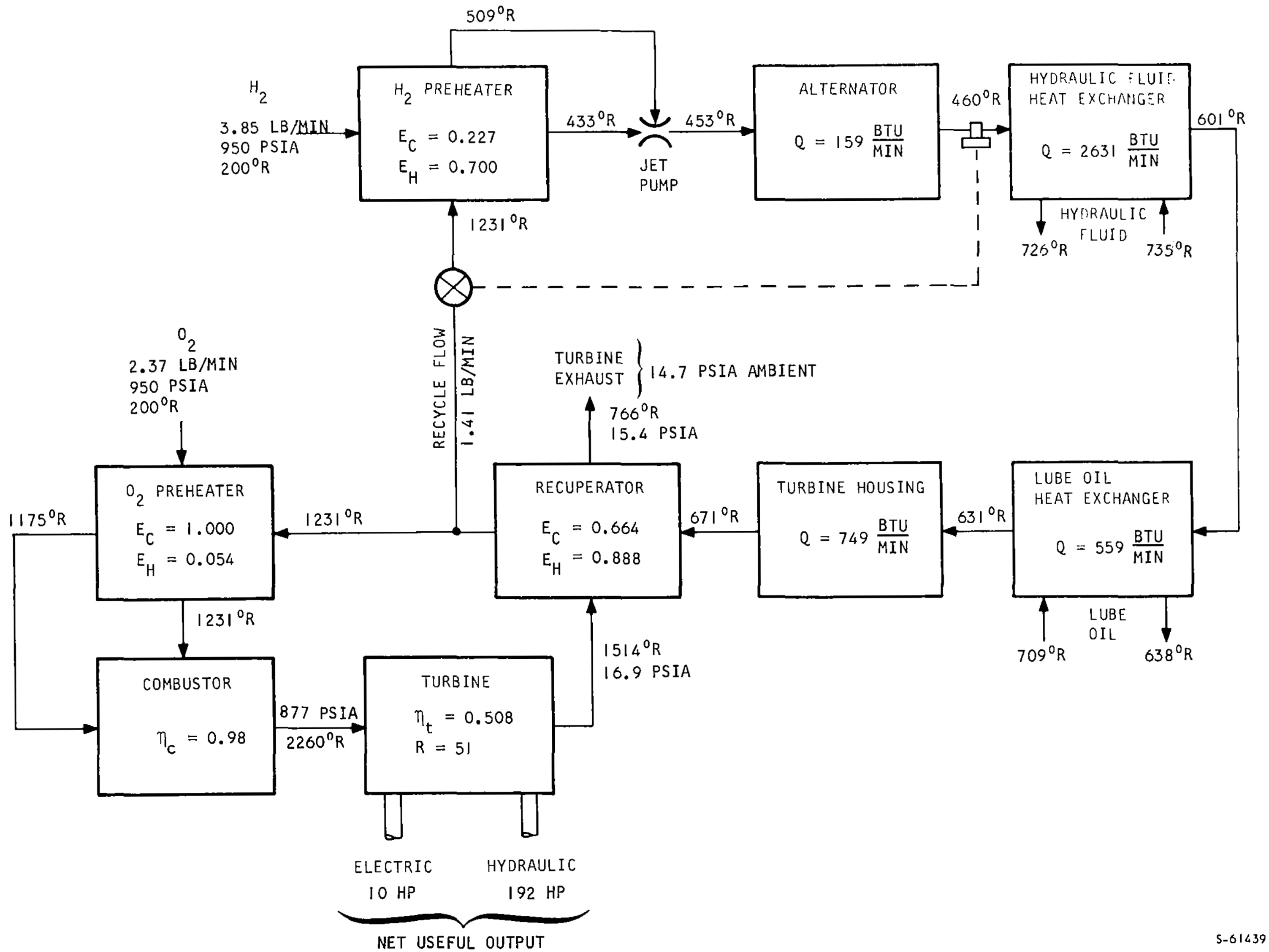


Figure 5-25. Sea Level, Full Power Cycle State Points, 200°R Gaseous H_2-O_2 Supplied System

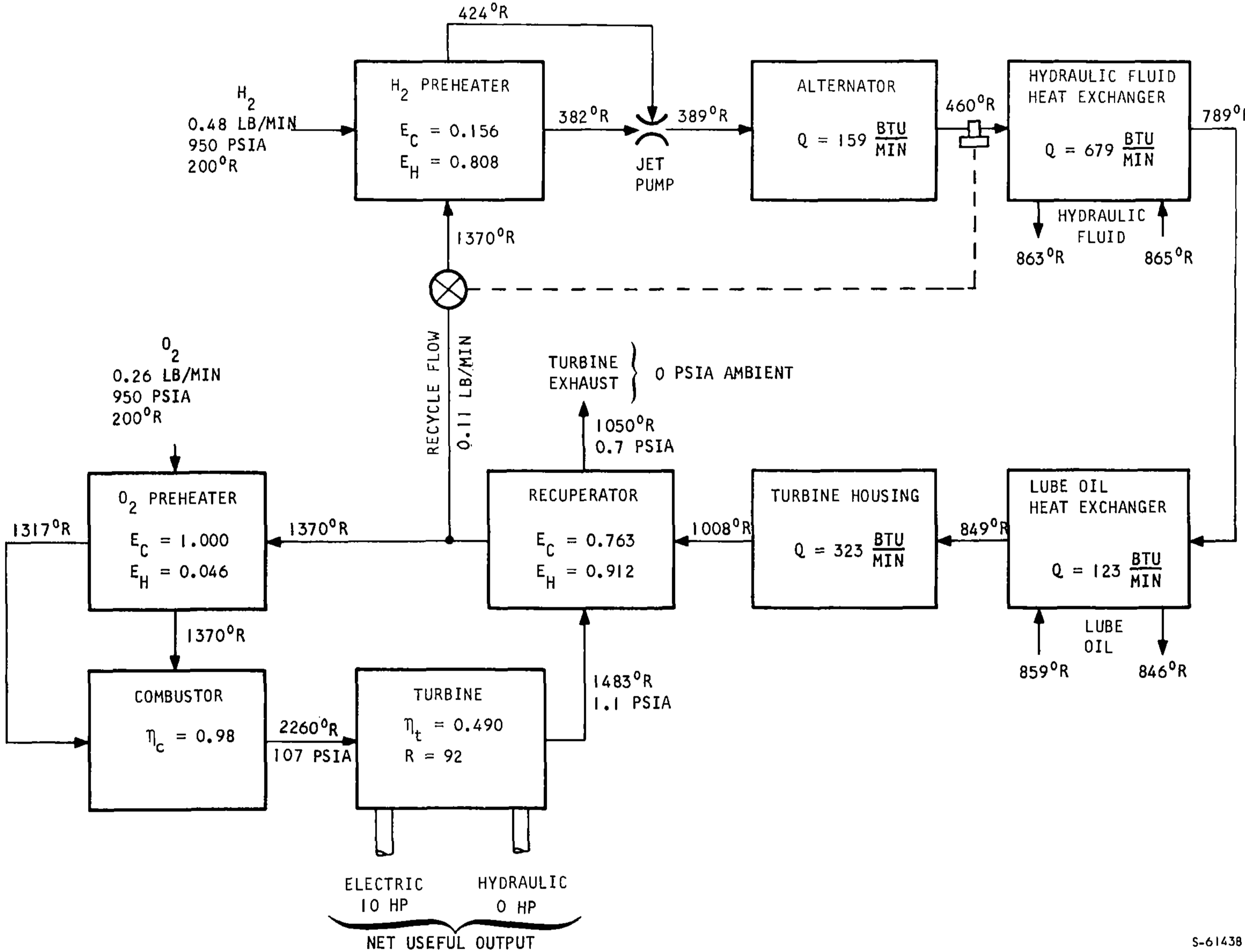


Figure 5-26. Space, Low Power Cycle State Points, 200°R Gaseous H_2-O_2 Supplied System

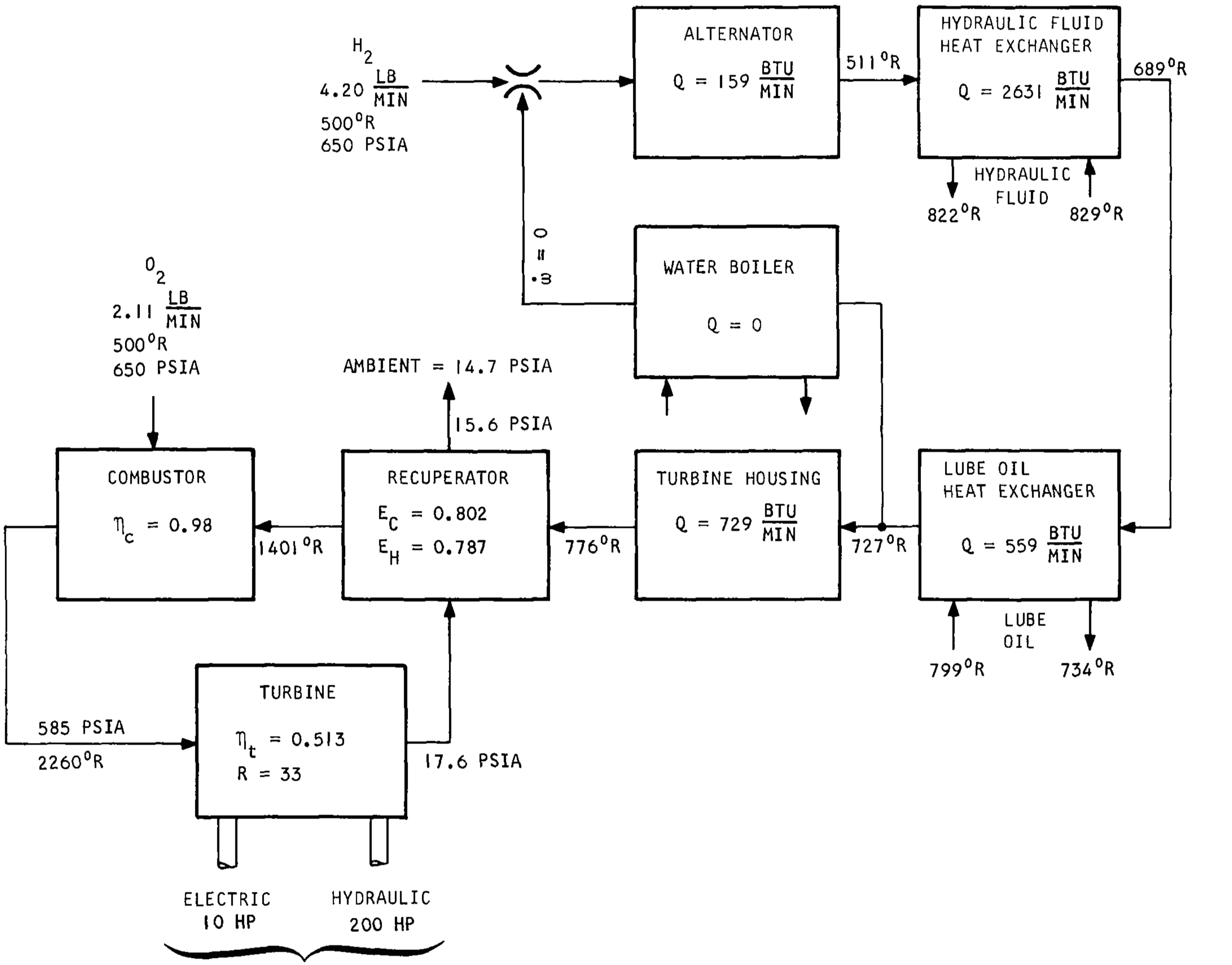


Figure 5-27. Sea Level, Full Power Cycle State Points, 500°R Gaseous H_2-O_2 Supplied System

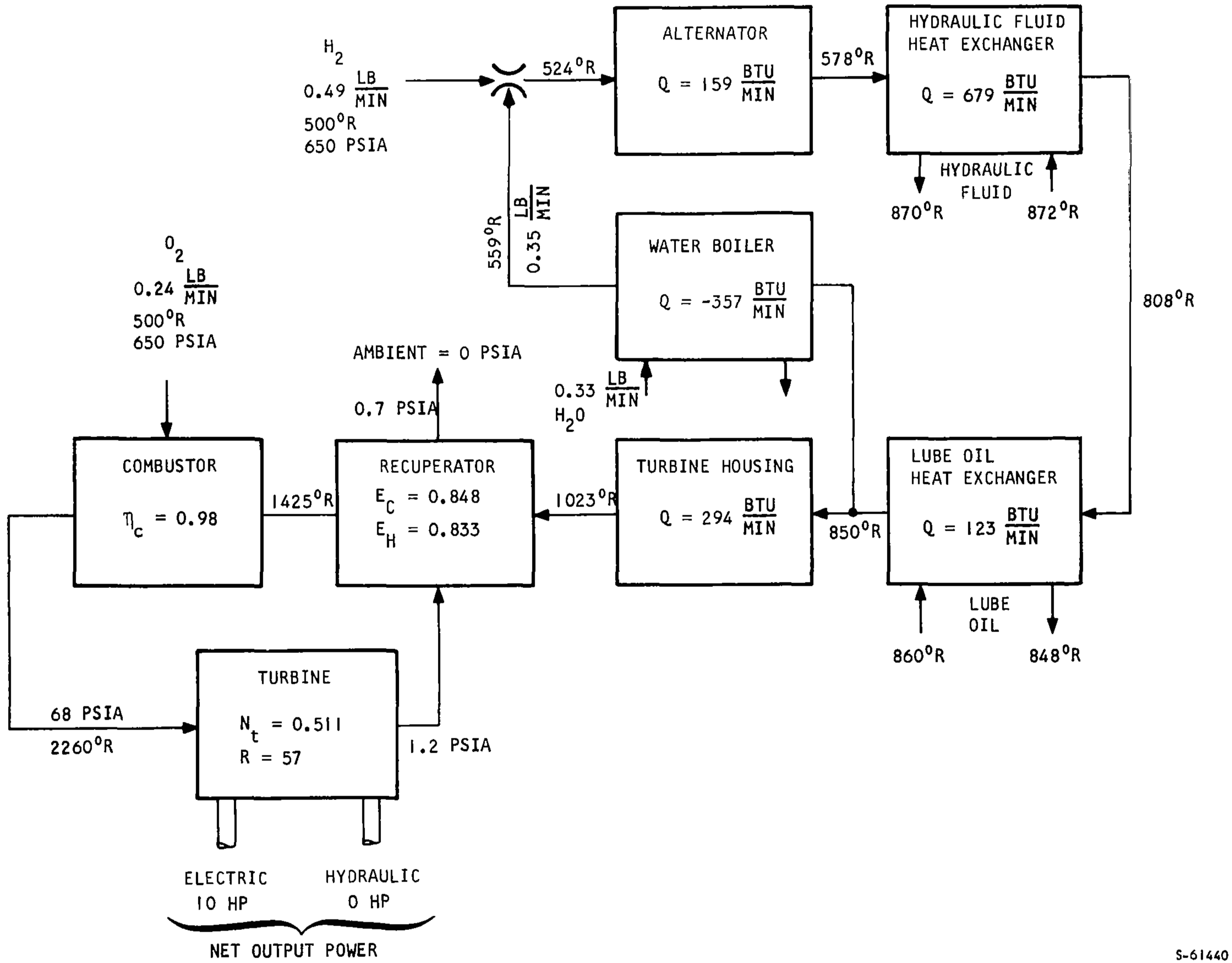


Figure 5-28 Space, Low Power Cycle State Points, 500°R Gaseous H_2-O_2 Supplied System

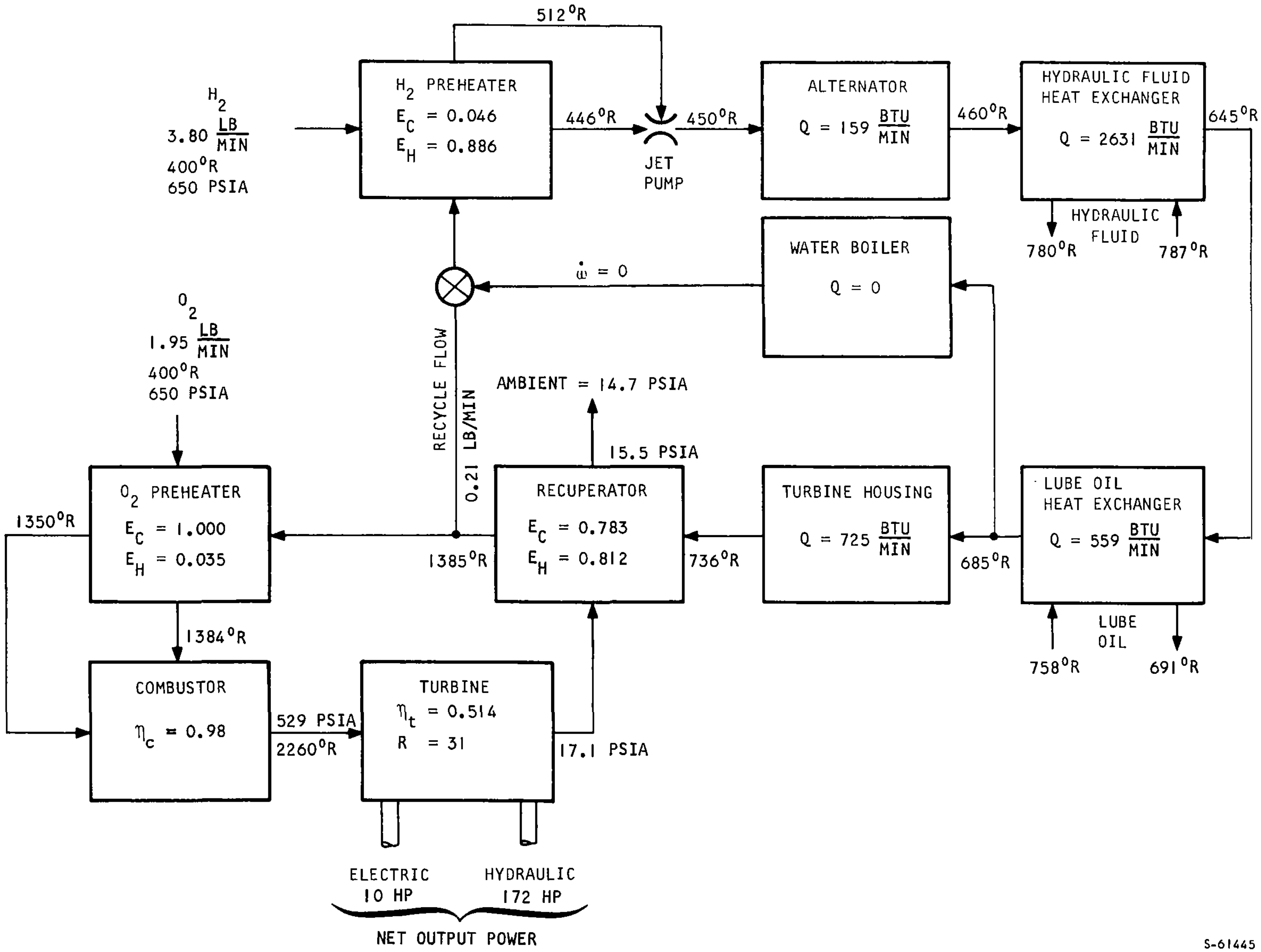


Figure 5-29. Sea Level, Full Power Cycle State Points, 400°R Gaseous H_2-O_2 Supplied System

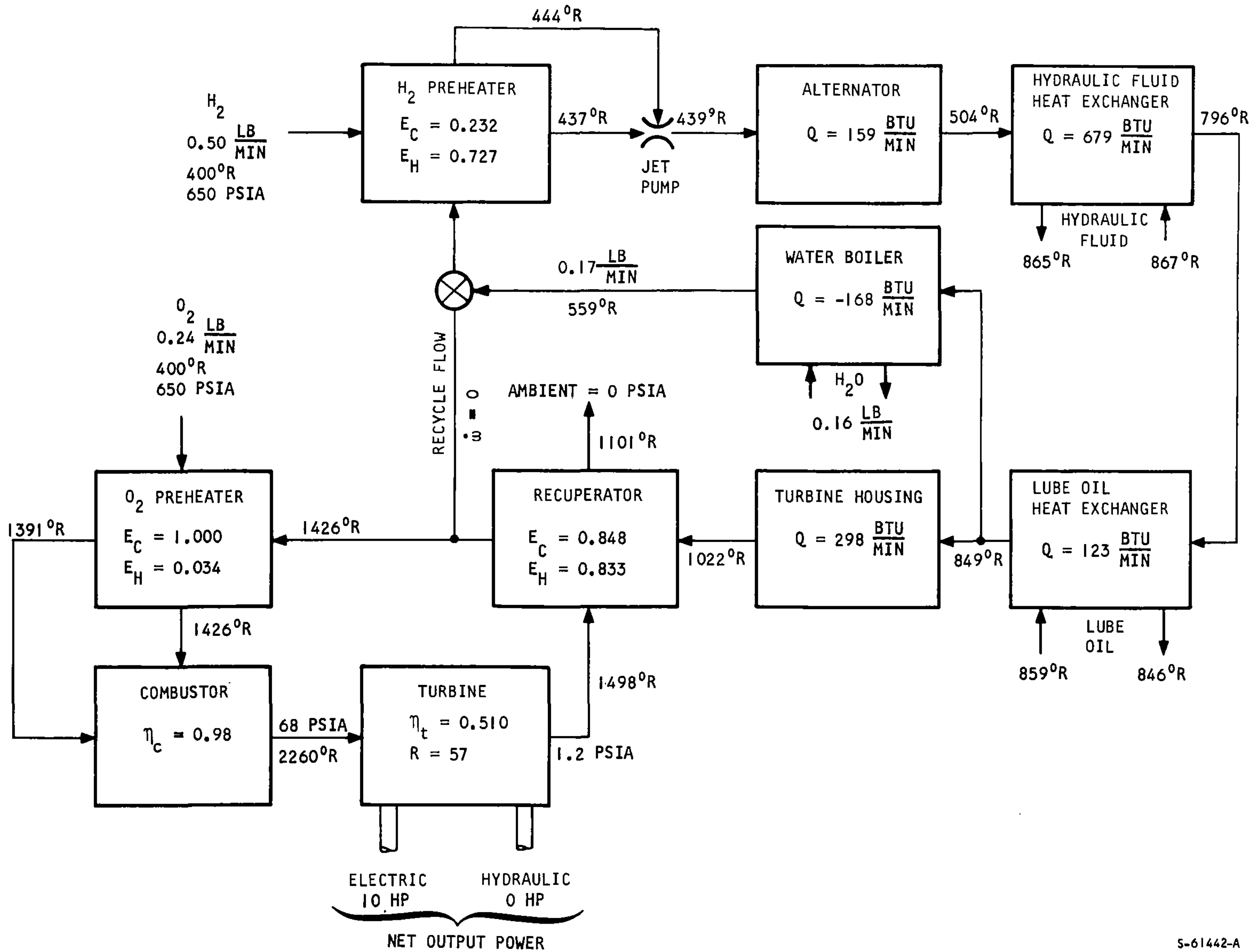


Figure 5-30. Space, Low Power Cycle State Points, 400°R Gaseous H_2-O_2 Supplied System

WEIGHT

Tables 5-5, 5-6, and 5-7 summarize the APU fixed weight as a function of the peak output power. The system weights differ slightly depending on the propellant inlet temperature. The data assume an inlet pressure of 650 psia; however, the fixed weight is nearly independent of pressure. Figure 5-31 shows a plot of fixed weight vs output power; the data are applicable to all inlet temperatures considered here. The inflection in the curve is due to the variation in available pump weights and speeds.

The fixed and variable weight of the booster APU system for 225 hp output at the gearbox, are tabulated in Table 5-8 for the various combinations of tank types (vacuum-jacketed hard shell, and soft shell) and propellant inlet temperature to the APU (200°, 300°, 400°, and 500°R). The table assumes a propellant inlet pressure of 650 psia for all cases except one. The weight data include the fluid required by the APS to condition the APU propellant to the desired inlet state and include a tankage allowance for that propellant. The tanks are assumed to be large tanks on the order of 10,000 lb each of hydrogen and oxygen.

The fixed and variable weight of the orbiter APU system are shown similarly in Table 5-9.

Finally, using the scaling criteria described in Section 2, it is possible to determine the APU system weight as a function of power level and total energy output. Such curves are shown in Figures 5-32 through 5-35. Figures 5-32 and 5-33 are the 500°R, 300 psia case requested by NASA and Figures 5-34 and 5-35 are for 200°R, 650 psia inlet conditions. They assume low-pressure hard shell tanks for the orbiter and low-pressure hard shell tanks for the booster.

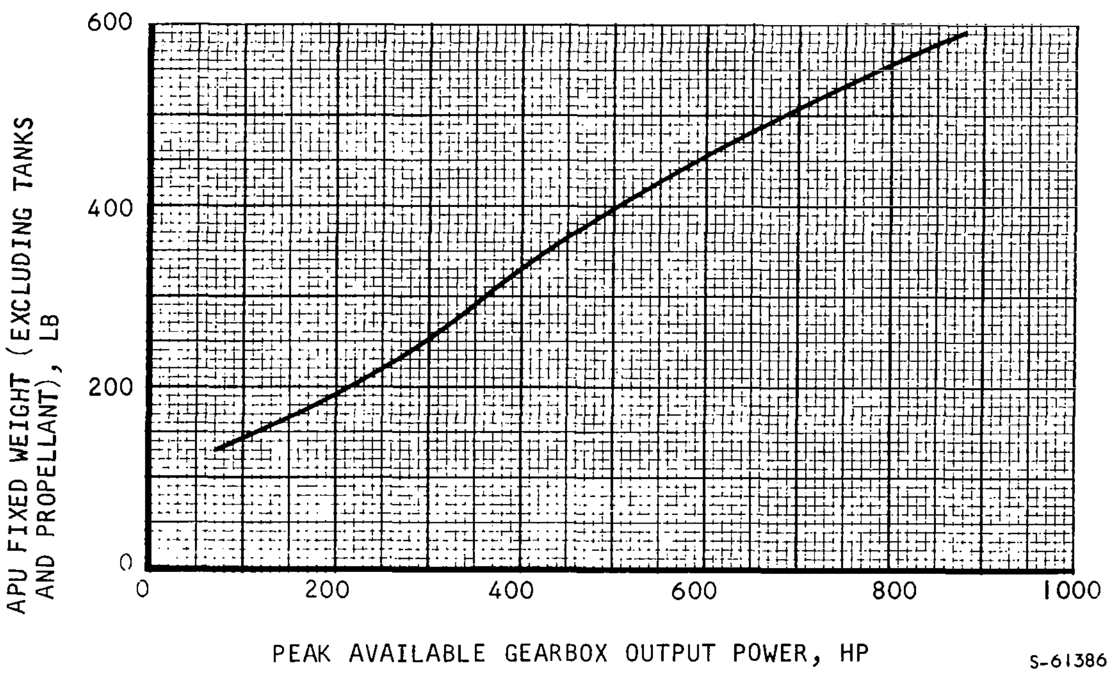


Figure 5-31. APU Fixed Weight; High Pressure Gaseous H_2-O_2 Supplied System

TABLE 5-5

APU FIXED WEIGHT HIGH PRESSURE 500°R GASEOUS H_2-O_2 SUPPLIED SYSTEM

Component \ Peak Power Required	100 hp	225-hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps	14.0	25.0	44.0	103.0	145.0
Generator	9.0	20.0	27.0	45.0	67.0
Turbine (with containment)	44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)	2.1	3.1	3.6	4.6	5.6
Control logic devices	6.0	6.0	6.0	6.0	6.0
Control valves	5.0	5.5	6.0	6.5	7.5
Pressure regulators	3.0	3.0	3.0	3.5	4.0
Heat exchangers					
Recuperator	10.2	16.5	19.5	26.8	34.2
Water boiler	2.0	4.5	6.0	10.0	15.0
O_2 preheater	0.4	0.5	0.6	0.7	0.9
Hydraulic oil cooler	2.4	4.8	5.5	6.8	8.1
Lube oil cooler	1.1	1.5	2.0	3.2	4.7
Gearbox with lube pump	5.7	17.4	26.6	58.9	92.5
Lube oil in sump	7.7	7.7	7.7	7.7	7.7
Instrumentation	1.0	1.0	1.0	1.0	1.0
Ducting	14.9	21.6	24.8	31.7	38.7
Subtotal	128.5	182.1	227.3	359.4	481.9
10 percent for vehicle support structure	12.9	18.2	22.7	35.9	48.2
Total fixed weight	141.4	200.3	250.0	395.3	530.1

TABLE 5-6

APU FIXED WEIGHT HIGH PRESSURE 400°R GASEOUS H_2-O_2 SUPPLIED SYSTEM

Component \ Peak Power Required	100 hp	225-hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps	14.0	25.0	44.0	103.0	145.0
Generator	9.0	20.0	27.0	45.0	67.0
Turbine (with containment)	44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)	2.1	3.1	3.6	4.6	5.6
Control logic devices	7.0	7.0	7.0	7.0	7.0
Control valves	5.5	6.0	6.5	7.0	8.0
Pressure regulators	3.0	3.0	3.0	3.5	4.0
Heat exchangers					
Water boiler	0.6	1.4	1.9	3.1	4.7
Recuperator	10.2	16.5	19.5	26.8	34.2
H_2 preheater	1.7	2.5	2.9	3.8	4.6
O_2 preheater	0.4	0.5	0.6	0.7	0.9
Hydraulic oil cooler	2.4	4.8	5.5	6.8	8.1
Lube oil cooler	1.1	1.5	2.0	3.2	4.7
Gearbox with lube pump	5.7	17.4	26.6	58.9	92.5
Lube oil in sump	7.7	7.7	7.7	7.7	7.7
Instrumentation	1.2	1.2	1.2	1.2	1.2
Ducting	16.4	23.8	27.3	34.9	42.6
Subtotal	132.0	185.4	230.3	361.2	481.8
10 percent for vehicle support structure	13.2	18.5	23.0	36.1	48.2
Total fixed weight	145.2	203.9	253.3	397.3	530.0

TABLE 5-7

APU FIXED WEIGHT HIGH PRESSURE 200° TO 300°R GASEOUS H_2-O_2 SUPPLIED SYSTEM

Component \ Peak Power Required	100 hp	225-hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps	14.0	25.0	44.0	103.0	145.0
Generator	9.0	20.0	27.0	45.0	67.0
Turbine (with containment)	44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)	2.1	3.1	3.6	4.6	5.6
Control logic devices	6.0	6.0	6.0	6.0	6.0
Control valves	4.5	5.0	5.5	6.0	7.0
Pressure regulators	2.0	2.0	2.0	2.5	3.0
Heat exchangers: Recuperator	10.2	16.5	19.5	26.8	34.2
Heat exchangers: H_2 preheater	1.7	2.5	2.9	3.8	4.6
Heat exchangers: O_2 preheater	0.4	0.5	0.6	0.7	0.9
Heat exchangers: Hydraulic oil cooler	2.4	4.8	5.5	6.8	8.1
Heat exchangers: Lube oil cooler	1.1	1.5	2.0	3.2	4.7
Gearbox with lube pump	5.7	17.4	26.6	58.9	92.5
Lube oil in sump	7.7	7.7	7.7	7.7	7.7
Instrumentation	1.0	1.0	1.0	1.0	1.0
Ducting	14.9	21.6	24.8	31.7	38.7
Subtotal	126.7	178.6	222.7	351.7	470.0
10 percent for vehicle support structure	12.7	17.9	22.3	35.2	47.0
Total fixed weight	139.4	196.5	245.0	386.9	517.0

TABLE 5-8

BOOSTER 225 HP APU SYSTEM WEIGHT - HIGH PRESSURE GASEOUS H_2-O_2 SUPPLIED SYSTEM (SINGLE APU)

Tank Type	Hard Shell					Soft Shell				
Propellant Inlet Condition	500°R 300 psia	500°R 650 psia	400°R 650 psia	300°R 650 psia	200°R 650 psia	500°R 300 psia	500°R 650 psia	400°R 650 psia	300°R 650 psia	200°R 650 psia
Fixed weight, lb	200.3	200.3	203.9	196.5	196.5	200.3	200.3	203.9	196.5	196.5
Hydrogen weight, lb	380.2	280.7	253.0	231.8	217.1	380.2	280.7	253.0	231.8	217.1
Hydrogen tank weight, lb	228	168.0	152.0	139.0	130.0	57.0	42.2	28.0	34.8	32.6
Oxygen weight, lb	215.1	168.6	148.2	132.3	122.7	215.1	168.6	148.2	132.3	122.7
Oxygen tank weight, lb	32.2	25.3	22.2	19.9	18.4	15.1	11.8	10.4	9.3	8.6
Water weight, lb	1.4	2.5	2.7	-	-	1.4	2.5	5.1	-	-
Water storage feed system weight, lb	14.1	14.1	14.1	-	-	14.1	14.1	14.1	-	-
Total system weight, lb	1071.3	859.5	796.1	719.5	684.7	883.2	740.2	672.7	604.7	572.5

TABLE 5-9

ORBITER 225 HP APU SYSTEM WEIGHT - HIGH PRESSURE GASEOUS H_2-O_2 SUPPLIED SYSTEM (SINGLE APU)

Tank Type	Hard Shell					Soft Shell				
Propellant Inlet Condition	500°R 300 psia	500°R 650 psia	400°R 650 psia	300°R 650 psia	200°R 650 psia	500°R 300 psia	500°R 650 psia	400°R 650 psia	300°R 650 psia	200°R 650 psia
Fixed weight, lb	200.3	200.3	203.9	196.5	196.5	200.3	200.3	203.9	196.5	196.5
Hydrogen weight, lb	94.7	76.7	70.3	64.3	59.0	94.7	76.7	70.3	64.3	59.0
Hydrogen tank weight, lb	58.8	47.5	43.5	39.9	36.6	18.9	15.3	14.1	12.8	11.8
Oxygen weight, lb	56.8	48.5	43.0	38.5	35.3	56.8	48.5	43.0	38.5	35.3
Oxygen tank weight, lb	9.1	7.8	6.9	6.2	5.7	6.8	5.8	5.2	4.6	4.2
Water weight, lb	9.5	12.5	13.0	-	-	9.5	12.5	13.0	-	-
Water storage feed system weight, lb	7.2	7.2	7.2	-	-	7.2	7.2	7.2	-	-
Total system weight, lb	436.4	400.5	387.8	345.4	333.1	393.2	366.3	356.7	316.7	306.8

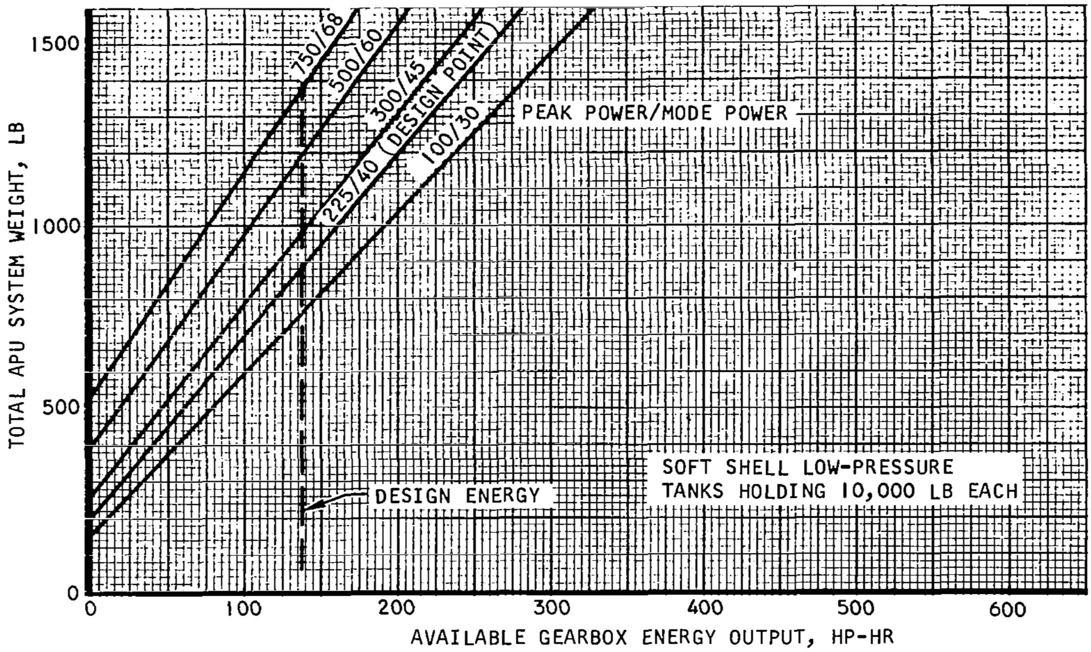


Figure 5-32. Booster Vehicle APU System Weight; 500°R, 300 psia Gaseous H_2-O_2 Supplied System

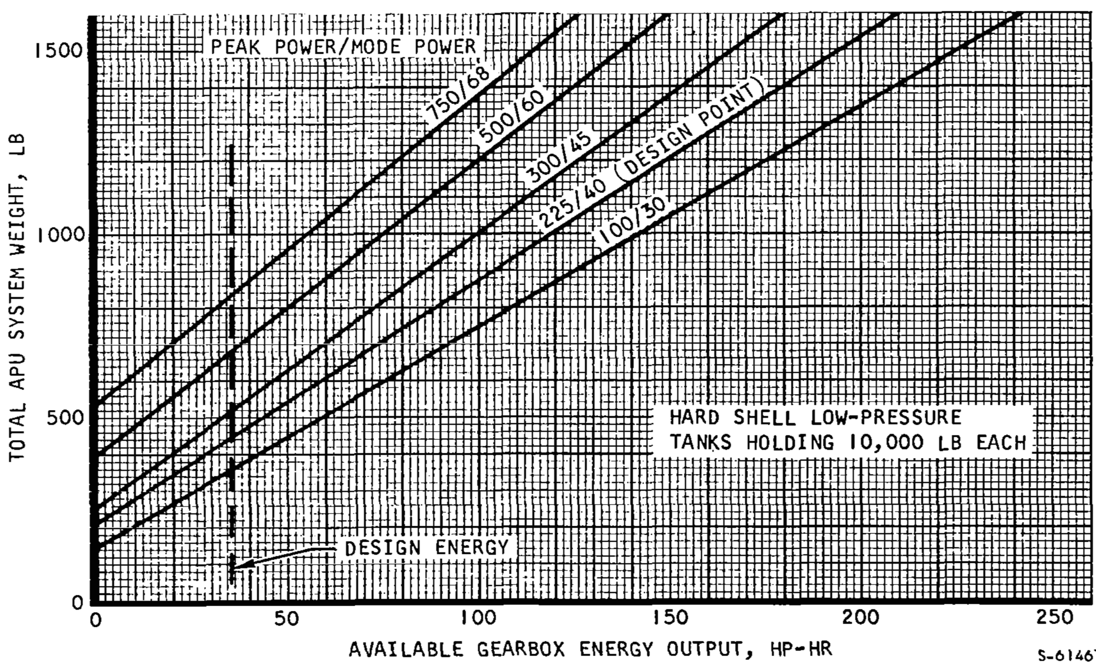


Figure 5-33. Orbiter Vehicle APU System Weight; 500° , 300 psia Gaseous H_2-O_2 Supplied System

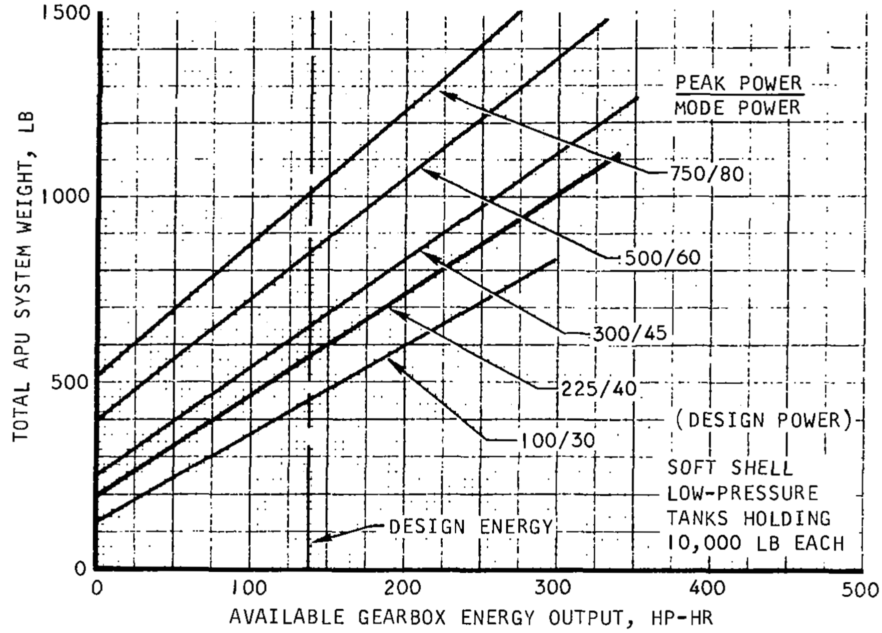


Figure 5-34. Booster Vehicle APU System Weight; 200°R, 650 psia Gaseous H_2-O_2 Supplied System

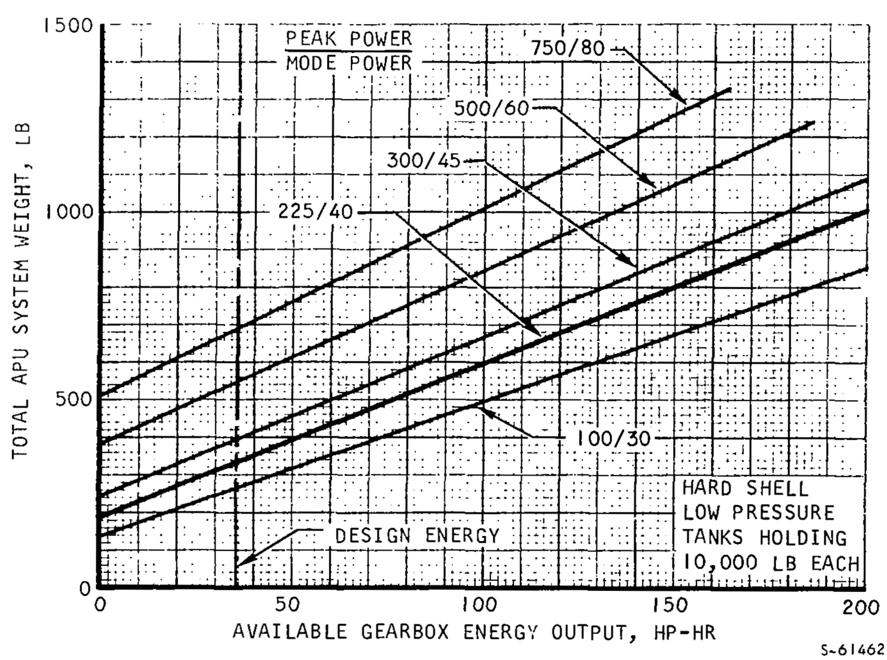


Figure 5-35. Orbiter Vehicle APU System Weight; 200°R, 650 psia Gaseous H_2-O_2 Supplied System

SECTION 6

DUAL-MODE AIRBREATHING/PROPELLANT SYSTEM

INTRODUCTION

The dual-mode system, shown in Figure 6-1, uses an airbreathing gas turbine power section for operation during atmospheric flight and a hydrogen-oxygen propellant turbine for operation outside the atmosphere. The gas turbine uses hydrogen (or alternately JP fuel) in a conventional open-Brayton-cycle with air as the working fluid. The propellant turbine cycle is similar to the integral high-pressure cryogenic supplied system described in Section 4 of this report. As a consequence, the system has two heat sinks corresponding to the two power sources (that is, ram air for gas turbine operation and hydrogen fuel flow for propellant turbine operation). A liquid thermal transport loop provides heat transport between the heat sources (generator, hydraulic fluid, gearbox lubricant, and turbine housing) and the two heat sinks (ram air or hydrogen). The propellant system uses a recirculating hydrogen loop with cycle regeneration to avoid excessively low temperatures in the hydrogen-to-transport fluid heat exchanger.

The two power turbines drive through fill-and-dump type fluid couplings which serve as clutches to connect the active turbine and disconnect the inactive turbine. During the clutching process, the fluid from one coupling is pumped to the other with both turbines running at rated speed to permit a smooth power transition between turbines.

SYSTEM TRADEOFFS

Gas Turbine Power Section

The gas turbine power section consists of a single-shaft turbine which drives the compressor and provides the useful shaft power output. Figure 6-2 shows the performance tradeoffs with turbine inlet temperature and compressor pressure ratio. For the present application, it will be desirable to design the gas turbine for a relatively low pressure ratio preferably for a single-stage compressor to simplify design and minimize fixed weight. It will be noted that specific fuel consumption is relatively flat with respect to these design parameters. Turbine inlet temperature, for example, has relatively little influence on SFC, but does greatly influence machine specific throughflow, which will be reflected in the size and weight of the gas turbine. For the present system studies, a compressor pressure ratio of six and a turbine inlet temperature of 2460°R will be assumed. A design point power level of 200 shp during the landing mode was selected for the turbine. Figure 6-3 shows the flowpath of the resulting gas turbine design.

Propellant Turbine Section

Propellant weight will vary with design turbine inlet pressure. As shown in Table 6-1, propellant weight remains relatively constant for the range of

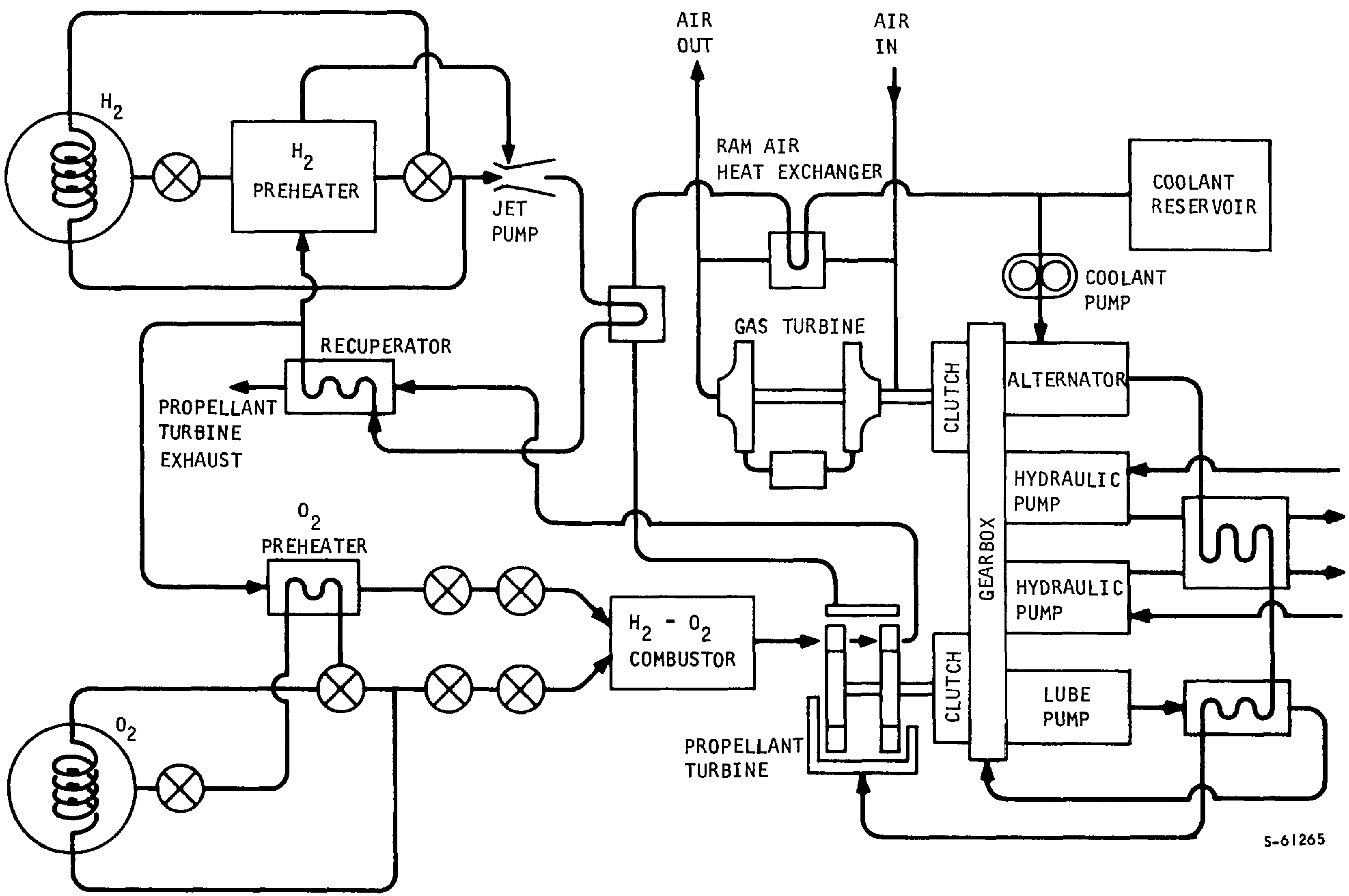


Figure 6-1. Dual-Mode System

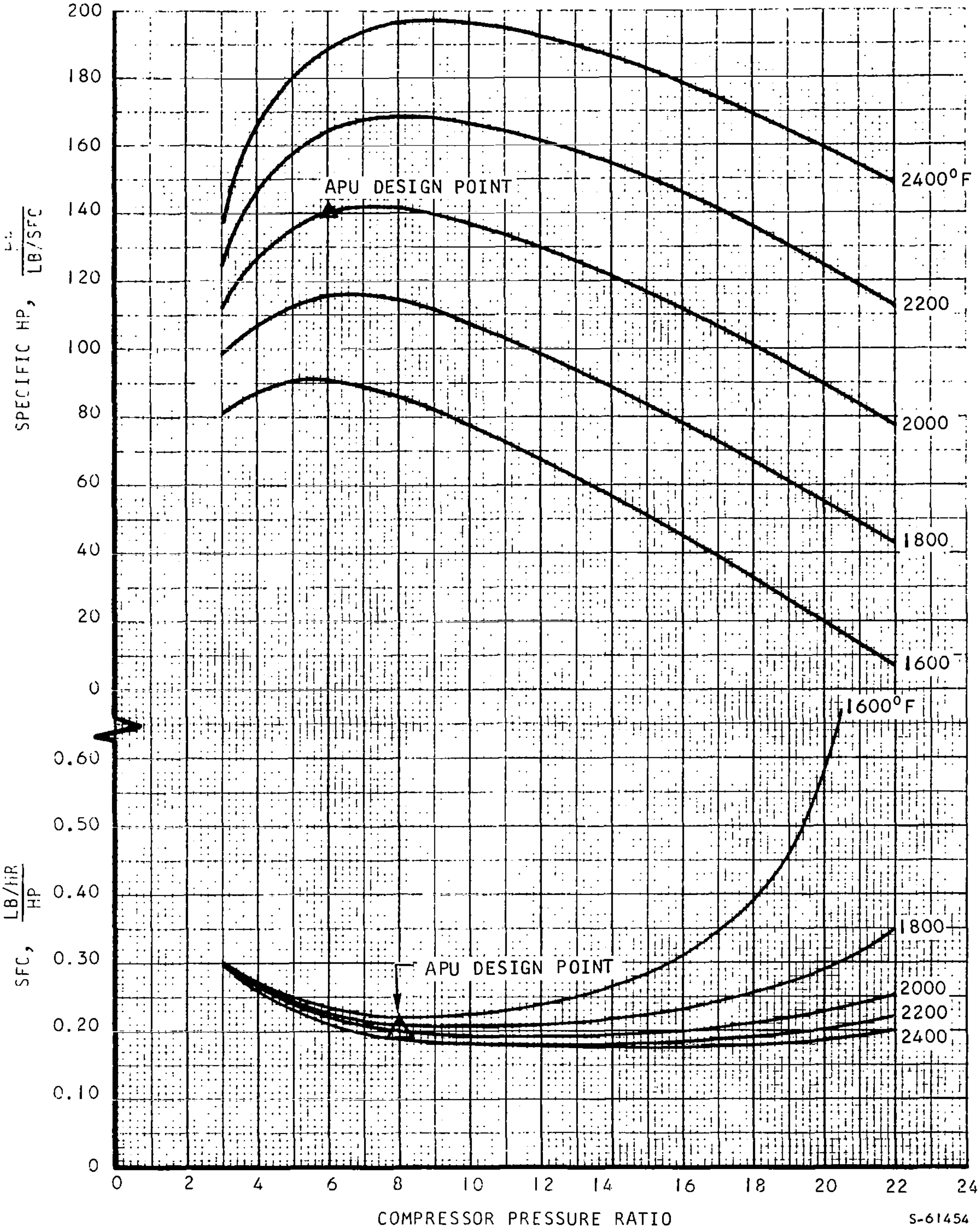


Figure 6-2. Open Cycle Gas Turbine Shuttle APU Parametric Study

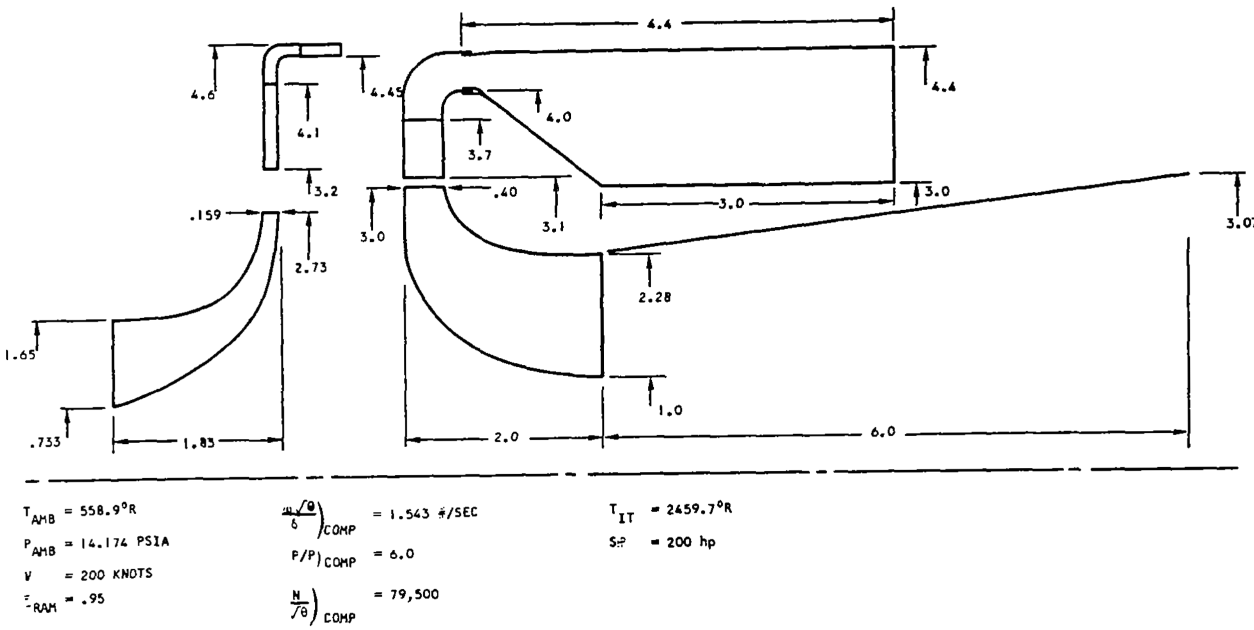


Figure 6-3. Shuttle Gas Turbine APU Flow Path

TABLE 6-1

H_2-O_2 TURBINE PROPELLANT WEIGHT
FOR BOOSTER AND ORBITER MISSIONS

Propellant Supply Pressure, psia		300	650	950
Booster Propellant Weight, lb	Hydrogen	24.4	21.1	20.6
	Oxygen	13.7	12.6	12.3
	Total	38.1	33.7	32.9
Orbiter Propellant Weight, lb	Hydrogen	58.8	47.6	46.3
	Oxygen	29.2	25.6	25.2
	Total	88.0	73.2	71.5

supply pressure between 650 and 950 psia. When tankage penalties are taken into account, minimum total system weight will be obtained at a design supply pressure near 650 psia.

SYSTEM PERFORMANCE

Performance of an airbreathing gas turbine engine depends upon output power and the temperature and pressure of the inlet air. Figures 6-4 through 6-6 show off-design performance as a function of altitude for three atmospheric models:

MIL-STD-210 Hot Atmosphere

U.S. Standard Atmosphere

MIL-STD-210 Cold Atmosphere

The gas turbine has been sized for the hot atmosphere condition. It will be noted that the power capability of the turbine will be greater for the other two atmospheric models. With this increase in power generating capacity, the turbine fuel consumption also increases with the colder inlet conditions. Operating altitude has a most significant effect on both fuel consumption and on power rating. Both power rating and fuel consumption decrease with increasing altitude. The altitude design point is an important consideration in gas turbine design. For the present mission profiles, the selected design point is 200 shp during the landing condition. (The 225 shp peak can be met by this turbine design on a transient basis.)

Figure 6-7 shows the gas turbine fuel consumption as a function of output power and altitude. The power limit shown in this curve is set by the turbine inlet temperature limits for continuous operation. A total of approximately 57.2 lb of hydrogen will be required for the atmospheric flight portion of the booster mission. The present orbiter mission has essentially no atmospheric flight and therefore requires all of the power to be supplied by the propellant turbine.

WEIGHT

Figure 6-8 gives the gas turbine size and weight as a function of output power. Table 6-2 summarizes system fixed weight for 100, 225, 300, 500, and 750 shp ratings. These data are plotted in Figure 6-9. For the present orbiter mission, no atmospheric flight is specified and the airbreathing elements (gas turbine power section and ram air heat exchanger) should be deleted.

The propellant turbine requires high-pressure cryogenic storage supplying the propellants at pressures on the order of 650 psia. The hydrogen used by the airbreathing gas turbine, on the other hand, can be supplied at a considerably lower pressure (on the order of 120 psia). At the design mission energy level, the tankage weights indicate approximately a 10-lb weight saving by combining both hydrogen propellant quantities in a single 650-psia tank.

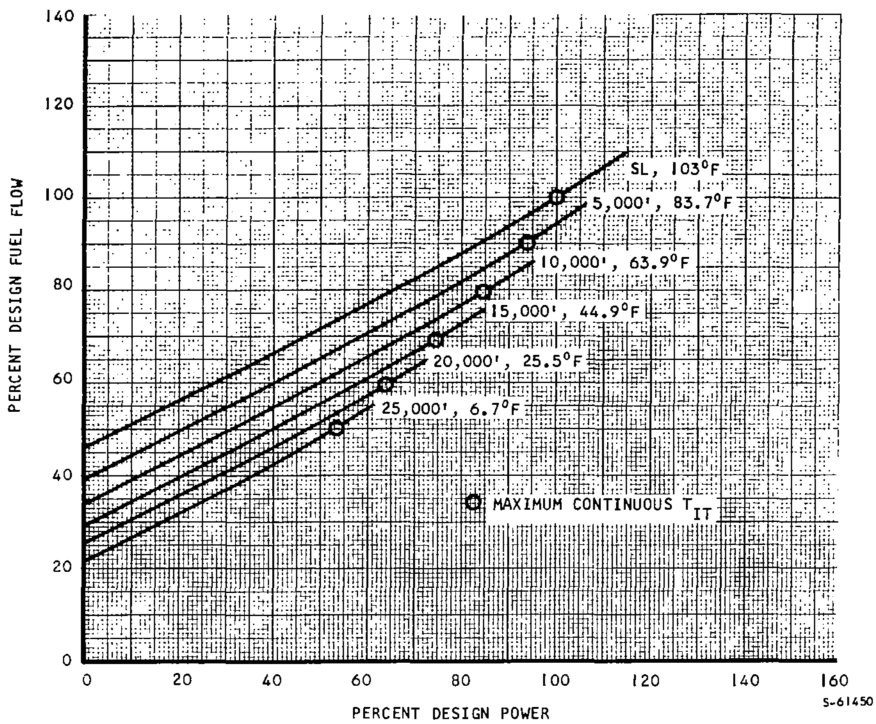


Figure 6-4. Shuttle Gas Turbine APU Estimated Performance - MIL-STD-210 Hot Atmosphere

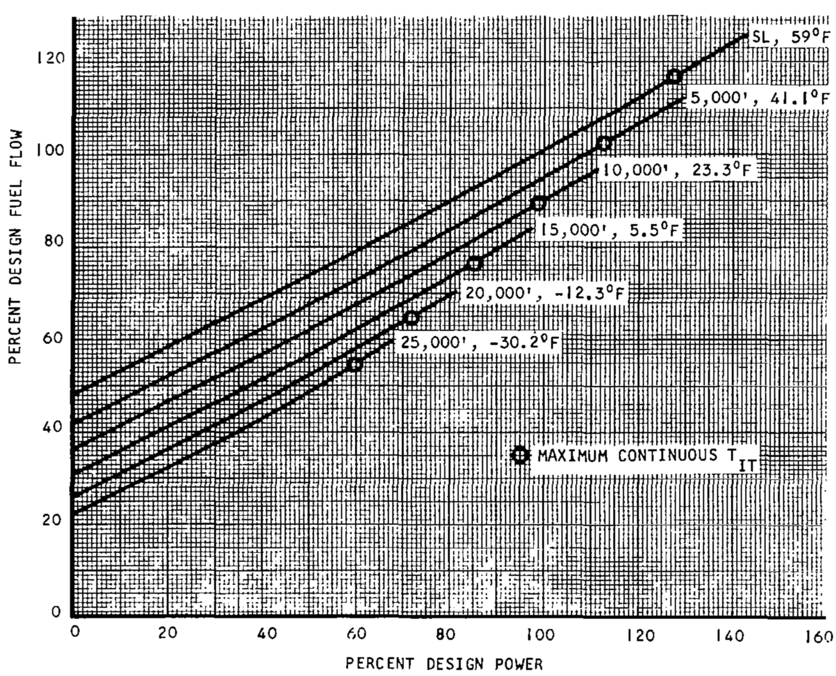


Figure 6-5. Shuttle Gas Turbine APU Estimated Performance - U.S. Standard Atmosphere

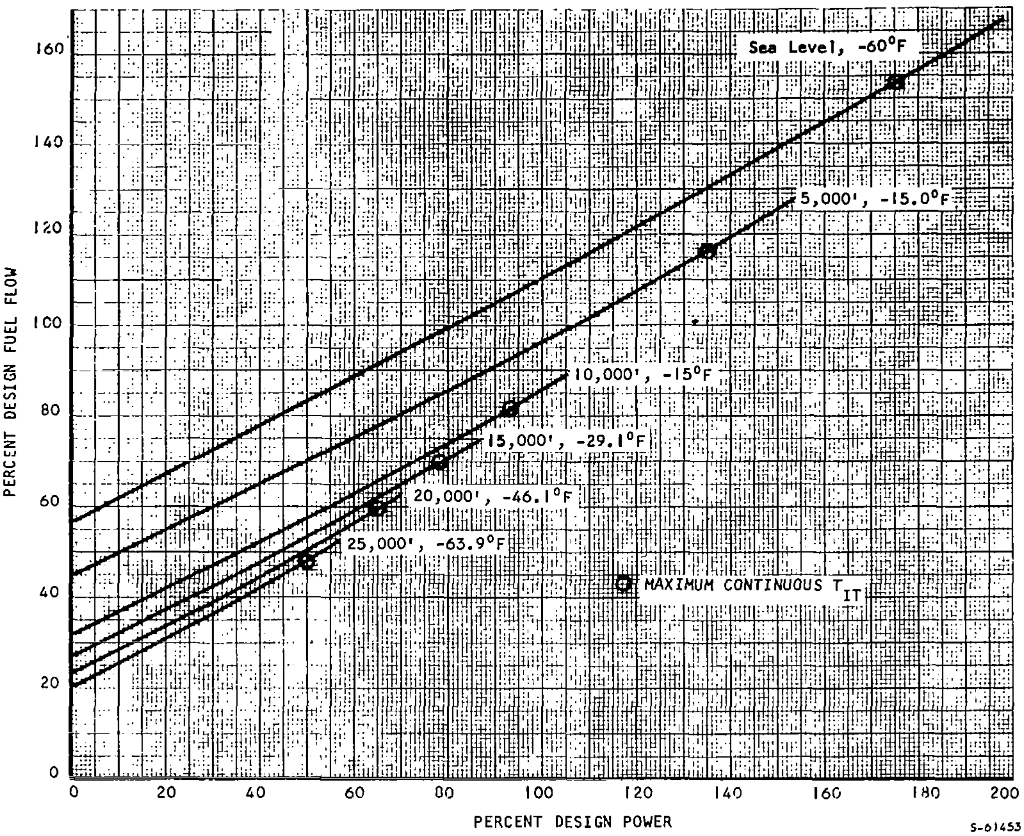


Figure 6-6. Shuttle Gas Turbine APU Estimated Performance, MIL-STD-210 Cold Atamosphere

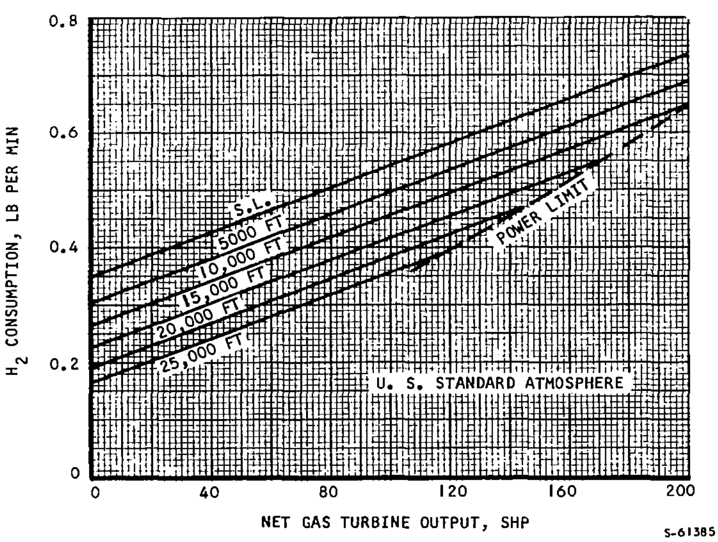


Figure 6-7. Shuttle Gas Turbine APU Fuel Consumption

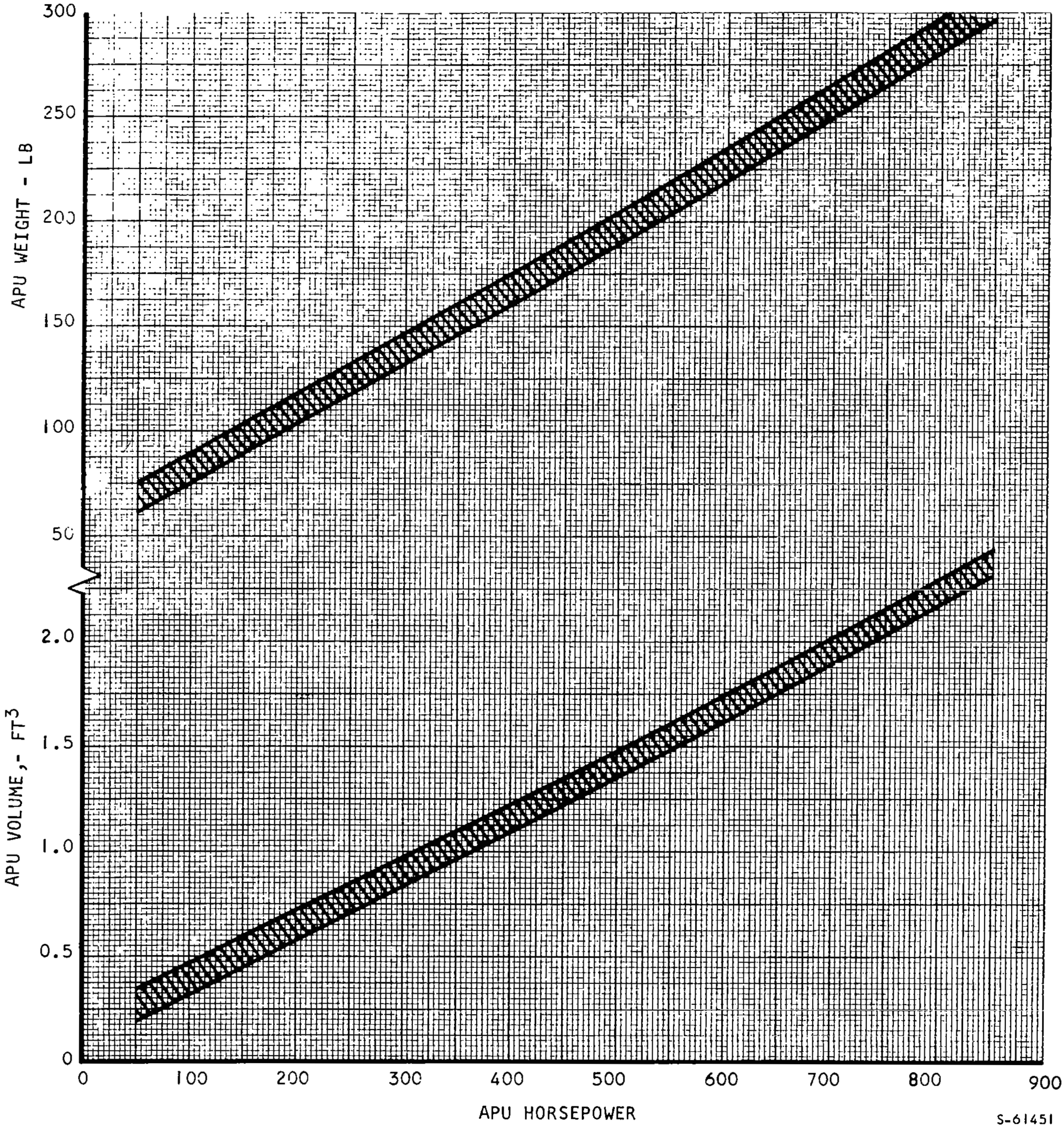


Figure 6-8. Shuttle Gas Turbine — Estimated Weight and Volume Curves

TABLE 6-2

APU FIXED WEIGHT DUAL-MODE SYSTEM

Component \ Peak Power Required	100 hp	225 hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps	14.0	25.0	44.0	103.0	134.0
Generator	10.1	22.0	30.5	53.3	78.6
Gas turbine	75.0	110.0	132.0	190.0	265.0
Propellant turbine (with containment)	44.0	44.0	44.0	44.0	44.0
Combustor (with insulation)	2.1	3.1	3.6	4.6	5.6
Control logic devices	7.0	7.0	7.0	7.0	7.0
Control valves	5.5	6.0	6.5	7.0	8.0
Pressure regulators	3.0	3.0	3.0	3.5	4.0
Heat exchangers					
Ram air heat exchanger	21.6	35.0	41.4	56.8	72.5
Recuperator	10.2	16.5	19.5	26.8	34.2
H_2 Preheater	1.7	2.5	2.9	3.8	4.6
O_2 Preheater	0.5	0.7	0.8	1.0	1.3
Coolant cooler	6.8	10.0	11.6	15.2	18.4
Hydraulic oil cooler	2.0	4.0	4.6	5.6	6.7
Lube oil cooler	0.7	1.0	1.3	2.1	3.1
Coolant pump	2.2	4.0	7.0	16.5	23.2
Coolant reservoir	9.4	20.0	26.2	43.1	64.1
Clutches	1.3	4.0	6.1	13.5	21.2
Gearbox with lube pump	5.7	17.4	26.6	58.9	92.5
Lube oil in sump	7.7	7.7	7.7	7.7	7.7
Instrumentation	1.2	1.2	1.2	1.2	1.2
Ducting	16.4	23.8	27.3	34.9	42.6
Subtotal	248.1	367.9	454.8	699.5	950.5
10 percent for vehicle support structure	24.8	36.8	45.5	70.0	95.1
Total fixed weight	272.9	404.7	500.3	769.5	1045.6

However, at higher energy levels, it is preferable for minimum tankage weight to obtain the gas turbine hydrogen from a separate low-pressure supply. The weight savings afforded by this approach must be evaluated against increased system complexity in maintaining two propellant supplies, unless one or the other can be integrated with another vehicle subsystem requirement.

Figures 6-10 and 6-11 show the total APU system weight as a function of power level and total energy output. Both curves assume hard shell tankage. These data use a scaling method somewhat similar to that described in Section 2; however, the hydrogen-oxygen turbine has been assumed to have a constant SPC regardless of the output power under the low backpressure conditions. This assumption is consistent with the turbine/system performance data presented in Figures 3-15 and 3-17 which show the turbine pressure ratio and efficiency at low back pressures are both almost constant.

It should be noted that using the dual-mode system on the orbiter (with gas turbine and ram air heat exchanger removed) makes that system weigh more than does an equivalent high-pressure cryogenic supplied system, which is described in Section 4. This is because the dual-mode system uses an intermediate cooling loop, whereas the high-pressure cryogenic system rejects heat directly to the cycle hydrogen flow.

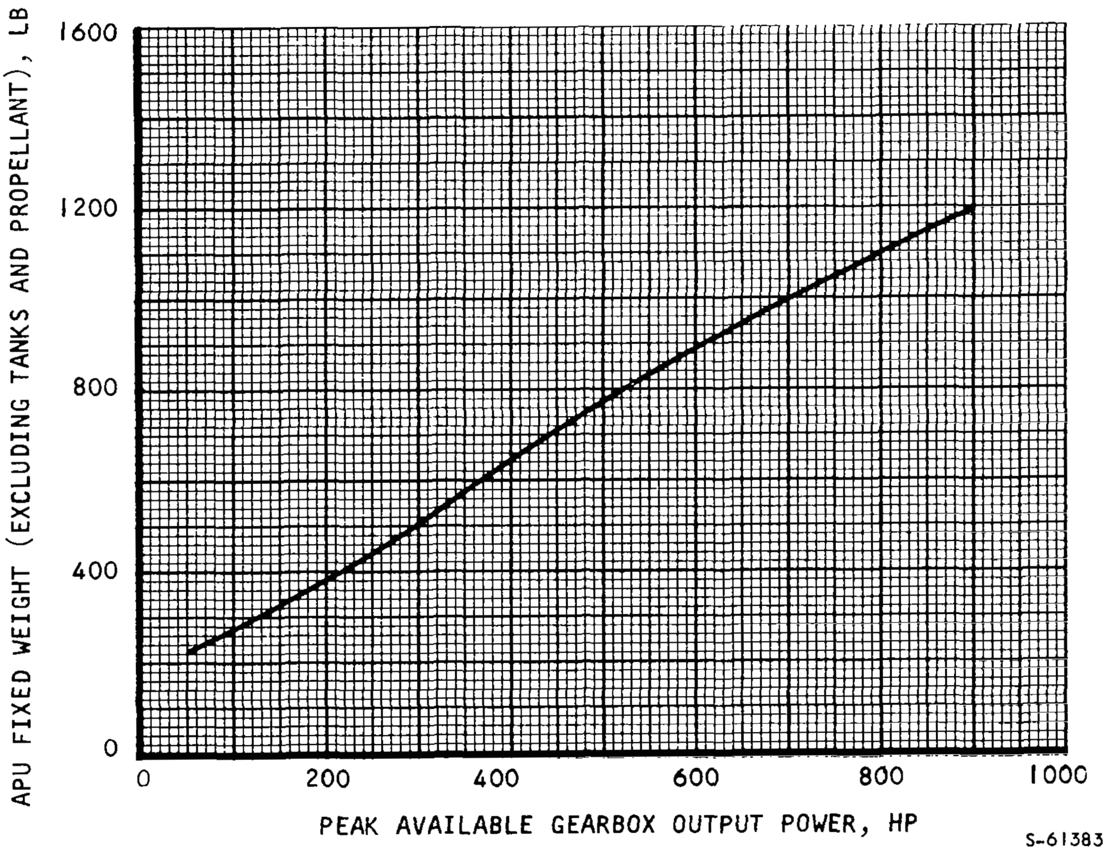


Figure 6-9. APU Fixed Weight; Dual-Mode System

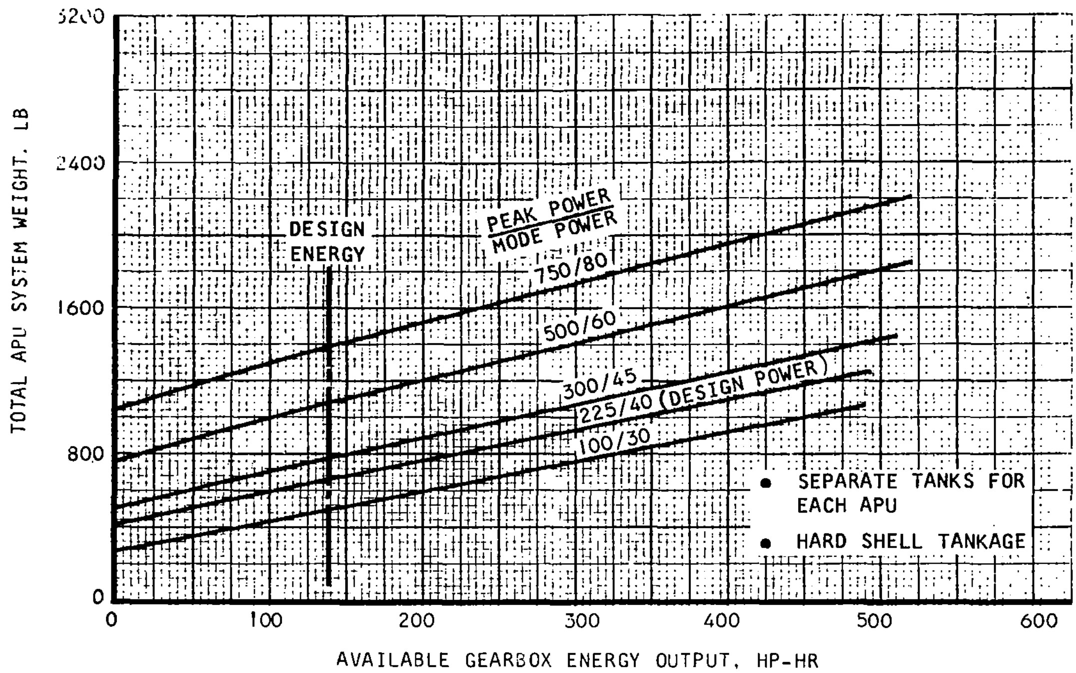


Figure 6-10. Booster Vehicle APU System Weight, Dual-Mode System

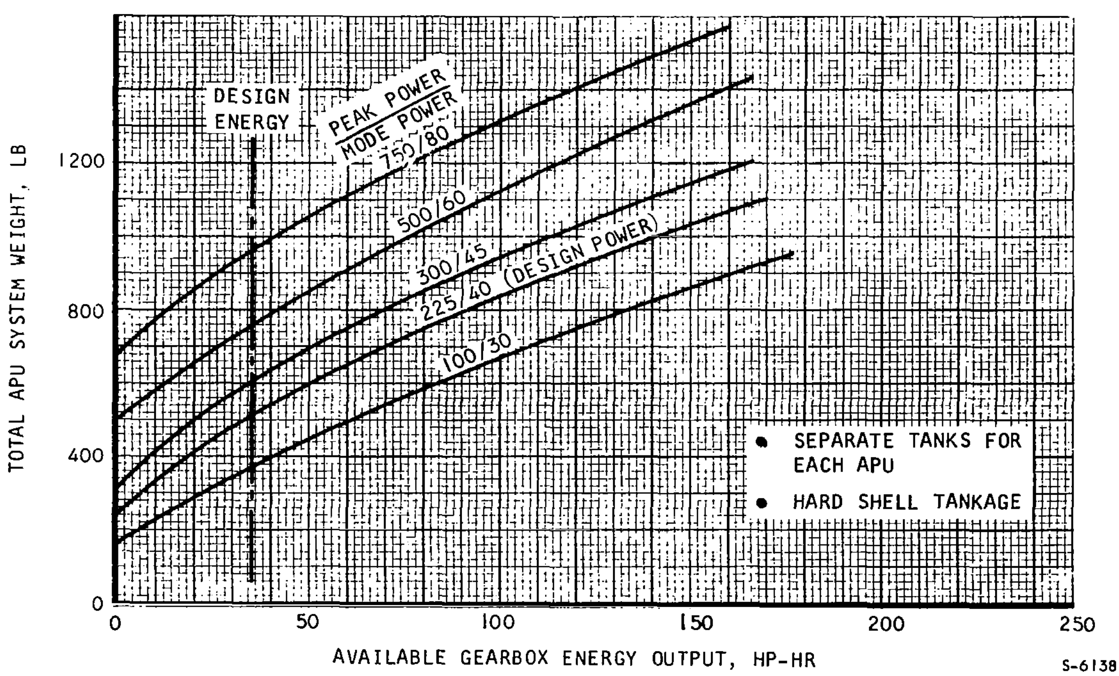


Figure 6-11. Orbiter Vehicle APU System Weight, Dual-Mode System

SECTION 7

MONOPROPELLANT SYSTEM

GENERAL DESCRIPTION

In the monopropellant system, turbine working fluid is supplied by a monopropellant gas generator. Cooling is provided by a recirculating thermal transport loop using a water boiler as the heat sink. The water and liquid monopropellant are stored in tanks provided with expulsion bladders which are pressurized by nitrogen. The monopropellant is additionally pressurized by an electric motor driven pump. Figure 7-1 shows a schematic of the system.

DESIGN CONSIDERATIONS

Storable Propellant Selection

Consideration was given to both storable monopropellants and bipropellants. Hydrazine mixtures were found to be the most suitable monopropellants in terms of performance and experience. Figure 7-2 is a comparison of several hydrazine-based monopropellants in terms of the design-point specific propellant consumption as a function of pressure ratio for 230 hp output and a representative turbine design. Propellant properties data are based on information presented in AFAPL-TR-67-167. It will be noted that the 75-24-1 (75-percent hydrazine, 24-percent hydrazine nitrate, 1 percent water) gives the best performance. This performance is obtained at a theoretical flame temperature of 2660°R, which is somewhat higher than the optimum temperature of 2260°R selected for the hydrogen-oxygen turbines. As will be shown, optimum turbine performance is obtained at a lower pitch line velocity (on the order of 1400 fps) as compared with the 1800 fps selected for the hydrogen-oxygen turbine. Consequently, it was established that the combination of a design pitch velocity of 1400 fps and a working fluid inlet temperature of 2660°R are compatible with design values selected for the hydrogen oxygen turbine.

Freezing point has been a major limitation in application of anhydrous hydrazine. Following are the freezing and boiling points of the four monopropellants shown previously:

	Freezing	Boiling or Decomposition
75-24-1	0°F	165°F
MHF-3	-65°F	165°F
MHF-5	-70°F	140°F
100 percent N_2H_4	36°F	236°F

To meet the required ambient temperature range of -65 to +300°F, thermal control provisions probably will be required for storage of any monopropellant. Clearly, the problem is less difficult with a monopropellant with a low freezing point. However, it is believed that the 0°F freezing point of the 75-24-1 monopropellant mixture is compatible with readily

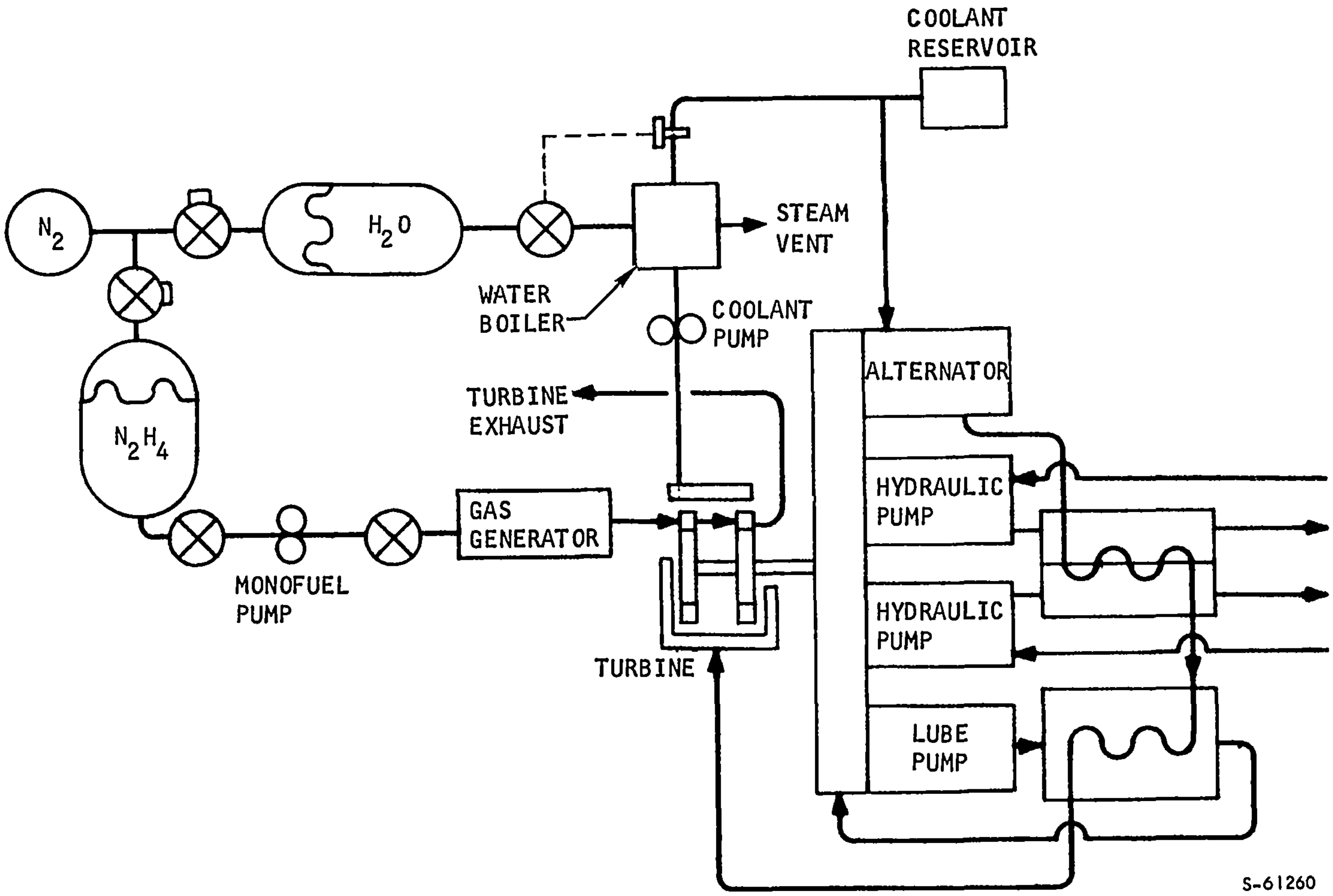


Figure 7-1. Monopropellant System

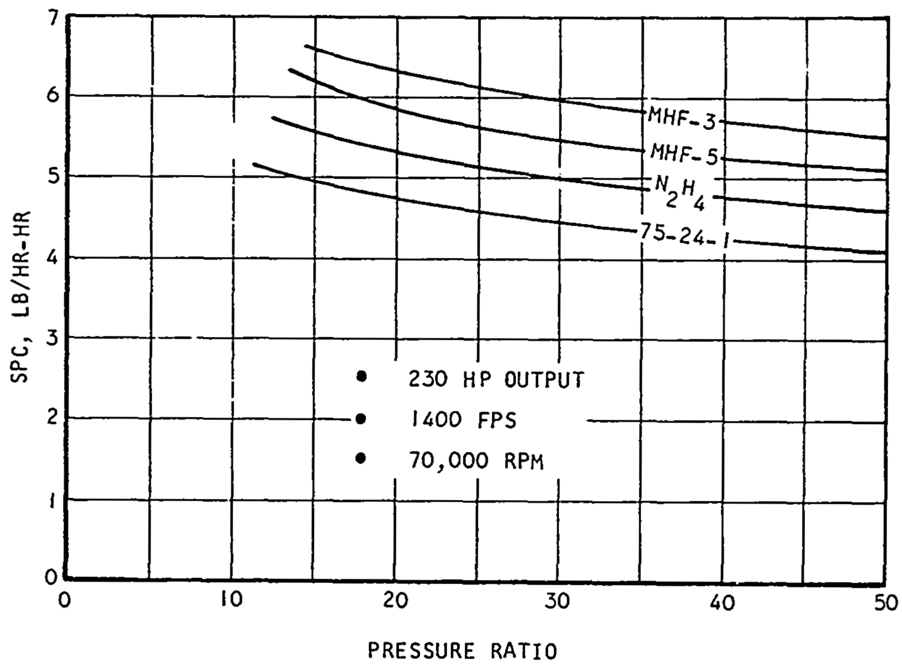


Figure 7-2. Storable Propellant Selection

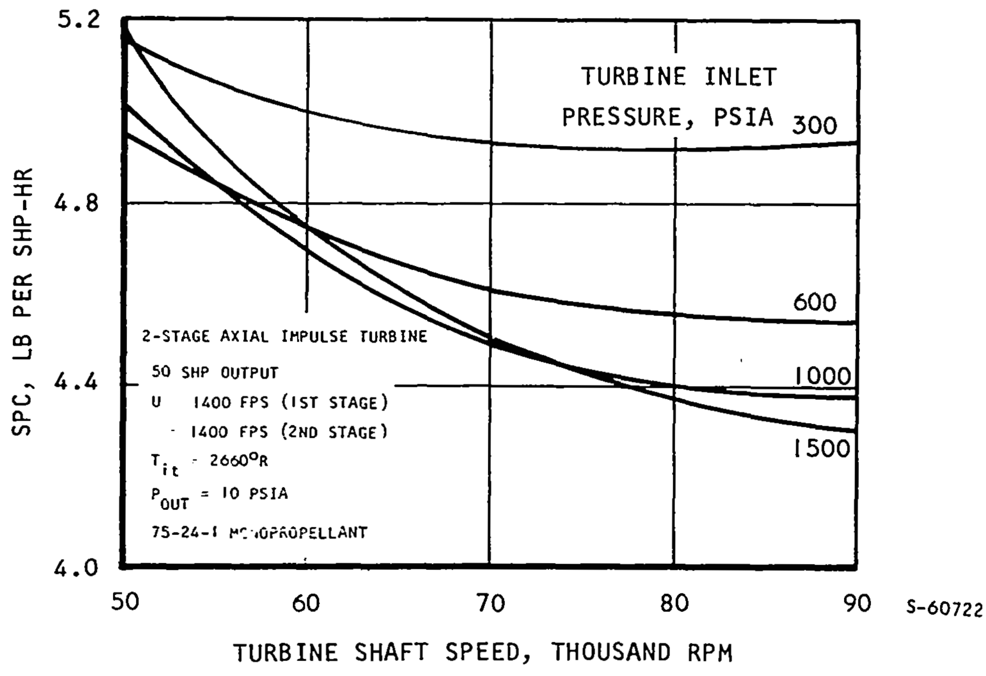


Figure 7-3. Monopropellant Turbine Rotational Speed

attainable conditions in the Space Shuttle vehicles. As a consequence of its superior performance, the 75-24-1 monopropellant was selected for the storable propellant system.

Consideration was also given to various possible bipropellant combinations. Nitrogen tetroxide-hydrazine (or hydrazine mixtures) was found to be the most suitable bipropellant combination. However, it was found that no performance advantage was obtained over monopropellants at the same gas temperature level. Since the 2660°R gas temperature obtained with 75-24-1 monopropellant was near the maximum set by turbine thermal and stress considerations, no incentive could be found for use of bipropellants over monopropellants.

Turbine Design

The thermodynamic properties of the working fluid provided by the monopropellant gas generator are significantly different from those obtained with a hydrogen-oxygen system. Therefore, it would be expected that an optimum monopropellant turbine design will be somewhat different than an optimum hydrogen-oxygen turbine. The lower monopropellant spouting velocities would tend to lead to lower pitch line velocities.

Figure 7-3 shows typical monopropellant design-point performance variation with turbine shaft speed and inlet pressure. For a turbine inlet pressure of 600 psia, optimum performance is obtained near 70,000 rpm.

Figure 7-4 shows the performance variation with pitch line velocity. A pitch line velocity of 1400 fps provides optimum turbine performance for a 600-psia inlet pressure. As mentioned previously, this lower velocity is compatible with the higher turbine inlet temperature obtained with the high-performance 75-24-1 monopropellant.

Turbine inlet pressure optimization is a function of the type of speed control. With pressure modulation, turbine inlet pressure is throttled by the speed control at part load and altitude operation. Therefore, it will be necessary to size the monopropellant pumping system for maximum power-sea level condition. At all other conditions, the pressure is reduced. Typically, for the booster mode power condition at 10,000-ft altitude, turbine inlet pressure is approximately 25 to 33 percent of that required for sea level full power. As a consequence, the system will pay a pumping power penalty for the relatively long duration mode power condition which represents 68 percent of the booster mission duration. Figure 7-5 shows a typical tradeoff between performance and turbine inlet pressure. If no pumping power penalty is paid, propellant consumption will continuously reduce with increasing inlet pressure. However, when the pumping power penalty is considered, optimum performance is obtained at a turbine inlet pressure for the mode power condition in the range from 600 to 800 psia (which corresponds to a pump design pressure from 1800 to 3200 psia).

Pulse modulation speed control leads to a different design inlet pressure level for the turbine. In this case, the part-load performance may not be sacrificed by reducing turbine pressure ratio. As previously indicated, the

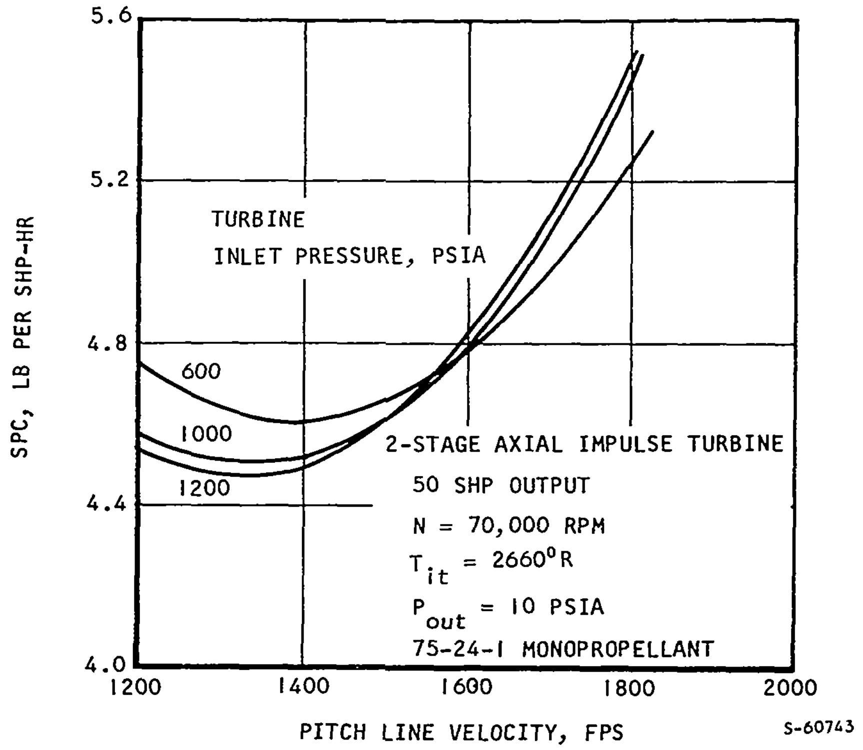


Figure 7-4. Monopropellant Turbine Pitch-Line Velocity

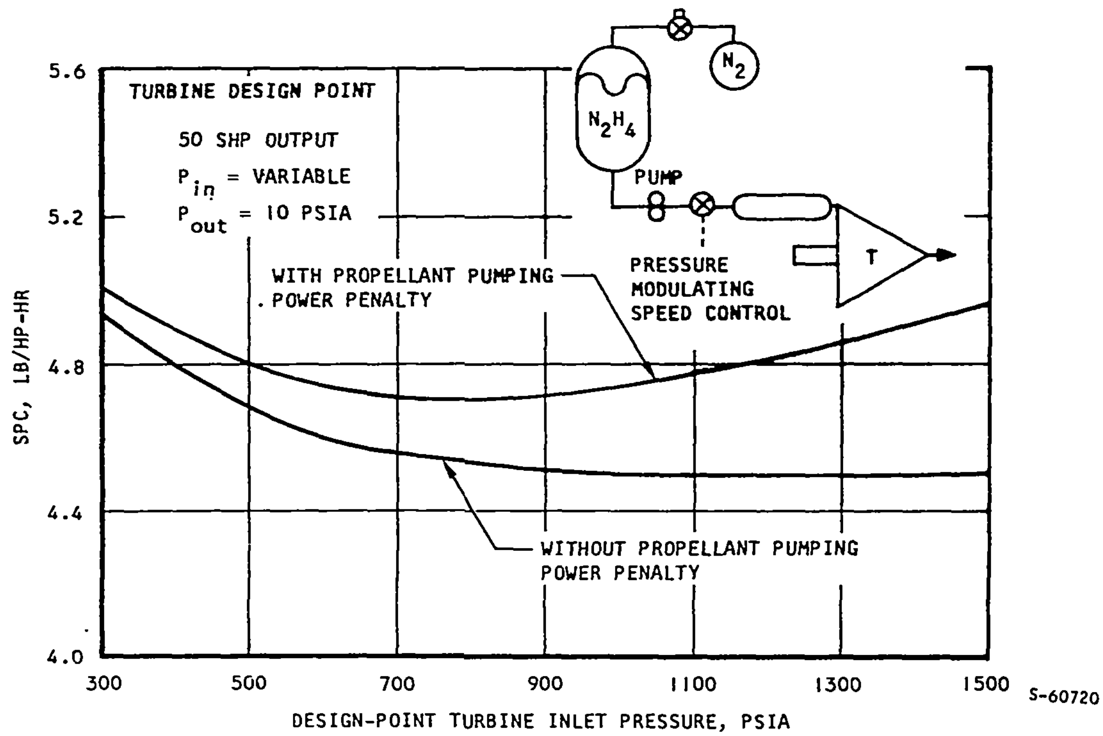


Figure 7-5. Monopropellant Turbine, Design-Point Inlet Pressure

relative merit of pulse modulation depends upon the speed control requirements. The volume of the gas generator is also important. It should be noted that monopropellants require large gas generator volumes as compared to hydrogen-oxygen combustors.

Propellant Exhaust Gas Properties

The exhaust products from a 75-24-I APU would consist of about 18 percent ammonia, 39 percent hydrogen, 12 percent water, and 31 percent nitrogen. Of these, only the ammonia represents a potential corrosion problem. However, AiResearch experience with monopropellant systems indicates that there will be no corrosion problems with any metals as long as the exhaust gas temperature is above the condensation point. When the exhaust gas is cooled sufficiently to allow water to condense (carrying dissolved ammonia with it), then minor corrosion problems may occur with copper or brass components. Thus, for the Space Shuttle APU, it appears that there will not be a corrosion problem if a monopropellant APU system is selected.

It should be noted that personnel must be kept isolated from the immediate vicinity of the APU exhaust gas flow, whether a hydrogen-oxygen or monopropellant APU is used because of the high gas temperatures and the absence of oxygen.

SYSTEM PERFORMANCE

Figure 7-6 shows the turbine performance map for a monopropellant turbine designed at altitude, mode power output. Unlike the hydrogen-oxygen systems, the monopropellant system can be benefit from recuperation since the propellant has little heat capacity. Therefore, system performance prediction can be obtained solely by using a variation of the mission integration program (described in Appendix F). This variant uses the following relationships:

Relation between turbine discharge pressure and ambient pressure (exhaust duct pressure drop equation)

Relation between turbine shaft power and power demanded at gearbox output pads

Relation between monopropellant pump drive power (assumed to be electrically driven using generator power) and pump throughflow

Turbine performance map (presented in Figure 7-6)

Fluid Requirements

Determination of the propellant flows involved a two-step iteration process, iterating first on turbine discharge pressure, and then on the power balance (propellant pump power, which is a function of flow, must be added to the power demanded at the gearbox output). This process was repeated for each of the mission segments for both the booster and orbiter. The total propellant and water requirements for the missions are shown in Table 7-I.

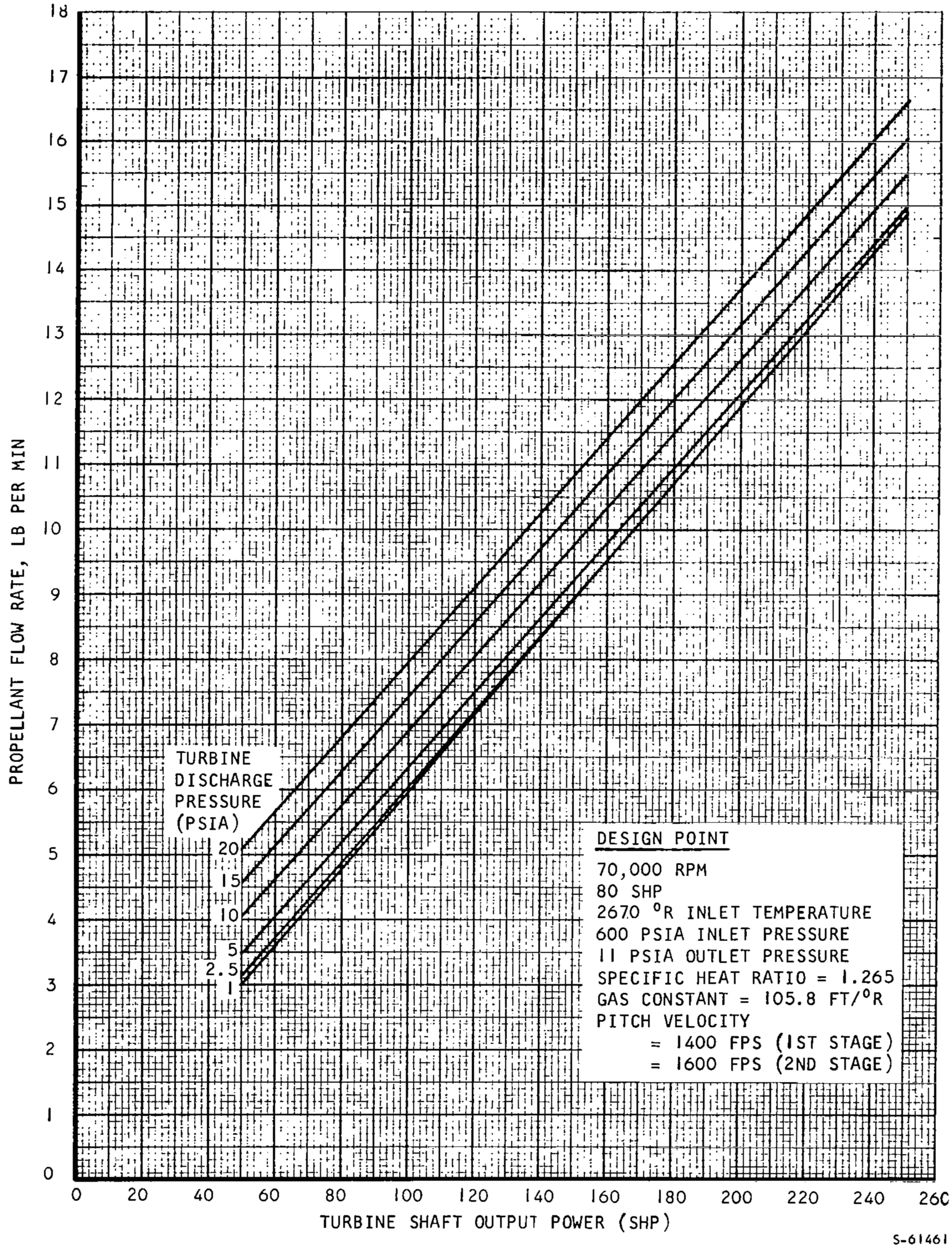


Figure 7-6. Monopropellant Turbine Performance Map

TABLE 7-1

MONOPROPELLANT SYSTEM FLUID REQUIREMENTS*

	Booster Vehicle	Orbiter Vehicle
75-24-1 monopropellant weight, lb	757.5	206.7
Water weight, lb	189.9	70.8

*Data include 5-percent residuals on both fluids.

Cycle State Points

Figures 7-7 and 7-8 show the state points in the monopropellant under the two extremes of operating conditions, sea level, full power, and space, zero hydraulic power. The schematics indicate that the water-glycol cooling flow is established by the sea level full power condition. At this high flow condition, the water boiler will only be able to cool the coolant to about 230°F because of the high ambient pressure, and the coolant flow out of the lube oil heat exchanger must be less than about 390°F (850°R--consistent with maximum temperature limitation placed on the hydrogen-oxygen systems at this point). Thus, the maximum temperature difference in the water-glycol loop is about 160°F. This difference, in conjunction with the maximum power heat loads, establishes the water-glycol flow rate.

WEIGHT

Table 7-2 summarizes the APU fixed weight as a function of the peak output power. Since the weight of monopropellant and water is dependent upon the details of the mission profile, these variable weights are excluded from the table. Figure 7-9 shows a plot of fixed weight vs output power. The inflection in the curve is due to the variation in available pump weights and speeds.

The fixed and variable weights of a 225 hp available gearbox output APU are presented in Table 7-3 for both the booster and orbiter vehicles.

Finally, using the scaling criteria described in Section 2, it is possible to determine the APU system weight as a function of power level and total energy output. Such curves are shown in Figures 7-10 and 7-11. The orbiter and booster parametric data are almost identical; however, there is a difference in the ratio of the amount of propellant to the amount of water required to perform the two missions. The booster mission requires about 0.251-lb water/lb propellant, whereas the orbiter mission requires 0.342-lb water/lb propellant.

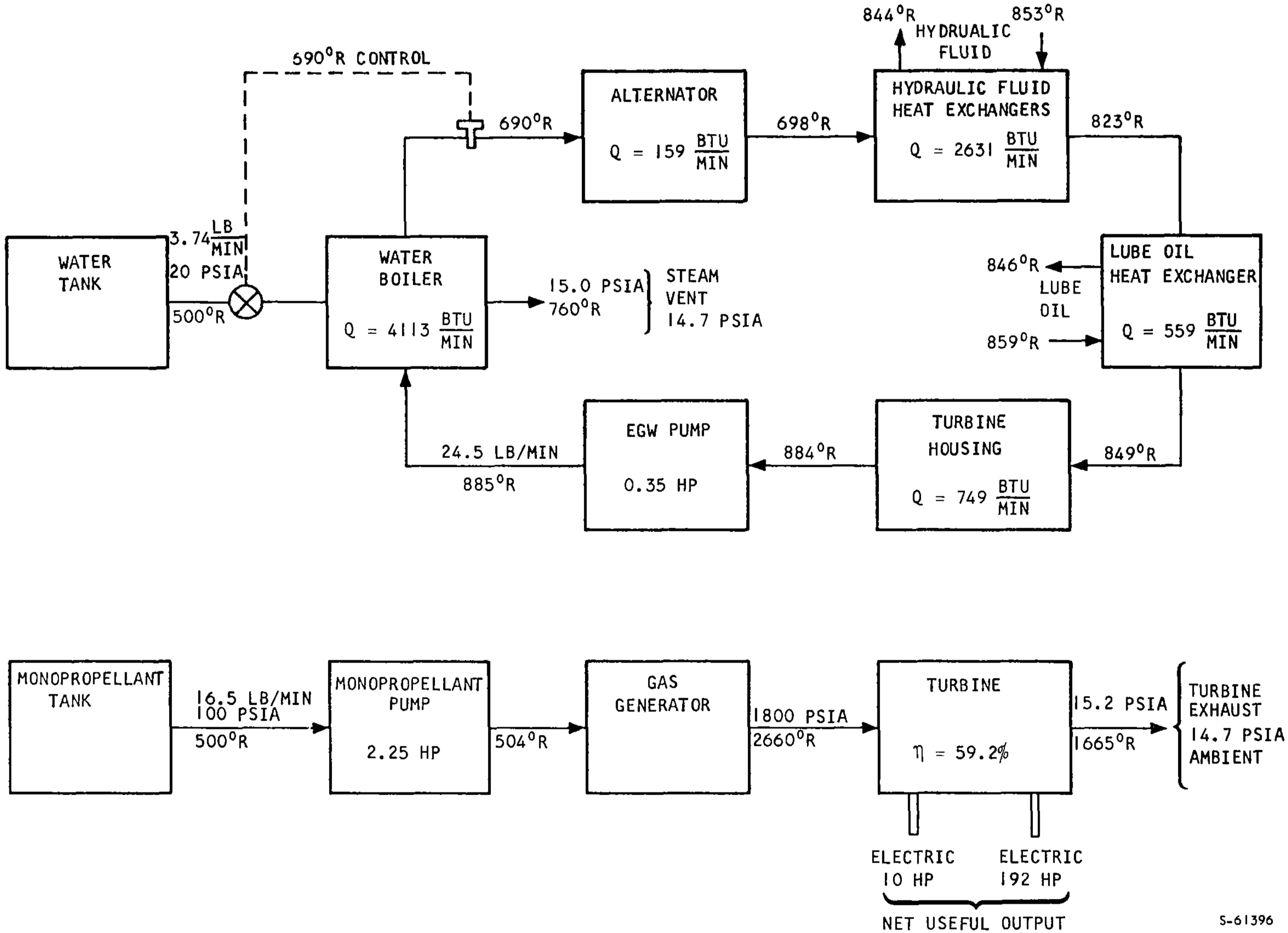


Figure 7-7. Sea Level, Full Power Cycle State Points Monopropellant APU System

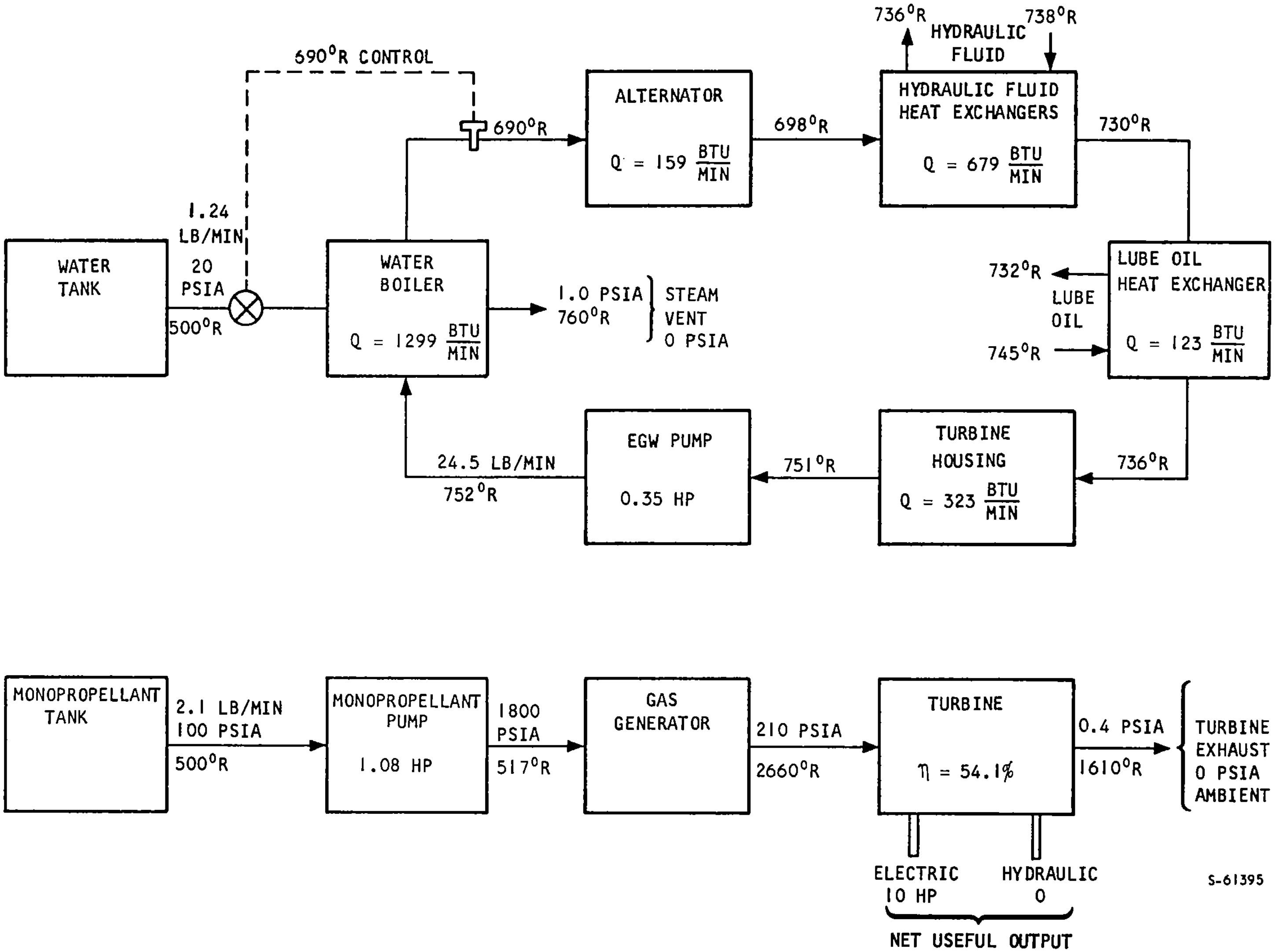


Figure 7-8. Space, Low Power Cycle State Points Monopropellant APU System

TABLE 7-2

APU FIXED WEIGHT MONOPROPELLANT SYSTEM

Component \ Peak Power Required	100 hp	225-hp Design Point	300 hp	500 hp	750 hp
Hydraulic pumps	14.0	25.0	44.0	103.0	145.0
Generator	11.2	24.0	34.0	61.5	90.2
Turbine (with containment)	44.0	44.0	44.0	44.0	44.0
Gas generator	2.0	3.0	3.5	4.5	5.4
Control logic devices	6.0	6.0	6.0	6.0	6.0
Control valves	4.5	5.0	5.5	6.0	7.0
Pressure regulators	2.0	2.0	2.0	2.5	3.0
Coolant reservoir	9.4	20.0	26.2	43.1	64.1
Heat exchangers: Water boiler	14.2	32.0	42.7	71.1	106.7
Heat exchangers: Hydraulic oil cooler	2.0	4.0	4.6	5.6	6.7
Heat exchangers: Lube oil cooler	0.7	1.0	1.3	2.1	3.1
Reactant pumps: Monopropellant pump	3.1	4.0	5.5	10.3	13.6
Reactant pumps: Coolant pump	2.2	4.0	7.0	16.5	23.2
Gearbox with lube pump	5.7	17.4	26.6	58.9	92.5
Lube oil in sump	7.7	7.7	7.7	7.7	7.7
Instrumentation	1.0	1.0	1.0	1.0	1.0
Ducting	27.8	40.2	46.1	59.0	72.0
Subtotal	167.5	240.3	307.7	502.8	691.2
10 percent for vehicle supporting structure	16.7	24.0	30.8	50.3	69.1
Total fixed weight	140.1	197.6	246.1	553.1	760.3

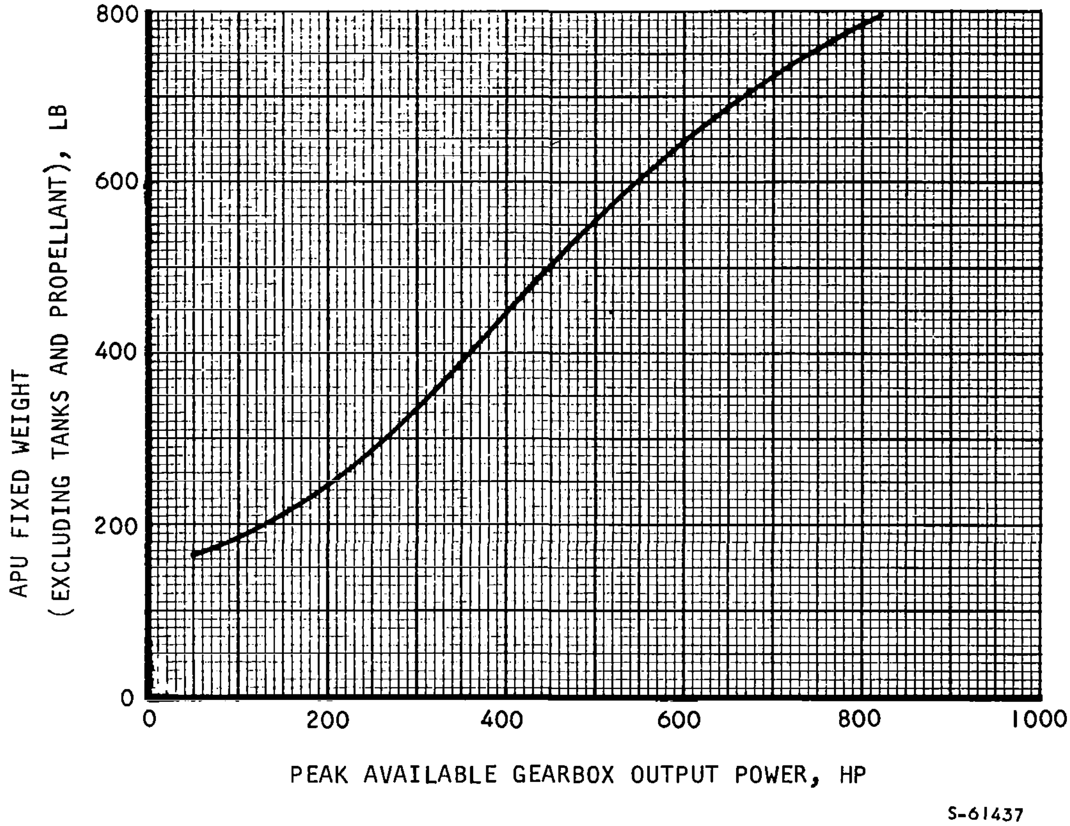


Figure 7-9. APU Fixed Weight, Monopropellant System

TABLE 7-3

BOOSTER AND ORBITER SYSTEM WEIGHTS FOR 225 HP APU
MONOPROPELLANT SYSTEM

Vehicle	Booster	Orbiter
Fixed weight, lb	264.3	264.3
Monopropellant weight, lb	756.5	206.7
Monopropellant tank weight, lb	16.1	7.0
Water weight, lb	189.9	70.8
Water tank weight, lb	7.0	4.1
Nitrogen tank and regulators weight, lb	8.3	4.3
Total system weight, lb	1242.1	557.2

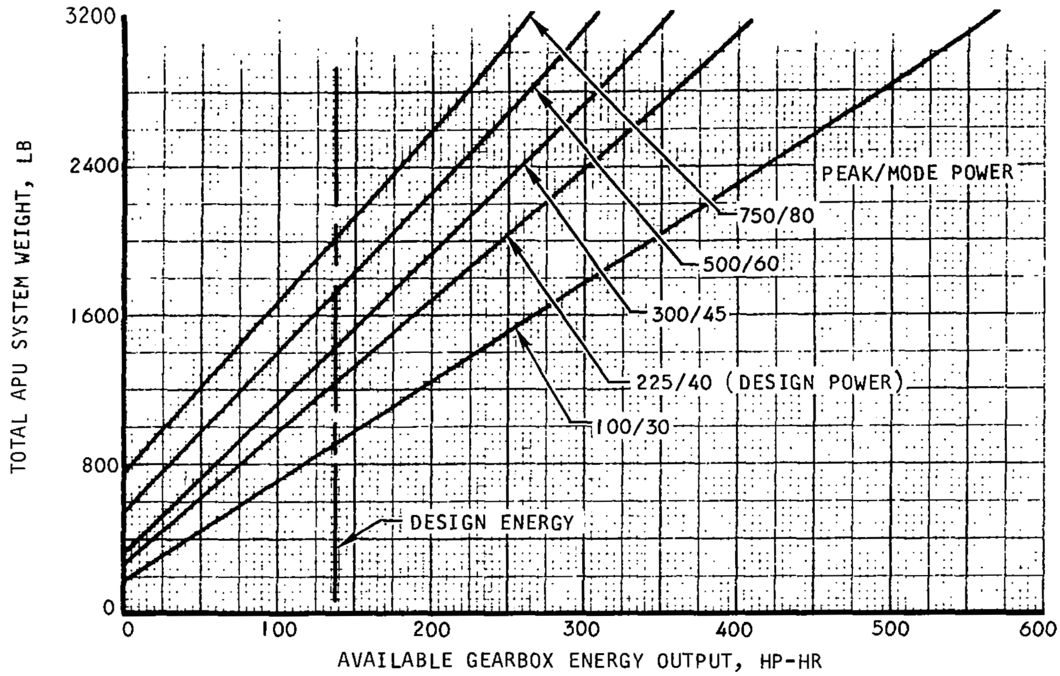


Figure 7-10. Booster Vehicle APU System Weight, Monopropellant System

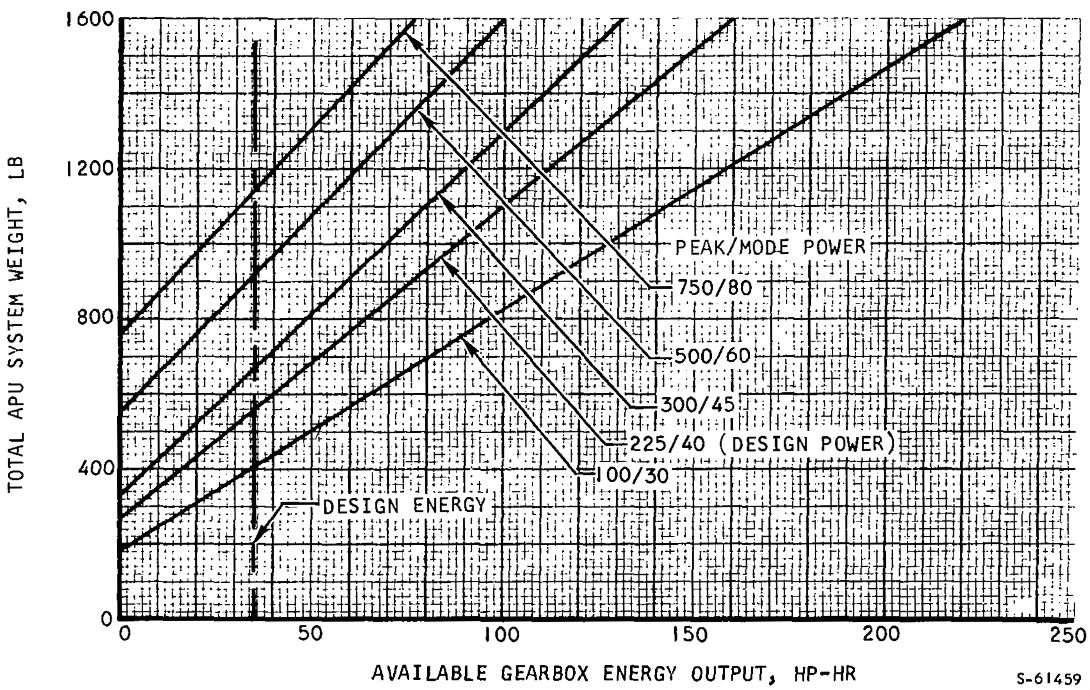


Figure 7-11. Orbiter Vehicle APU System Weight, Monopropellant System

SECTION 8

SYSTEM COMPARISON AND EVALUATION

INTRODUCTION

Sections 3 through 7 have established the performance capabilities of the five candidate systems that have been studied during the second half of Phase I of the APU system study. These data are compared with one another in this section to establish the relative incentive for selection of the various system candidates. The system candidates are as follows:

Low-pressure cryogenic liquid supplied system (Section 3). Uses cryogenic pumps to achieve an optimum turbine inlet pressure).

Integral high pressure cryogenic supplied system (Section 4). Cryogens are stored at a high pressure with pressure being established by trading off tank weight and turbine performance as functions of system pressure.

High pressure gaseous hydrogen-oxygen supplied system (Section 5). Cryogens are supplied as high-pressure gases from another system on-board the vehicle.

Dual-mode system (Section 6). Uses a hydrogen-oxygen turbine for space operation and a hydrogen-fueled gas turbine for atmosphere operation.

Monopropellant system (Section 7). Monopropellant turbine using an expendable evaporant, water, as the system heat sink.

SYSTEM COMPARISON

Table 8-I is a summary comparison of the final series of APU systems studied for the Space Shuttle and described in previous sections of this report. To the basic five types of systems, two variants are shown, one reflecting the difference between shared and separate tankage (for the low-pressure cryogenic supplied system), the other reflecting a difference in inlet propellant conditions (for the high-pressure gaseous supplied system). The weight comparisons shown in the table were made on the basis of the following ground rules:

(a) System weight includes propellant and propellant tankage penalty.

(b) Propellant tankage penalty for separate tanks based upon mission requirements for one APU with separate tankage.

(c) Propellant tankage penalty for shared tanks based upon incremental penalties for adding to a system containing 10,000 lb of hydrogen and 10,000 lb of oxygen.

(d) Gaseous supplied systems penalized for propellants required for thermal conditioning and tankage for storage of thermal conditioning propellants.

(e) Systems weights are given for both soft shell and hard shell cryogenic tankage.

The data given in this table established the basis for the system evaluations to be discussed subsequently.

SYSTEM EVALUATION

NASA has established the evaluation criteria and weighting factors as the following:

Cost: Item	Weighting	Reliability: Item	Weighting
Low Weight	25	Simplicity	30
High Flexibility	20	Experience	5
Ease of Development	10		
Ease of Manufacturing	5		
Ease of Maintenance	5		

The basis for establishing system ratings for these various considerations will be discussed in the paragraphs following. It should be noted that minor changes in the method of evaluation (such as placing added emphasis on key areas such as controls and pumps) do not effect the relative ranking of the candidate systems.

Weight Evaluation

Table 8-2 lists system weight for the booster and orbiter missions and the ratings based upon (1) booster mission, (2) orbiter mission, and (3) a total mission in which the orbiter system weight is given a weighting of six (to reflect the weight cost to the booster of orbiter weight). The total mission rating most accurately represents relative system merit. This weight evaluation is based upon the assumption of system commonality between the booster and orbiter, which at the present time appears to be appropriate.

System Flexibility Evaluation

System flexibility represents the ability of the system to meet changing mission requirements with minimum modification. Two types of change can be anticipated; one involves increasing APU operating time; the other involves increasing the output power level. Changes of the first type will be reflected primarily in the expendable requirements. If the expendables are supplied by integral storage in the APU system, changes in tank size will be required. If

TABLE 8-I

APU SYSTEM COMPARISON SUMMARY

APU System Configuration	System Weight,* lb: Booster	System Weight,* lb: Orbiter	Tankage	Propellant Conditioning: Pressure	Propellant Conditioning: Temperature	Heat Sink	Primary Controls	No. Major Compo-nents	Interface Considerations: With Vehicle	Interface Considerations: With Ground Support	Development Problem Areas	Booster/Orbiter System Commonality
Low-pressure cryogenic liquid supplied system (integral low-pressure tanks)	604 (689)	388 (417)	Separate low-pressure	Electric motor driven pumps	Recuperative in cycle	Propellant flow	• H_2 flow control • O/F (O_2) control • H_2 recycle control • H_2 tank pressure control • O_2 tank pressure control	23	• APU installation • Exhaust duct installation • Tankage installation	• Cryogenic H_2 supply • Cryogenic O_2 supply • Gearbox lubricant • Hot N_2 for ground runup	• Low-flow, high-head H_2 and O_2 pumps • Pump cavitation with saturated inlet • Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Same except for cryogenic storage tanks
Low-pressure cryogenic liquid supplied system	537 (629)	310 (333)	Shared low-pressure	Electric motor driven pumps	Recuperative in cycle	Propellant flow	• H_2 flow control • O/F (O_2) control • H_2 recycle control	17	• Cryogenic delivery from tanks to pumps • APU installation • Exhaust duct installation	• Gearbox lubricant • Hot N_2 for ground runup	• Low-flow, high-head H_2 and O_2 pumps • Pump cavitation with saturated inlet • Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Same
Integral high-pressure cryogenic supplied system	684 (779)	425 (458)	Separate high-pressure	Direct feed from tanks	Recuperative in cycle	Propellant flow	• H_2 flow control • O/F (O_2) control • H_2 recycle control • H_2 tank pressure control • O_2 tank pressure control	14	• APU installation • Tank installation • Exhaust duct installation • Evaporation exit duct installation	• Cryogenic H_2 supply • Cryogenic O_2 supply • Gearbox lubricant • Hot N_2 for ground runup	• Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Same except for cryogenic storage tanks
High-pressure gaseous supplied system (500°R, 300 psia)	883 (1071)	393 (436)	Shared	Shared	Shared	Propellant flow plus expendable evaporant H_2O	• H_2 flow control • O/F (O_2) control • H_2 recycle control (cold) • Evaporator H_2O flow control	10	• Propellant delivery state • APU installation • Exhaust duct installation • Evaporator exit duct installation	• Distilled H_2O supply • Gearbox lubricant • Hot N_2 for ground runup	• Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Same except for water storage tank
High-pressure gaseous supplied system (200° to 300°R, 600 psia or greater)	573 (685)	307 (331)	Shared	Shared	Recuperative in cycle	Propellant flow	• H_2 flow control • O/F (O_2) control • H_2 recycle control (hot)	12	• Propellant delivery state • APU installation • Exhaust duct installation	• Gearbox lubricant • Hot N_2 for ground runup	• Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Same
Dual mode system Propellant turbine Air breathing gas turbine	629 (677)	473 (526)	Separate high-pressure H_2 from propellant turbine tanks	Direct feed from tanks Direct feed from tanks	Recuperative in cycle Vaporizer in engine	Propellant flow Ram air	Propellant turbine: • H_2 flow control, • O/F (O_2) control • H_2 recycle control gas turbine: • GT fuel flow control, • GT overtemperature limit • Ram airflow control • Clutch control	24	• Ram air inlet and exit ducting installation • APU installation • Propellant turbine exhaust duct installation	• Cryogenic H_2 supply • Cryogenic O_2 supply • Heat transport fluid • Gearbox lubricant • Gaseous H_2 for ground runup	• Temperature sensor for O/F control • High turndown ratio for pressure modulating type control	Gas turbine power section and ram air heat exchanger deleted from orbiter system
Monopropellant system	1242	557	Integral low-pressure tanks	Electric motor driven pumps	Not required	Expendable evaporant	• Monofuel flow control • Evaporator H_2O flow control	14	• Water storage tank thermal control • APU installation • Turbine exhaust duct installation • Evaporator exit duct installation	• Monofuel supply • Distilled H_2O supply • Heat transport fluid • Gearbox lubricant • Hot N_2 for ground runup	• Long-life gas generator design	Same except for monopropellant and evaporant storage tanks

*Weights without parentheses are for soft shell tanks.
Weights with parentheses are for hard shell tanks.

TABLE 8-2

SYSTEM RATING FOR WEIGHT

System	System Weight, lb		System Rating		
	Booster Mission	Orbiter Mission	Booster Mission	Orbiter Mission	Total Mission
Low-pressure cryogenic supplied system (integral tanks)	604	388	0.890	0.790	0.818
Low-pressure cryogenic supplied system (shared tanks)	537	310	1.000	0.990	1.000
Integral high-pressure supplied system	684	425	0.786	0.723	0.741
Gaseous supplied system (500°R, 300 psia)	883	393	0.609	0.780	0.754
Gaseous supplied system (200°R, 650 psia)	573	307	0.938	1.000	0.993
Dual mode system	629	473	0.854	0.650	0.691
Monopropellant system	1242	557	0.433	0.551	0.523

TABLE 8-3

SYSTEM RATING FOR FLEXIBILITY

	Number of Changes Required for Increased:			
	Mission Duration	Output Power	Total	Rating
Low-pressure cryogenic supplied system (integral tanks)	2	23	25	0.440
Low-pressure cryogenic supplied system (shared tanks)	0	17	17	0.646
Integral high-pressure supplied system	2	14	16	0.688
Gaseous supplied system (500°R, 300 psia)	1	10	11	1.000
Gaseous supplied system (200°R, 650 psia)	0	12	12	0.915
Dual mode system	2	24	26	0.423
Monopropellant system	2	14	16	0.688

the expendables are obtained from shared tankage, it will probably not be necessary to make changes in the APU system. In this case, systems using shared tankage will rate higher than those having integral tankage. With changes in system requirements, which involve significant increases in system output power, it will be necessary to resize most of the system components. As a consequence, the criteria for this type of flexibility involve the total number of major components comprising the system. Table 8-3 gives the system flexibility evaluation.

Ease of Development Evaluation

As shown in Table 8-4, the following three factors were assumed to be indicative of the relative ease of development:

Number of rotating components in system

Number of control functions

Number of development problem areas

The sum of these factors should provide a relative index of the development effort required for the various systems. As might be expected, the monopropellant system emerges with the highest rating here.

Ease of Manufacturing Evaluation

In this evaluation (shown in Table 8-5), the total number of major components, the number of rotating components, and the number of complex assemblies provide the basis for the evaluation. Largely because of having a fewer number of components, the gaseous supplied systems have the highest rating for this factor.

Ease of Maintenance Evaluation

The maintenance evaluation given in Table 8-6 assumes that the primary activities will involve system checkout of control functions and resupplying expendable materials. The low-pressure cryogenic supplied system and the $200^{\circ}R$, 650 psi gaseous supplied system are tied for the higest ratings in this category.

System Simplicity Evaluation

The system simplicity evaluation (Table 8-7) is based upon the total number of major components, the number of control functions, and the number of rotating components. The gaseous supplied systems show the highest ratings in this evaluation.

System Experience Evaluation

The primary criteria in the system experience evaluation (Table 8-8) is based upon the number of development problem areas. The monopropellant system with the fewest development problem areas has the highest rating in this category.

TABLE 8-4

SYSTEM RATING FOR EASE OF DEVELOPMENT

	Number of Rotating Components	Number of Control Functions	Number of Development Problem Areas	Total Number	System Rating
Low-pressure cryogenic supplied system (integral tanks)	6	5	6	17	0.470
Low-pressure cryogenic supplied system (shared tanks)	6	3	6	15	0.533
Integral high-pressure supplied system	4	5	2	11	0.726
Gaseous supplied system (500°R, 300 psia)	4	4	2	10	0.800
Gaseous supplied system (200°R, 650 psia)	4	3	2	9	0.890
Dual mode system	6	7	2	15	0.534
Monopropellant system	5	2	1	8	1.000

TABLE 8-5

SYSTEM RATING FOR EASE OF MANUFACTURE

	Number of Major Components	Number of Rotating Components	Number of Complex Assemblies	Total Number	System Rating
Low-pressure cryogenic supplied system (integral tanks)	23	6	9	38	0.500
Low-pressure cryogenic supplied system (shared tanks)	17	6	7	30	0.634
Integral high-pressure supplied system	14	4	7	25	0.760
Gaseous supplied system (500°R, 300 psia)	10	4	5	19	1.000
Gaseous supplied system (200°R, 650 psia)	12	4	5	21	0.904
Dual mode system	24	6	9	39	0.488
Monopropellant system	14	5	6	25	0.760

TABLE 8-6

SYSTEM RATING FOR EASE OF MAINTENANCE

	Number of Control Functions	Number of Materials Supplied	Total Number	System Rating
Low pressure cryogenic supplied system (integral tanks)	5	4	9	0.555
Low-pressure cryogenic supplied system (shared tanks)	3	2	5	1.000
Integral high-pressure supplied system	5	4	9	0.555
Gaseous supplied system (500°R, 300 psia)	4	3	7	0.714
Gaseous supplied system (200°R, 650 psia)	3	2	5	1.000
Dual mode system	7	5	12	0.416
Monopropellant system	2	5	7	0.714

TABLE 8-7

SYSTEM RATING FOR SIMPLICITY

	Number of Major Components	Number of Control Functions	Number of Rotating Components	Total Number	System Rating
Low-pressure cryogenic supplied system (integral tanks)	23	5	6	34	0.529
Low-pressure cryogenic supplied system (shared tanks)	17	3	6	26	0.693
Integral high-pressure supplied system	14	5	4	23	0.783
Gaseous supplied system (500°R, 300 psia	10	4	4	18	1.000
Gaseous supplied system (200°R, 650 psia)	12	3	4	19	0.945
Dual mode system	24	7	6	37	0.486
Monopropellant system	14	2	5	21	0.855

TABLE 8-8

SYSTEM RATING FOR EXPERIENCE

	Number of Development Problem Areas	System Rating
Low-pressure cryogenic supplied system (integral tanks)	6	0.167
Low-pressure cryogenic supplied system (shared tanks)	6	0.167
Integral high-pressure supplied system	2	0.500
Gaseous supplied system (500^{0}R, 300 psia)	2	0.500
Gaseous supplied system (200^{0}R, 600 psia	2	0.500
Dual mode system	2	0.500
Monopropellant system	1	1.000

SYSTEM RATING

Table 8-9 summarizes the ratings discussed previously and applies the weighting factors to establish total weighted ratings for each system. Following is a listing of the relative standing of the various systems together with the total weighted rating (based on a maximum possible total score of 100).

System	Total Weighted Rating
200^0R, 650 psia gaseous supplied system	93.39
500^0R, 300 psia gaseous supplied system	87.92
High-pressure cryogenic supplied system (integral tanks)	73.11
Low-pressure cryogenic supplied system (shared tanks)	73.04
Monopropellant system	69.85
Low-pressure cryogenic supplied system (integral tanks)	55.93
Dual mode system	52.68

CONCLUSIONS

Based upon the foregoing evaluation, the gaseous supplied system is recommended for the Space Shuttle APU. Considerable incentive is shown for operation at low inlet temperature and the highest available pressure. This system has the disadvantage of depending upon another system (APS) that has not yet been defined. As a consequence, the APU system will be required to have a measure of flexibility with respect to the ability to accommodate a range of inlet conditions.

TABLE 8-9

SYSTEM EVALUATION

Evaluation Category	Weighting Factor	Low-Pressure Cryogenic Supplied System (Integral Tanks)		Low-Pressure Cryogenic Supplied System (Shared Tanks)		High-Pressure Cryogenic Supplied System (Integral Tanks)		Gaseous Supplied System (500°R, 300 psia)		Gaseous Supplied System (200°R, 650 psia)		Dual Mode System		Monopropellant System	
		Rating	Weighted Rating	Rating	Weighted Rating	Rating	Weighted Rating	Rating	Weighted Rating	Rating	Weighted Rating	Rating	Weighted Rating	Rating	Weighted Rating
Weight	25	0.818	20.45	1.000	25.00	0.741	18.52	0.754	18.85	0.993	24.82	0.691	17.28	0.523	13.07
Flexibility	20	0.440	8.80	0.646	12.92	0.688	13.76	1.000	20.00	0.915	18.30	0.423	8.46	0.688	13.76
Ease of development	10	0.470	4.70	0.533	5.33	0.726	7.26	0.800	8.00	0.890	8.90	0.534	5.34	1.000	5.00
Ease of manufacturing	5	0.500	2.50	0.634	3.17	0.760	3.80	1.000	5.00	0.904	4.52	0.488	2.44	0.760	3.80
Ease of maintenance	5	0.555	2.78	1.000	5.00	0.555	2.78	0.714	3.57	1.000	5.00	0.416	2.08	0.714	3.57
Simplicity	30	0.529	15.87	0.693	20.79	0.783	23.49	1.000	30.00	0.945	28.35	0.486	14.58	0.855	25.65
Experience	5	0.167	0.83	0.167	0.83	0.500	2.50	0.500	2.50	0.500	2.50	0.500	2.50	1.000	5.00
Total		-	55.93	-	73.04	-	73.11	-	87.92	-	93.39	-	52.68	-	69.85
Relative system rating		6		4		3		2		1		7		5	

SECTION 9

TECHNOLOGY EVALUATION

INTRODUCTION

Critical technology development areas can be defined as having one or more of the following features:

(a) Design is complex

(b) There are no similar components in proven use

(c) A major component characteristic is not amenable to analysis and test verification is not available

(d) Manufacturing processes are not proven

(e) Performance, life, and reliability have a major impact on system tradeoffs and selection processes

(f) Development risk is high

To a very considerable extent, the Phase I activities have been directed to establishing candidate system concepts with a minimum of these characteristics. Then, in the system comparison and evaluation studies summarized in the previous section, the selection criteria led to recommendation of a system with the fewest problem areas.

For the recommended gaseous hydrogen and oxygen supplied system, two problem areas were identified:

(a) High turndown ratio for the flow controls

(b) Reliable long-life temperature sensors for O/F mixture ratio control (past experience indicates temperature sensor signals have a tendency to lose calibration when operated in a hydrogen environment for extended periods.)

The low-pressure cryogenic supplied system was penalized because of the problem areas anticipated with the propellant pumps. These problems resulted from the low flows and high heads, the desire to use centrifugal pump designs for reliability, and pump operation with saturated cryogenic liquids and the resulting cavitation at the inlet. Work with small capacity cryogenic pumps will proivde a useful technological base for future systems.

Pressure modulating control was assumed for the systems described in this report. Pulse modulating control may offer performance advantage at the very high turndown ratios required for the Space Shuttle APU. However, pulse modulating control requires a large accumulator (weight on the order of 50 to 100 lb) for stability in driving a variable-delivery hydraulic pump. In addition, pulse modulating control imposes a large number of cycles on the control valves and requires continuous operation of the combustor igniter. Both of these characteristics tend to limit component operating life. If pulse modulation control is selected, technology work in the various areas discussed above will be desirable.

APPENDIX A

H_2-O_2 TURBINE PARAMETRIC DESIGN STUDIES

Parametric turbine studies were conducted to obtain the optimum turbine configuration for the Space Shuttle APU application. These studies included selection of the type turbine most appropriate, optimization of the various parameters associated with the turbine design and evaluation of competitive turbine designs over the assumed power-altitude profile to obtain the minimum propellant requirement. The results of these studies are presented below.

A summarization of the turbine design selected as being best for the pumped propellant system in the Space Shuttle APU application is presented in Table I. A performance map of this turbine is also shown in Table I.

SELECTION OF TURBINE TYPE

A preliminary screening was made of all turbine types which might be competitive in the space shuttle APU application. The following candidate turbine types were eliminated for the reasons stated.

Radial Flow--Eliminated on the basis of stress and thermal problems relative to other competitive types.

Reaction--Eliminated on the basis of optimum specific speed considerations.

The preliminary screening resulted in the following three types of turbines being competitive in this application:

Pressure-compounded, multi-stage, single-disk (reentry), axial impulse

Pressure-compounded, multistage multiple disk, axial impulse

Velocity-compounded, multistage, multiple-disk, axial

Final selection of the turbine type was made on the basis of performance studies described in this appendix. This evaluation emphasized two particular criteria: (1) minimize the integrated mission propellant requirements, and (2) maximize the advantages in the design of the hardware. The final turbine types eliminated were

Reentry--Eliminated on the basis of no significant advantage relative to the other competitive types and known problems with seals and close clearances.

Velocity-compounded--Eliminated on the basis of no significant advantage relative to the other competitive types and has long blades leading to problems with blade vibration and blade root stresses.

TABLE I

TYPICAL SPACE SHUTTLE APU TURBINE OPTIMUM FOR PUMPED PROPELLANT SYSTEM

A. DESIGN SUMMARY

Turbine Type:	2-Stage, 2-Disk, Pressure-Compounded, Axial Impulse
Control Method:	Turbine Inlet Pressure Modulation with Load
Propellants:	Hydrogen and Oxygen
Design Point:	*Mode Power at 10,000-ft altitude* 80 shp 70,000 rpm 2260°R inlet temperature 220 psia inlet pressure 11 psia outlet pressure 600 psia max. inlet pressure (at Max. Power) O/F = 0.45

Design Parameters:

	First Stage	Second Stage
Nozzle throat area, sq in.	0.0447	0.2351
Nozzle exit area, sq in.	0.0869	0.2937
Nozzle exit angle, deg	16.0	16.0
Blade inlet angle, deg	19.2	21.5
Pitch diameter, in.	5.89	6.54
Arc of admission, percent	9.2	31.0
Blade height, in.	0.265	0.239
Blade chord, in.	0.350	0.350
Tip gap, in.	0.010	0.010
Number of blades	85	85

B. PERFORMANCE MAP

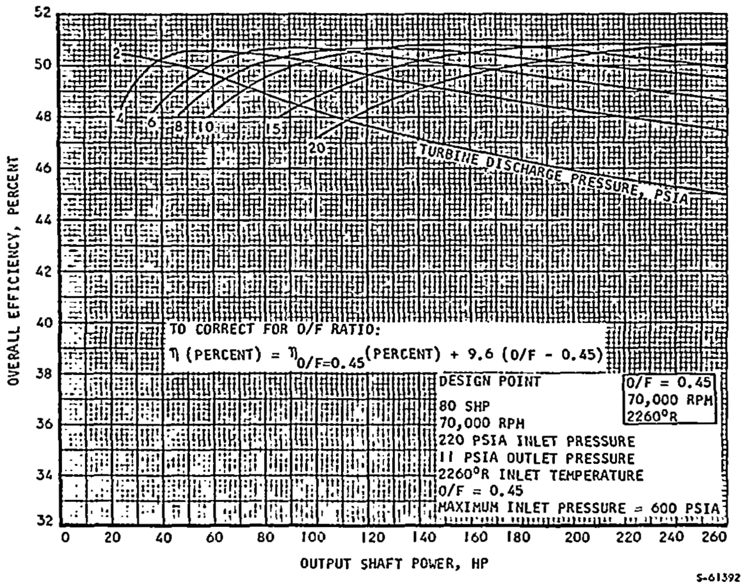


Figure 1a shows a typical design-point performance comparison of the pressure-compounded reentry turbine and the two-disk pressure compounded turbine. The data indicate that the reentry turbine offers no performance advantages in comparison to the pressure-compounded turbine.

The performance of two-stage pressure-compounded and velocity-compounded turbines is compared in Table 5 (presented later in this appendix) for an assumed mission profile. The data indicate that the pressure-compounded turbine shows a performance advantage over the velocity-compounded turbine, particularly when the two turbines are designed at sea level, full power output. When the design point is at altitude, mode power, the pressure-compounded turbine shows about 6 lb propellant weight savings over the velocity-compounded turbine. Thus, the pressure-compounded turbine offers superior aerodynamic performance (lower propellant consumption rate) regardless of the design point selected for the turbine.

The velocity-compounded turbine design used in the performance comparison of Table 5 is one having a near-optimum blade h/D ratio on the first stage of about 0.05, or a blade height of 0.3 in. on the first stage. At this condition, although the efficiency is maximized, the second-stage has extremely long blades which would create mechanical design problems (excessive root stresses and vibrational excitation, as discussed below). Figure 1b shows cross-sections of the turbine stages for both the pressure-compounded and velocity-compounded turbine designs. Although the first stage blading heights are about equal (0.265 in. for pressure-compounded and 0.300 in. for velocity-compounded), the difference in the second-stage blading heights is readily apparent.

In an attempt to reduce the height of the second-stage blading, a series of velocity-compounded turbine designs have been investigated. These designs are summarized in Table 2. In these designs, it has been possible to reduce the second-stage blading height by reducing the first-stage blading height. Thus, by halving the first-stage blade height to 0.150 in., the second-stage blade height is reduced from 1.114 in. to 0.557 in. The resulting decrease in efficiency is only about 0.3 percent, or about 4 lb of propellant weight increase for the mission assumed in Table 5. As discussed below, a blade height of 0.557 in. is acceptable mechanically. So, with the reduced blade height, the velocity-compounded turbine can be made to achieve an acceptable mechanical design; however, the difference between its performance and that of the pressure-compounded turbine has been increased to about 10 lb of propellant.

Two separate analyses were conducted to establish the acceptability of a given blading height. These are:

- effects of blade height on the stresses existing at the blade root
- effects of blade height on the blade natural frequency and on the blading excitation frequencies caused by passing stator vanes

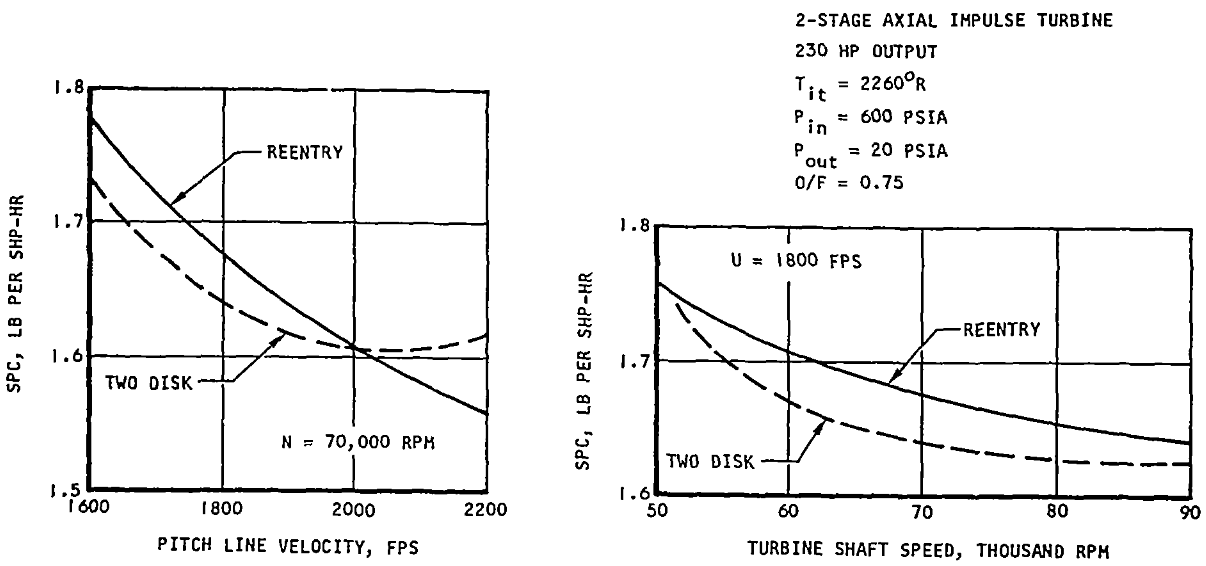
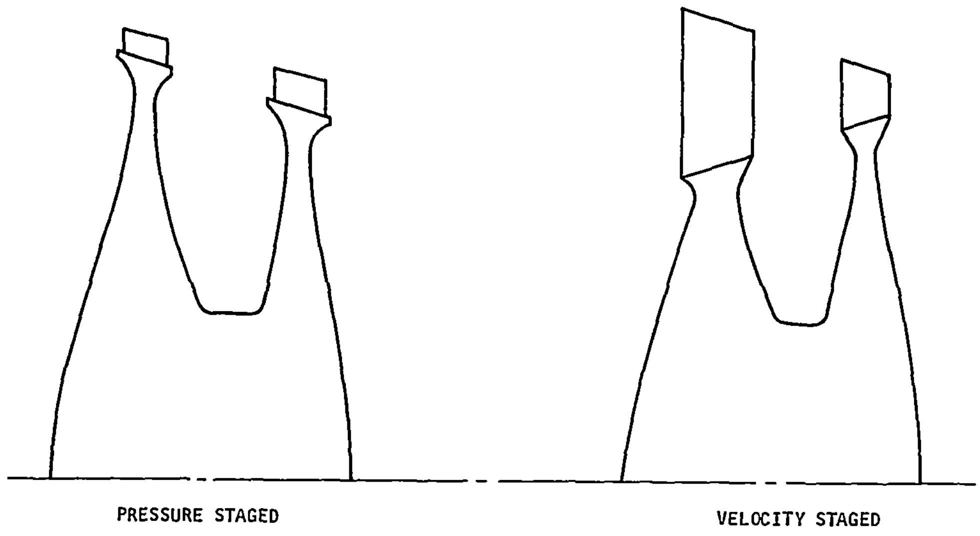


Figure la. Comparison of 2-Disk and Reentry 2-Stage Turbines



- WHEEL WEIGHTS AND INERTIAS ARE ESSENTIALLY THE SAME FOR PRESSURE STAGE AND VELOCITY STAGED TURBINES
- VELOCITY STAGED BLADE ROOT TENSILE STRESS IS 3 TIMES HIGHER THAN PRESSURE STAGED
- PRESSURE STAGED TURBINE HAS BLADE BENDING FREQUENCIES 50 TIMES HIGHER THAN VELOCITY STAGED
- VELOCITY STAGED TURBINE HAS AT LEAST 3 MODES OF BLADE STATOR EXCITATION FREQUENCIES BELOW THE OPERATING SPEED

Figure lb. Comparison of Pressure-Compounded and Velocity-Compounded 2-Stage Turbine Wheel Designs

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TABLE 2

VELOCITY-COMPOUNDED TURBINE DESIGNS

	Turbine A	Turbine B	Turbine C
Design Point Conditions	Identical for all turbines; 80 hp output, 11 psia discharge pressure, 496 psia inlet pressure, 0.45 O/F ratio, inlet temperature 1800°F, tip speed 1800 fps, 70,000 rpm		
First-stage Blade Height, in.	0.150	0.200	0.300
Second-stage Blade Height, in.	0.557	0.75	1.114
Nozzle Admission, percent	31.21	23.25	15.50
Efficiency, percent	46.41	46.72	46.73

The blade root stress analysis indicates that the blade root stress for blades having a taper ratio of 2 will be about 90,000 psi per in. of blade height. Thus, if the blading is to meet the creep criteria discussed in Appendix B (Figure 8 of Appendix B), the root stress must be less than about 50,000 psi to meet the required combination of creep and operating endurance. Thus, the maximum blade height should not exceed about 0.56 in. The blading for the pressure-compounded design is well within this limitation. That for the velocity-compounded design just meets this requirement.

The blade natural frequency analysis (assuming constant blade cross-section and ignoring stiffening due to the centrifugal forces) shows that blades having heights of about 0.26 in. (those for the pressure-compounded turbine) will have a natural frequency in excess of 100,000 Hz. At a height of 0.557 in., the natural frequency is about 11,500 Hz. The most likely possible blading excitation source, the stator vanes (assuming 85 stator vanes) will cause the velocity-compounded blading to become excited at speeds well within the normal operating range of the turbine. However, the pressure-compounded blading can be designed so that there will be no blading excitation within the normal speed capability of the turbine. Therefore, although it would be possible to design the velocity-compounded blading successfully, the pressure-compounded blading provides a straight-forward design in which there will be no concern about possible blading excitation.

In summary, for a typical mission, the pressure-compounded turbine has a performance advantage of about 10 lb of propellant (about 15 lb of stored propellant weight, assuming very large low pressure hard shell tanks) over the velocity-compounded turbine. Additionally, if the propellants are supplied to the APU as gases, then the total performance advantage, including the propellant conditioning penalty incurred to supply the APU gas, is about 20 lb, assuming 300°R gas supply temperature at the APU. Part of this weight advantage is due to the performance degradation required on the velocity-compounded turbine to achieve acceptable mechanical design. However, even at this weight penalty, the velocity-compounded turbine will have higher blade root stresses and considerably lower blade natural frequencies. Therefore, it can be concluded that the pressure-compounded turbine should be selected for the Space Shuttle APU application. It shows a performance advantage, and has a slight mechanical design advantage.

PARAMETRIC STUDIES

Extensive investigations were conducted to determine the sensitivity of design point performance to various turbine design variables. These investigations resulted in the establishment of optimum values for some of the turbine design variables while appropriate values of other variables were obtained in conjunction with the stress and thermal analyses. The turbine inlet temperature and pitch line velocity were in this latter category.

The power level-altitude-duration profile used in the studies conducted (including the off-design performance comparison of this appendix , as well) is presented in Table 3.

TABLE 3

POWER LEVEL-ALTITUDE-DURATION PROFILE

Power Level	Ambient Pressure, psia	Duration, min
Maximum, 100 percent	14.7	9
Maximum, 100 percent	0	9
Mode, 22 percent	10	144
Idle, 7 percent	0	18

Of the four operating conditions shown in Table 2, two are of special significance from the standpoint of the turbine design. They are

> Maximum Power (100 Percent) at Sea Level--This operating condition establishes the APU power rating. In addition, because of the high propellant consumption rate at this condition, it must be considered as a possible design point even though the operating duration is relatively short.

Mode Power (22 Percent) at Altitude (10 psia)--Operation at this condition consumes the majority of the total integrated mission propellant requirement because of the long duration. Therefore, this operating condition must be considered a possible design point.

The operating condition at which the turbine should be designed is not clear-cut. In fact, changing the relative durations of the above two operating conditions can change the design point from one condition to the other. In the studies conducted it was always assumed that one or the other of these two operating conditions was the design point. However, it is not inconceivable that the total integrated mission propellant requirement could be a minimum with a design between that obtained at each operating condition.

Table 4 presents a list of the turbine design variables investigated for the pressure-compounded, multistage, multiple-disk, axial impulse type turbine. The range covered for each variable and the value selected for the final turbine design are also listed in the table. The values selected were based on the following:

Pitch line velocity and inlet temperature determined by stress considerations

Design power level, inlet pressure, and discharge pressure determined on a system level for minimum propellant consumption

O/F ratio determined by cycle heat balance

Other parameters optimized on an SPC basis considering various limitations

In general, the data presented are indicative of turbine design point performance. Consequently, they are applicable to both pressure-modulated and pulse-modulated turbine designs (this is true of Figures 2 through 12, excepting Figure 3, which shows off-design data assuming pressure modulation).

Number of Stages

As shown by the representative results presented in Figure 2, there is a substantial performance incentive associated with the use of a multistage turbine. However, there is no incentive to use more than three stages. The decision between two- and three-stage turbines required an off-design point performance evaluation.

Figure 3 shows the off-design point performance at the various power operating conditions of interest for both two- and three-stage turbines. Based on this performance, it was estimated that the three-stage turbine would save approximately 6 lb in propellant weight. However, the three-stage

TABLE 4

TURBINE DESIGN VARIABLES INVESTIGATED
PRESSURE-COMPOUNDED, MULTISTAGE, MUTIPLE-DISK, AXIAL IMPULSE TURBINE

Variable	Range Covered	Value Selected
Number of stages	1 to 4	2
Rotational speed, rpm	50,000 to 90,000	70,000
Pitch line velocity, fps	1200 to 2200	2000
Inlet temperature, ^{0}R	2060 to 2660	2260
Inlet pressure (maximum), psia	35 to 2000	Depends on system configuration
Discharge pressure, psia	0 to 25	*
Power level (maximum/mode), hp	230/50 to 239/80	239/80
Chord width, in.	0.30 to 0.70	0.35
Blade height/diameter ratio	0.030 to 0.100	0.045
Stage pressure ratio split	1:1 to 5:1	3:1
Tip clearance, in.	0.005 to 0.020	0.010
Number of blades	75 to 130	85

*16 psia at sea level with the 239 hp power level and 11 psia at 10,000 ft with the 80 hp power level.

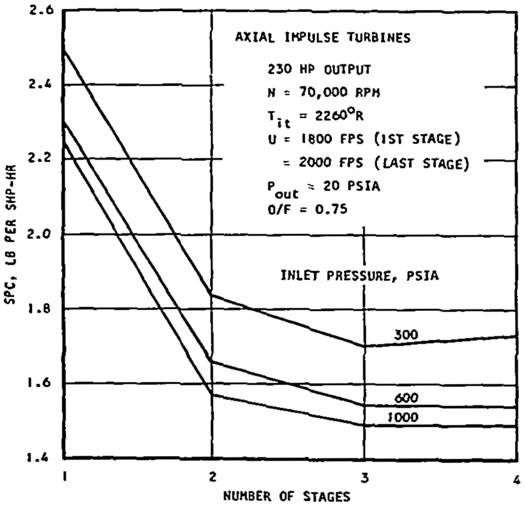

Figure 2. Effect of Number of Stages on Design Point Performance

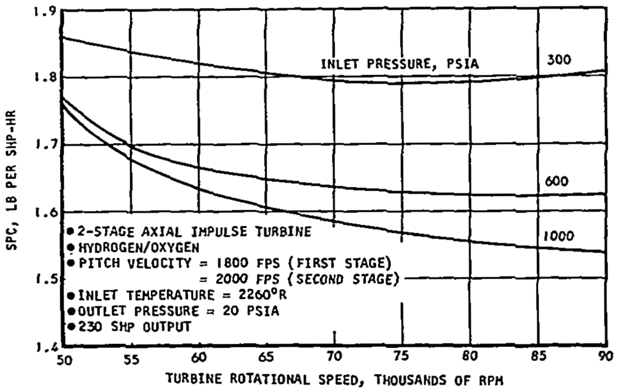

Figure 4. Typical Turbine Design Point Performance as a Function of Rotational Speed with Inlet Pressure as a Parameter

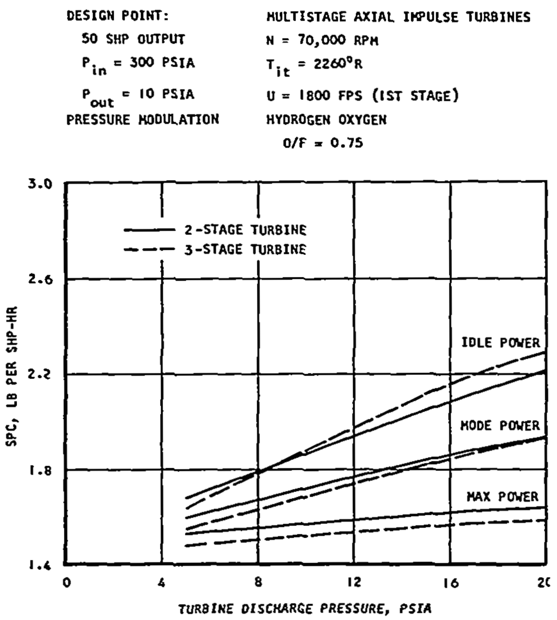

Figure 3. Comparison of Performance of 2- and 3-Stage Turbines at the Various Power Conditions

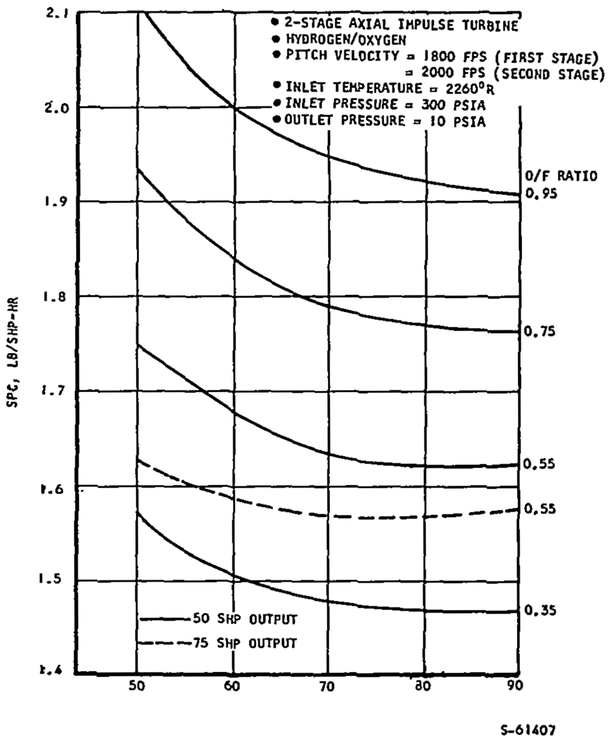

Figure 5. Typical Turbine Design Point Performance as a Function of Rotational Speed with O/F Ratio as a Parameter

turbine would outweigh the two-stage turbine by about 9 lb. This slight net weight disadvantage of the three-stage turbine coupled with the more complex design resulted in the selection of the two-stage turbine for the Space Shuttle APU application.

Shaft Speed

Typical two-stage turbine design point performance as a function of shaft speed is presented in Figures 4 and 5. Figure 4 has turbine inlet pressure as a parameter and is typical of performance at the maximum power at sea level design point. Figure 5 uses O/F ratio as a parameter and is typical of performance at the mode power at altitude design point. Later studies have shown that the O/F ratio at the mode power design point will be approximately 0.45. Based on the results of these figures, a turbine rotational speed of 70,000 rpm was selected as being near optimum.

Turbine Pitch Velocity and Inlet Temperature

These parameters are lumped together since they are linked with stress limitations for the turbine design. As shown by Figure 6, the optimum pitch velocity increases with inlet temperature. This is in opposition to limitations imposed by stress considerations. However, the energy available per unit flow rate of gas increases linearly with increasing turbine inlet temperature while the optimum pitch line velocity (and, hence, turbine efficiency) increases at a much slower rate with temperature. Therefore, the best solution is to maximize the turbine inlet temperature consistent with an adequately stressed turbine wheel design.

Based on this consideration and using the best high temperature materials available, a turbine inlet temperature as high as 2260^{o}R is felt to be usable in the Space Shuttle APU application. The pitch line velocity corresponding to this temperature is 2000 fps. A 2260^{o}R inlet temperature and 1800 fps pitch-line velocity on the first stage have been used in these parametric studies.

Turbine Back Pressure

The APU will operate over a range of altitude from essentially space vacuum to sea level. In general, better turbine performance will be obtained at reduced back pressure, although there are practical limits on the usable pressure ratio. Figure 7 shows the effect of turbine back pressure on APU performance. This tradeoff is important to the optimum sizing of the components and turbine discharge ducting.

As can be seen from the significant variation of SPC with discharge pressure in Figure 7, the operating altitude and exhaust duct pressure loss are important factors in turbine performance. The approximate tradeoff factor for optimization of the exhaust gas ducting and heat exchangers is 6.3 lb/psi of turbine back pressure.

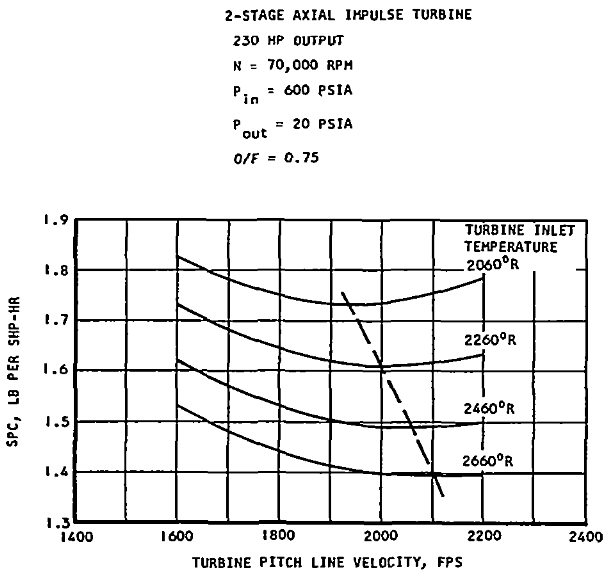

Figure 6. SPC Variation with Design Inlet Temperature and Pitch Velocity

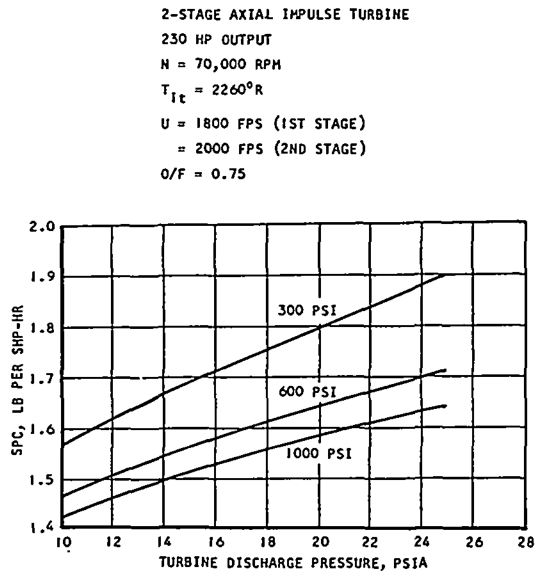

Figure 7. Effect of Turbine Discharge Pressure on Performance

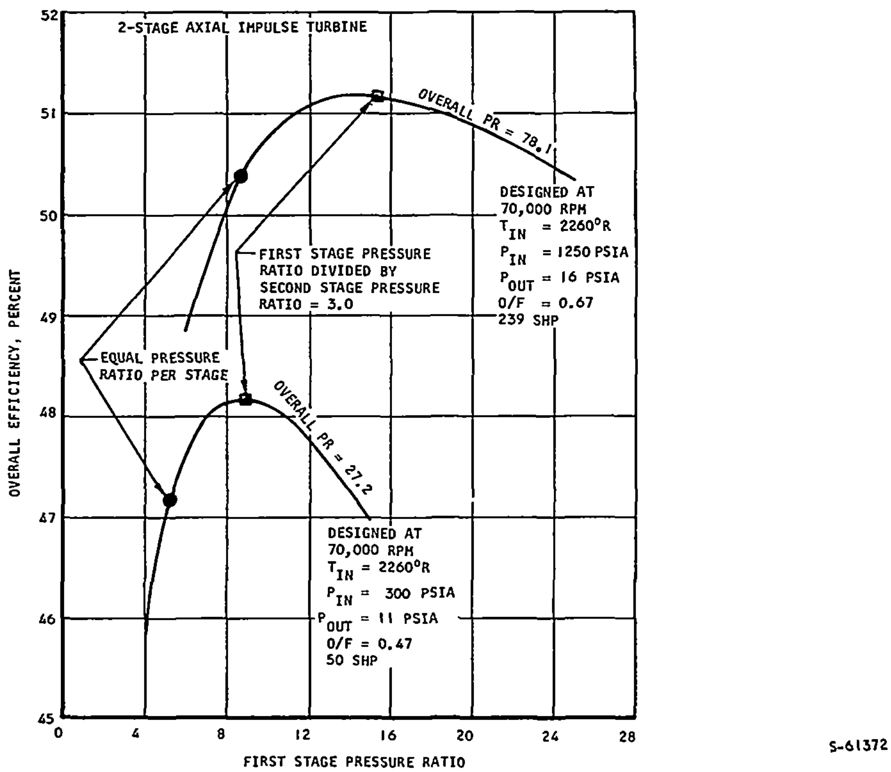

Figure 8. Effect of Pressure Ratio Split Between Stages on Efficiency

In the latest studies conducted, the back pressures shown below were used.

Power Level	Ambient Pressure, psia	Turbine Back Pressure, psia
Maximum, 100 percent	14.7	16
Maximum, 100 percent	0	4
Mode, 22 percent	10	11
Idle, 7 percent	0	1

These back pressures are consistent with good turbine performance and reasonably small weight penalties associated with the ducting and heat exchangers.

Blade Height and Pressure Ratio Split

Figure 8 shows the effect on turbine efficiency that the pressure ratio split between stages has at two different design points. The maximum efficiency occurs when the ratio of first stage pressure ratio to second-stage pressure ratio is three. This value has been used for all the turbine configurations in the off-design performance analysis studies conducted.

The effect of turbine blade height on efficiency is shown in Figure 9. Here the turbine efficiency is plotted as a function of the first stage blade height to pitch line diameter ratio (h/D) with first stage to second-stage pressure ratio as a parameter. There is an optimum h/D ratio for each pressure ratio split. Since the optimum pressure ratio split has been shown to be three in Figure 8, the optimum h/D is seen to be 0.045 from Figure 9. This value has been used in the various turbine designs studied.

Blade Chord Width

The variation of turbine efficiency with blade chord width is shown in Figure 10. The variation is small but does have an optimum value of approximately 0.35 in. This is the value used in the turbine studies conducted.

Generally, the efficiency drops as the chord gets larger because the rotor coefficient decreases. However, Reynold's number effects take over at small chords and the result is an optimum chord width. The recommended lower limit of chord width is shown in Figure 10 and is determined by the minimum acceptable chord Reynold's number of 20,000.

Rotor Tip Clearance

Figure 11 shows the effect of rotor tip clearance on turbine efficiency. The decreasing turbine efficiency with increasing tip clearance is due to leakage losses. The larger the tip clearance, the larger the leakage area. The minimum clearance that can be used is determined by the relative expansions

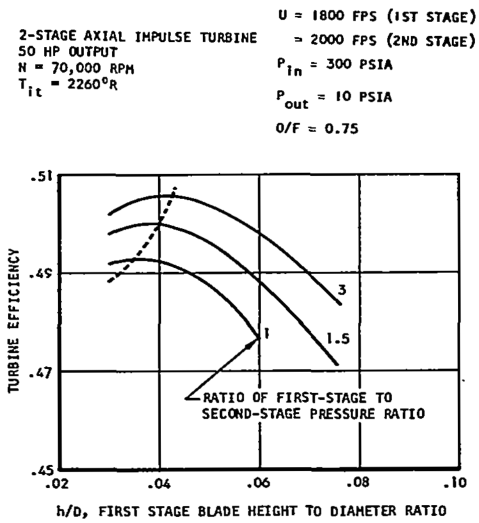

Figure 9. Effect of Blade Height on Efficiency

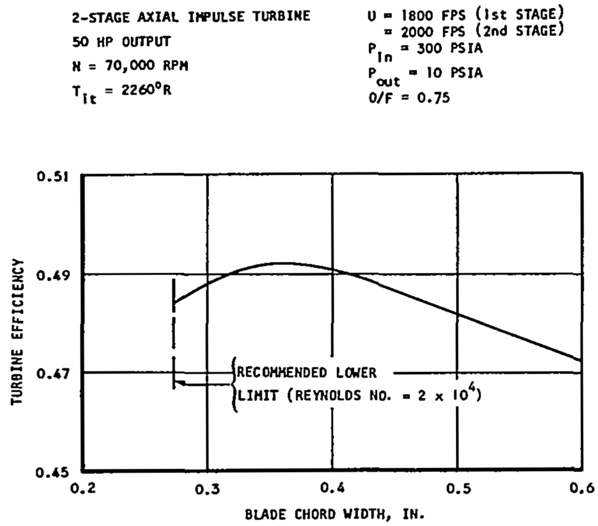

Figure 10. Effect of Blade Chord Width on Turbine Efficiency

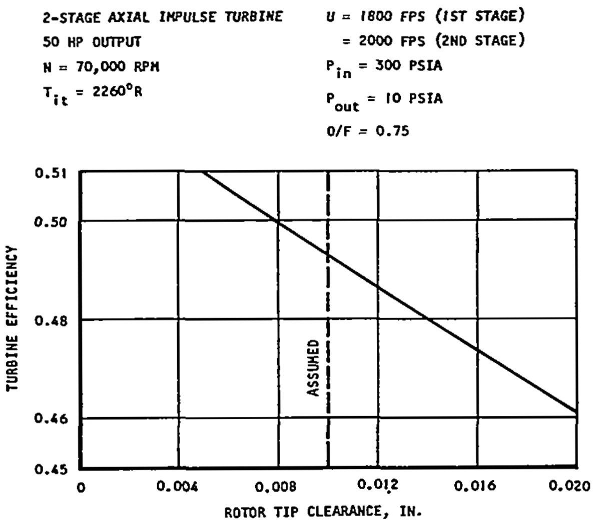

Figure 11. Effect of Rotor Tip Clearance on Turbine Efficiency

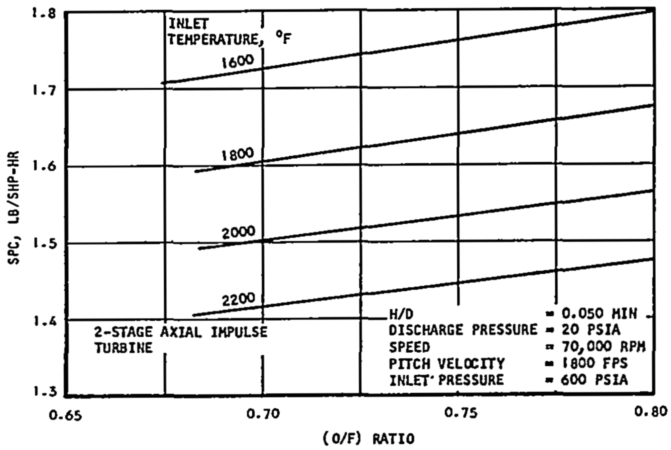

Figure 12. Variation of SPC with O/F Ratio with Inlet Temperature as a Parameter

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of the turbine rotor and turbine housing. Both the housing and rotor are subject to thermal expansion and, in addition, the rotor is subject to centrifugal expansion. The relative expansion under all operating conditions, both transient and steady state must be considered. A rotor tip clearance of 0.010 in. has been used in the studies conducted.

O/F Ratio

As shown by Figure 12, SPC will increase with O/F ratio for a given turbine inlet temperature. The O/F ratio will be determined by an energy balance for the cycle and will depend upon turbine inlet temperature, propellant inlet state, and the waste heat fed back into the cycle. By means of recuperation (using the turbine exhaust gas to preheat the propellants), the O/F ratio can be reduced for a given temperature. Using other waste heat (from the generator, gearbox, hydraulic fluid, combustor, turbine housing, etc.) for propellant preheating serves as an efficient means of disposing of waste heat while improving cycle performance.

PERFORMANCE STUDIES

The objective of this study was to determine the optimum turbine design for the space shuttle APU application using hydrogen and oxygen propellants. As stated previously, there were three basic types of turbines in contention for use in this application:

Pressure compounded, multistage, single-disk (reentry), axial impulse

Pressure-compounded, multistage, multiple-disk, axial impulse

Velocity-compounded, multistage, multiple-disk, axial

Also there were two possible design points for the turbine:

Maximum power (100 percent) at sea level

Mode power (22 percent) at altitude

Since it was not clear-cut which turbine type was superior or which operating condition was the design point, it was necessary to design turbines of all competitive types at each of the possible design points. These turbine designs were then evaluated on the basis of propellant consumed over a specified power level-altitude-duration profile. The results are presented in Table 4. The data assume that pressure modulation is used for turbine power/speed control.

The studies presented in this table were conducted at two different times and, as a result of updating, two different power profiles were used. The first four columns in Table 4 compare the various candidate pressure-compounded turbine types with a power profile as follows:

Maximum power	230 shp
Mode power	50 shp
Idle power	16 shp

A comparison of pressure-compounded and velocity-compounded turbines is made in the last four columns of Table 5. The power profile used here was:

Maximum power	239 shp
Mode power	80 shp
Idle power	50 shp

As can be seen in the comparison of the pressure-compounded turbines, the 3-stage, 3-disk turbine has the lowest propellant requirement. However, it will weigh approximately 9 lb more than the 2-stage, 2-disk turbine and hence, it has no advantage for the increased complexity involved. The 2-stage, 1-disk reentry turbine also results in a propellant consumption less than the 2-stage, 2-disk turbine but the small potential weight saving using a reentry turbine does not warrant the complexity and known problems involved. Therefore, the use of a ?-stage, 2-disk pressure-compounded turbine appears to be optimum for the power profile used. It also appears that the 2-stage, 2-disk turbine designed at the mode power point has a lower propellant requirement than the same turbine designed at the sea level, maximum power operating condition.

The last four columns of Table 5 show a comparison of pressure-compounded vs velocity-compounded turbines. All four turbines are of the 2-stage, 2-disk variety. Two are designed at the sea level maximum power point and two at the mode power point at altitude. Once more it is found that the sea level maximum power point results in significantly more propellant required than the mode power point.

A comparison of the pressure-compounded and velocity-compounded turbines shows that the pressure-compounded variety requires less propellant than the velocity-compounded for corresponding design points. Thus, on the basis of the comparisons made in Table 5, the 2-stage, 2-disk, pressure-compounded axial impulse turbine designed at the mode power conditions has been selected for use in the Space Shuttle APU application.

TURBINE INLET PRESSURE

Since it is desirable to find an optimum value of turbine inlet pressure, a series of turbine designs were obtained to determine the effect of inlet pressure and pressure ratio on the turbine performance. Figures 13 and 14 show the design point SPC variation with inlet pressure and pressure ratio, respectively. Curves are shown for designs at the mode power point and the maximum power operating condition at sea level. As can be seen, an optimum value is not obtained and it is concluded that the turbine design alone does not establish the optimum inlet pressure.

The optimum turbine inlet pressure is determined by the APU system and can only be obtained by evaluating the APU performance with turbines designed for operation at various pressure levels. Therefore, nine turbines have been designed and system performance obtained for each one (see Sections 3 through 5 of this report for details). The result of this study is that the optimum

TABLE 5

INTEGRATED MISSION PROPELLANT CONSUMPTION OF VARIOUS CANDIDATE TURBINES

		MAX. POWER = 230 SHP, MODE POWER = 50 SHP, IDLE POWER = 16 SHP				MAX. POWER = 239 SHP, MODE POWER = 80 SHP, IDLE POWER = 50 SHP			
	Type of axial impulse turbine	2-stage, 2-disk, pressure compounded	2-stage, 2-disk, pressure compounded	3-stage, 3-disk, pressure compounded	2-stage, 1-disk reentry, pressure compounded	2-stage, 2-disk, pressure compounded	2-stage, 2-disk, pressure compounded	2-stage, 2-disk, velocity compounded	2-stage, 2-disk, velocity compounded
	Design point	230 shp at sea level	50 shp at altitude	50 shp at altitude	50 shp at altitude	239 shp at sea level	80 shp at altitude	239 shp at sea level	80 shp at altitude
	Design pressures, psia — inlet	600	300	300	300	1250	465	1250	465
	Design pressures, psia — outlet	20	10	10	10	16	11	16	11
Operating Conditions and Duration	Max. power at sea level, 9 min.	SPC = 1.55 W = 53.5	SPC = 1.57 W = 54.2	SPC = 1.51 W = 52.1	SPC = 1.55 W = 53.5	SPC = 1.47 W = 52.6	SPC = 1.46 W = 52.3	SPC = 1.59 W = 57.0	SPC = 1.46 W = 52.4
Operating Conditions and Duration	Max. power at altitude 9 min.	SPC = 1.39 W = 48.0	SPC = 1.47 W = 50.8	SPC 1.43 W = 49.4	SPC = 1.50 W = 51.8	SPC = 1.29 W = 46.2	SPC = 1.35 W = 48.5	SPC = 1.32 W = 47.2	SPC = 1.32 W = 47.2
Operating Conditions and Duration	Mode power at altitude 144 min.	SPC = 1.75 W = 210.0	SPC = 1.59 W = 190.9	SPC = 1.57 W = 188.5	SPC = 1.55 W = 186.0	SPC = 1.57 W = 301.7	SPC = 1.48 W = 284.1	SPC = 1.84 W = 352.5	SPC = 1.52 W = 291.5
Operating Conditions and Duration	Idle power at altitude, 18 min.	SPC ≈ 1.53 W = 7.4	SPC = 1.71 W = 8.2	SPC = 1.70 W = 8.2	SPC = 1.63 W = 7.8	SPC = 1.16 W = 17.4	SPC = 1.23 W = 18.4	SPC = 1.22 W = 18.3	SPC = 1.21 W = 18.2
	Total propellant consumed, lb	W = 318.9	W = 304.1	W = 298.2	W = 299.1	W = 417.9	W = 403.3	W = 475.0	W = 409.3
	Integrated SPC lb/shp-hr	1.647	1.57	1.538	1.544	1.50	1.445	1.703	1.468

SPC = Specific propellant consumption, lb per shp-hr.
W = Propellant consumed, lb

NOTE: The results presented above are based on an O/F ratio of 0.67 at the two max. power conditions, and O/F ratio of 0.45 at the mode power condition, and an O/F ratio of 0.40 at the idle power condition.

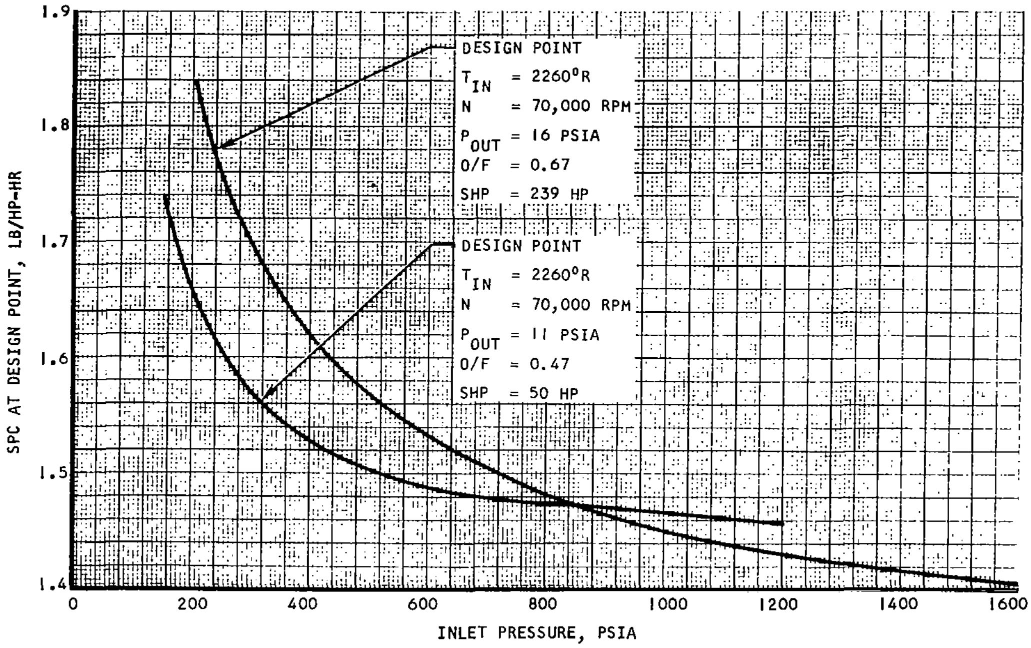

Figure 13. SPC as a Function of Turbine Inlet Pressure

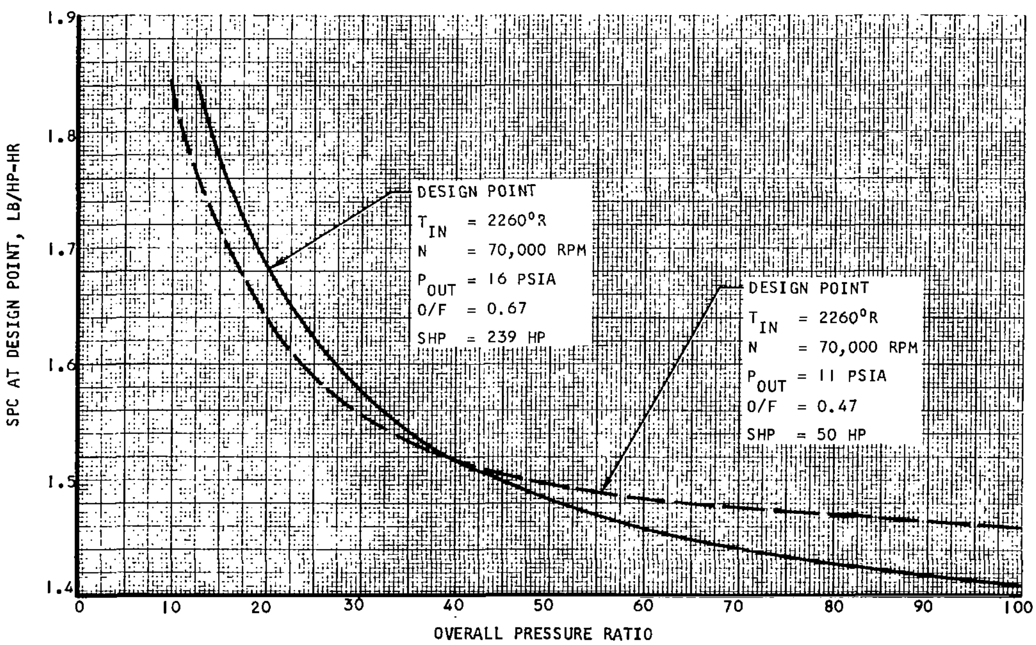

Figure 14. SPC as a Function of Turbine Pressure Ratio

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turbine is designed at the mode power point with the inlet pressure at the sea level maximum power point being 600 psia for the hydrogen-oxygen systems with high pressure tanks or low pressure tanks with pumps. Clearly, if the APU is supplied with high pressure gas from another vehicle system, there is no optimum pressure; the higher the pressure, the better the performance.

Figures 15 through 23 present the performance maps of the nine turbines evaluated in the system performance program. All nine turbines are designed at 70,000 rpm and an inlet temperature of 2260°R. The other design conditions are as follows:

Figure No.	Power		Pressures, psia			O/F Ratio
	Level	Value, hp	Inlet	Outlet	Maximum	
15	Maximum	239	1500	16	1500	0.67
16	Maximum	239	1250	16	1250	0.67
17	Maximum	239	900	16	900	0.67
18	Maximum	239	600	16	600	0.67
19	Maximum	239	280	16	280	0.67
20	Mode	80	540	11	1500	0.45
21	Mode	80	464	11	1250	0.45
22	Mode	80	330	11	900	0.45
23	Mode	80	220	11	600	0.45

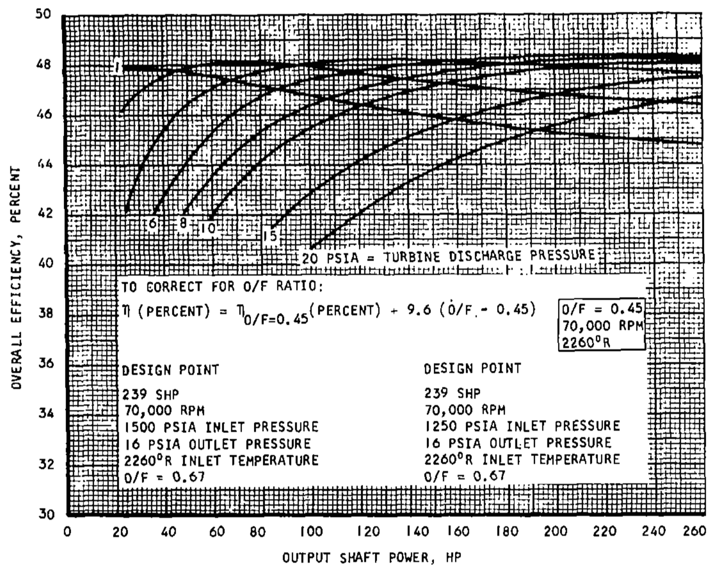

Figure 15. Turbine Performance Map

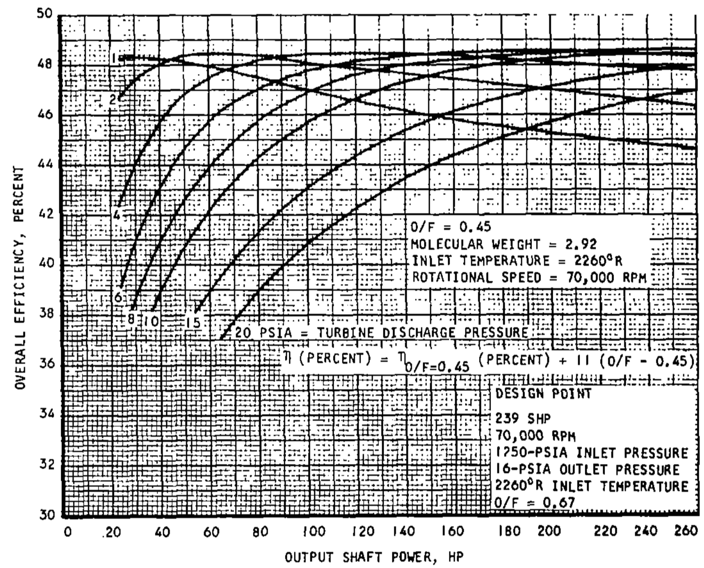

Figure 16. Turbine Performance Map

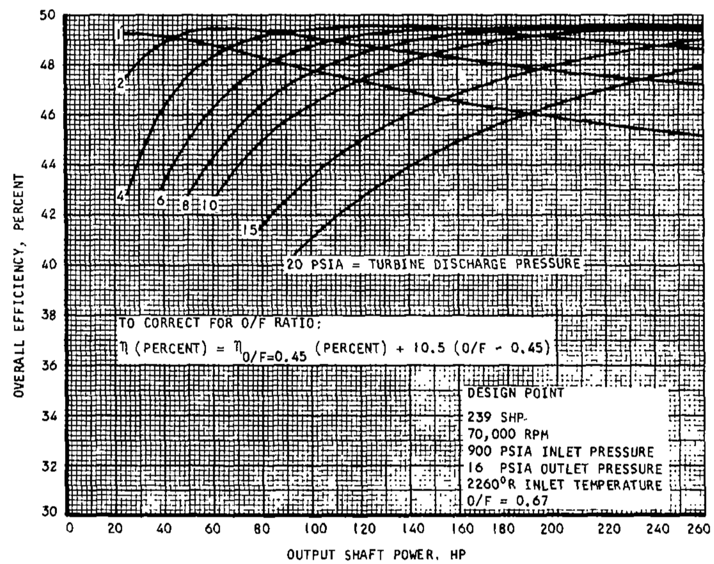

Figure 17. Turbine Performance Map

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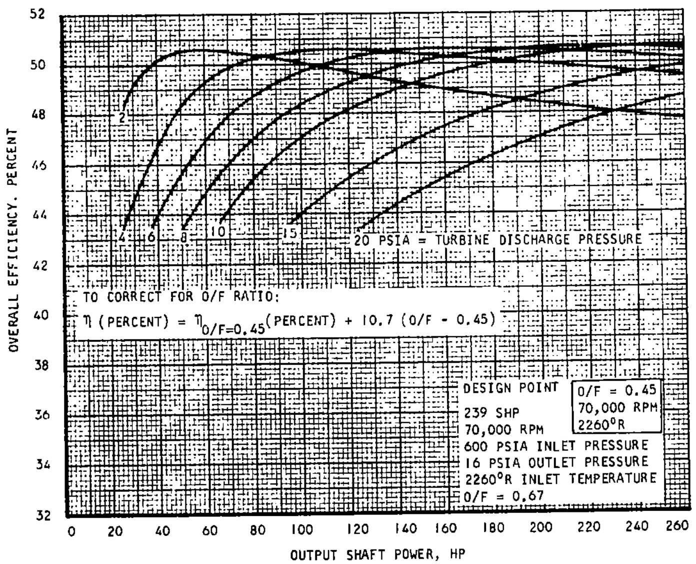

Figure 18. Turbine Performance Map

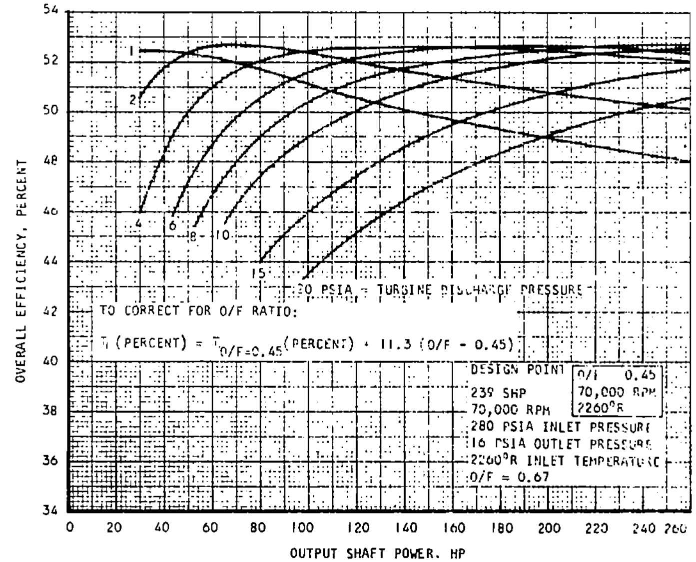

Figure 19. Turbine Performance Map

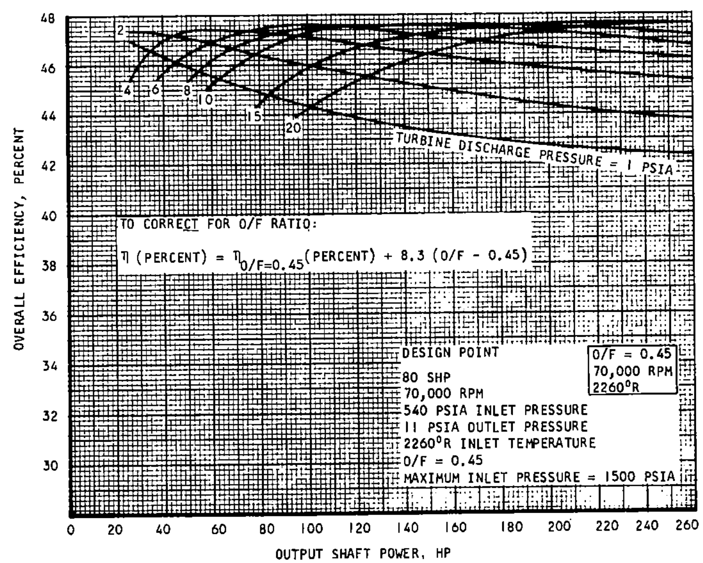

Figure 20. Turbine Performance Map

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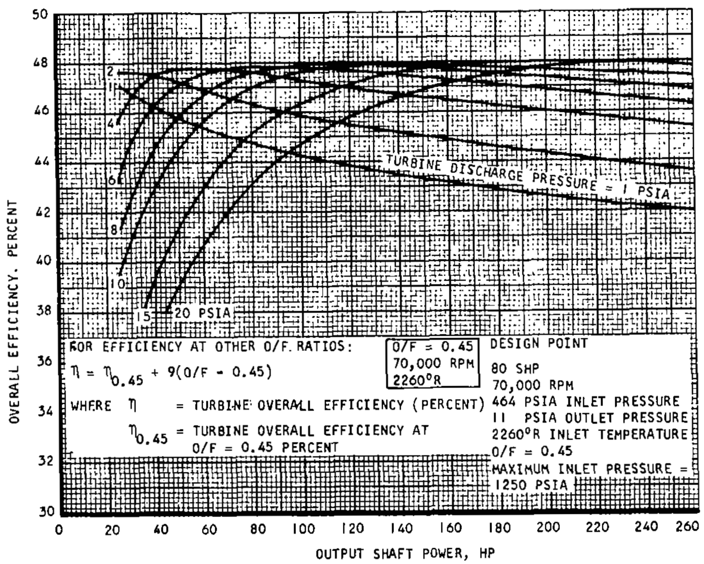

Figure 21. Turbine Performance Map

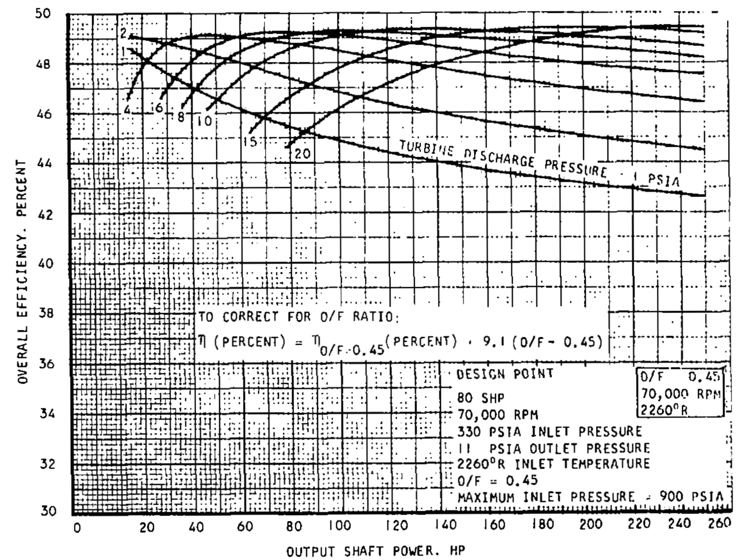

Figure 22. Turbine Performance Map

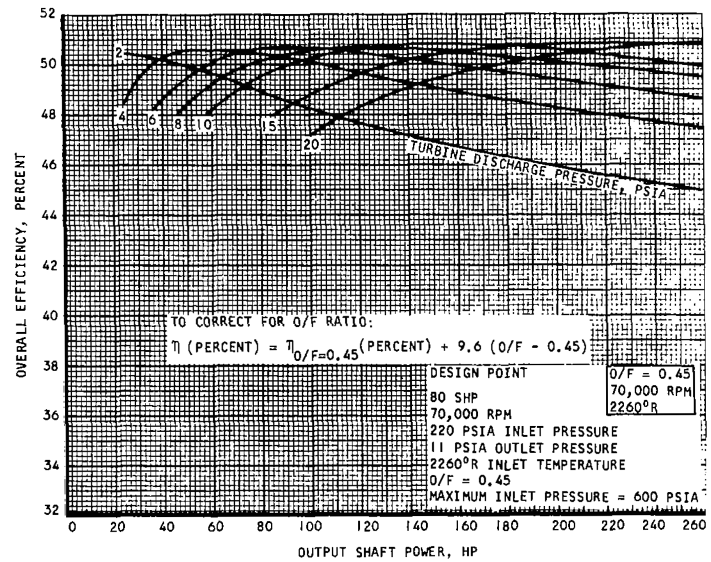

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Figure 23. Turbine Performance Map

APPENDIX B

TURBINE MECHANICAL DESIGN

INTRODUCTION

Although most of the Phase I work was oriented towards establishing the effect of various component design parameters (such as rotational speed, number of turbine blades, etc.) on the system performance, it is also necessary to perform sufficient mechanical design to ensure the feasibility of obtaining the desired component design parameters. This appendix presents the mechanical design that has been accomplished.

The mechanical design activities have been concentrated on the turbine rotating assembly and on the overall turbine power unit configuration. In support of these activities, it has been necessary to establish design criteria in such areas as overspeed criteria, material allowable stresses, and turbine containment requirements. Thus, the appendix is divided into the following major topics:

- Design criteria
- Rotating assembly design
- Turbine power unit configuration

DESIGN CRITERIA

The primary design criteria of interest in Phase I are those effecting the turbine performance potential. In particular, these are:

Overspeed Criterion--The centrifugal stresses on the turbine disks are proportional to the square of the rotational speed. Selection of a high burst speed will necessitate lowering the allowable stress of the disk material at the normal operating conditions.

Disk Growth--The turbine disk will expand due to the action of the centrifugal forces on the disk. Most of this expansion will be elastic so that the disk will return to the same shape at zero speed. However, the high temperature portion of the disk will be subject to a plastic-like phenomena called creep in which a gradual expansion with time occurs.

Low Cycle Fatigue--Even at relatively low material stresses, there is some hysteresis in the material stress-strain curve which will tend to cumulatively act to produce a fatigue effect.

Maximum Allowable Stress--Independently of the above considerations, good design practice dictates that some safety factor, or confidence factor be applied to the tested values of the material yield and ultimate stresses.

Overspeed/Containment Criteria

Table I shows the overspeed and containment criteria used for rotating machinery in various aircraft and space vehicle applications. Based on these data, the recommended overspeed criterion for the Space Shuttle APU is 130 percent of the nominal speed. At this speed, the turbine housing should be capable of containing the turbine disks should a disk burst occur. For the APU, the nominal speed, based on aerodynamic design considerations (described in Appendix A), is 70,000 rpm for optimum performance. Allowing up to a 5 percent variation in turbine speed (although the final turbine controls may provide closer speed regulation), the design speed is 73,500 rpm. This is the maximum turbine rotational speed expected under any normal operation, and, applying the overspeed criterion, the burst speed becomes 91,000 rpm.

TABLE I

OVERSPEED/CONTAINMENT CRITERIA

Type of Application	Design Overspeed	Containment	Speed Control	Overspeed Trip
Aircraft gas turbine APU's	125 to 135 percent	110 percent	Yes	Yes (110 percent)
Aircraft cooling turbines	135 to 150 percent	135 to 150 percent	No	No
Missile APU's	150 percent	None	Yes	No
Closed Brayton cycle power turbine	150 percent	None	Yes	No
Recommended Space Shuttle APU turbine	130 percent of Nominal design	130 percent of Nominal design	Yes	Yes

Candidate Disk Material Properties

There are a number of candidate materials for the turbine disks. However, based on the yield strength, ductility, and fatigue strength data presented in Figures 1, 2, 3, and 4, the most likely candidate material is Udimet 700. Although the manufacturer's published yield strength data shown in Figure 1 indicate that IN-100 has a slightly higher yield strength than Udimet 700, AiResearch testing of typical material samples (Figure 2) indicates that the obtainable strength in cast IN-100 specimens is considerably less than the manufacturer's quoted values. Most probably this difference is strongly dependent upon the specimen shape and the details of the casting process. Additionally, Udimet 700 has excellent ductility, allowing it to absorb the high local temperature gradients occurring on startup.

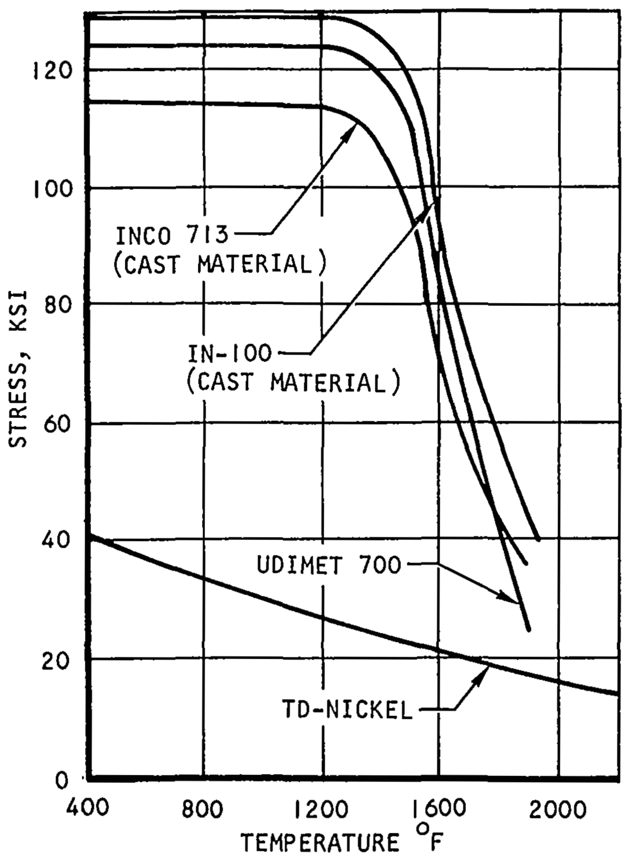

Figure 1. Typical Manufacturers' Quoted 0.2 Percent Yield Strengths

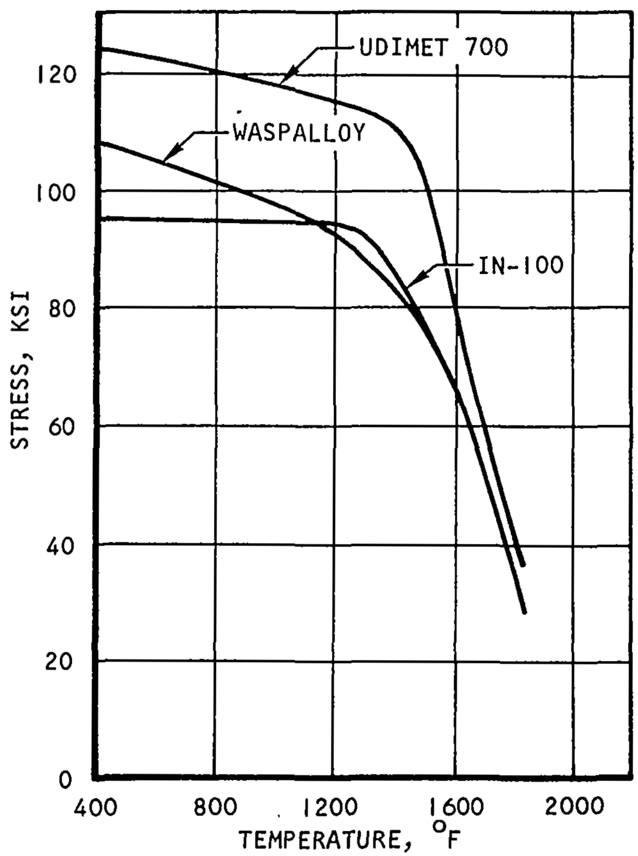

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Figure 2. 0.2 Percent Yield Strength Design Minimum based on AiResearch Test Data

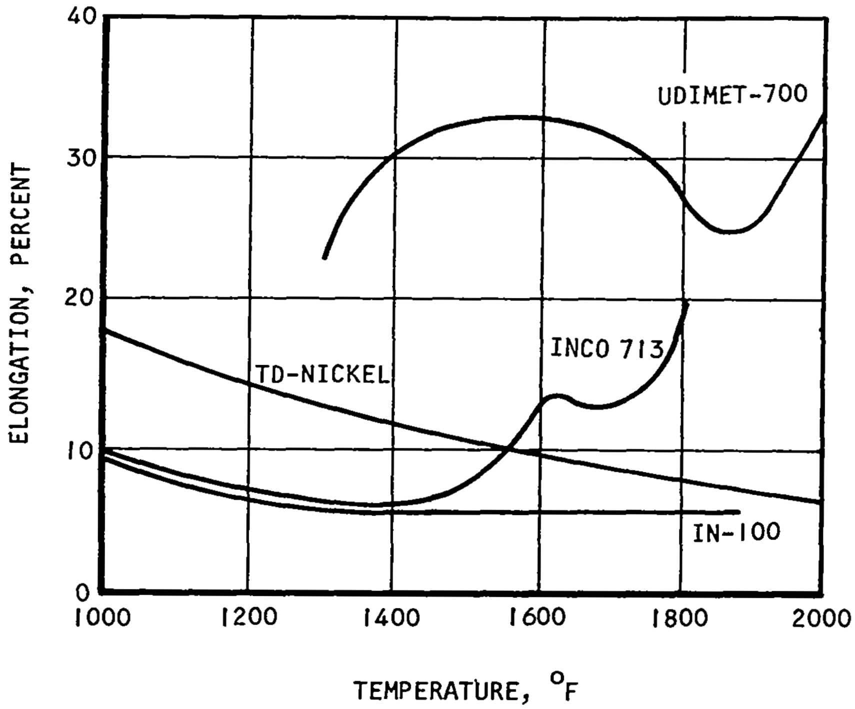

Figure 3. Candidate Disk Material Ductility

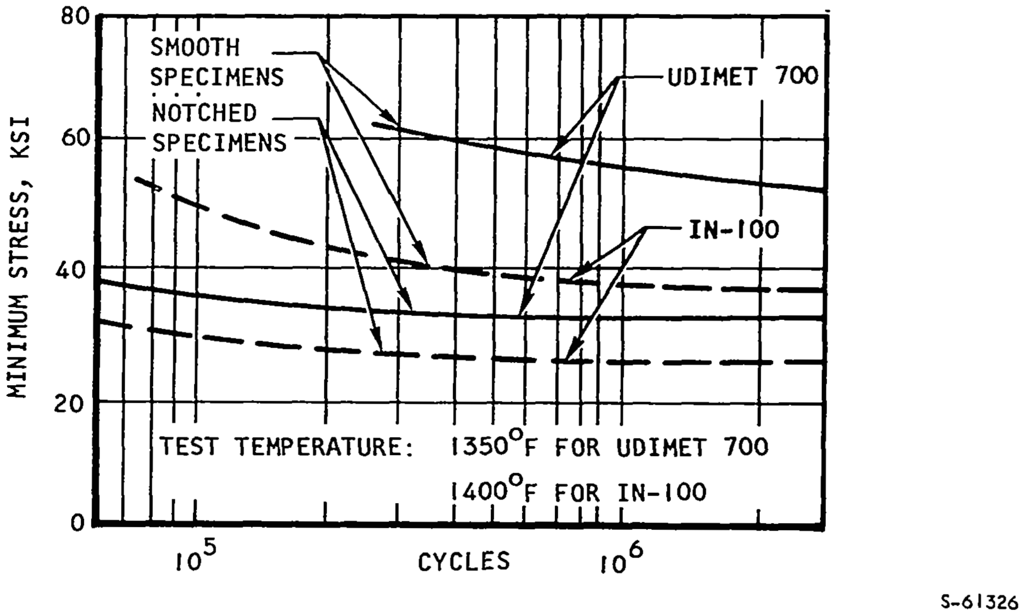

Figure 4. Candidate Disk Material Fatigue Strength

The major disadvantage of Udimet 700 is its high cost and its poor manufacturability. In these respects, IN-100 appears superior. Thus, for turbine wheels in which the rim is slotted so that the temperature gradients do not cause high hoop stresses in the rim, an IN-100 wheel would be desirable. Final selection of the turbine wheel material will not be made until Phase II of the study is initiated.

Using Udimet 700 as the baseline material, it is possible to apply the selected design criteria (shown at the top of Table 2) to determine the allowable stress at the design speed of 73,500 rpm. Table 2 shows this process, indicating that the governing criterion for the allowable stress is the low cycle fatigue requirement of 1500 cycles of startup/shutdown transients. Figure 5 shows the Soderberg diagram used to establish the low cycle fatigue allowable stress. This diagram is based on the Udimet 700 stress-strain curve shown in Figure 6. The data are for a temperature of 1000°F, which is approximately the maximum temperature occurring at the neck of the turbine disk. Thus, the maximum stress in the turbine disk neck portion (at 1000°F) should be limited to 80,000 psi at the design speed. Although the turbine rim is hotter, it is much more lightly stressed. Similarly, because the center portion of the disk operates at temperatures well below 1000°F, higher stresses are allowable there. Figure 7 shows the variation in second-stage weight with the allowable stress. The second-stage is the heavier of the two turbine stages since it operates at a higher pitch line velocity (2000 fps, as compared to 1800 fps for the first stage). The data indicate a substantial weight penalty occurs for low-allowable stresses. They also show that the weights of slotted and solid disks are equivalent.

Turbine Blading Design

Because most of the turbine disk operates at temperatures well below those at which significant creep occurs, only the blading creep need be established. That portion of disk growth due to expansion of the disk under the centrifugal loads can be accommodated in the design of the turbine casing. However, the gradual growth of the blades with time then becomes the determining factor on the clearance between the blade tip and the casing.

The blading height for pressure-compounded turbines is on the order of 0.25 in. for optimum designs, and the blade tip-casing clearance obtainable in turbines of this size and type is about 0.010 in. Thus, the maximum possible blading growth is about 4 percent. However, the creep data shown in Figure 8 indicate that at a temperature of 1960°R (about 100°R above the maximum blading temperature), the blading will have 1 percent creep growth in 1000 hr when operated at a stress of 35,000 psi. Studies of the blading, assuming a tapered blade section (accomplished by placing a tapered hollow in the blade center as shown in Figure 9), indicate that the blade root stress will be about 40,000 psi. This stress rapidly decreases along the blade, becoming zero at the blade tip. Therefore, the integrated creep along the blade is less than 0.2 percent. Thus, because of its short blading, the pressure-compounded turbine will be well within the disk growth creep criterion of 0.1 percent overall (equivalent to about 1.5 percent allowable creep on the blading alone).

TABLE 2

UDIMET 700 TURBINE DISK ALLOWABLE STRESS

DISK ALLOWABLE STRESS AT DESIGN SPEED AND MAXIMUM TEMPERATURES SELECTED AS THE LOWEST OF THE FOLLOWING:

- 90% OF THE 0.2% YIELD STRESS (APPLICABLE TO COMBINED CENTRIFUGAL AND THERMAL STRESSES)
- (80% OF THE ULTIMATE STRESS) $\times \left(\frac{\text{DESIGN SPEED}}{\text{BURST SPEED}}\right)^2$ (APPLICABLE ONLY TO THE CENTRIFUGAL STRESSES)
- STRESS AS DICTATED BY 1500 CYCLES FROM ZERO TO DESIGN SPEED AND BACK TO ZERO WITH FULL TRANSIENT TEMPERATURE GRADIENTS
- STRESS AS DICTATED BY 0.1% ALLOWABLE DISK GROWTH DUE TO EXPANSION FROM CENTRIFUGAL FORCES AT DESIGN SPEED

AS APPLIED TO A UDIMET 700 DISK OPERATING AT 73,500 RPM DESIGN AND 91,000 RPM BURST WITH MAXIMUM DISK TEMPERATURE = 1000°F (EXCLUDING BLADES AND RIM, BOTH OF WHICH ARE ONLY LIGHTLY STRESSED):

- 90% OF THE 0.2% YIELD = 106,000 PSI
- (80% ULTIMATE) $\times \left(\frac{\text{DESIGN SPEED}}{\text{BURST SPEED}}\right)^2$ = 83,500 PSI (APPLICABLE ONLY FOR CENTRIFUGAL STRESSES)
- LOW CYCLE FATIGUE STRESS = 80,000 PSI
- ALLOWABLE DISK GROWTH STRESS = 110,000 PSI (ASSUMING CONSTANT-STRESS DISK)

CONCLUSION: FOR UDIMET 700 DISK THE ALLOWABLE STRESS SHOULD BE 80,000 PSI

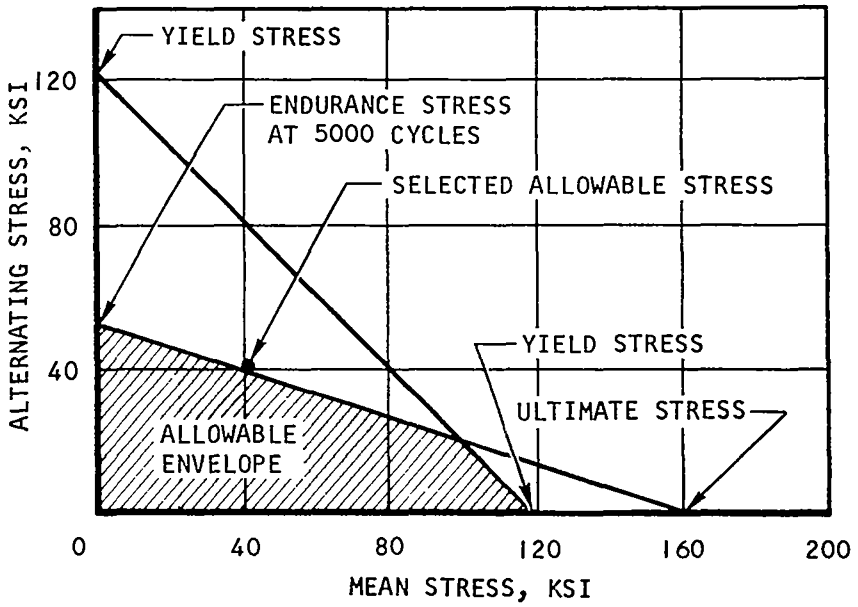

Figure 5. Udimet 700 Soderberg Diagram

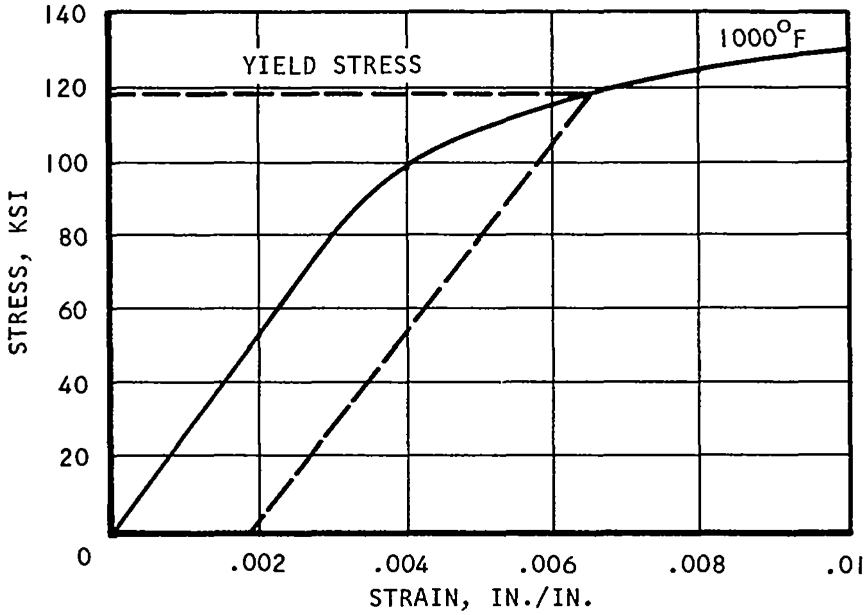

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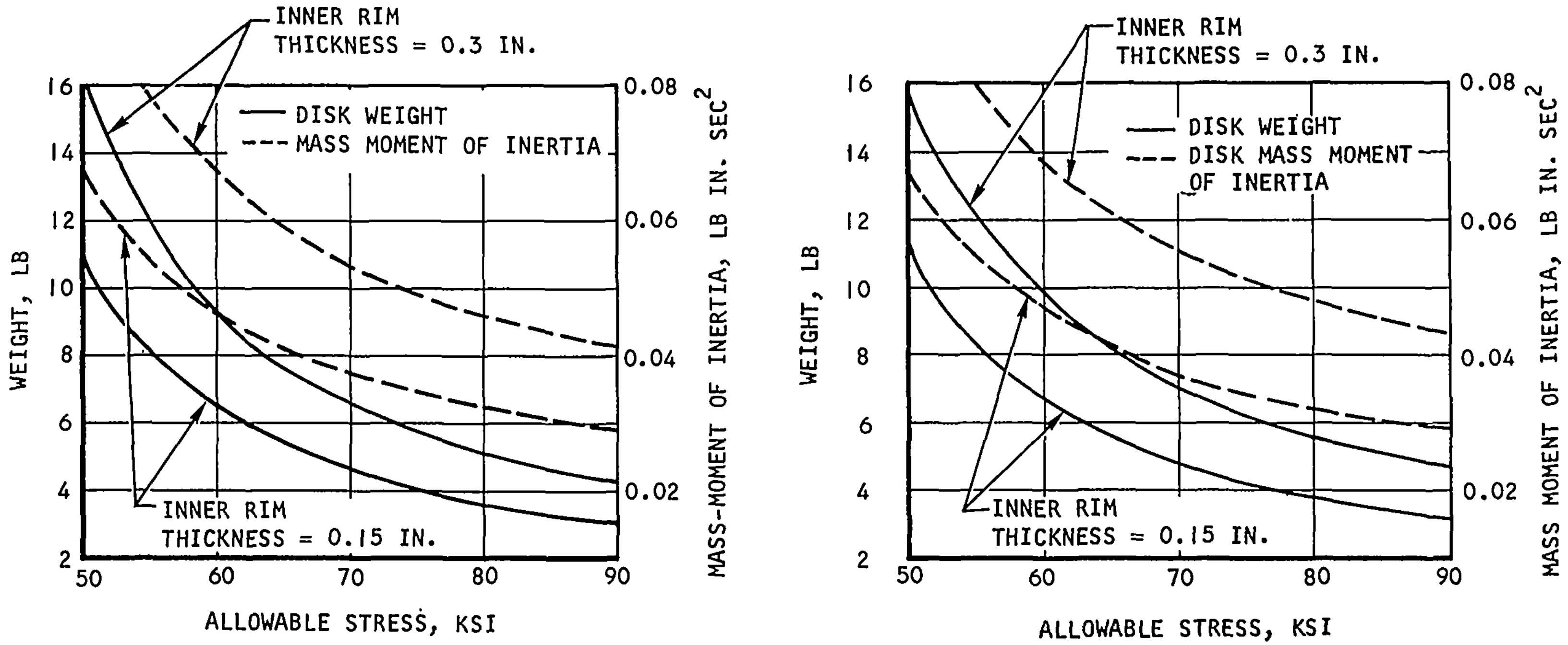
Figure 6. Udimet 700 Stress-Strain Curve at 1000°F



SOLID DISK

SLOTTED DISK

DISK OF UDIMET 700 FOR 2000 FPS PITCH LINE VELOCITY,
70,000 RPM, 85 BLADES, 0.35 IN. RIM TOP WIDTH

DISK O.D. = 6.9 IN.

RIM O.D. = 6.18 IN.

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Figure 7. Second-Stage Disk Weight and Inertia vs Allowable Stress

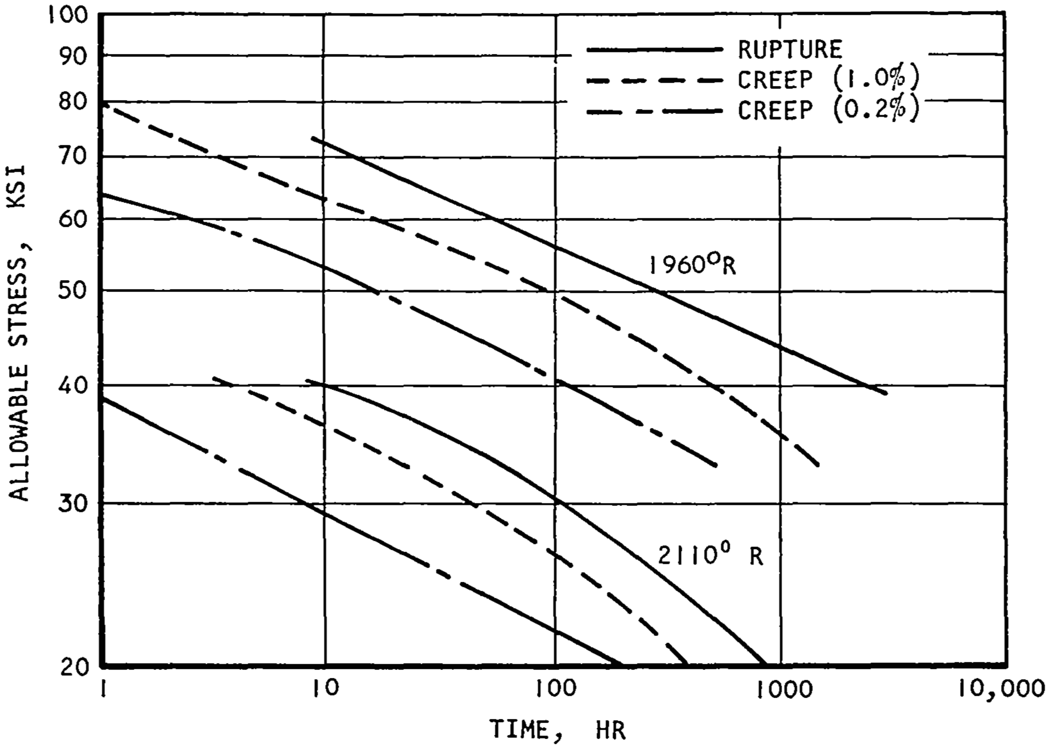


Figure 8. Udimet 700 Creep Allowable Stress

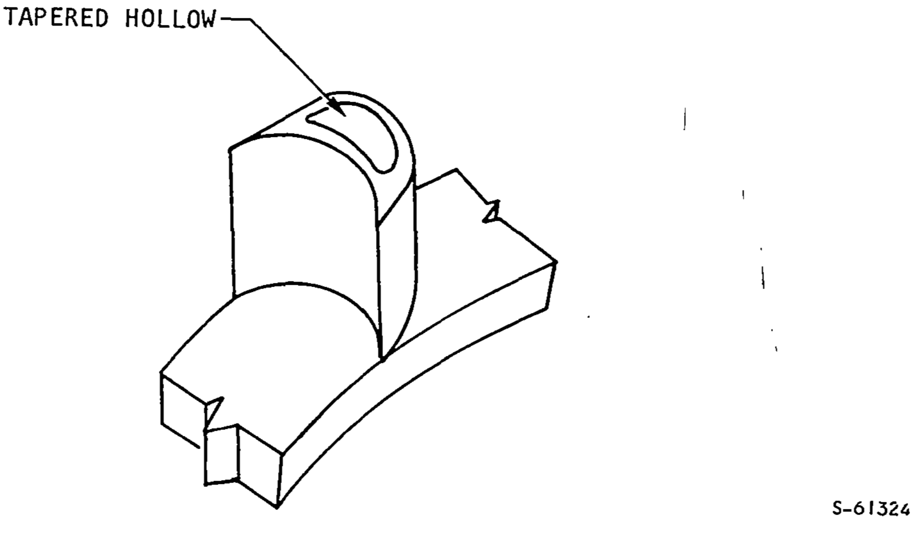


Figure 9. Tapered Blade Configuration

ROTATING ASSEMBLY DESIGN

The rotating assembly design has resulted in a preliminary layout with accompanying analyses necessary to establish design validity. The design process consisted of examinations of the turbine operating conditions (based on mechanical design considerations, as opposed to aerodynamic, or performance considerations), and the stage assembly/support concepts prior to initiation of the layout. These studies establish the overall mechanical design and the layout and then translate this into a final, detailed configuration. The discussion below parallels the design activity, presenting the materials as follows:

- Turbine operating conditions
- Stage assembly/support concepts
- Rotating assembly layout

Turbine Operating Conditions

The aerodynamic performance studies presented in Appendix A show that there is incentive to operate the turbine at a high speed with a high inlet temperature. Although there is a definite optimum for both the pitch-line velocity (occurring at about 2000 fps) and the rotational speed (occurring at about 70,000 rpm), performance continues to improve as the turbine inlet temperature is increased. Thus, the limitation on inlet temperature is established by the mechanical desigm of the rotating assembly. Additionally, since there is a strong intertie between the turbine inlet temperature and the pitch-line velocity (stress is proportional to the square of the pitch-line velocity), it is necessary to consider the parameter combination best meeting the mechanical design limitations while still offering good performance.

Thermal studies presented later in this appendix show that the neck portion of the turbine disk (having both high stresses and high temperatures) operates about 600^{0}R below the turbine inlet temperature. The material yield strength data presented in Figure 1 indicate that there is a rapid decrease in material strength for temperatures exceeding 1200^0 to 1400^{0}F. This would correspond to a turbine inlet temperature of 1800^0 to 2000^{0}F. Thus, 2000^{0}F can be taken as the maximum desirable turbine inlet temperature.

Similarly, investigation of the effect of pitch-line velocity indicates that increasing the pitch-line velocity from 1800 fps to 2000 fps causes about a 37 percent increase in the disk stresses for a fixed disk shape. Thus, there is a significant increase in the difficulty of mechanical design while only a slight increase in the performance (about 2.5 percent reduction in propellant consumption). Therefore, a pitch-line velocity of 1800 fps has been selected for use during the Phase I studies. Also, the turbine inlet temperature has been set at 1800^{0}F, although about 8 percent propellant consumption reduction could be obtained by using 2000^{0}F. During Phase II, further studies will be performed to establish the final selection of turbine inlet temperature and pitch-line velocity.

Stage Assembly/Support Concepts

Since the selected turbine uses only two stages, it is possible to support it as a cantilever. Thus, it is desirable to place the second stage on the inboard side of the cantilever since it operates at a lower temperature than the first stage and since it is heavier than the first stage (because the second stage operates at a higher pitch-line velocity).

Table 3 shows some candidate stage assembly concepts that have been considered. Of these concepts, the preferred configurations are either the electron-beam welded configuration or the off-centerline bolts configuration. Electron-beam assembly requires no special fastening parts, but does not facilitate single-stage replacement. Unfortunably, the primary turbine disk materials have a low degree of weldability; thus welding is not recommended. The off-centerline bolting concept will have a slightly higher weight but single-stage replacement is simple. Therefore the preferred assembly concept is bolting.

Rotating Assembly Layout

Building on the selected turbine operating conditions and the preferred stage assembly concepts, a rotating assembly layout has been made. This layout is shown in Figure 10. The turbine stages are cantilevered with the second stage inboard. Hydrogen is used to cool the turbine disks--accomplished by flowing hydrogen gas through cooling passages in the turbine case. A combination of radiation and conduction in the clearance between the casing and the disk ensures effective heat transfer. The bearings are lubricated and cooled by oil flowing from jets in the turbine housing. An alternate method of bearing cooling combines the jet cooling with cooling obtained by passing oil through the center of the rotating shaft and out across the surface of the inner race. This is shown in the layout. However, thermal analyses presented in this appendix show that adequate cooling is provided by the jet oil flow alone.

An alternate method of transferring torque from the turbine support shaft to a high speed pinion is shown in Figure 11. This method, although it requires additional bearings, is preferred over that shown in Figure 10 since it reduces the machining operations on the turbine shaft and allows incorporation of a shear section to eliminate damage to the gearbox and/or turbine in event of overtorquing or sudden jam-up.

To ensure the practicability of the proposed design, the following supporting analyses have been performed:

Thermal analysis

Stress analysis (first stage disk only)

Critical speed analysis

Bearing operating condition analysis

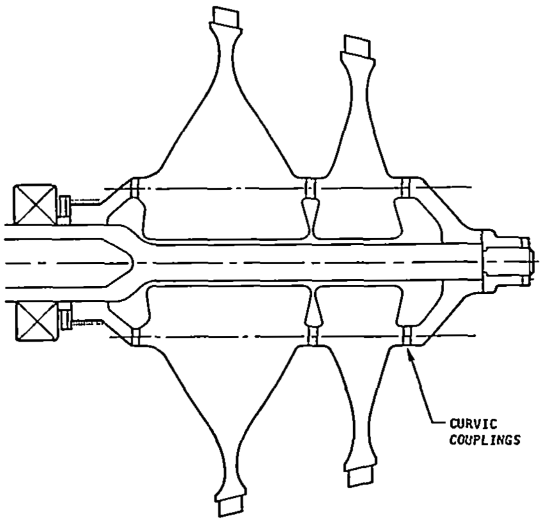
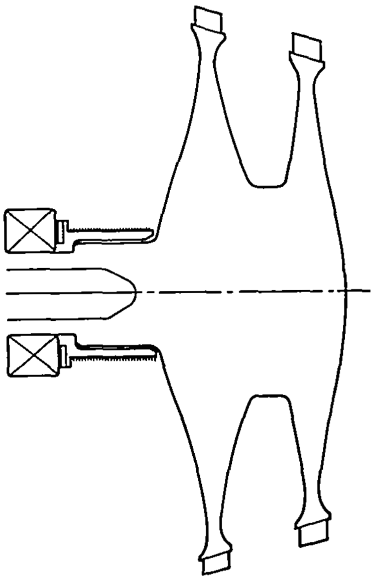
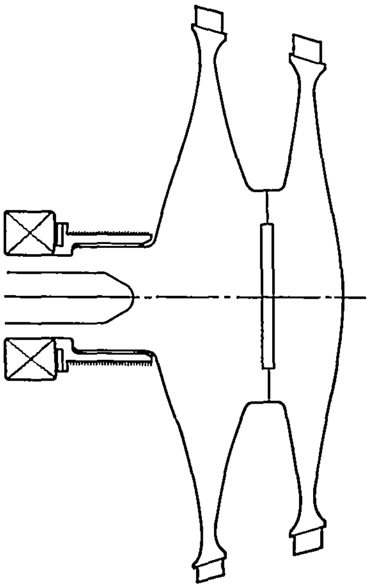
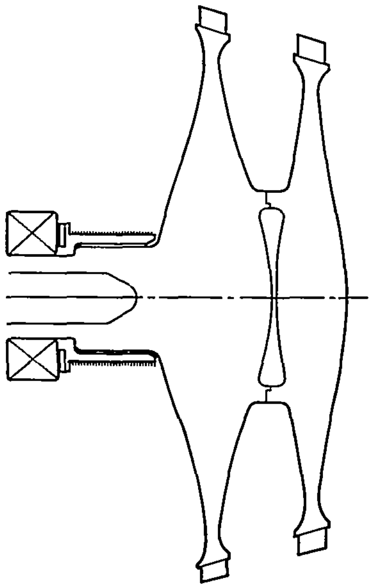
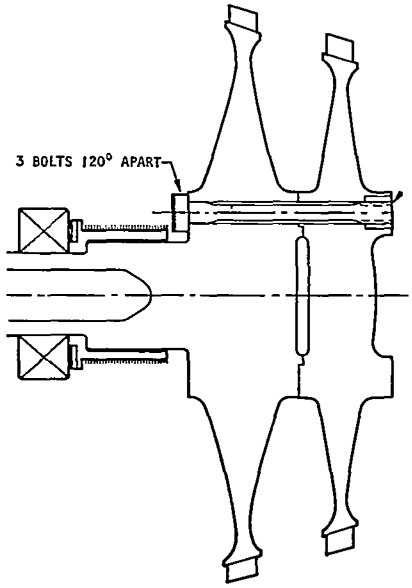
THROUGH-BOLTS ON CENTERLINE	SINGLE PIECE
	
• DIFFICULT TO MACHINE CURVIC COUPLINGS IN UDIMET 700 • DISK THICKNESS GREATLY INCREASED IN ORDER TO OFFSET STRESS CONCENTRATION AT HOLE • INCREASED DISK WEIGHT AND INERTIA • LOW HEAT TRANSFER BETWEEN STAGES	• MINIMUM WEIGHT CONFIGURATION • CAN NOT OBTAIN IDEAL MATERIAL GRAIN FLOW IN BOTH STAGES • HIGH COST DUE TO PROBABLE HIGH REJECTION RATE (BECAUSE OF EXTENSIVE MACHINING) • HIGH HEAT TRANSFER BETWEEN STAGES

TABLE 3

CONCEPT COMPARISON

INERTIA WELDING	ELECTRON - BEAM WELDING	OFF-CENTERLINE BOLTS
		
• DISKS MUST BE IN ROUGH FORM WITH LARGE DIAMETERS FOR CLAMPING CHUCKS - LATER FINISHING MUST BE DONE ON ENTIRE ASSEMBLY • MATERIAL FLOW IS UNEVEN AND EXACT POSITIONING IS NOT POSSIBLE • RELATIVELY LOW HEAT TRANSFER BETWEEN STAGES	• DISKS CAN HAVE ALL MACHINING EXCEPT BLADES DONE PRIOR TO WELDING • PRECISE POSITIONING IS POSSIBLE • DISK SHAPE FACILITATES ACCESS TO WELD SURFACE FOR FINISH GRINDING • SINGLE-STAGE REPLACEMENT IS DIFFICULT • RELATIVELY LOW HEAT TRANSFER BETWEEN STAGES	• REQUIRES REINFORCING PADS ON TURBINE DISKS • SIMPLE ASSEMBLY • REPLACEMENT OF A SINGLE STAGE IS EASY • LOW HEAT TRANSFER BETWEEN STAGES

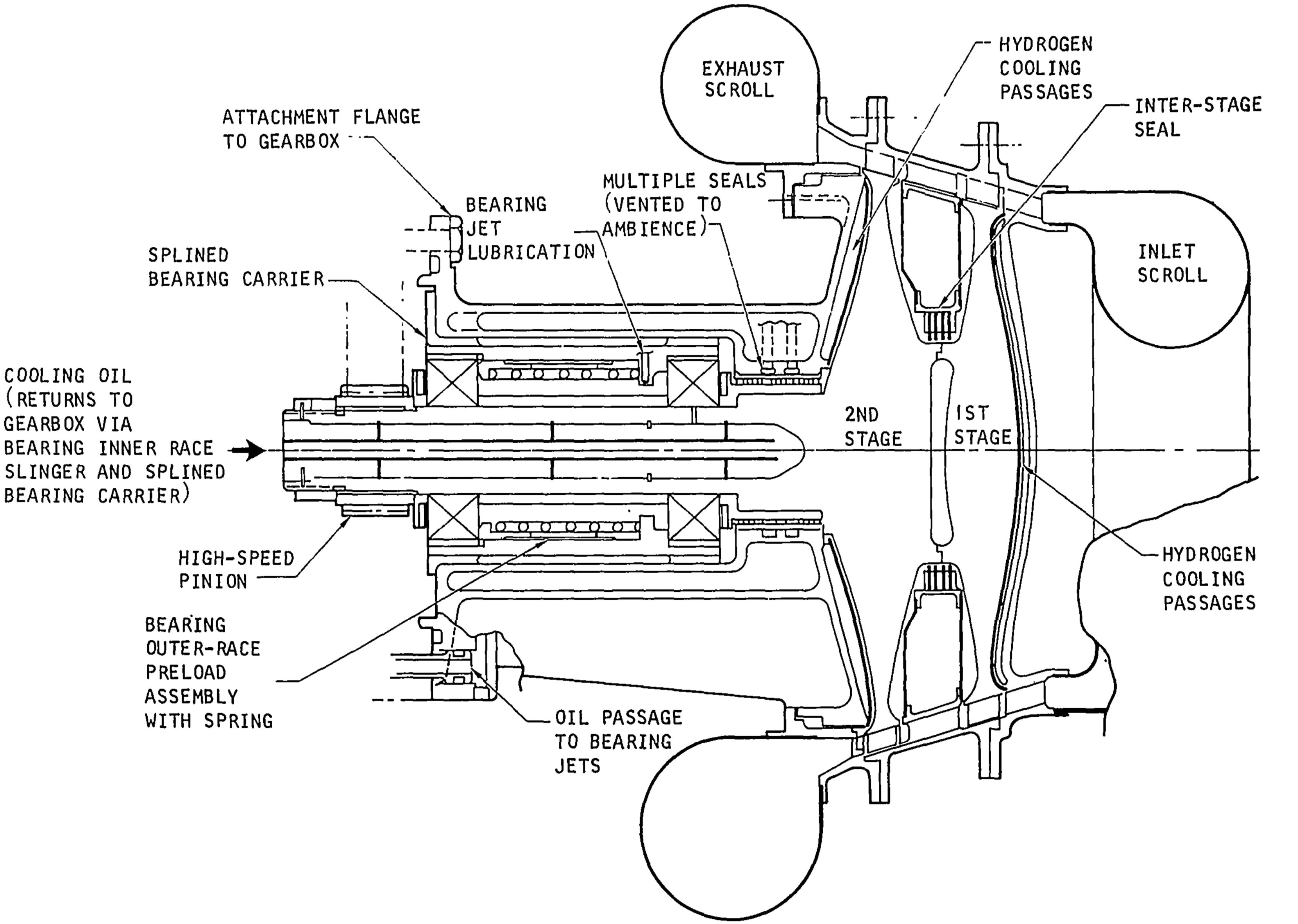


Figure 10. Rotating Assembly Layout

- MINIMIZES MACHINING OPERATIONS ON TURBINE SHAFT
- QUILL JOINT OFFERS A DEGREE OF TORSIONAL DAMPING AND FLEXIBILITY
- ALLOWS INCORPORATION OF A SHEAR SECTION TO MINIMIZE OVER-TORQUE DAMAGE
- REQUIRES PRECISE ALIGNMENT OF 4 BEARING SURFACES
- REDUCES RADIAL LOADS ON SUPPORT BEARINGS
- COMPLICATES GEARBOX

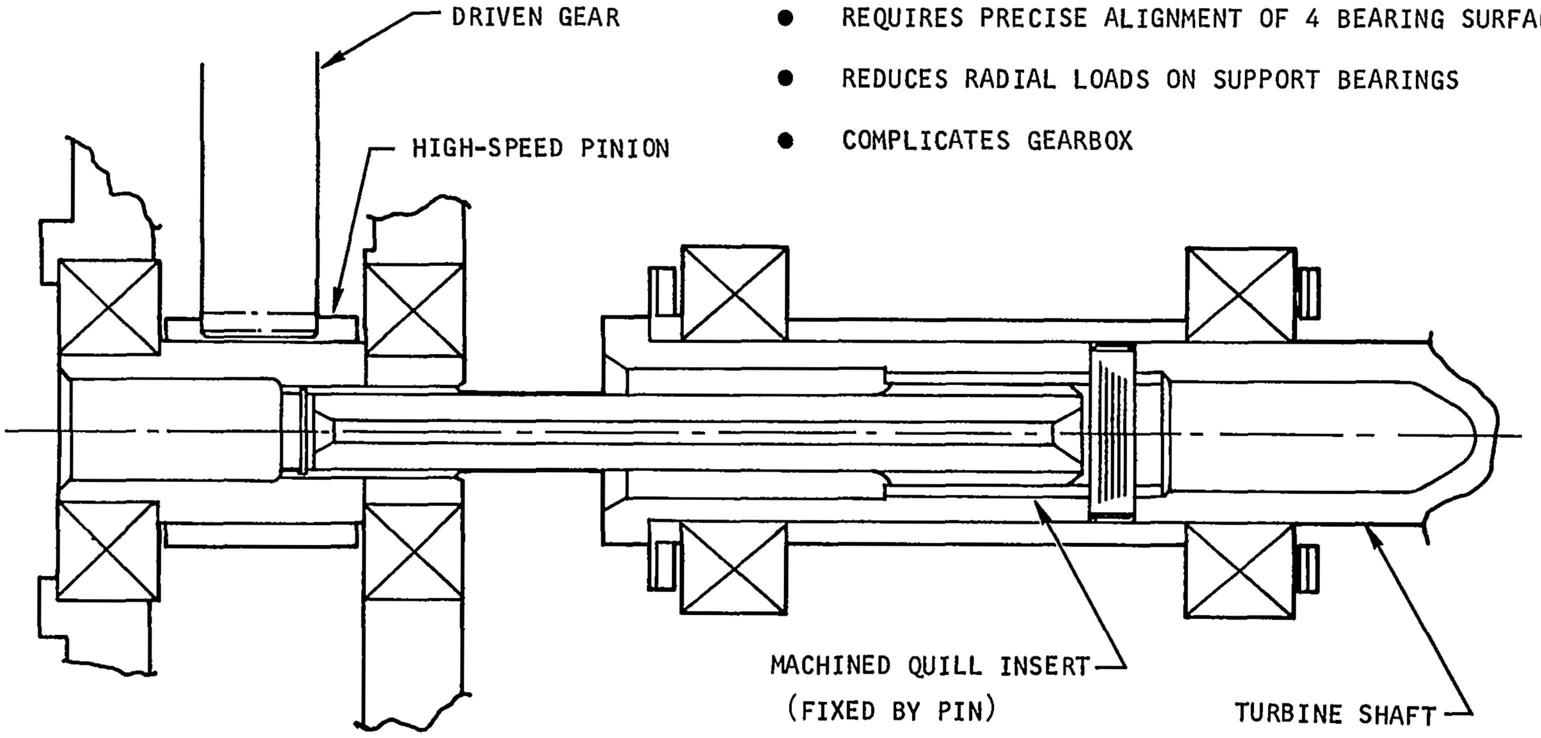


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Figure 11. Alternate High-Speed Pinion Splined Quill Shaft Drive Concept

1. Thermal Analysis

Available computer programs have been used to establish the steady-state and transient temperatures occurring throughout the rotating assembly. Two areas are of particular interest, the outboard bearing, and the first-stage turbine disk.

a. Hot End Bearing

Figure 12 shows the outboard bearing heat generation as a function of the axial and radial load on the bearing. The bearing configuration is as shown in Figure 10. The heat generation calculations are based on a method presented in Rolling Bearing Analysis (pages 421-450) by T. A. Harris published by John Wiley and Sons in 1966. Figure 13 presents the bearing operating temperature as a function of the heat generated for various cooling schemes. These schemes consist of a combination of the bearing jet lubricating flow and oil flow through the center shaft. The data indicate the acceptable bearing temperatures are obtained solely by cooling with the bearing jet oil flow of 20 lb/hr. Transient studies also indicate that the bearing temperature remains within the limitations of the lubricant during the heat soakback occurring after shutdown. In operation, the bearing temperature is primarily dependent upon the temperature of the jet oil. During shutdown, the bearing temperature is primarily dependent upon the temperature of the second-stage turbine disk.

b. First-Stage Turbine Disk

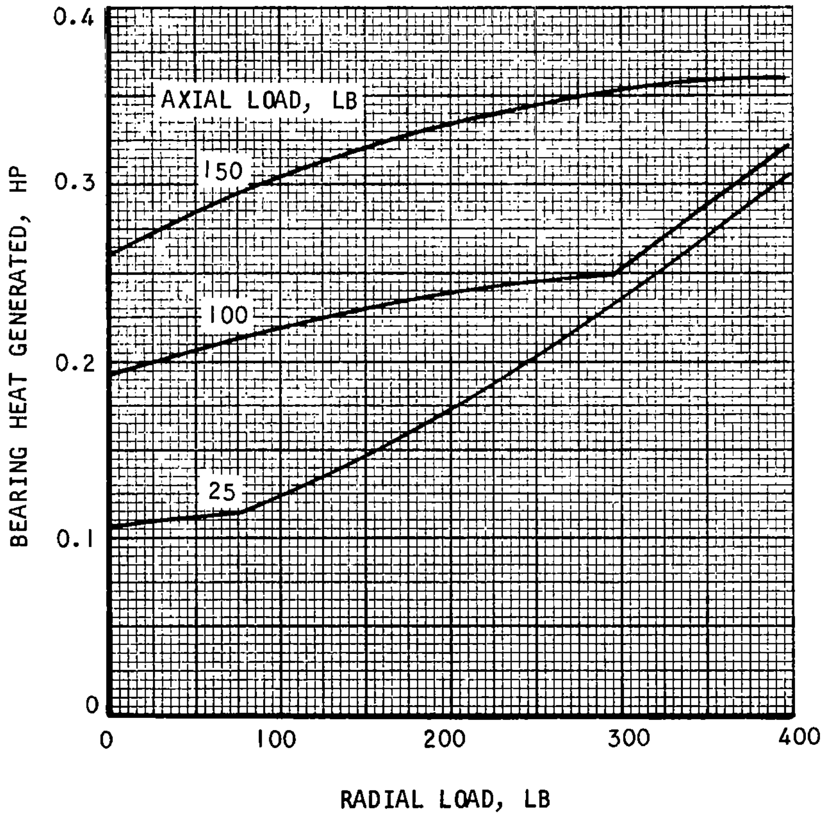
Figure 14 shows the steady-state and transient temperatures for the first-stage turbine disk. These data assume that the turbine housing is cooled by cold hydrogen gas passed through the casing as shown in Figure 10. The data indicate that the maximum blade temperature is about 1400°F, about 400°F below the turbine inlet temperature of 1800°F. Most of the turbine disk operates at a temperature of about 700°F, with the neck portion at about 1200°F.

2. Stress Analysis

The data of Figure 14 also show the steady-state stress distribution on the first-stage turbine disk. Although it operates at a higher pitch-line velocity, the second stage has considerably lower operating temperatures and thermal gradients. During the Phase II studies, detailed analyses will be performed on both turbine stages.

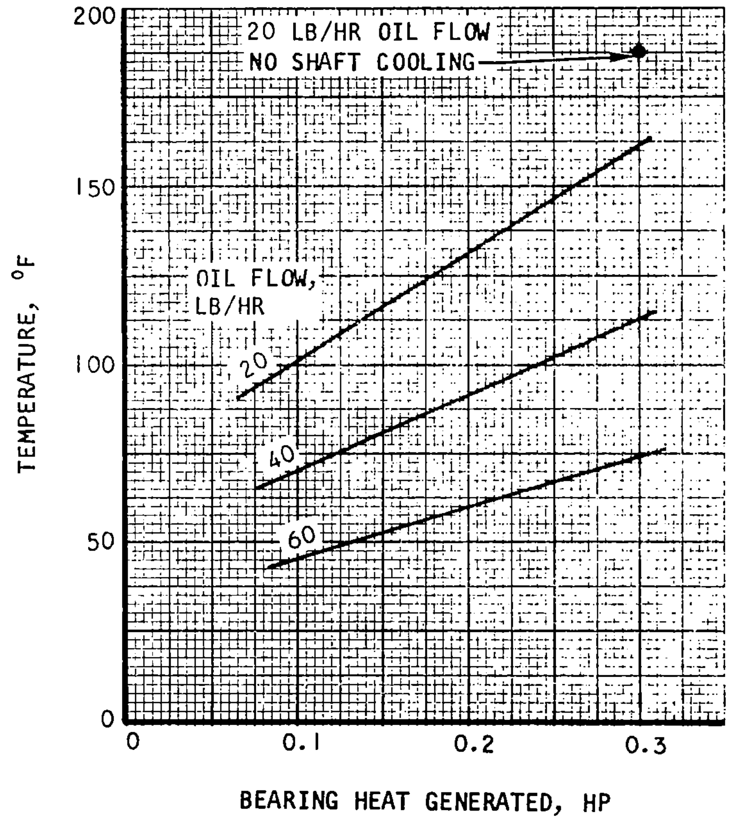
The stress data of Figure 14 show a maximum stress of about 85,000 psi occurring at the center of the disk. At the neck, the tangential stress is about 30,000 psi (compressive) and the radial stress is about 50,000 psi (tensile). These values are well within the 80,000 psi allowable stress that has been established by the Soderberg diagram of Figure 5.

It should be noted that the turbine disk is assumed to be a solid rim disk. The stresses due to thermal gradients can be partially relieved if the rim portion of the disk is slotted. However, the data of Figure 7 indicate



- 25 MM BEARING
- 15^{0} CONTACT ANGLE
- MIL-L-7808 LUBRICANT

Figure 12. Heat Generated in Bearing



- 240 LB/HR OIL COOLING FLOW THROUGH SHAFT AT 20^{0}F
- 400 LB/HR HYDROGEN FLOW THROUGH TURBINE HOUSING AT -60^{0}F

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Figure 13. Hot-End Bearing Steady-State Operating Temperatures

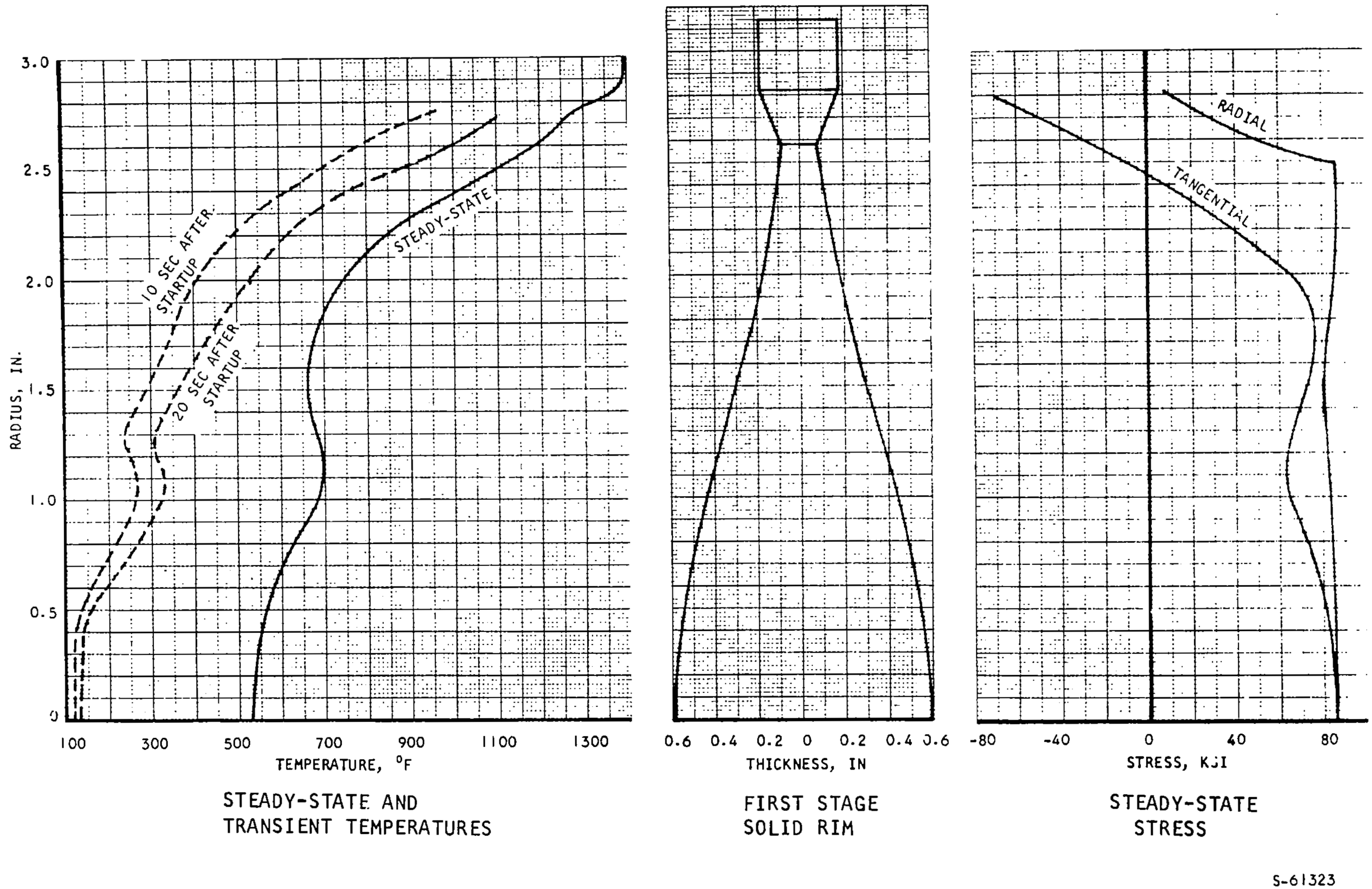

Figure 14. First-Stage Disk Temperatures and Stresses

that there is little weight incentive to do this. During Phase II, stress analyses for the transient temperature distributions will also be performed. These analyses may indicate that a slotted rim disk, such as is used in some aircraft turbines, will be desirable.

3. Critical Speed Analysis

Figure 15 shows the analytical model of the rotating assembly that is used as the basis of the critical speed study. Figure 16 shows the variation in the rigid body natural frequencies as a function of the bearing mount spring rate. The data indicate that a spring rate on the order of 15,000 lb/in. will place the rigid body modes substantially below the turbine operating speed.

For the selected bearing mount spring rate (equal to that commonly used by AiResearch in design of high-speed rotating equipment of this size), Figure 17 shows the vibrational amplitude vs frequency and gives the mode shapes of the two rigid body modes. The data are based on a rotating assembly balancing accuracy of about 0.18 gr-in. They indicate that the maximum deflection of the turbine disk is well within the disk-shroud clearance of 0.010 in. at maximum speed.

4. Bearing Operating Condition Analysis

The bearings used on the rotating assembly layout of Figure 10 have an operating DN number of 1.84 million at the design speed of 73,500 rpm. This DN is well within the present state of the art, as demonstrated by the bearing test data summarized in Table 4. The data were obtained as part of AiResearch IR and D activity in support of development of an engine-driven compressor for the Boeing Model 2707 (SST) aircraft. Four bearings of each type were tested.

TABLE 4

BEARING TEST DATA

Bearing construction	Balls and races: M-50 separator: Silver plated AISI 4340
Bearing lubrication	Oil: 1 gpm/bearing supplied at 300^{0}F
Bearing manufacturer	Industrial Tecktonic, Inc., Torrance, California
Test times	Bearing 204 (20 mm x 47 mm x 14 mm): 3640 hr at 74,000 rpm Bearing 205 (25 mm x 52 mm x 15 mm): 3800 hr at 75,000 rpm Bearing 305 (25 mm x 62 mm x 17 mm): 2450 hr at 65,000 rpm

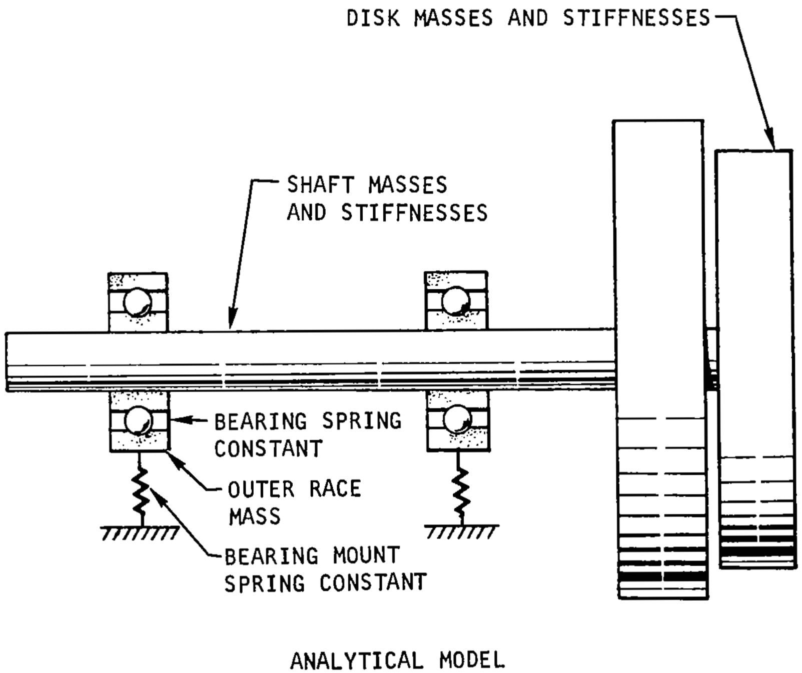

Figure 15. Critical Speed Analytical Model

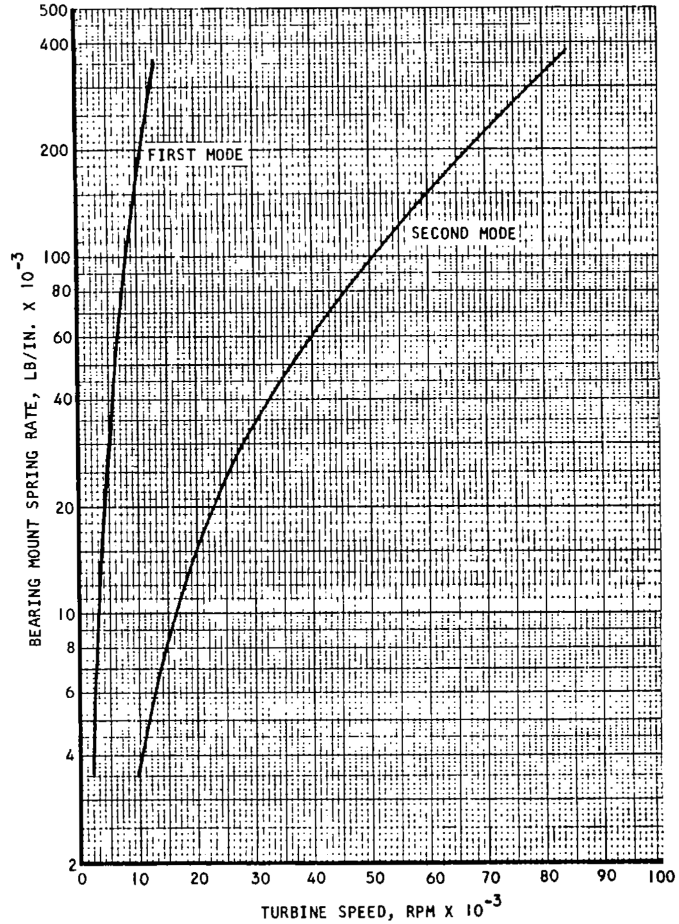

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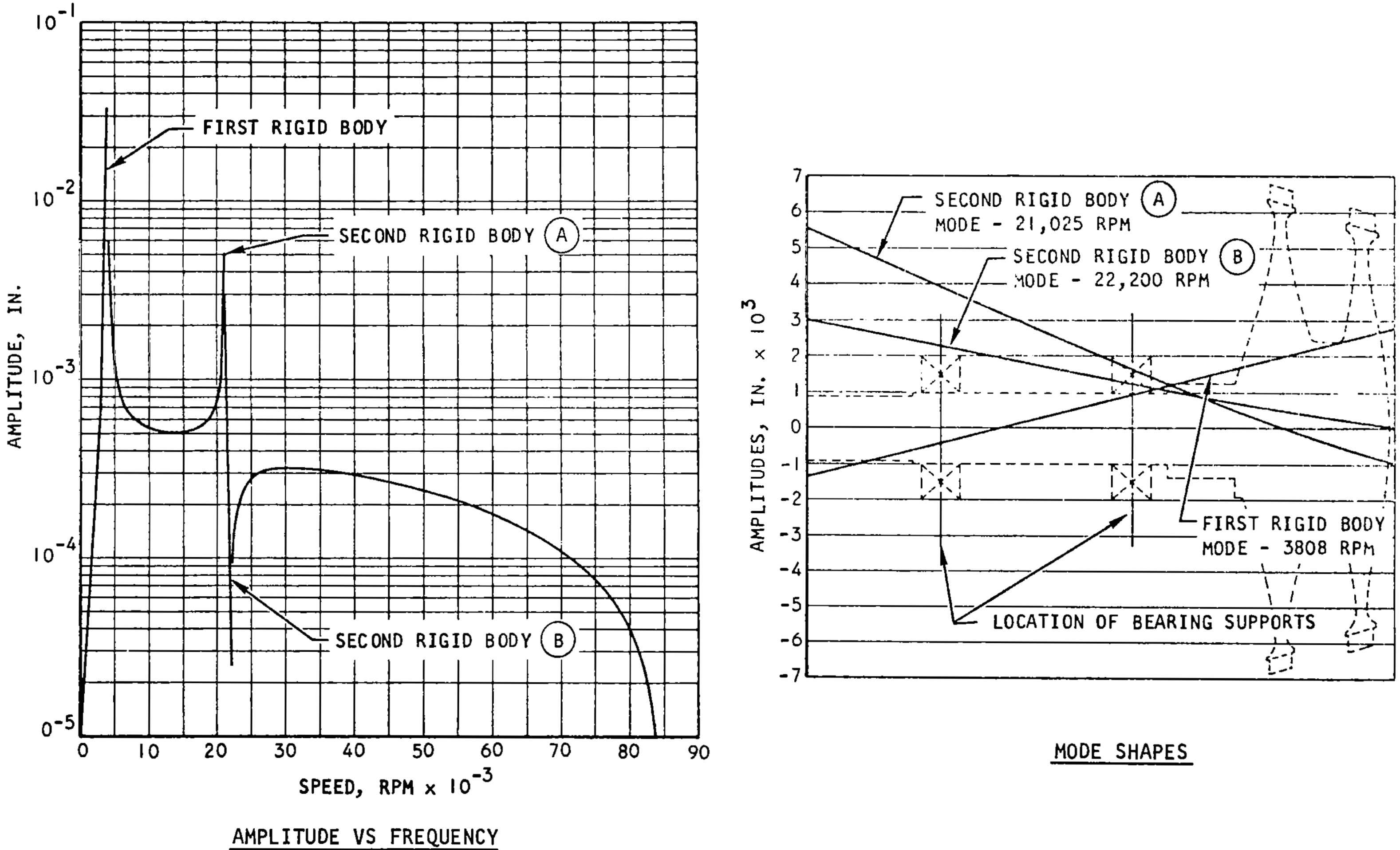
Figure 16. Selection of Bearing Mount Stiffness



- BEARING MOUNT SPRING CONSTANT = 15,000 LB/IN.
- BEARING SPRING CONSTANT = 800,000 LB/IN.

S-61321

Figure 17. Rotating Assembly Vibrational Modes

TURBINE POWER UNIT CONFIGURATION

The other major component of the turbine power unit, in addition to the rotating assembly, is the gearbox. The gearbox must provide the following functions:

Output pad for electrical generator

Output pad for hydraulic pumps

Means of flowing oil through the gearbox and the rotating assembly

Input pad for the rotating assembly

Structural support for the entire turbine power unit

It should be noted that the design of the hydraulic pump and the generator is outside the scope of this study contract. However, it is necessary to briefly consider these units when designing the gearbox. For this study, the hydraulic pumps are assumed to be Vickers designs (PV3-300 for a 70 gpm pump operating at 5000 rpm, and PV3-115 for a 35 gpm pump operating at 7000 rpm), and the electric generator is assumed to reassemble the Bendix 28B262-5 rotating rectifier generator operated at 12,000 rpm, although modifications would be required for cooling with hydrogen gas instead of air.

Single vs Twin Hydraulic Pumps

Studies of the gearing required for the single pump configuration indicate that the gearbox weight will be about 50 percent higher than for the two-pump gearbox. Schematics of the gearing arrangements for the two concepts are shown in Figures 18 and 19. The added weight penalty for the single pump configuration is largely due to the fact that the single pump operates at a lower speed and consequently requires more than twice the torque required to drive one of the 35 gpm pumps. Although the single pump weighs slightly less than two 35 gpm pumps, the total weight of pumps and gearbox favors the twin pump configuration. Additionally, using twin pumps allows one of the pumps to be depressurized at low power levels, thus reducing the pump parasitic power requirements below those of the single pump configuration.

Type of Gearing

Both spur and planetary gearing were considered for this application. However, the speed reductions are such as to allow driving the alternator with only a single stage of spur gearing. Thus, spur gears show a decided weight advantage over planetary gearing. If a higher turbine speed had been selected, such as 90,000 rpm, planetary gearing would probably show a weight advantage since two stages of spur gearing would be required to drive the alternator.

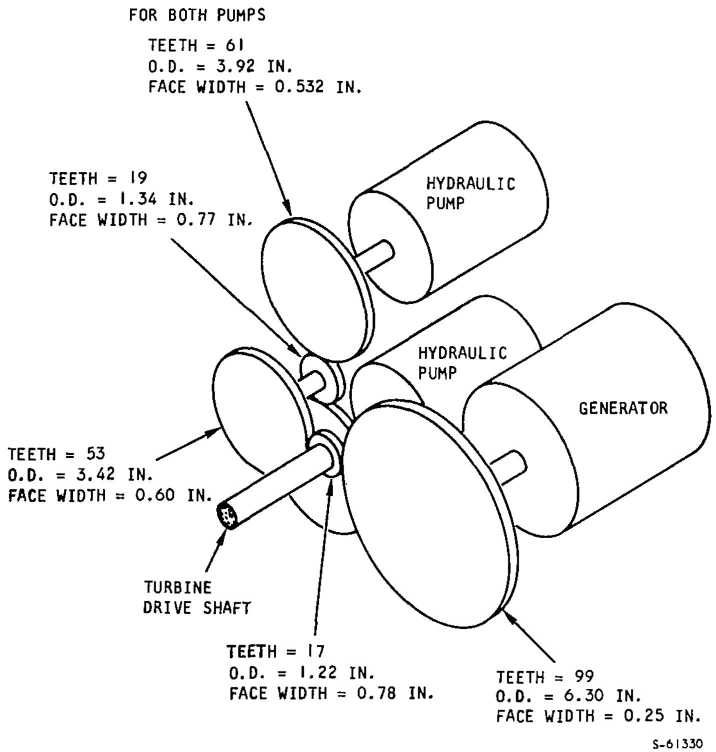


Figure 18. Twin Pump Gearing Configuration

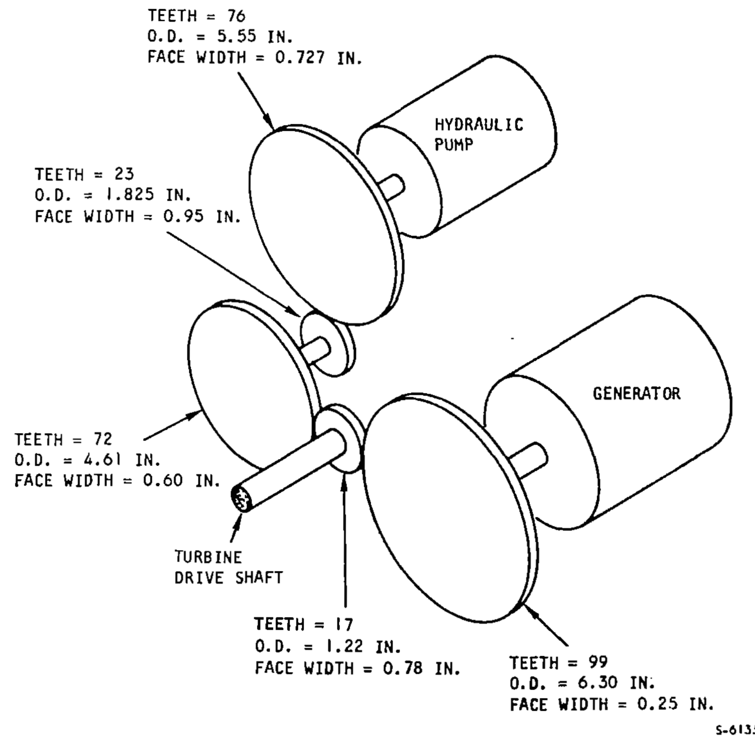


Figure 19. Single Pump Gearing Configuration

Turbine Power Unit Layout

AiResearch Drawing SK64481 shows a layout of the turbine power unit, presenting a cross-section of the gearbox. The layout is for an APU having two 35 gpm hydraulic pumps. The turbine rotating assembly is located on one side of the gearbox with the pumps and generator on the other side. The gearbox casing provides mounting flanges for the rotating assembly, pumps, and generator, and has three points for attaching the turbine power unit to vehicle structure. Lubrication of the gearbox and turbine bearings is accomplished by passing oil through a gear pump (located at the top of the gearbox, driven from the hydraulic pump gearing) which feeds the oil to the various gear meshs, cooling passages, and bearings. Pickup and return of the oil to the gearbox is accomplished by use of a rotating sump which provides a pressure head to drive the oil back to the gear pump. The oil flow passages between the various gears and the return lines to the gear pump are shown in the gearbox end view on the drawing, and schematically in Figure 20.

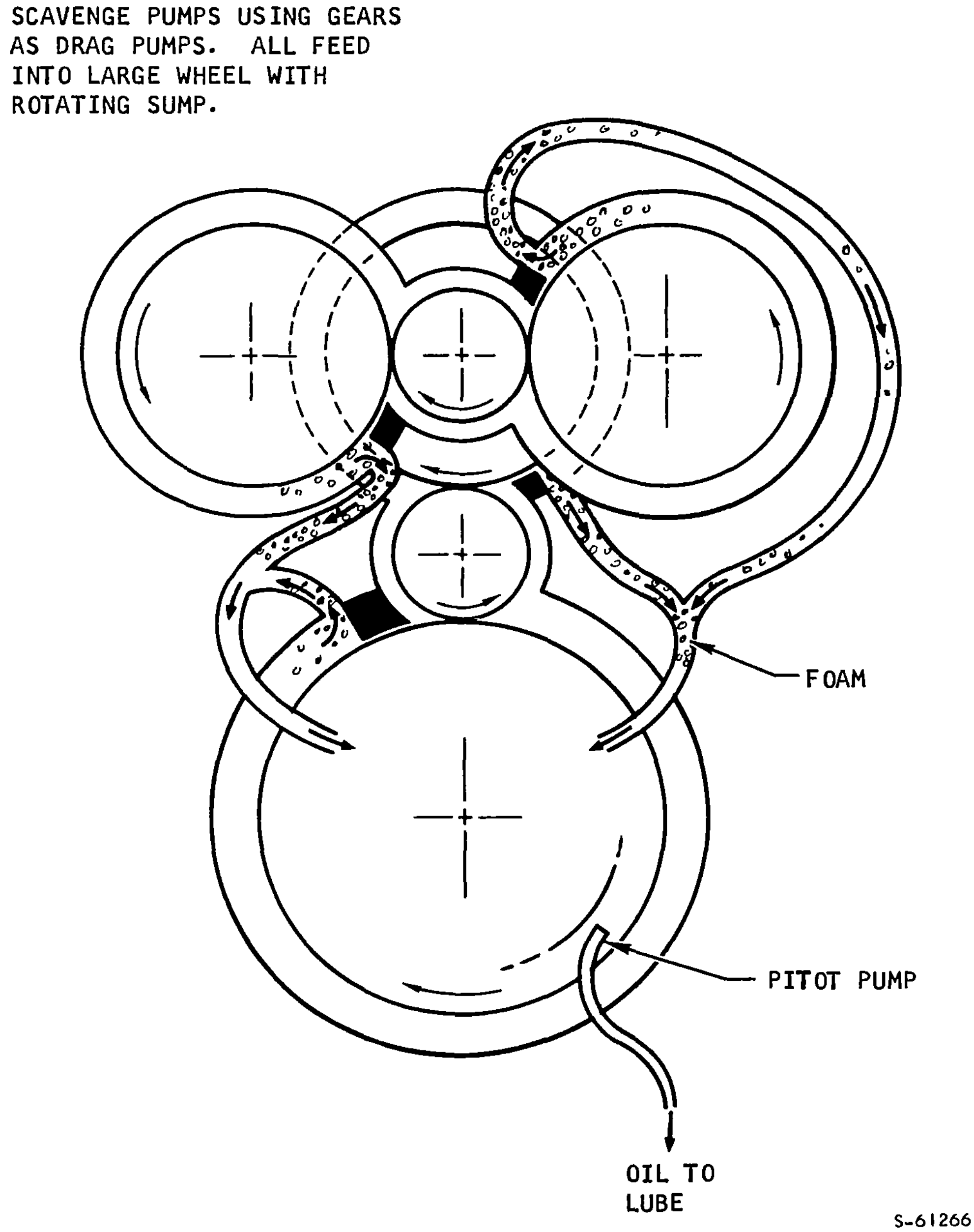
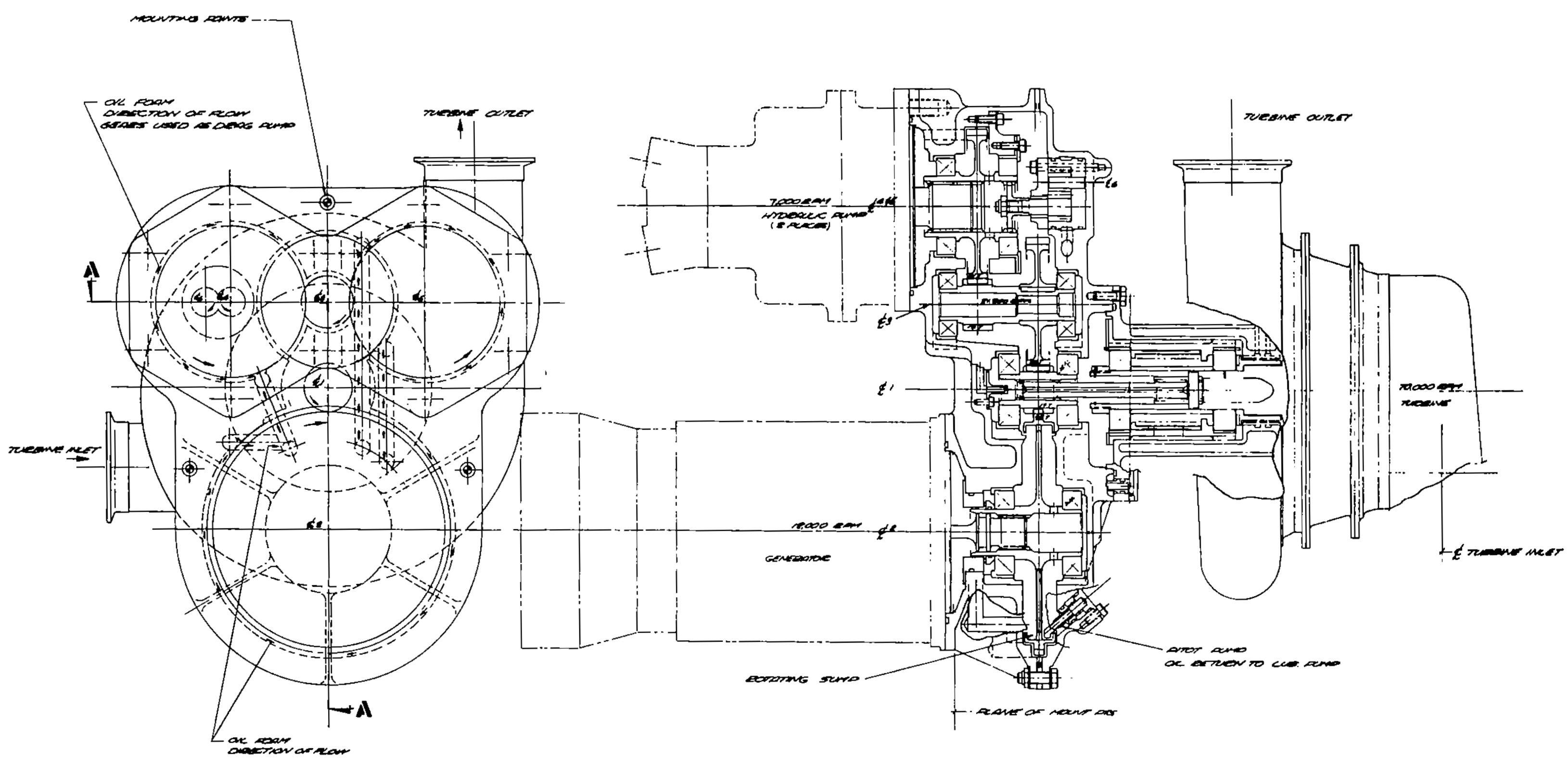


Figure 20. Zero "G" Lube System



Cross section, power unit, gear box.

APPENDIX C

CRYOGENIC TANKAGE

INTRODUCTION

This appendix presents the parametric study conclusions and results for the APU cryogenic tank system. The study parametrically evaluated tank optimization and performance for various orbiter and booster configurations.

An existing AiResearch computer program was used to evaluate the optimum vent pressure for various delivery pressures for both the hydrogen and oxygen tanks. These optimum vent pressures were then used to evaluate tank performance. The performance calculations considered parameters such as hard and soft shell tank design, percent of contents used on first day, orbiter and booster standby times, and also the various tank sizes. The range of parameters investigated was from 100 to 2000 lb deliverable contents; and from 35 to 800 psia delivery pressure.

ASSUMPTIONS

General

The tanks were designed with 5 percent residuals and 5 percent ullage volume.

The fill factor used was 95 percent.

Structural mounting fittings constitute 15 percent of inner shell weight.

Oxygen tanks were designed considering the lightest configuration between aluminum and Inconel

Hydrogen tanks were designed considering the lightest configuration between aluminum and titanium

Orbiter calculations considered a 6-day quiescent period after the first day's fluid withdrawal

Booster calculations considered minimum flight time and no quiescent period

2.5 Btu/hr allowed for line heat leak

Ambient temperature was 500°R

Minimum annulus for hard shell design was 1.5 in.

10 g acceleration

0.007-in. manufacturing tolerance

Safety factor was 1.5 except for supercritical where safety factor was 2.2.

Equations

$P_{CR} = 1.6 \frac{Et^2}{D^2}$ buckling mode for outer shell

$P_{VENT} = \frac{2t\sigma}{R}$ burst mode for inner shell

$$\text{LOAD FACTOR} = \frac{\text{TANK WET WEIGHT}}{\text{DELIVERABLE CONTENTS}}$$

Structural Parameters

Aluminum $\rho = 0.1$ lb/in.^{3} $\sigma = 39{,}000$ psi

Inconel $\rho = 0.3$ lb/in.^{3} $\sigma = 158{,}000$ psi

Titanium $\rho = 0.161$ lb/in.^{3} $\sigma = 144{,}000$ psi $E = 16 \times 10^6$ psi

Insulation = 1.5 lb/ft^3

Fiberglass = 22.4 lb/ft^3

Protective outer cover on soft shell = 0.08 lb/in.^{3}

Minimum Thickness

Outer shell = 0.035 in.

Inner shell = 0.035 in.

Soft shell outer cover = 0.010 in.

Foam insulation = 0.25 in.

INSULATION CONCEPTS

Two insulation concepts were considered:

Vacuum Jacketed Superinsulation--This concept consists of an inner tank shell designed to fail in a burst mode and an outer shell designed to fail in a buckling mode due to extreme ambient pressure. A vacuum is maintained between the two shells. This hard shell design lends itself to very predictable insulation characteristics. For extended ground prelaunch wait or extended atmospheric flight, the advantages of the vacuum insulation may justify the weight penalties associated with this shell design.

Ambient Pressure Superinsulation--This concept consists of insulation that operates at ambient pressure and is covered by a soft nonpressure carrying shell. For operation occurring principally in space, this type of insulation can be very efficient. For details of both concepts see Figure 1.

OPTIMIZATION

Tank optimization consisted of five sections, each section being interdependent on the other. Optimization yields the most efficient pressure band for a given delivery pressure and a given tank size. The optimization also considers the proper material selection for the different hydrogen and oxygen tanks and it considers the proper insulation requirements for the different orbiter and booster mission requirements. Throughout the optimization study, reference is made to the term load factor. Load factor is defined as the tank wet weight divided by the tank deliverable contents. The optimizations investigated are:

Material selection

Vent pressure range

Insulation thickness

Fluid usage profile

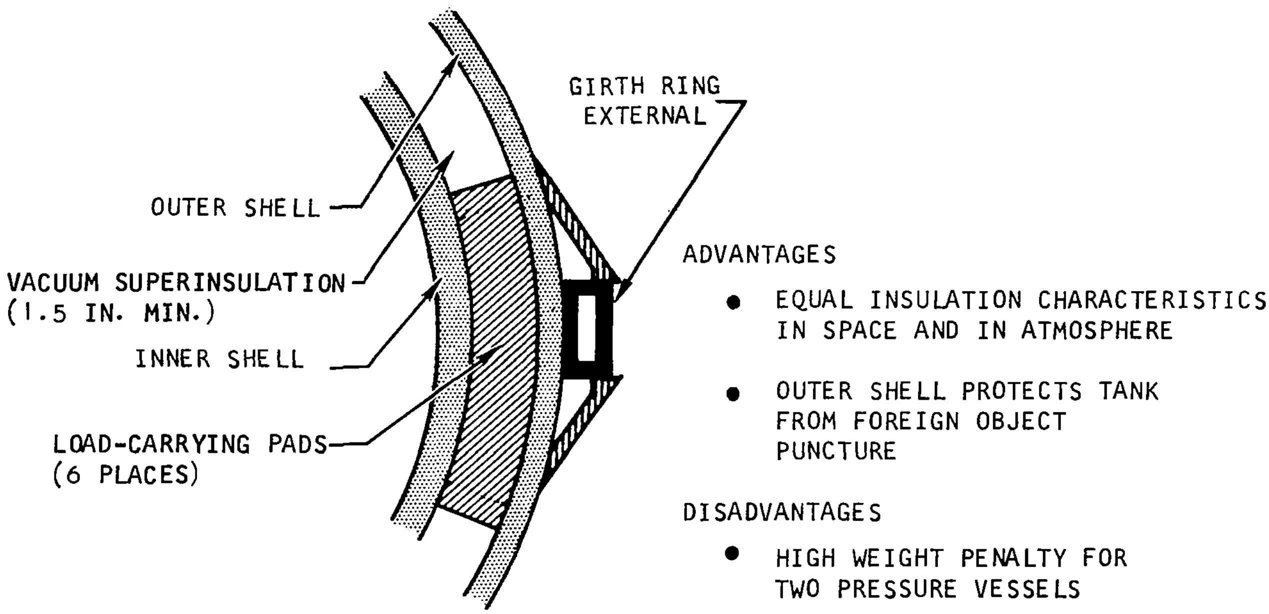
Delivery pressure

Material Selection

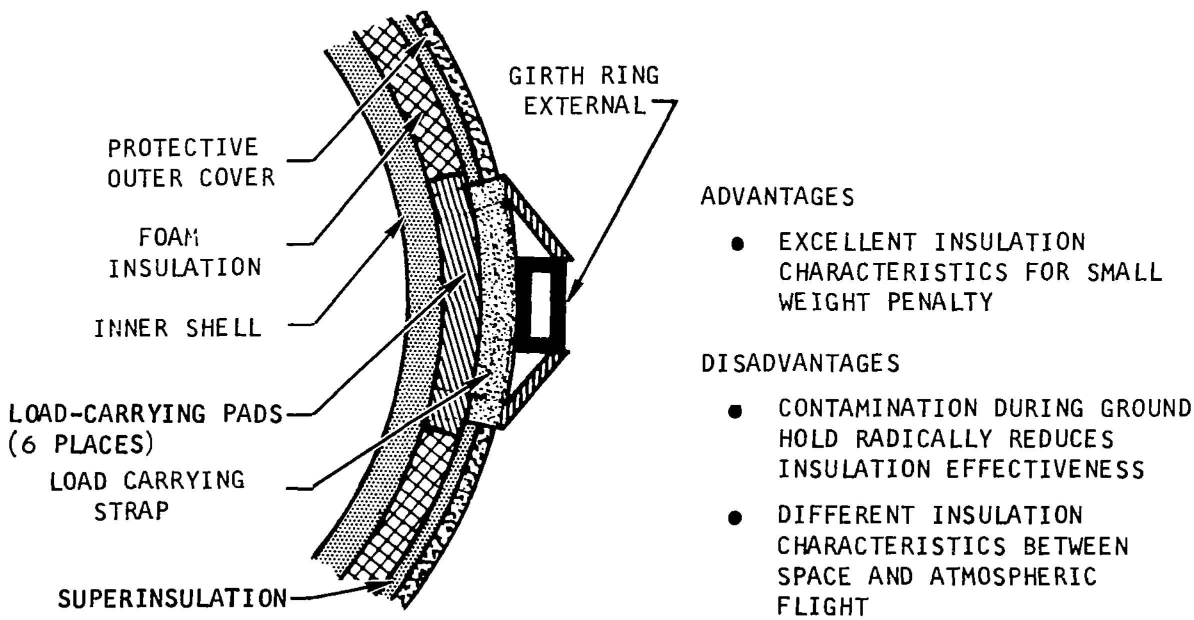
The main difference between oxygen and hydrogen tank material selection is that the highly corrosive atmosphere of oxygen must be considered. Past AiResearch experience dictates use of Inconel or aluminum for oxygen tanks. For hydrogen tanks aluminum or titanium are efficient materials. Figures 2 and 3 compare the inner shell weight for hydrogen and oxygen tanks for the materials evaluated. The minimum material thickness of 0.035 in. makes aluminum preferable at low pressures, and titanium or Inconel preferable at high pressures.

Vent Pressure

It is important to parametrically study the effect of a variable vent pressure (pressure band) on the total tank wet weight and thus the tank load factor. For any given energy (heat input and standby time) requirement there exists a tradeoff between increased insulation weight and increased tank shell weight due to the higher tank pressures that arise when more heat leak is allowed into the tank. Tables 1 and 2 show the results of the high pressure case when the pressure band was varied for two different size tanks. Note that for constant tank deliverable contents, the tank wet weight is proportional to the load factor. Figures 4 and 5 are typical low pressure results of this tradeoff.



a. HARD SHELL DESIGN



b. SOFT SHELL DESIGN

Figure 1. Insulation Design Concepts

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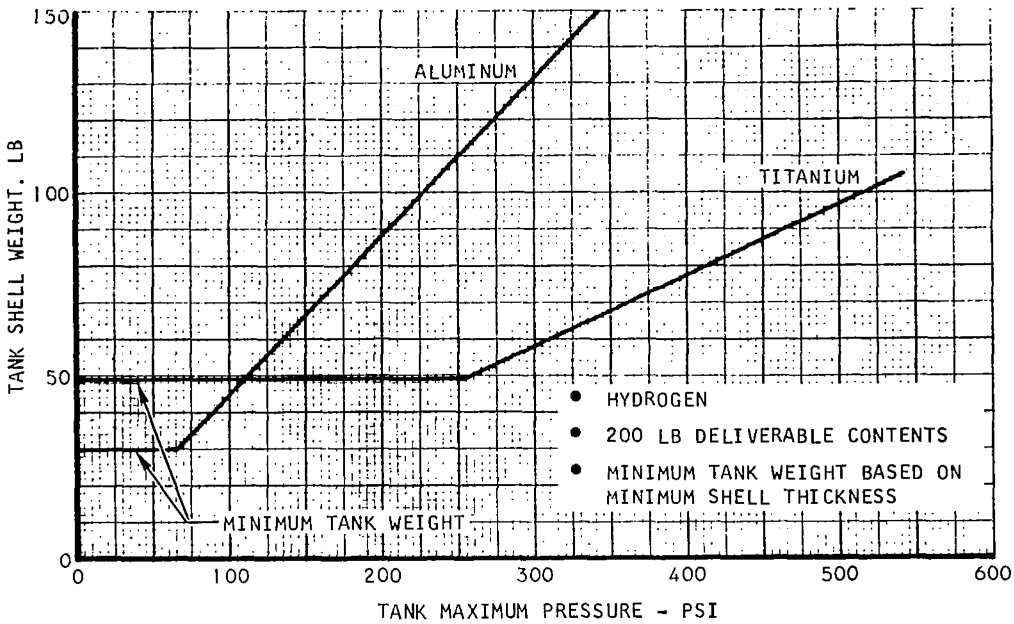

Figure 2. Hydrogen Inner Shell Weight vs Maximum Tank Pressure

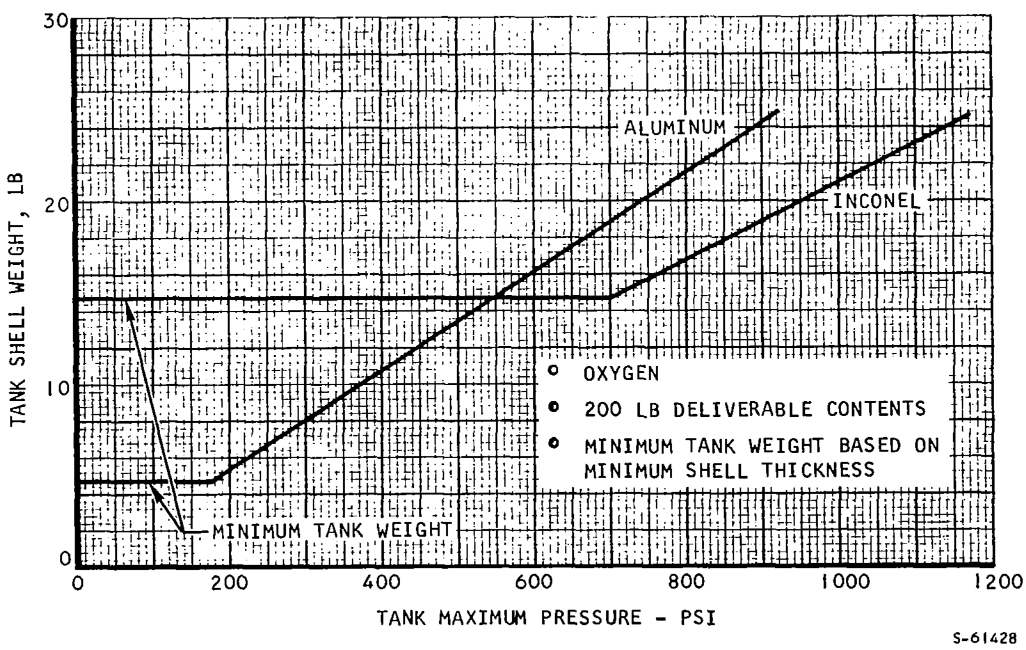

Figure 3. Oxygen Inner Shell Weight vs Maximum Tank Pressure

TABLE 1

HYDROGEN TANK OPTIMUM PRESSURE BAND VS WET WEIGHT

	250-lb Deliverable Contents				750-lb Deliverable Contents			
Operating pressure, psia	200	400	600	800	200	400	600	800
Pressure band, psi	50	100	23	75	30	100	25	75
Wet weight, lb	590	621	674	722	1622	1736	1889	2043

TABLE 2

OXYGEN TANK OPTIMUM PRESSURE BAND VS WET WEIGHT

	250-lb Deliverable Contents				750-lb Deliverable Contents			
Operating pressure, psia	200	400	600	800	200	400	600	800
Pressure band, psi	300	200	200	600	300	200	100	600
Wet weight, lb	320	312	311	357	913	911	921	998

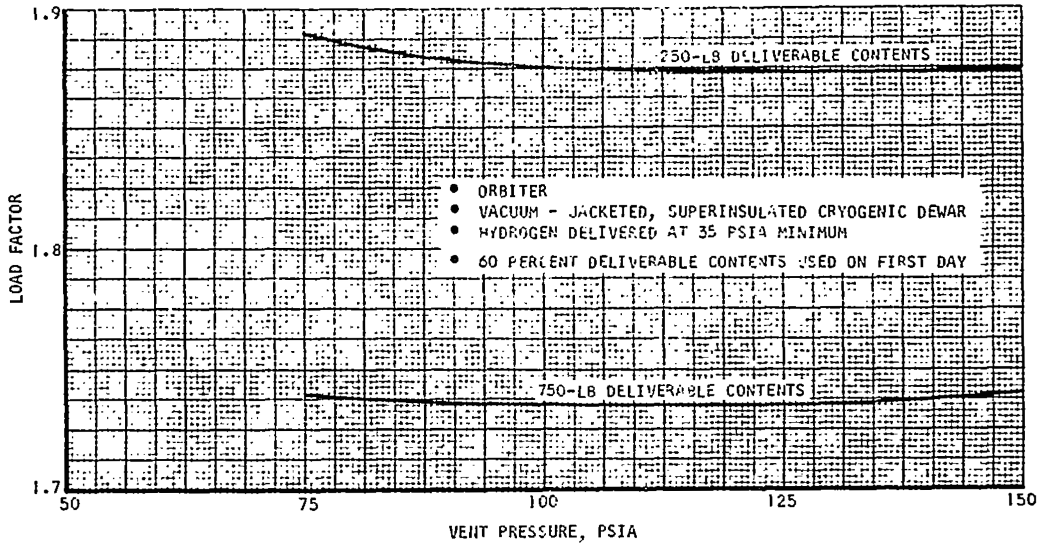

Figure 4. Hydrogen Tank Vent Pressure vs Load Factor

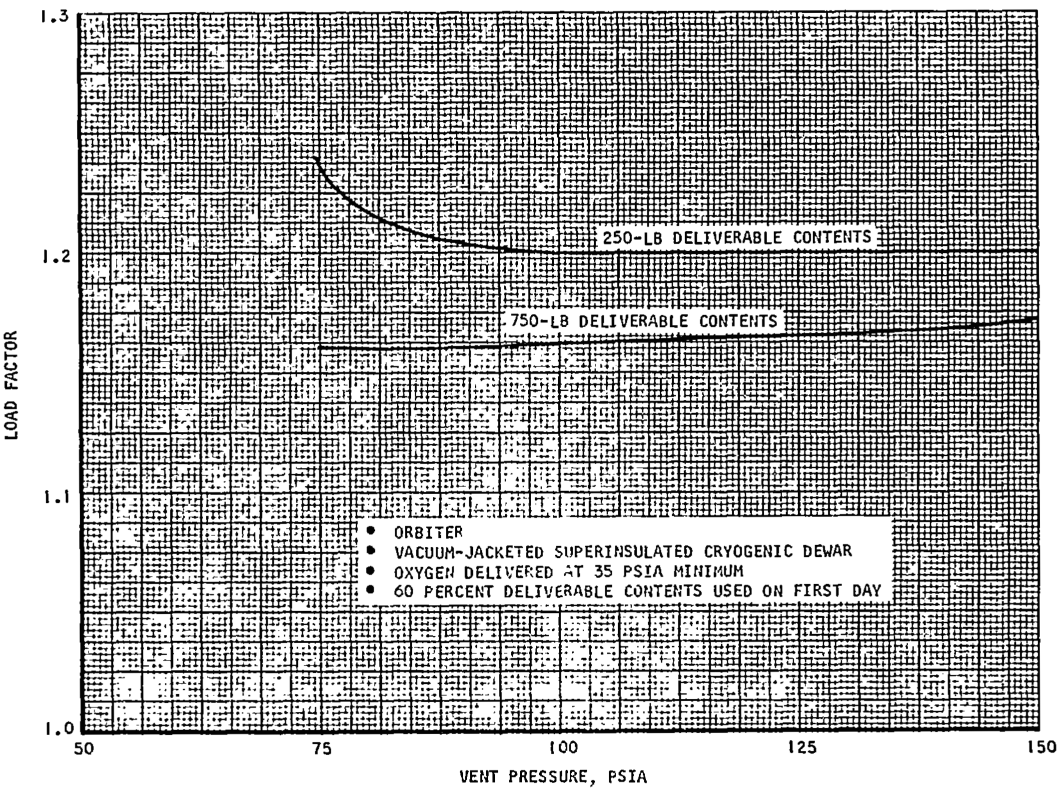

Figure 5. Oxygen Tank Vent Pressure vs Load Factor

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Insulation Thickness

There is a tradeoff for hard-shell tank designs between increased insulation thickness and percent of deliverable contents vented. Figures 6 and 7 show the tradeoff for an orbiter hard-shell design.

Fluid Usage Profile

One of the important parameters in tank optimization is the fluid usage profile. For the orbiter the first large withdrawal of propellant occurs on the first day and then there is a 6-day quiescent period before the remaining propellant is used. Thermodynamically, the amount of propellant used on the first day directly affects the amount of heat that the tank can absorb during the quiescent period before venting occurs. The booster mission requires no quiescent period and thus the insulation requirements are sufficiently met at minimum thickness values. Figures 8 and 9 show a typical result of the relationship between percent of contents used on first day vs load factor for large and small tanks.

Delivery Pressure

Figure 10 shows the effect of delivery pressure on the tank load factor. These data are required to select the operating conditions for the supercritical tank system described in Section 4.

TANK PERFORMANCE

The following discussion is based on the paramtric data obtained in the optimization study discussed earlier in this appendix. Table 3 outlines the results of a typical tank optimization. The following figures represent extensive tank performance evaluation based on the criteria listed in the assumptions section of this appendix. This section outlines tank performance in terms of load factor. Load factor is equal to wet weight divided by deliverable contents. Figures 11 through 18 show these results for high and low pressures for both orbiter and booster configurations. Figures 19 through 22 show the performance comparison between the tank insulation concepts.

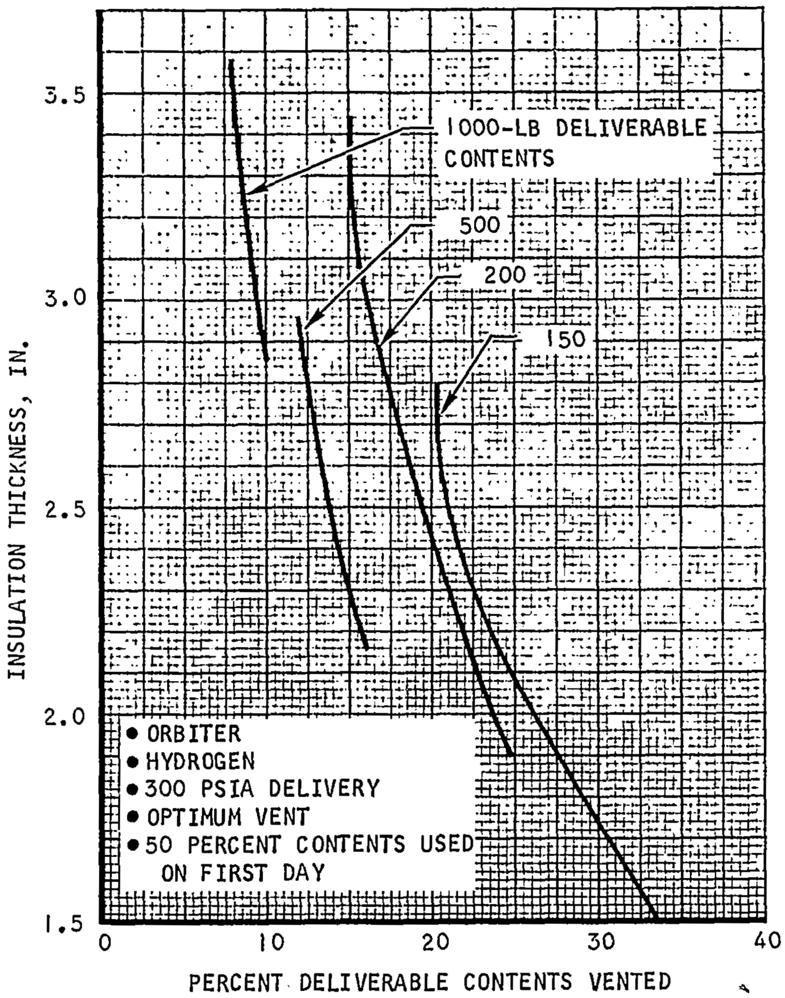

Figure 6. Insulation Thickness vs Contents Vented

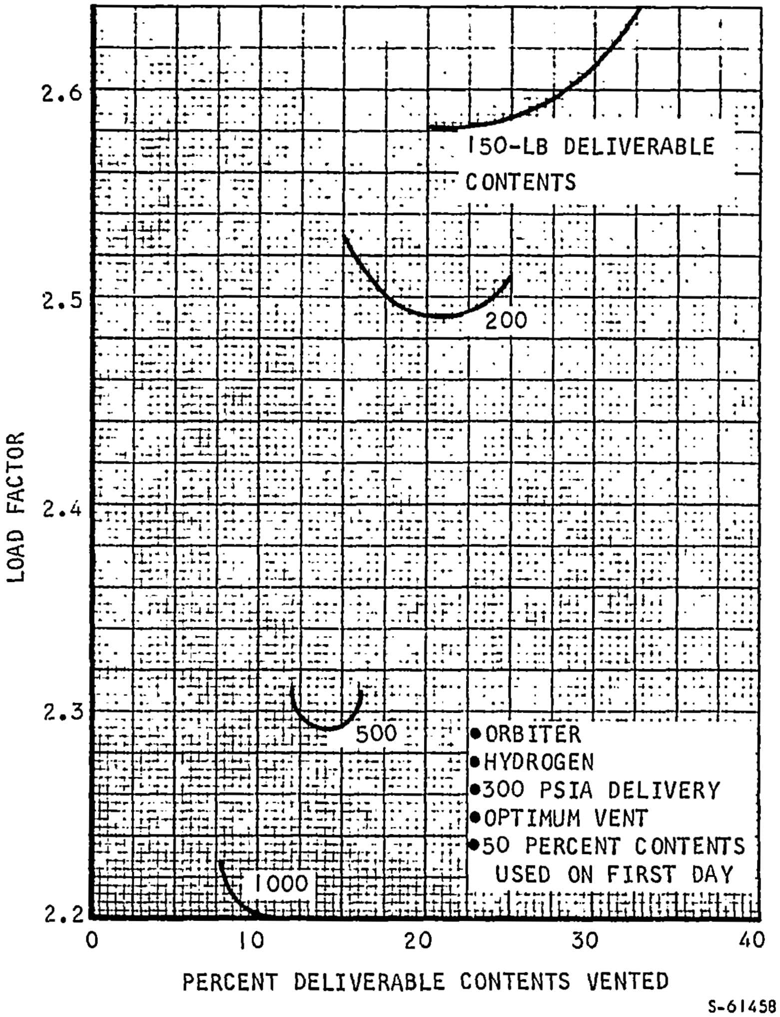

Figure 7. Load Factor vs Contents Vented

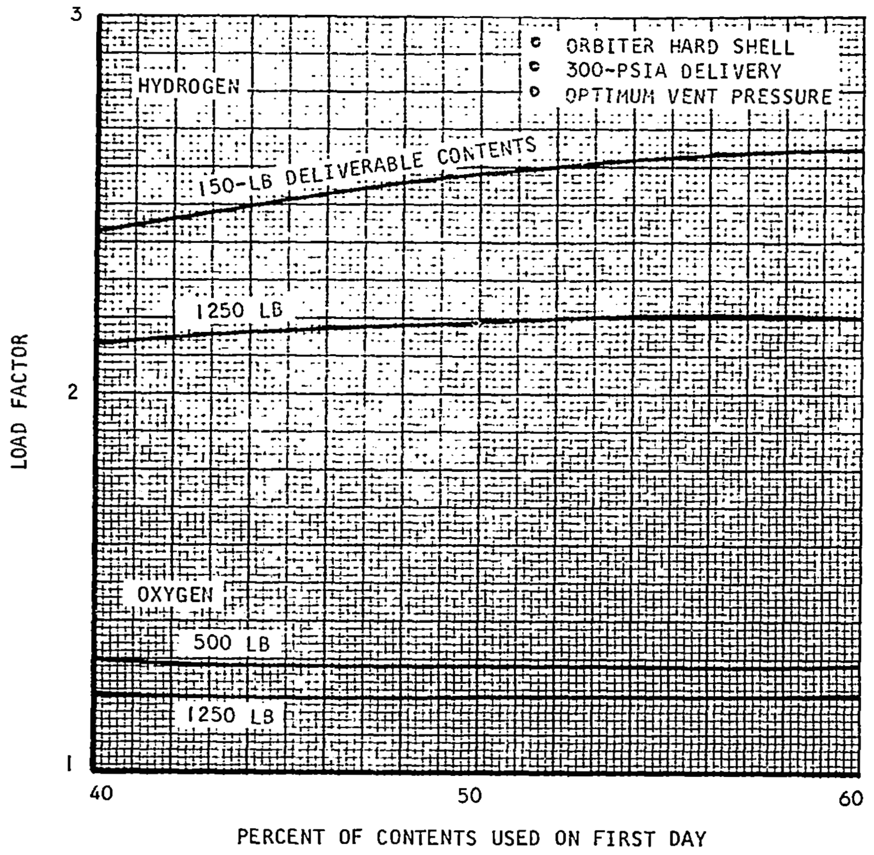

Figure 8. Load Factor vs Percent of Contents Used First Day

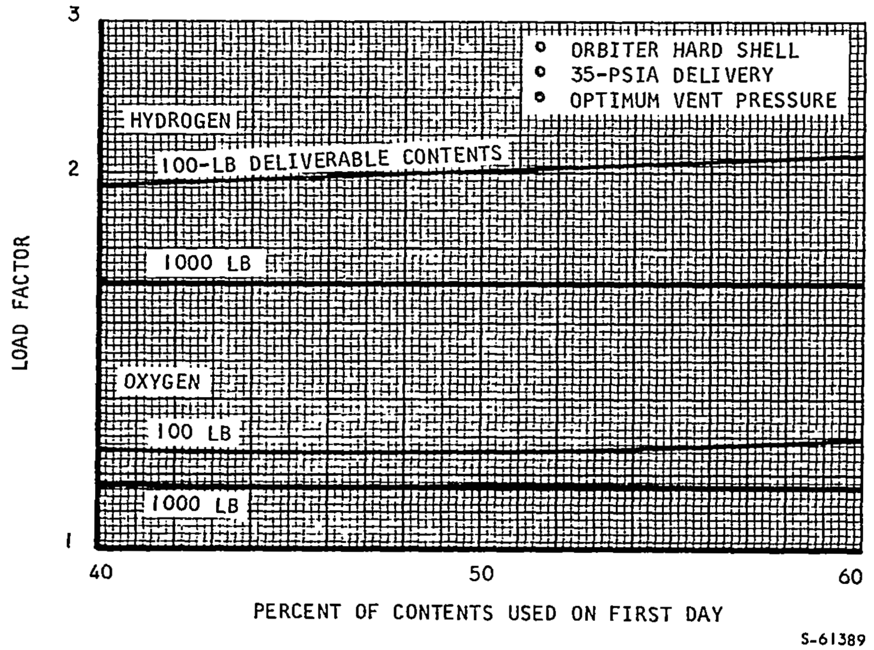

Figure 9. Load Factor vs Percent of Contents Used First Day

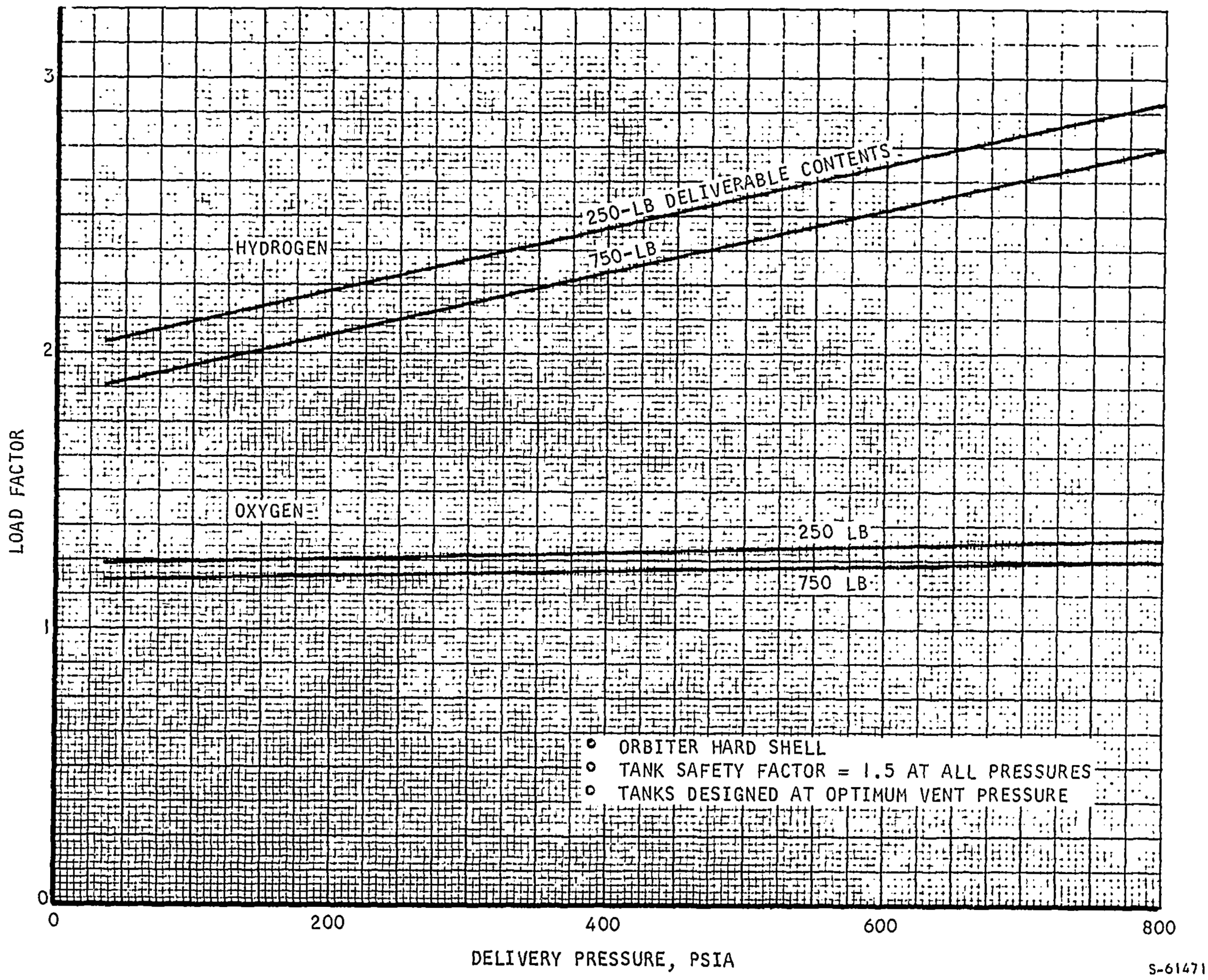

Figure 10. Load Factor vs Delivery Pressure

TABLE 3

ORBITER HARD-SHELL OPTIMIZATION DATA

	Oxygen	Hydrogen	Oxygen	Hydrogen
Delivery pressure	35 psia	35 psia	300 psia	300 psia
Inner shell	Aluminum	Aluminum	Aluminum	Titanium
Outer shell	Titanium	Titanium	Titanium	Titanium
Inner shell thickness, in.*	0.035 at 200 lb 0.06 at 1000 lb	0.053 at 200 lb 0.035 at 1000 lb	0.116 at 200 lb 0.048 at 1000 lb	0.047 at 200 lb 0.08 at 1000 lb
Outer shell thickness, in.*	0.035 at 200 lb 0.037 at 1000 lb	0.053 at 200 lb 0.090 at 1000 lb	0.035 at 200 lb 0.037 at 1000 lb	0.054 at 200 lb 0.09 at 1000 lb
Insulation thickness, in.	1.519	1.519	1.519	1.519
Fluid vented, lb*	0 at 200 lb 0 at 1000 lb	0 at 200 lb 10 at 1000 lb	0 at 200 lb 0 at 1000 lb	10 at 200 lb 10 at 1000 lb
Maximum operating pressure, psia	150	100	550	325
Wet weight, lb	242 at 200 lb 1168 at 1000 lb	378 at 200 lb 1710 at 1000 lb	255 at 200 lb 1206 at 1000 lb	418 at 200 lb 1934 at 1000 lb

*Reference to 200 lb at 1000 lb is tank deliverable contents.

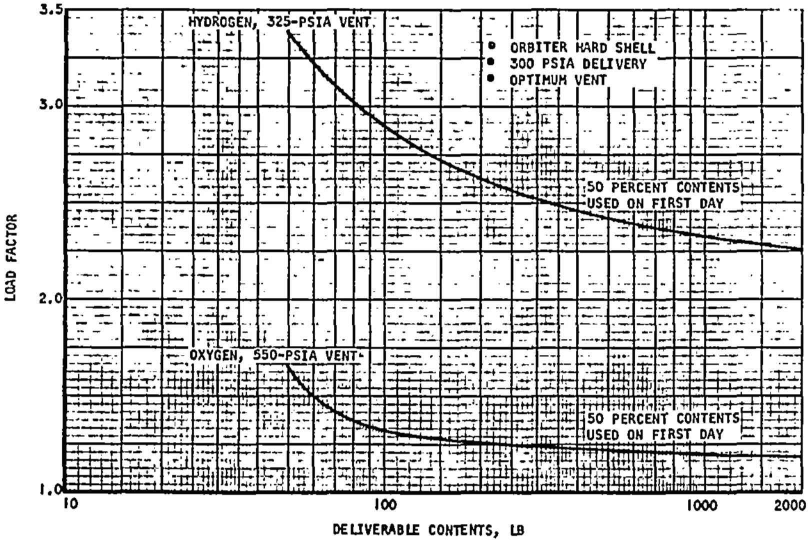

Figure 11. Load Factor vs Deliverable Contents

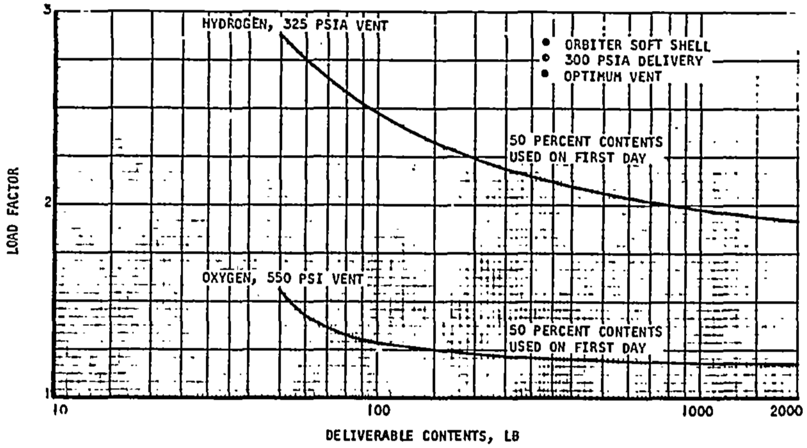

Figure 12. Load Factor vs Deliverable Contents

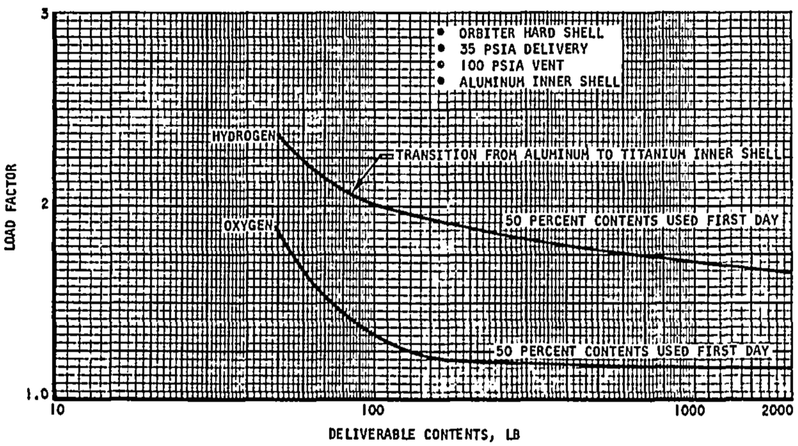

Figure 13. Load Factor vs Deliverable Contents

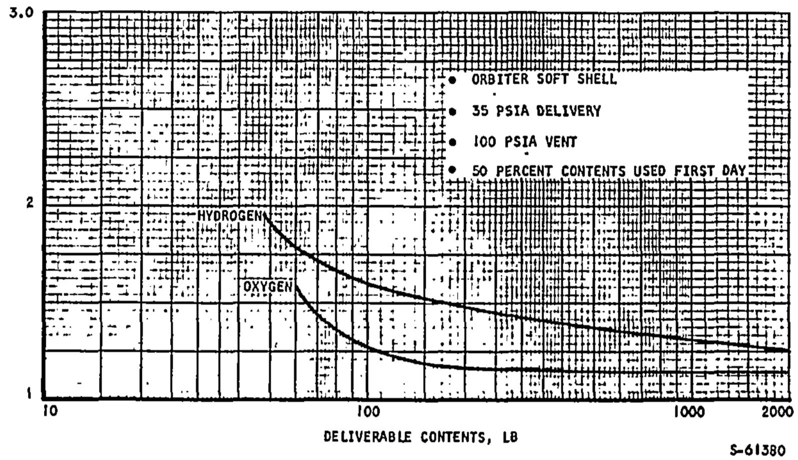

Figure 14. Load Factor vs Deliverable Contents

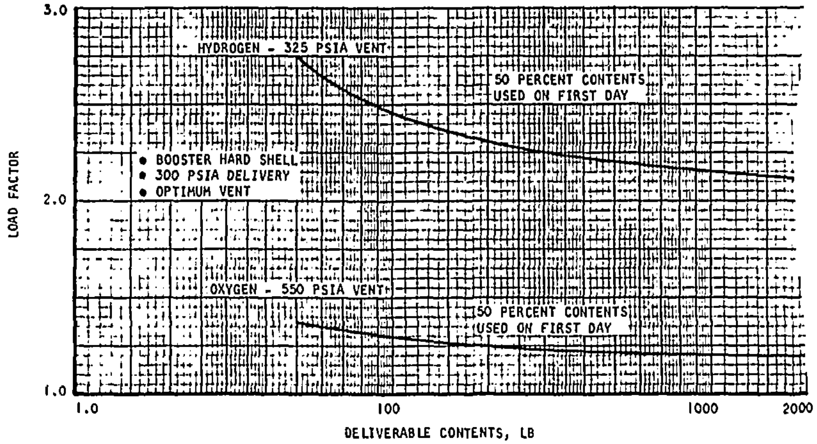

Figure 15. Load Factor vs Deliverable Contents

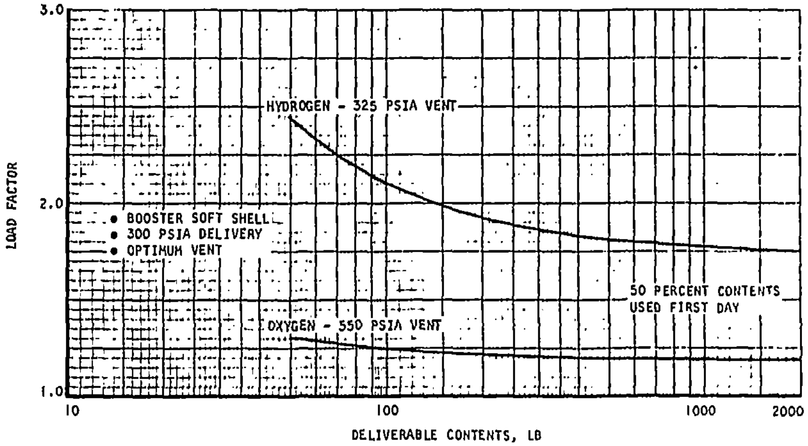

Figure 16. Load Factor vs Deliverable Contents

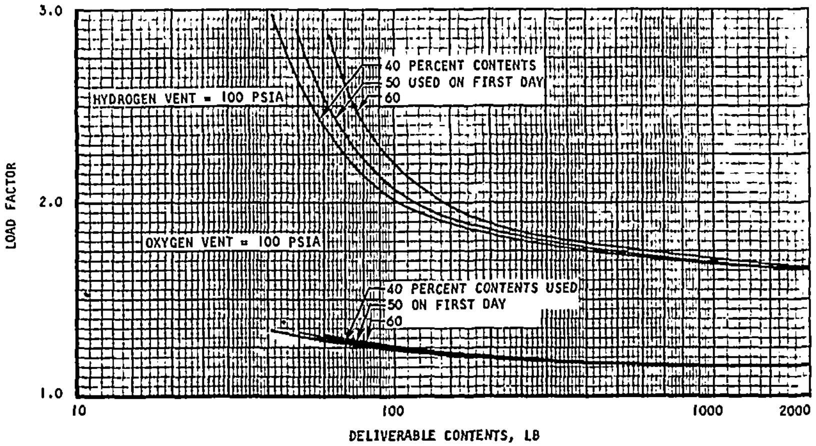

Figure 17. Load Factor vs Deliverable Contents

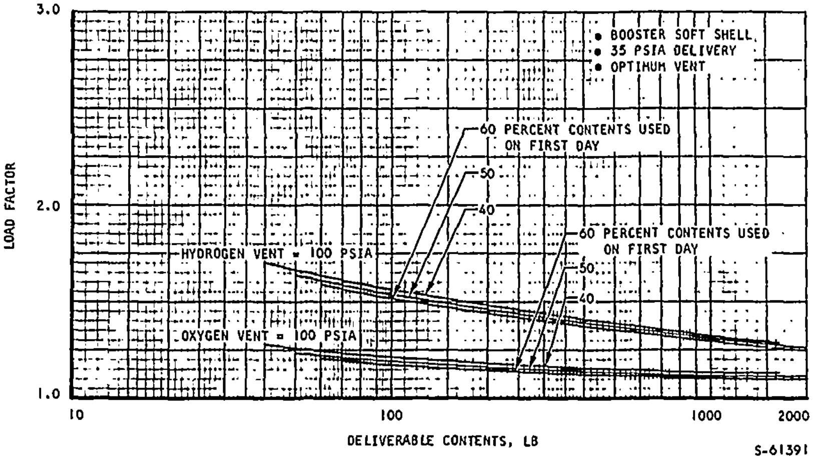

Figure 18 Load Factor vs Deliverable Contents

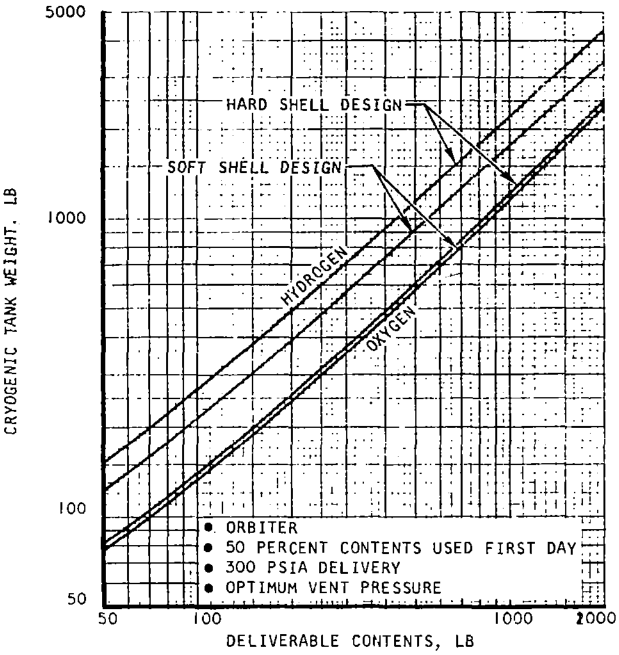

Figure 19. Cryogenic Tank Weight vs Deliverable Contents

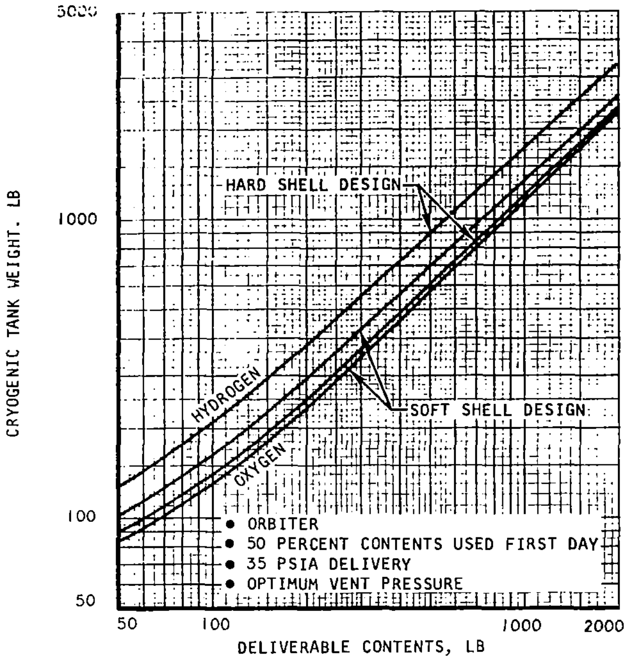

Figure 20. Cryogenic Tank Weight vs Deliverable Contents

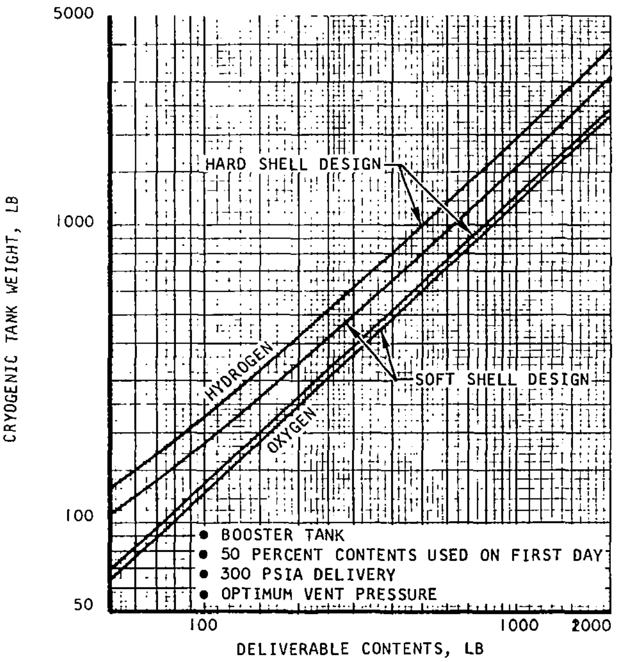

Figure 21. Cryogenic Tank Weight vs Deliverable Contents

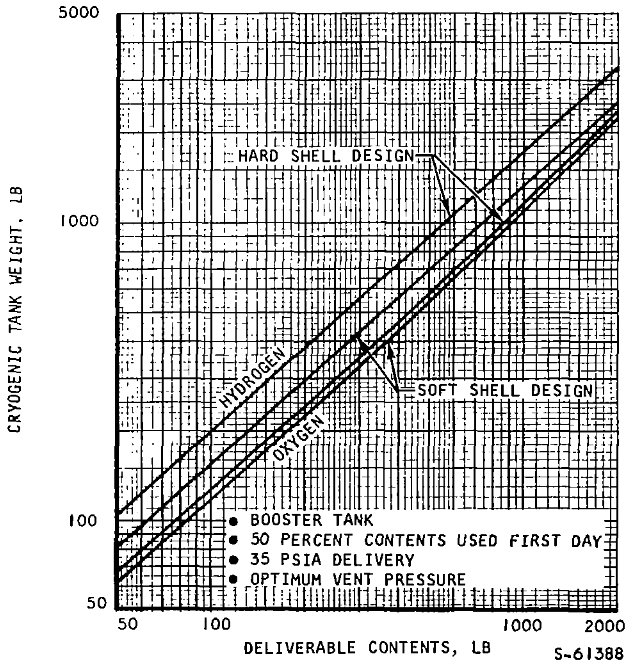

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Figure 22. Cryogenic Tank Weight vs Deliverable Contents

APPENDIX D

HEAT EXCHANGERS

INTRODUCTION

This appendix describes the design studies of the five heat exchangers used in the preferred propellant conditioning system, the recuperative system with recycle. Figure I shows the relative system locations of these units.

DESIGN CONSIDERATIONS

When making a optimization design study, the many factors to be considered include:

Type of heat exchanger--plate fin or tubular, with or without buffer zone

Service life and safety requirements--including corrosion and fouling considerations

Maintainability

Material

Type of tubing--plain tube, finned tube or dimpled tube

Type of surface matrix for plate fin and outside the tubes

Size and thickness of plate fin and tubing

Cost--construction cost, development cost and maintenance cost

Unit weight and volume

Shape of unit--cylindrical shell or box type shell, dimensional ratios, and dimensional limitations, if any

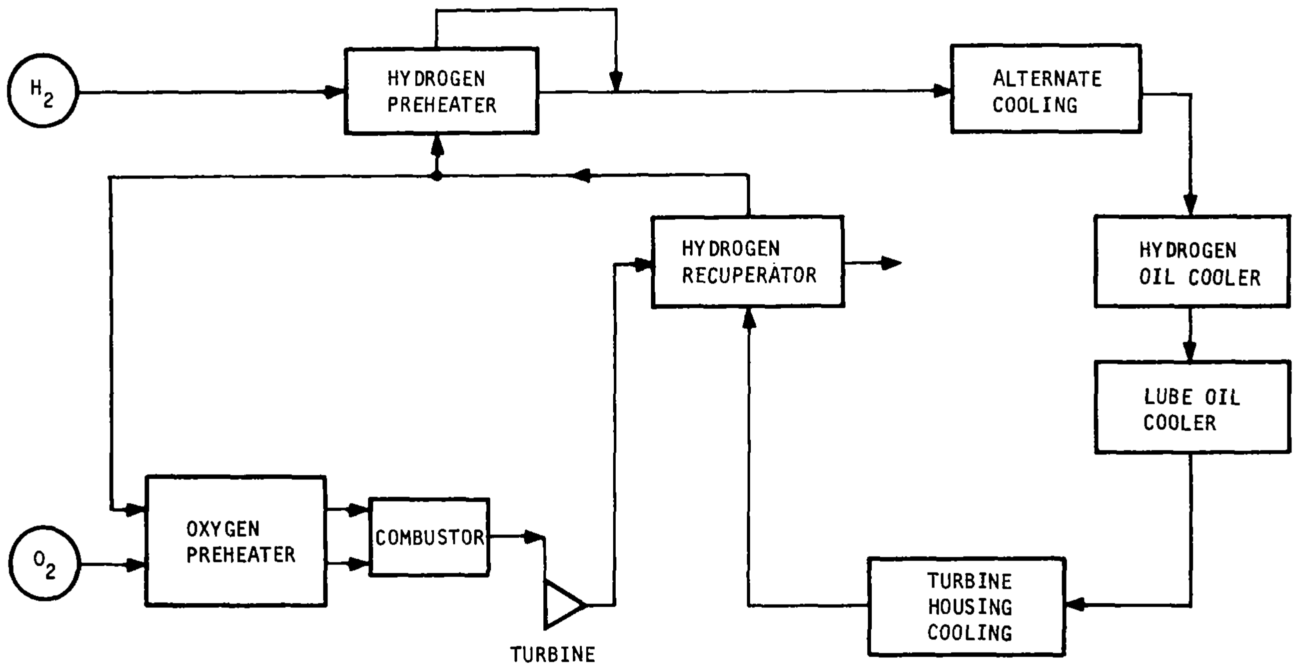
Flow arrangement--cross flow, cross counter or cross parallel flow

Location of the fluids--hot fluid in shell or cold fluid in shell

Number of passes--hot side and cold side

Pressure drop allowance and effectiveness--a tradeoff consideration for the system; optimizing improved system performance vs heat exchanger weight

It is almost impossible to optimize every factor listed above. Many of these factors must be predetermined when making parametric studies.



HEAT EXCHANGER	OXYGEN PREHEATER	HYDROGEN PREHEATER	HYDRAULIC OIL COOLER	LUBE OIL COOLER	HYDROGEN RECUPERATOR
HOT SIDE FLOW	HYDROGEN AT 1200 PSIA	HYDROGEN AT 1200 PSIA	MIL-H-5606 HYDRAULIC OIL	MIL-L-7808 LUBE OIL	H_2 - H_2O HOT GAS (O/F=0.65)
COLD SIDE FLOW	OXYGEN AT 1250 PSIA	HYDROGEN AT 1200 PSIA	HYDROGEN AT 1200 PSIA	HYDROGEN AT 1200 PSIA	HYDROGEN AT 1200 PSIA
HOT SIDE EFFECTIVENESS	0.036	0.656	0.04	0.649	0.917
COLD SIDE EFFECTIVENESS	0.873	0.274	0.49	0.255	0.589
HEAT TRANSFERED $(\frac{BTU}{MIN})$	474.0	4482.0	2562.0	510.0	10866.0

Figure 1. Schematic Diagram Showing the Relative System Locations of the Heat Exchangers Studied

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Plate fin heat exchanger designs were excluded from consideration because each of the system heat exchangers has at least one high pressure fluid stream. The plate-fin heat exchanger is usually optimum for low pressure service only.

Most of the APU heat exchangers will see large temperature gradients, particularly during startup. However the shell-and-tube heat exchangers are well adapted to these gradients. The selected tubing can take pressures of 600 psid with stress of only 3750 psi. By placing expansion bends in the tubes, the differential deflection between the tubes and the shell can be accommodated by slight flexing of the tube.

The service life and safety requirements determine whether a buffer zone is needed and also partially determine material and thickness of the parts. It is assumed that a buffer zone is not required for any of the heat exchangers. Dimpled stainless steel tubing with a 0.1-in. OD, 0.008-in. wall thickness is used for all heat exchangers except the oxygen preheater which uses plain tubing. Shell side surface matrix (tube spacing and pitch, in-lined or staggered) is so chosen that good dimensional ratios and minimum unit weight are obtained for the desired pressure drops.

PARAMETRIC STUDIES

An AiResearch tubular heat exchanger design program (H0424A) was used for the design study. This program takes the physical property data of the fluids, the tested friction factor and Colburn modulus (F and J curves) data, and the given problem statements. The program iterates to a solution by using the physical properties of the fluids at average film temperatures in the heat exchanger.

Figures 2, 3, and 4 present typical parametric studies performed when sizing these heat exchangers. Figure 2 shows the number of tube passes vs heat exchanger core weight. Figures 3 and 4 show the effect of design point pressure drops on the hydrogen recuperator weight when the number of passes are selected in Figure 2. Pressure drops and number of passes have been optimized in terms of unit weight and volume for all the heat exchangers.

Figure 5 shows weight increase as a function of effectiveness for the oxygen preheater. The selection of the design point effectiveness is a tradeoff consideration for the system performance. Since the oxygen preheater is a small unit, with a small weight increase, high effectiveness (thus better combustor performance) is selected as a design point condition.

HEAT EXCHANGER DESCRIPTION

Oxygen Preheater

This is a 6-pass cross-counterflow shell and tube heat exchanger with hot hydrogen inside the tubes and cold oxygen outside the tubes. Plain tubes are used instead of dimpled ones for this heat exchanger to ensure high reliability. A buffered heat exchanger could be used if even higher safety precaution is required.

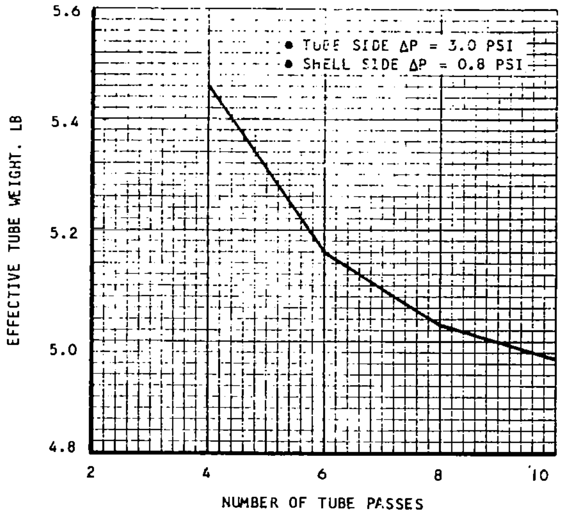

Figure 2. Parametric Study of Heat Exchanger Weight vs Number of Tube Passes for Hydrogen Recuperator

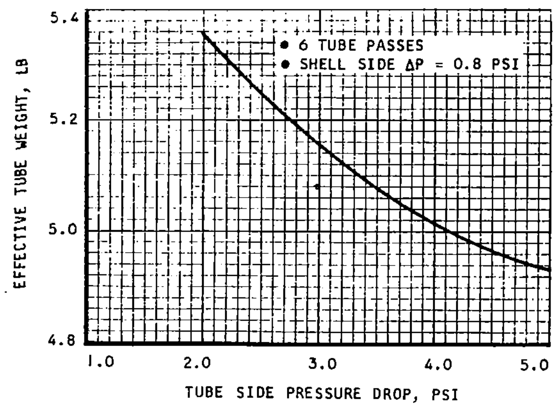

Figure 4. Parametric Study of Heat Exchanger Weight vs Tube Side Pressure Drop for Hydrogen Recuperator

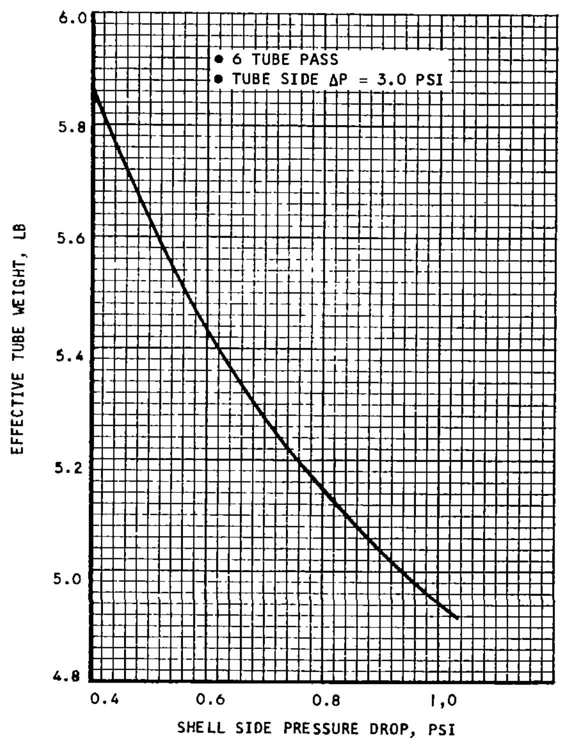

Figure 3. Parametric Study of Heat Exchanger Weight vs Shell Side Pressure Drop for Hydrogen Recuperator

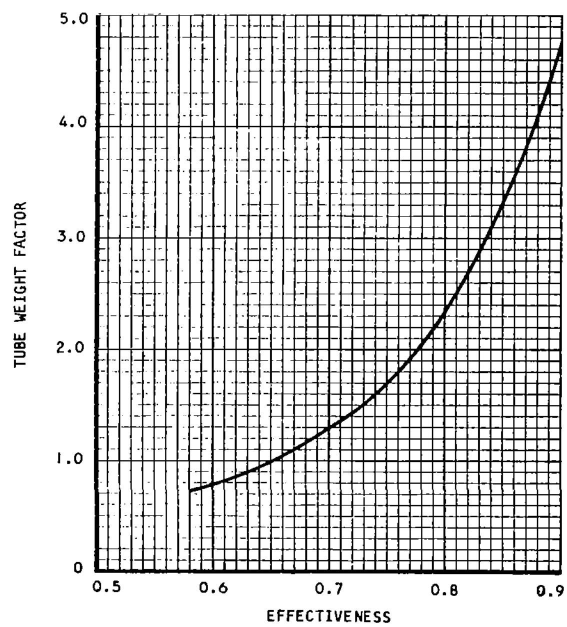

Figure 5. Tube Weight vs Effectiveness for Oxygen Preheater

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This heat exchanger is designed for the following operating conditions:

	Hot Fluid (H_2 at 1200 psi)	Cold Fluid (O_2 at 1250 psi)
Flow, lb/min	3.6	2.16
Inlet temperature, ^0R	1233.0	193.0
Outlet temperature, ^0R	1195.2	1100.9
Inlet pressure, psia	1200.0	1250.0
Pressure drop, psi	≤ 1.0	≤ 5.0
Effectiveness	0.036	0.873
Heat transferred, Btu/min	474.0	

Hydrogen Preheater

This is a 2-pass cross-parallel shell and tube heat exchanger with hot hydrogen inside the tubes and cold hydrogen outside the tubes. This heat exchanger is designed to equalize the temperatures of the incoming and recycle hydrogen flow streams to ensure efficient ejector performance. A cross-parallel flow arrangement ensures equalization at all off-design conditions.

The design point conditions for this heat exchanger are:

	Hot Fluid (H_2 at 1200 psia)	Cold Fluid (H_2 at 1200 psia)
Flow, lb/min	1.68	3.6
Inlet temperature, ^0R	1233.0	72.0
Outlet temperature, ^0R	471.4	390.0
Inlet pressure, psia	1200.0	1200.0
Pressure drop, psi	≤ 0.1	≤ 1.0
Effectiveness	0.656	0.274
Heat transferred, Btu/min	4482.0	

Hydraulic Oil Cooler

This is a one-one pass cross-flow shell and tube or box type tubular heat exchanger with hydrogen inside the tubes and hydraulic oil outside the tubes. Because of the low effectiveness, the cross flow arrangement is selected to obtain better pressure drop and dimensional ratio combinations. This heat exchanger is designed for the following operating conditions:

	Hot Fluid (MIL-H-5606 Hydraulic Oil)	Cold Fluid (H_2 at 1200 psi)
Flow, lb/min	546.0	5.52
Inlet temperature, ^0R	720.5	460.0
Outlet temperature, ^0R	710.0	587.7
Inlet pressure, psia	200.0	1200.0
Pressure drop, psi	$\leq$ 2.5	$\leq$ 1.5
Effectiveness	0.04	0.49
Heat transferred, Btu/min	2562.0	

Lube Oil Cooler

This is a 6-pass cross counterflow shell and tube heat exchanger with hydrogen inside the tubes and lube oil outside the tubes. The design point conditions for this heat exchanger are:

	Hot Fluid (MIL-L-7808 Lube Oil)	Cold Fluid (H_2 at 1200 psi)
Flow, lb/min	15.0	5.28
Inlet temperature, ^0R	660.0	554.0
Outlet temperature, ^0R	591.2	581.0
Inlet pressure, psia	200.0	1200.0
Pressure drop, psi	$\leq$ 4.0	$\leq$ 1.0
Effectiveness	0.649	0.255
Heat transferred, Btu/min	510.0	

Hydrogen Recuperator

This is a 6-tube-pass, 1-shell-pass cross-counterflow box type tubular heat exchanger with hydrogen inside the tubes and hot combustion products outside the tubes. Since the pressure drop on the shell side is more important than that on the tube side, this heat exchanger is designed to minimize the shell side pressure drop with a box type construction and multipass on the tube side. This heat exchanger is designed for the following operating conditions:

	Hot Fluid (H_2-H_2O Hot Gas, O/F = 0.65)	Cold Fluid (H_2 at 1200 psia)
Flow, lb/min	6.6	6.0
Inlet temperature, ^{o}R	1500.0	628.0
Outlet temperature, ^{o}R	700.0	1141.6
Inlet pressure, psia	20.0	1200.0
Pressure drop, psi	$\leq$ 1.0	$\leq$ 5.0
Effectiveness	0.917	0.589
Heat transferred, Btu/min	10866.0	

HEAT EXCHANGER PERFORMANCE

Heat exchanger performance is computed with AiResearch Tubular Heat Exchanger Performance Computer Program (H0415P). The performance curves including ηhA and $\rho \Delta p$ vs $\dot{w}$ of each flow for every heat exchanger except the oxygen preheater are presented in Figures 6 to 9. These curves are plotted at given inlet temperatures of the fluids and at the given flow rate of the other fluid. This means that these curves are obtained at the design point film temperatures. When using these curves at conditions other than specified, necessary corrections must be made such that corrected flow rates are used.

FUTURE HEAT EXCHANGER DESIGN ANALYSES

The tubular heat exchanger design and performance programs (H0424A and H0415P) use fluid property values at arithmetic average film temperatures and the tested f and J factor curves.

The physical properties of hydrogen and oxygen at the working pressures are strong functions of the temperature and the slopes of these functions change irregularly (especially at temperatures below 500^{o}R). The conventional method of using arithmetic average film temperatures to determine fluid properties becomes doubtful because the property values at this temperature may be far different from the actual mean property values. This deviation

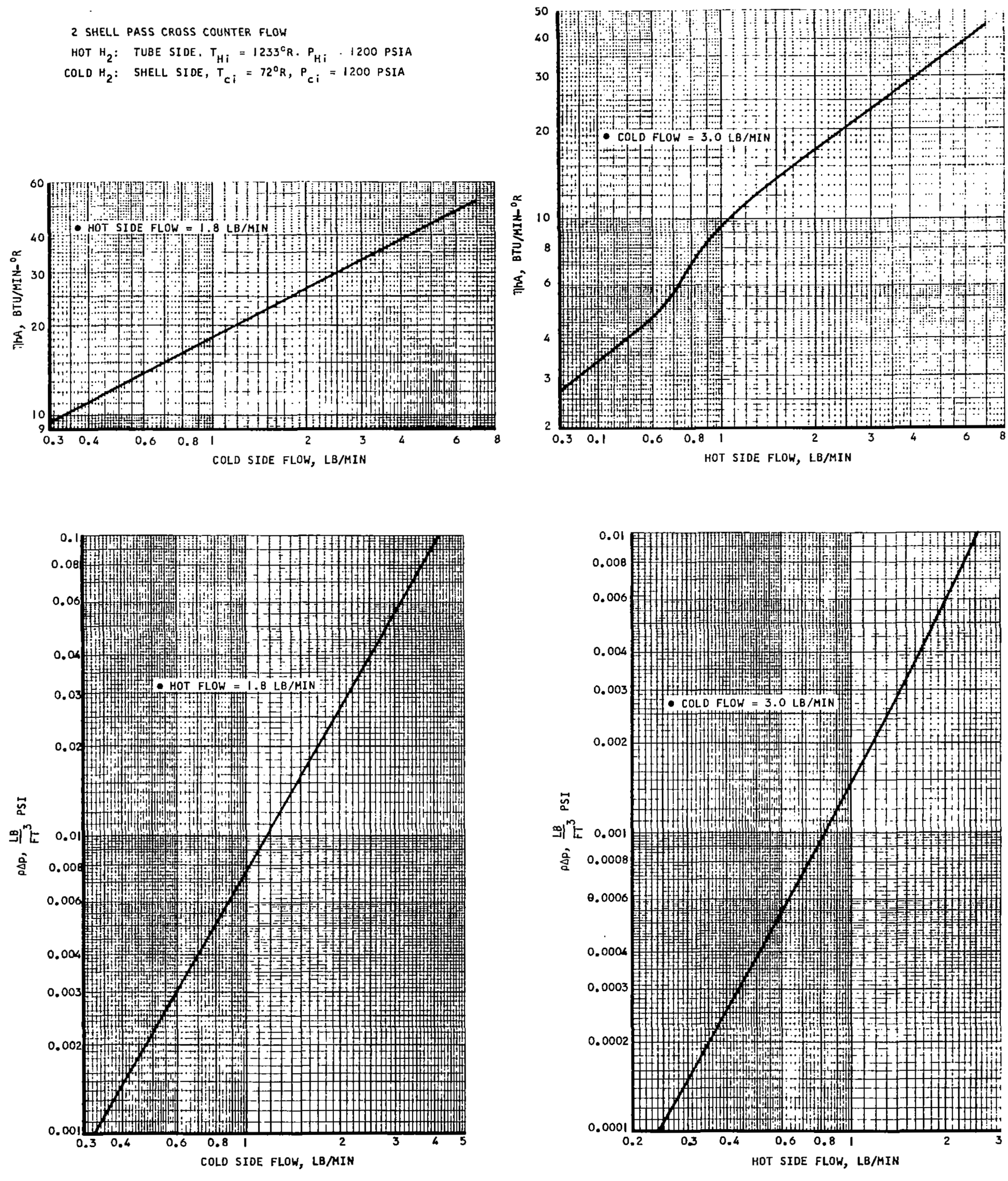


Figure 6. Hydrogen Preheater Performance Curves

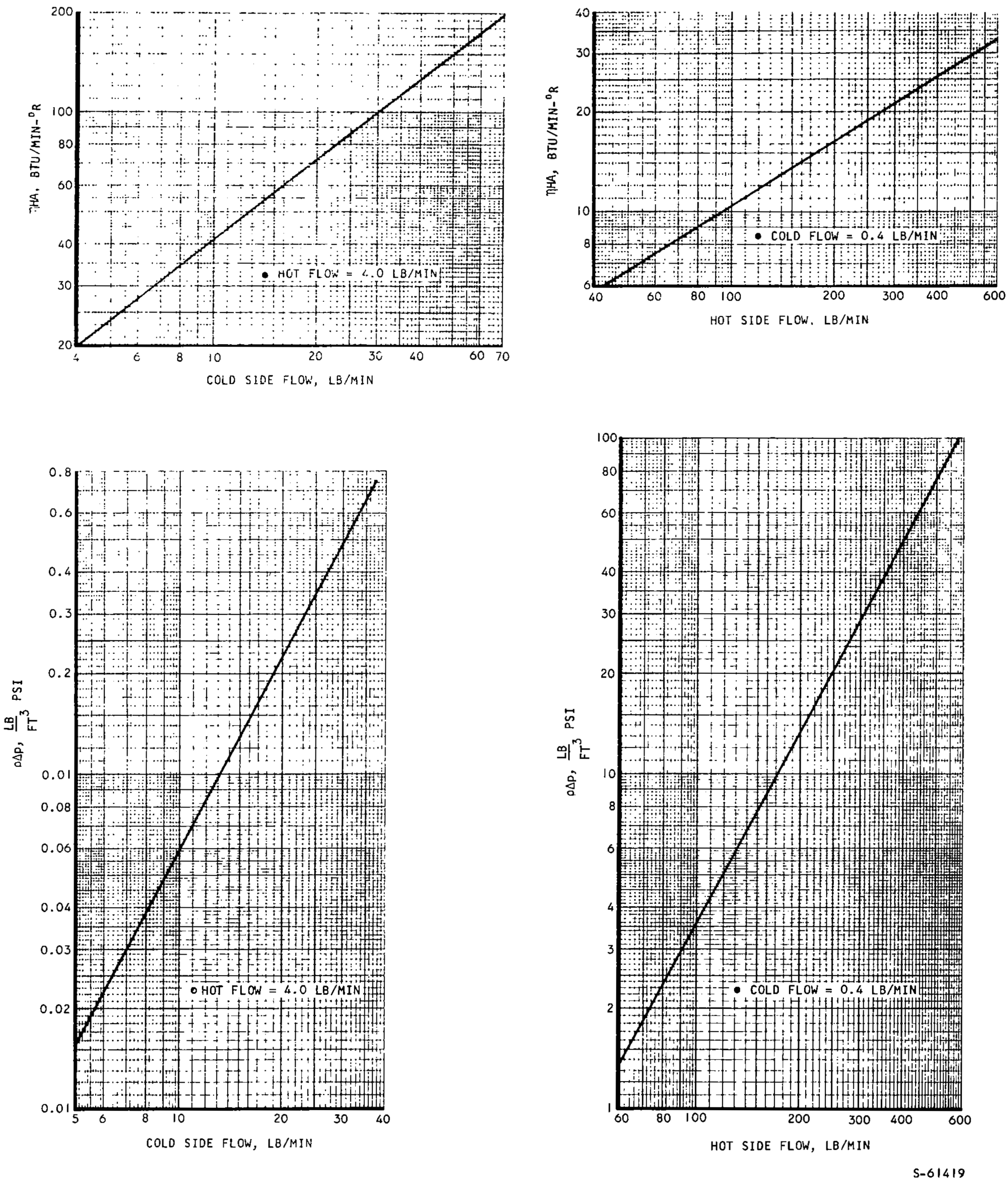


Figure 7. Hydrualic Oil Cooler Performance Curves

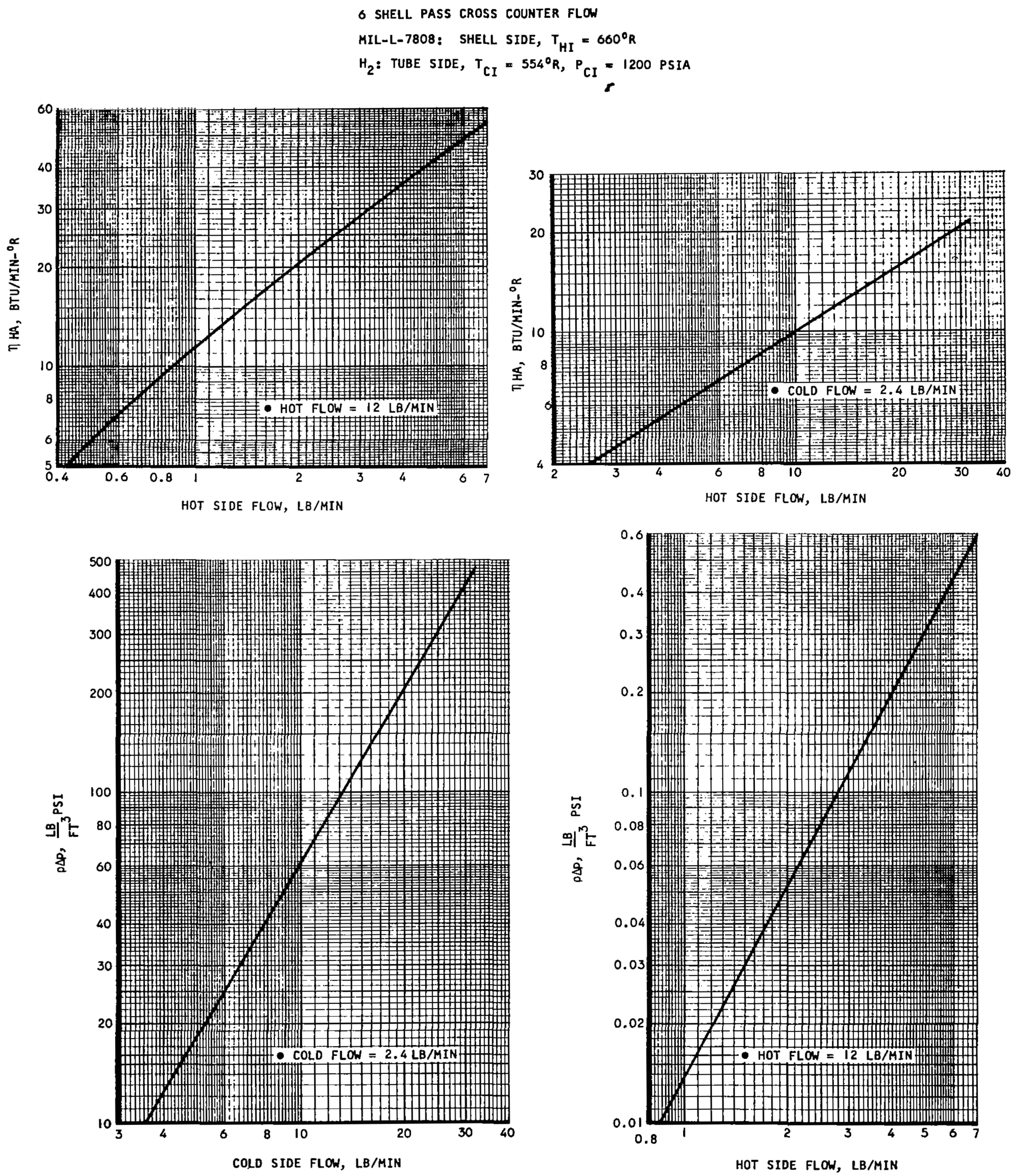
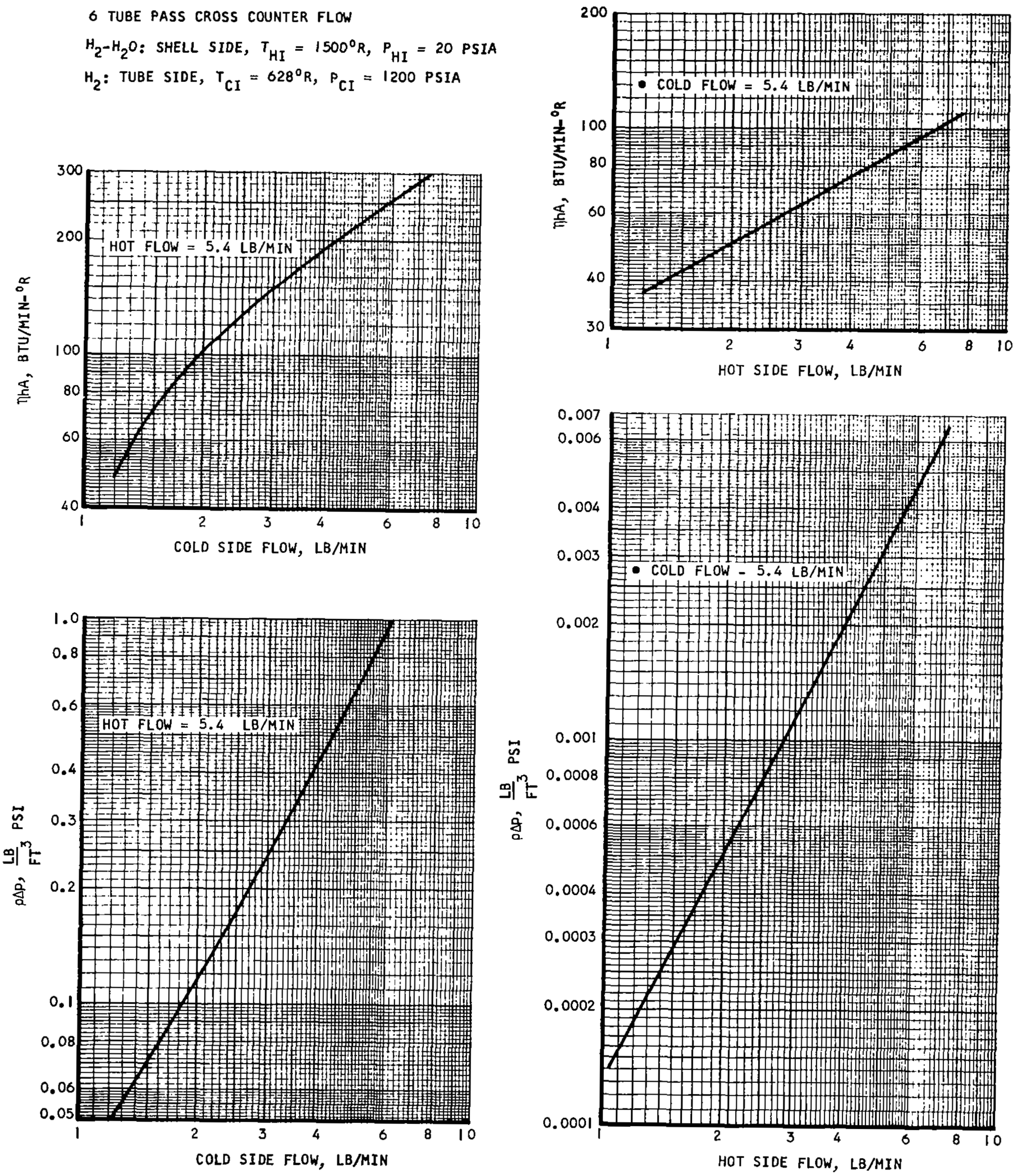


Figure 8. Lube Oil Cooler Performance Curves

S-61404



S-61405

Figure 9. Hydrogen Recuperator Performance Curves

may or may not cause error in the design work, and if there is an error introduced, the magnitude of this error is not easily predictable. This possible error will not effect the overall system performance study significantly since the total heat quantities on most of the units will be unaltered. The probable result will be an inaccurate sizing of the heat exchangers which can be corrected at a later date.

Work is being done on advanced performance prediction techniques that will eliminate the possible error. A nodal heat exchanger program is being developed so that the internal heat exchanger performance can be studied. Each node of the heat exchanger is formed as a small heat exchanger. The temperature spans across these nodes are small enough that the fluid property variation will be insignificant and the conventional method can be applied. This advanced program will be made for final design of all system heat exchangers.

Because the oxygen temperature change in the oxygen preheater is large, the property values change severely. This change makes performance curves obtained from normal calculation methods incorrect. A fixed performance is assumed at this time when making system studies. This fixed performance assumption does not effect the overall system performance since the total combustor inlet enthalpy is unaffected.

APPENDIX E

CYCLE PERFORMANCE ANALYSIS

INTRODUCTION

This appendix describes a computer program used to calculate APU performance at various operating conditions. The program outputs the component performance (efficiency, effectiveness, etc.) and the state points throughout the system. These data establish the feasibility of the selected cycle configuration and provide a basis for further refining the component design problem statements. Additionally, by operating the program at a series of points, it is possible to establish the APU performance solely as a function of the output power required and the ambient pressure. These performance data can then be used to establish the propellant requirements for an entire mission (as described in Appendix F).

SYSTEM CONFIGURATION

Figure 1 shows the system configuration used as the basis for the program logic. This configuration uses cryogenic pumps to compress the hydrogen and oxygen to the required pressure. The pumps are electrically driven (0.85 motor efficiency used) with power being supplied from the generator on the gearbox. Thus, it is necessary to use a slightly larger generator with this cycle than with one having high-pressure tankage. Although this appendix describes the program version used for the pumped cryogenic system, minor modifications have been made to create program versions applicable to all the hydrogen-oxygen systems considered in the study.

Figures 2 through 8 present the component performance data used in the program. Figure 2 shows the hydrogen pump performance and Figure 3 gives the oxygen pump performance. (These are for an APU system operating at a maximum pressure of about 1250 psia.) Figure 4 shows the ejector performance. The heat loads (power losses due to component inefficiencies or cooling needs) are given in Figures 5 through 8 for the turbine, the hydraulic pumps, the generator, and the gearbox. Additionally, it is assumed that 15 percent of the net output hydraulic power is returned to the APU system in the form of waste heat (hydraulic fluid temperature rise). Performance maps for the hydrogen preheater, the hydraulic fluid heat exchanger, the lube oil heat exchanger, the recuperator, and the oxygen preheater are given in Appendix D.

The turbine performance map consists of a plot of turbine efficiency as a function of the output power, the discharge pressure, and the O/F ratio. Appendix A shows the various turbine performance maps that have been used in this cycle analysis program. Assessing system performance in terms of the turbine design point (discharge pressure and output power) has made it possible to optimize the system peak pressure. For systems having peak pressures other than 1250 psia, the pump performance maps and the system line pressure drop relationships are also altered.

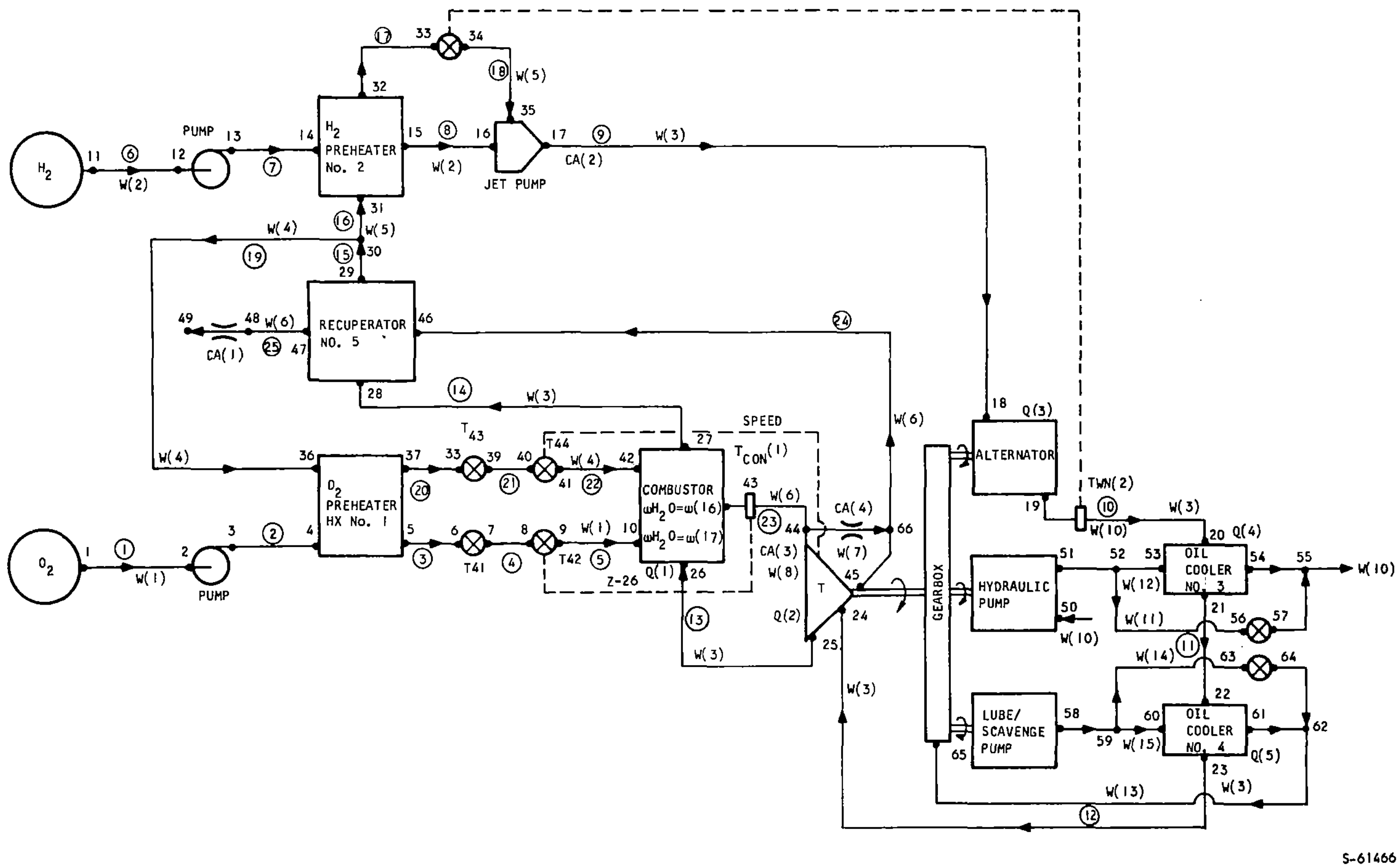


Figure 1. Space Shuttle APU System Configuration

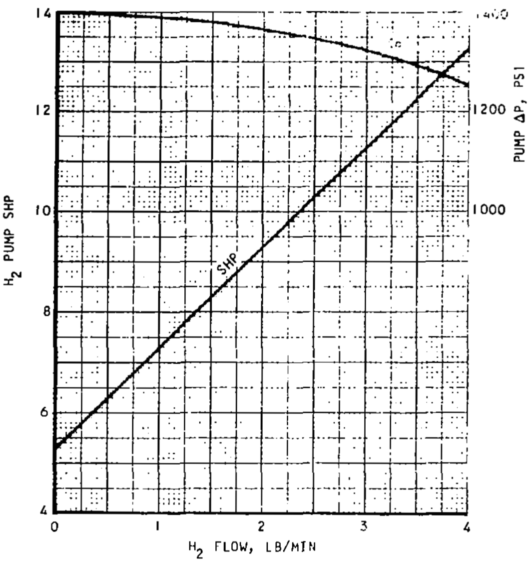


Figure 2. Hydraulic Pump Performance Maps for 1250 psia Delivery

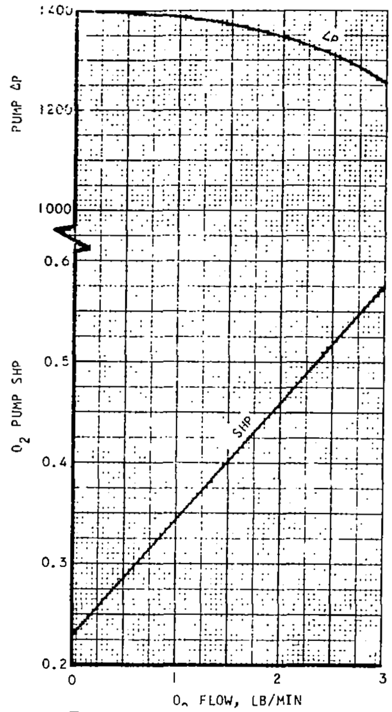


Figure 3. Oxygen Pump Performance Maps for 1250 psia Delivery

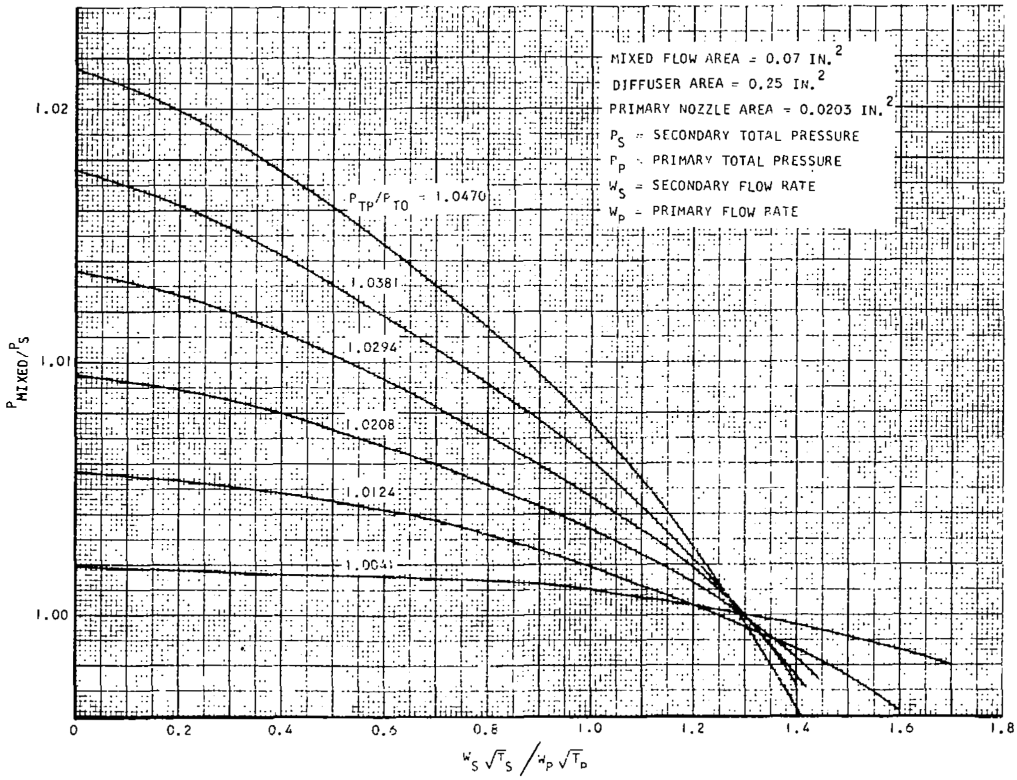


Figure 4. Ejector Performance Map

S-61464

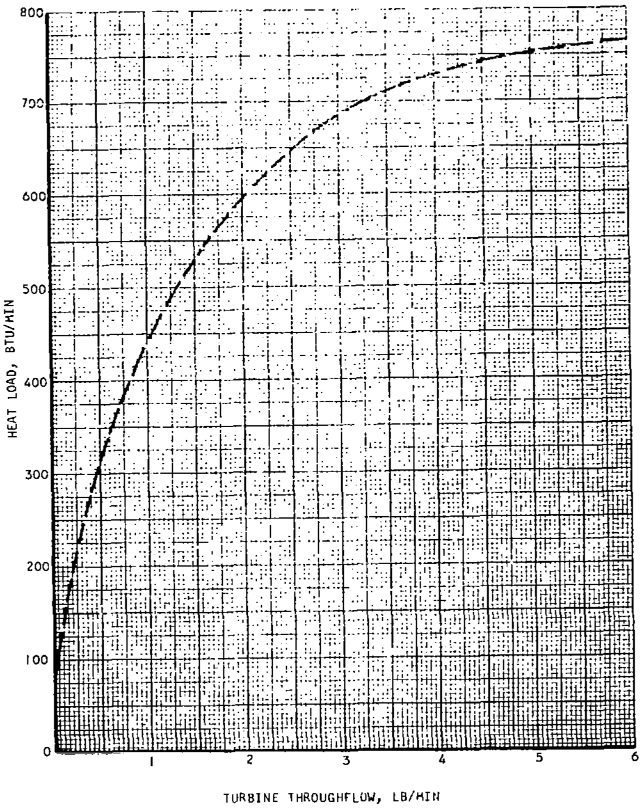


Figure 5. Turbine Cooling Load vs Throughflow

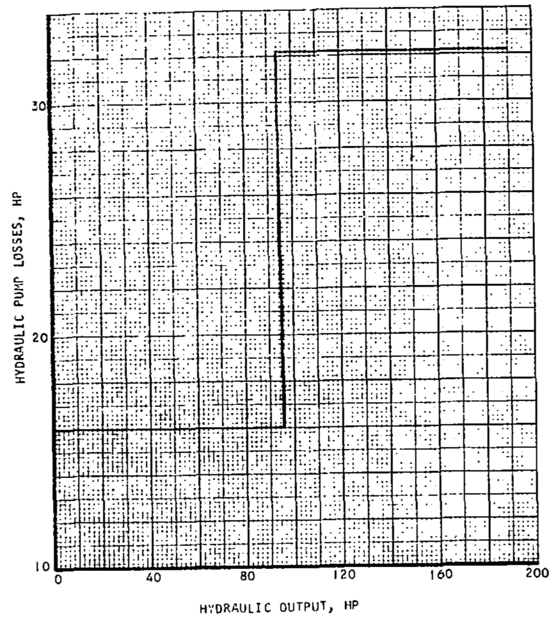


Figure 6. Hydraulic Pump Heat Generation

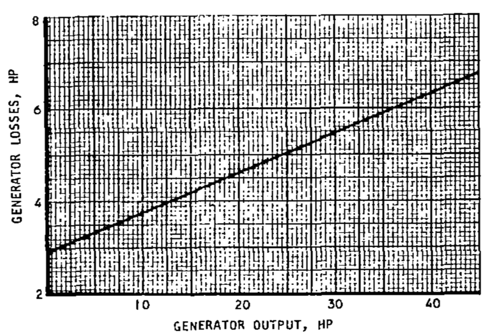


Figure 7. Generator Heat Generation

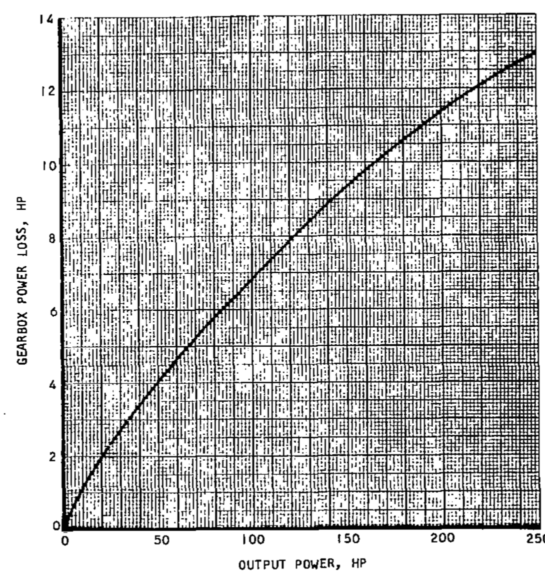


Figure 8. Gearbox Heat Generation

S-61465

Table I shows the line pressure drops (in terms of the density-pressure drop product divided by the square of the line flow) for a system operating at 1250 psia. For lower pressure systems where pressure drop is more critical, it is necessary to use larger line sizes. Thus, lowering the system pressure results in an increase in the system line weight.

CALCULATION PROCEDURE

The procedure used to determine the system performance is one originally developed for prediction of aircraft environmental control system performance. It consists of an iterative convergence to a set of state points satisfying the component performance capabilities and the required system boundary conditions (output power, ambient pressure, etc.). Initially, the first guess system flows are obtained by calculating the approximate output power (since the hydrogen and oxygen pumps are driven from the gearbox, an exact power calculation is not possible until the system flows are established), and by using the flow relationship through the turbine nozzle and the relationship between turbine throughflow, pressure ratio, O/F ratio, and turbine output work to calculate approximate flows for the guessed O/F ratio. Then a series of five separate, nested convergence loops are applied to make the system meet the various boundary conditions imposed upon it. The five convergence loops are as follows:

(1) Turbine Discharge Pressure--Iterate to make the turbine discharge pressure, less the line losses in the exhaust line, equal to the ambient pressure.

(2) Generator Cooling Discharge Temperature--Iterate by altering the amount of flow recycled from the recuperator to the ejector to make the fluid temperature at the generator discharge equal to 460^{0}R (this provides an ideal heat rejection temperature for the hydraulic and lube oil heat exchangers).

(3) Turbine Nozzle Inlet Pressure--Iterate to make turbine flowthrough equal to first-guess flow required for power; flow can be reduced by lowering the nozzle inlet pressure, which occurs when the control valves in front of the combustor are partially closed. If the calculated flow with the throttle valves full open is less than the first-guess required flow, then the program sets the required flow equal to the calculated flow (which, as will be established by iteration loop 5, probably means that the system cannot provide the required output power).

(4) Turbine Inlet Temperature--Adjust the O/F ratio to make the combustor discharge temperature equal to the design turbine inlet temperature (1800^{0}F).

(5) Power Balance--Adjust the propellant flow to provide the required output power (if throttle valves are wide open and power generated is less than required, then printout to indicate this).

TABLE I

DUCT PRESSURE DROP FACTORS FOR 1250 PSIA SYSTEM

Fluid	Duct*	Diameter, in.	Length, in.	No. of 90 Deg Turns	Nominal Pressure, psia	Nominal Temperature, °R	$Z \times 10^4$ **
Oxygen	1-2	0.5	240	4	45	193	1,795
	3-4	0.25	60	1	1260	193	13,330
	5-6	0.25	12	1	1260	540	3,510
	7-8	0.25	12	1	1255	540	3,510
	9-10	0.25	24	2	1250	540	7,000
Hydrogen	11-12	0.75	240	4	40	72	253
	13-14	0.375	60	1	1280	72	1,840
	15-16	0.375	24	1	1275	390	851
	17-18	0.625	60	1	1275	417	228
	19-20	0.625	60	2	1275	427	228
	21-22	0.625	24	1	1275	514	102
	23-24	0.625	60	3	1275	540	256
	25-26	0.625	24	1	1270	562	102
	27-28	0.625	24	2	1270	573	125
	29-30	0.75	12	1	1265	1184	22.8
	30-31	0.75	12	1	1265	1184	22.8
	32-33	0.375	12	1	1265	475	53
	34-35	0.375	12	1	1265	475	53
	30-36	0.75	120	3	1260	1184	140
	37-38	0.5	12	1	1255	1184	141
	39-40	0.5	12	1	1255	1184	141
	41-42	0.5	24	2	1250	1184	285
Hydrogen-Steam Exhaust	43-44	0.75	12	2	1250	2260	35.6
	45-46	2.50	24	1	16	1570	0.151
	47-48	2.50	240	4	15.5	700	0.9

*Duct numbers refer to duct locations shown in Figure 1.
**Z factor is (density x pressure drop)/(flow squared) where density is in lb per cu ft, pressure drop is in psi and flow is in lb per min.

In each of these iterations, only a single variable is being altered. The other variables are held constant. Thus, in iterative loop 1 for example, the propellant flow remains constant as the discharge pressure is varied to obtain compatibility with the ambient pressure. Consequently, each iteration on loop 2 requires a series of iterations on loop 1 in order to obtain discharge pressure convergence. Similarly, each iteration on loop 5 requires iteration to convergence on all of the prior iterative loops.

FLUID PROPERTIES

Because the APU system operates with fluids at cryogenic temperatures, it is not possible to use perfect gas laws. Therefore, it becomes necessary to provide the computer program with maps of the various fluid properties (density, pressure, temperature, and enthalpy data are required). Figures 9 and 10 show typical fluid property maps. The data sources for the fluid properties used in the cycle performance program are as follows:

Oxygen--"The Thermodynamic Properties of Oxygen," by Richard Byron Stewart, a PhD thesis at the University of Iowa, 1966.

Hydrogen Below 100^{0}K--A computer program by Hans Roder of NBS (the source for the data of NBS Nonograph 94)

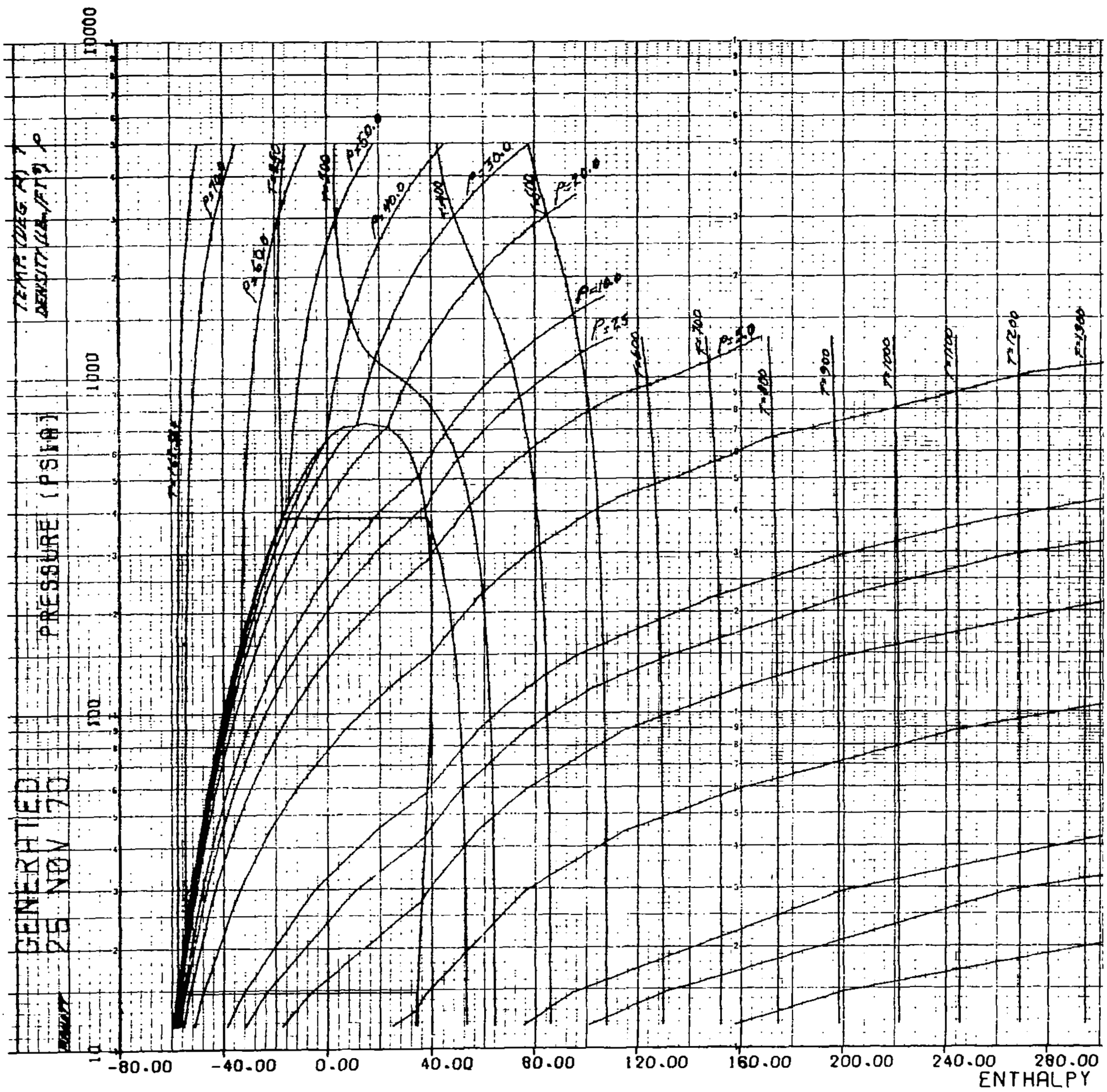
Hydrogen Above 100^{0}K--A computer program by McCarty of NBS

Water--"Thermodynamic Properties of Steam," by Keenan and Keyes, 1959 printing

Data retrieval of the fluid property information is based on a map reading program developed as a result of AiResearch IR and D activity.

OUTPUT INFORMATION

Figure 11 shows a sample of the program output. The data are sufficient to specify the state points at each location in the system and to establish performance, both at the component and system level.



10000
1000
100
10
PRESSURE (PSIA)
GENERATED
25 NOV 70
-80.00
-40.00
0.00
40.00
80.00
120.00
160.00
200.00
240.00
280.00
ENTHALPY

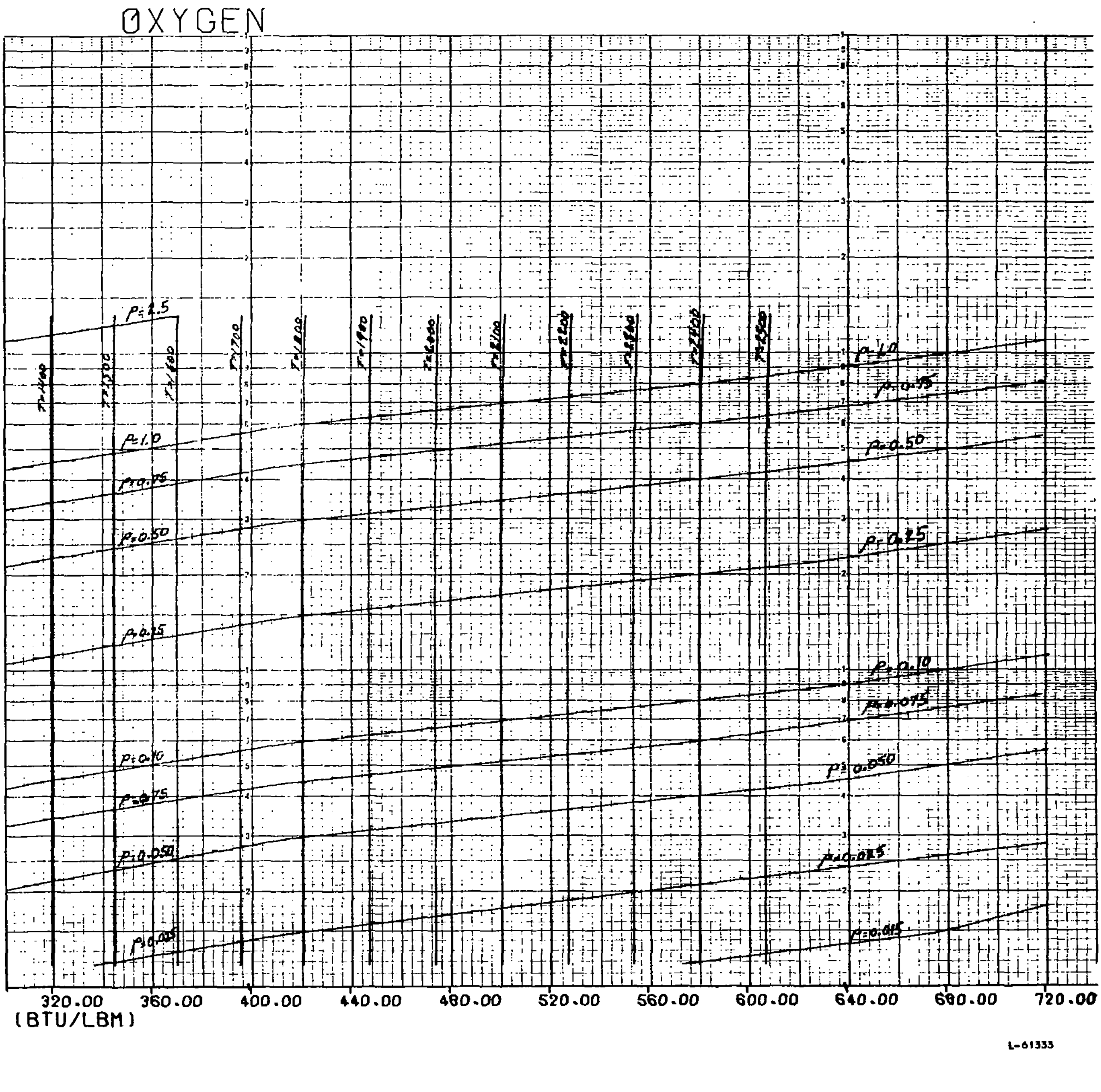
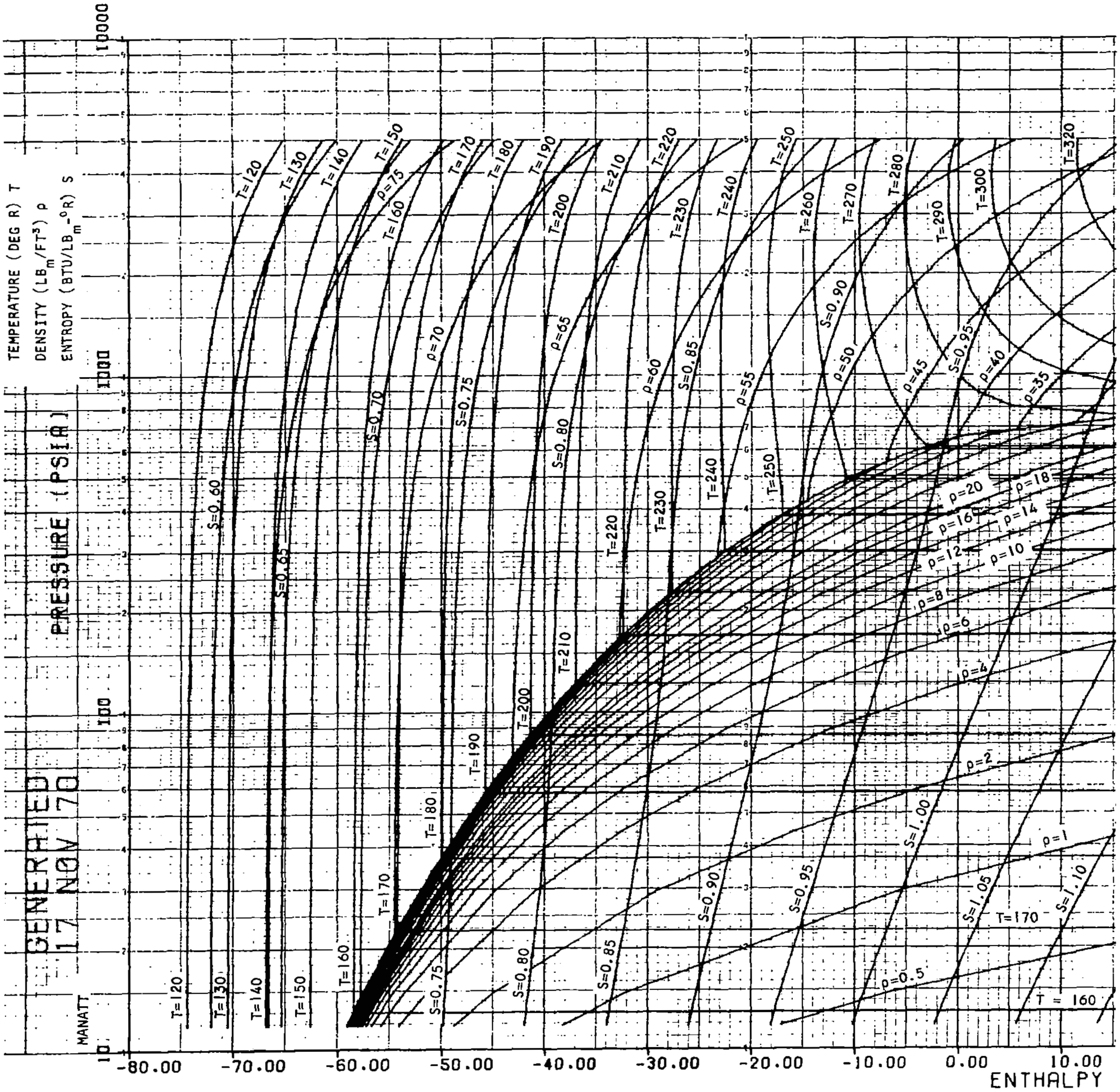


Figure 9. Oxygen Thermodynamic Data
(Pressure from 1 to 5000 psia,
Temperature from 162° to 2500°R)


TEMPERATURE (DEG R) T
DENSITY (LB$_m$/FT^3) ρ
ENTROPY (BTU/LB$_m$-°R) S
PRESSURE (PSIA)
GENERATED
17 NOV 70
MANATT
10
100
1000
10000
-80.00
-70.00
-60.00
-50.00
-40.00
-30.00
-20.00
-10.00
0.00
10.00
ENTHALPY

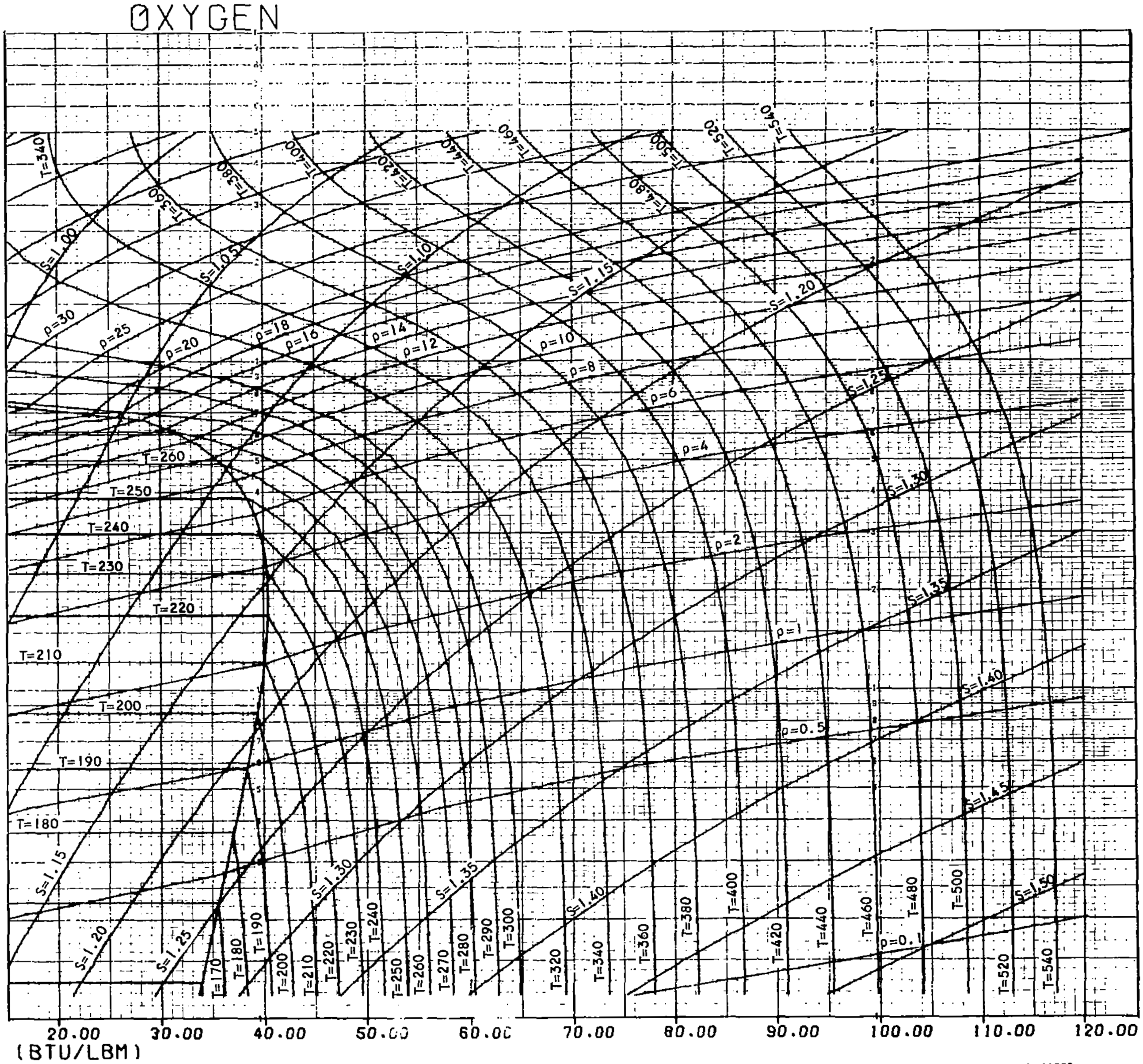


Figure 10. Low Temperature Oxygen Thermodynamic Data (Expanded Scale)

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SPACE SHUTTLE H2-02 RECUPERATIVE APU SYSTEM PERFORMANCE  NO, 9 TURBINE
                  DECEMBER 7, 1970         12105152          PAGE 1 OF 2
 • CONDITION •  CASE 6

 • UNITS • AREA=SQ IN, H=BTU/LB, P=PSIA, Q=BTU/MIN, T=DEG R, W=LB/MIN

                         I N P U T   D A T A
                   DUCT PRESSURE LOSS COEFFICIENTS *10E+4
   1805,0    451,0    141,0    141,0    282,0     63,7     16,0
      8,8     20,1     20,1      8,8     24,0      8,8     12,9
      6,4     22,7    141,0    141,0     13,0      6,4      6,4
   12,900   10,400     ,151     ,900

   SFT     TUR ON F  SF JP P   SF JP W   TUR OBCA  JP PRICA  TURB NOZ
  1,000      ,000     2,000     1,000     5,000     ,0203     ,1178
TUR LKCA  ETA COMB  LUBE P HP  ETA LOPP  T TUR IN  T ALTE O
  ,0002      ,98       ,20       ,50     2260,0     460,0

                        POINT INPUT DATA
                  HYDRA HP    ELECT HP    P AMB
                   200,00      10,00      14,70

                        O U T P U T   D A T A
                     DRIVE POWER                           HP LOSS
   HYD PUMP  ALTERTOR  LUB PUMP   H2 PUMP   O2 PUMP  GEAR BOX
    232,00     16,98       ,40      2,40       ,13     12,88

   TURBINE OUTPUT POWER =  262,17    O/F =  ,632    SPC = 1,781

   PROPELLANT                    HYDROGEN       OXYGEN
   FLOW RATE                      4,77           3,01
   PRESSURE                       35,00          35,00
   TEMPERATURE                    41,60         176,60
   ENTHALPY                      -96,6          -51,5
                                  PUMP INFORMATION
                                 HYDROGEN       OXYGEN
   PRESSURE IN                    35,0           35,0
   PRESSURE OUT                  384,5          384,3
   TEMPERATURE IN                 41,6          176,6
   TEMPERATURE OUT                46,4          179,9
   ENTHALPY IN                   -96,6          -51,5
   ENTHALPY OUT                  -75,3          -49,6
   EFFICIENCY                      ,72            ,51

                      O2 PREHEATER INFORMATION
           FLOW   PRE IN  P OUT    T IN    T OUT    H IN   H OUT   EFF
COLD SIDE  3,01   384,3   384,3   179,9  1202,2   -49,6   270,0 1,000
HOT SIDE   4,77   334,7   334,4  1202,3  1144,2  4119,5  3917,5  ,057

                                  CONTROL VALVE
   IS IT IN CONTROL              YES            YES
   PRESSURE IN                   333,9          384,0
   PRESSURE OUT                  300,7          300,7
   TEMPERATURE IN               1144,2         1202,2
```

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SPACE SHUTTLE H2-02 RECUPERATIVE APU SYSTEM PERFORMANCE  NO, 9 TURBINE
                  DECEMBER 7, 1970         12105152          PAGE 2 OF 2
 • CONDITION •  CASE 6

                        COMBUSTOR INFORMATION
   HYDROGEN FLOW              4,77     OXYGEN FLOW                3,01
   PRESSURE IN               300,1     PRESSURE OUT              297,0
   TEMPERATURE IN H2        1144,2     TEMPERATURE IN O2        1202,2
                     FLOW   P IN    P OUT    T IN    T OUT    H IN    H OUT
   COOLING SIDE      7,66   375,1   375,1   608,0   608,0  2049,8  2049,8
   HEAT REJECTED                ,0

                         TURBINE INFORMATION
   INLET FLOW                 7,78     SPECIFIC HEAT RATIO        1,359
   PRESSURE IN              295,5     PRESSURE OUT                18,2
   TEMPERATURE IN          2260,0     TEMPERATURE OUT           1634,5
   ENTHALPY IN             5317,1     ENTHALPY OUT              3886,3
   PRESSURE RATIO            16,25     EFFICIENCY                  ,531
                     FLOW   P IN    P OUT    T IN    T OUT    H IN    H OUT
   COOLING SIDE      7,66   375,5   375,5   578,7   608,0  1947,4  2049,8
   HEAT REJECTED               784,9

                       RECUPERATOR INFORMATION
           FLOW   PRE IN  P OUT    T IN    T OUT    H IN   H OUT   EFF
COLD SIDE  7,66   374,4   336,0   608,0  1202,3  2049,8  4119,5  ,579
HOT SIDE   7,78    17,9    15,7  1634,5   703,4  3886,2  1848,7  ,907
   • TURBINE EXHAUST OVERBOARD PRESSURE •      14,91
                      H2 PREHEATER INFORMATION
           FLOW   PRE IN  P OUT    T IN    T OUT    H IN   H OUT   EFF
COLD SIDE  4,77   384,5   384,5    46,4   411,2   -75,3  1341,8  ,316
HOT SIDE   2,89   334,9   334,8  1202,3   532,1  4119,5  1783,0  ,980

                        JET PUMP PERFORMANCE
                   FLOW PARA      FLOW       PRESSURE    TEMPERATURE
   PRIMARY JET        ,00         4,77       384,34      411,18
   SECONDARY JET      ,00         2,89       332,73      532,05
   RESULTANT                      7,66       384,34      455,94
   FLOW PARA RATIO          ,000    P PRI/P SEC                  ,0000
   JET PUMP RISE           ,0000

                       ALTERNATOR INFORMATION
                     FLOW   P IN    P OUT    T IN    T OUT    H IN    H OUT
   COOLING SIDE      7,66   383,6   383,6   455,9   462,0  1508,3  1530,5
   HEAT REJECTED              169,8
                   HYDRAULIC OIL COOLER INFORMATION
           FLOW   PRE IN  P OUT    T IN    T OUT    H IN   H OUT   EFF
COLD SIDE  7,66   382,8   382,6   462,0   557,6  1530,5  1873,9  ,433
HOT SIDE 544,00    30,0    30,0   682,9   673,8      ,0      ,0  ,041
   HEAT REJECTED             2630,5
                      LUBE OIL COOLER INFORMATION
           FLOW   PRE IN  P OUT    T IN    T OUT    H IN   H OUT   EFF
COLD SIDE  7,66   382,2   376,6   557,6   578,7  1873,8  1947,4  ,199
HOT SIDE  14,80    30,0    30,0   663,3   587,5      ,0      ,0  ,717
   HEAT REJECTED              563,5
```


Figure 11. Typical Cycle Performance Program Output

APPENDIX F

VEHICLE POWER PROFILE/ALTITUDE/PROPELLANT CONSUMPTION

INTRODUCTION

This appendix describes the method used to determine the APU performance during any specified vehicle mission. It also presents the mission power/altitude/time profiles supplied by NASA for the booster and orbiter vehicles.

INPUT INFORMATION

The computer program requires four inputs:

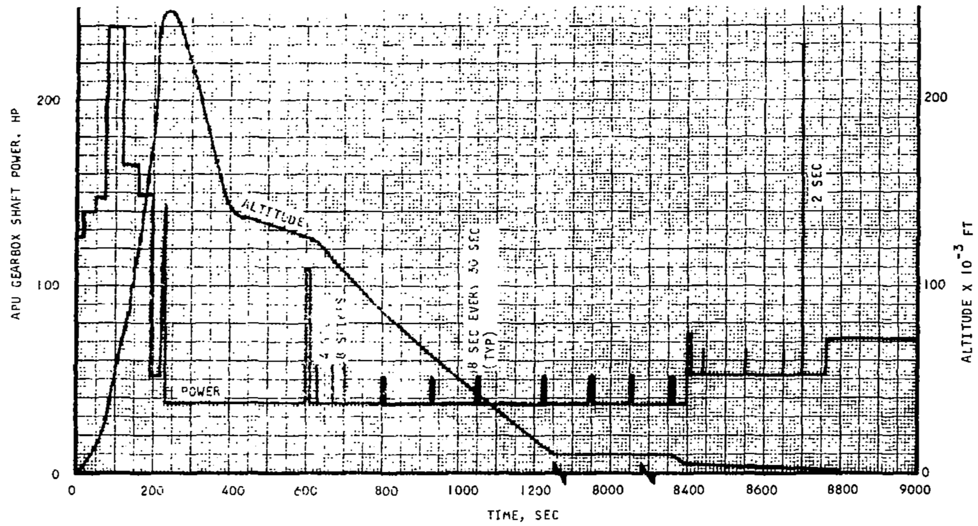
- A map of the APU propellant consumption as a function of the output power and the ambient pressure
- A map of the vehicle power requirements as a function of time
- A map of the vehicle altitude as a function of time
- A map of the ambient pressure as a function of altitude

APU Propellant Consumption Map

The APU performance can be specified solely as a function of the output power and the ambient pressure. Although there is a slight difference in the map depending upon the relative split between output hydraulic and electric power, the electric power level of 10 hp has been assumed as the basis for the performance map. This power level primarily effects the output power level at which it is necessary to operate with both hydraulic pumps pressurized (thus causing a step increase in the propellant consumption). Sample APU performance maps are shown in Section 3 through 5 for both the hydrogen and oxygen flow requirements. These sample maps are for APU's having turbines designed at various power outputs/discharge pressures and maximum inlet pressures. The maps were obtained by using the program described in Appendix E to establish performance at various power levels and ambient pressures.

Mission Profile and Power Requirements Maps

As with the APU performance maps, it is possible to place the mission profile and power requirements in a tabular form for input to the program. Figures 1 and 2 show the mission power/altitude/time profiles specified by NASA for use during the latter part of the Phase I study. These data are given in terms of the gearbox output power required for driving the pumps and the generator. (The sample APU performance maps of Figure 1 are given in terms of the net power available after allowance for the hydraulic pump and generator losses.)



Time, sec From	To	Period, sec	Hydraulic hp Output	Hydraulic Shaft hp Input	Electric hp Output	Electric Shaft hp Input	Total Shaft hp	Input Energy, hp-sec
-1000	-800	200	7	23	13	17	40	8,000
-800	-500	300	75	95	13	17	112	33,600
-500	0	500	7	39	13	17	56	28,000
0	5	5	50	82	36	42	124	720
5	45	40	90	122	13	17	139	5,560
45	75	30	140	172	13	17	147	4,410
75	115	40	190	222	13	17	239	9,560
115	155	40	117	149	13	17	166	6,640
155	195	40	100	132	13	17	149	5,960
195	230	35	7	39	13	17	56	1,960
230	240	10	100	132	13	17	149	1,490
240	600	360	7	23	10	14	37	13,320
600	610	10	80	96	10	14	110	1,100
610	700	58	7	23	10	14	37	2,146
610	700	32	27	43	10	14	57	1,824
700	8400	5700	7	23	10	14	37	210,900
700	8400	2000	15	31	10	14	45	90,000
8400	8410	10	45	61	10	14	75	750
8400	8700	20	20	52	10	14	66	1,320
8410	8700	270	7	39	10	14	53	14,310
8700	8702	2	180	216	10	14	230	460
8702	8760	58	7	39	10	14	53	3,074
8760	9000	140	25	57	10	14	71	9,940
9000	9600	600	20	52	10	14	66	39,600
Total energy hp-hr = 137.37 hp-hr								

Notes:
1. Hydraulic power conversion based on two pumps per APU with one pump depressurized during prolonged low power need.
2. Pump losses 16 to 20 hp for one pump and 32 hp for 2 pumps. Pump loss reference from Vickers PV3-300 at 5000 rpm.
3. Electric power based on 30 kw shared among 3 alternators.

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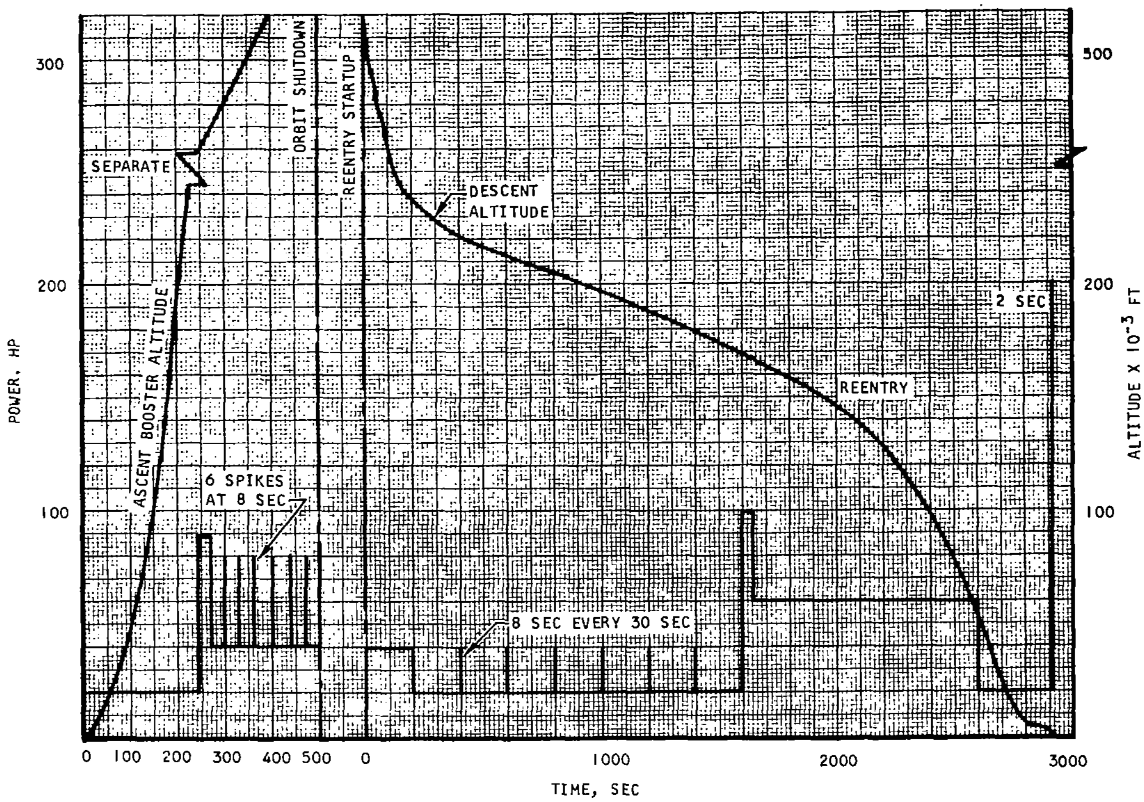
Figure 1. Booster Mission APU Power Schedule



ORBITER POWER AND TIME SCHEDULE

Function	Gearbox Output Power, hp	Time, sec	Input Energy, hp-sec
Boost	20	250	5000
Propellant valves punch	90	10	900
TVC and valves	40	240	9600
TVC - spikes	40	12	480
Descent checkout	40	200	8000
Quiescent	20	1400	28000
Spikes	20	112	2240
Reentry	60	1000	60000
Reentry spike	40	10	400
Quiescent	20	300	6000
Approach	200	2	400
Landing	50	100	5000
			125,920

Total Energy = 35 hp-hr + 20 percent startup and shutdown = 42 hp-hr

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Figure 2. Orbiter Mission APU Power Schedule

Ambient Pressure/Altitude Map

Since the turbine performance maps are presented as a function of ambient pressure, and since the mission profile data are given in terms of ambient altitude, it is necessary to provide a map specifying the relationship between pressure and altitude. The ICAO standard atmosphere and the NASA tentative upper atmosphere have been used as the basis of these data.

OUTPUT

Figure 3 shows a sample output obtained from the computer program. The data give the propellant consumption for each mission segment, showing the quantities attributable to the base power level and these due to power spikes occurring during the segment. The total energy output, the average power output, and the total propellant requirements for the mission are also provided. All data are in terms of net output power, after accounting for hydraulic pump and generator losses. The total propellant is unaffected by this method of accumulating energy output.

PROPELLANT AND POWER STUDY - - BOOSTER MISSION-- 300 PSI PUMP TURB 9

SEG	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE E GEN'D	SPIKE E GEN'D	HYDROGEN CONSUMED BASE HP	HYDROGEN CONSUMED SPIKE HP	OXYGEN CONSUMED BASE HP	OXYGEN CONSUMED SPIKE HP
1	-1000.0	-800.0	20.0	-.0	-.0	-.0	.0	66.7	.0	5.21	.00	2.29	.00
2	-800.0	-500.0	88.0	-.0	-.0	-.0	.0	440.0	.0	13.35	.00	7.41	.00
3	-500.0	.0	20.0	-.0	-.0	-.0	.0	166.7	.0	13.02	.00	5.73	.00
4	.0	5.0	86.0	-.0	-.0	-.0	700.0	7.2	.0	.22	.00	.12	.00
5	5.0	45.0	103.0	-.0	-.0	-.0	6484.8	68.7	.0	1.83	.00	1.06	.00
6	45.0	75.0	153.0	-.0	-.0	-.0	19650.1	76.5	.0	1.77	.00	1.11	.00
7	75.0	115.0	203.0	-.0	-.0	-.0	40521.0	135.3	.0	2.75	.00	1.81	.00
8	115.0	155.0	130.0	-.0	-.0	-.0	77713.0	86.7	.0	1.90	.00	1.21	.00
9	155.0	195.0	113.0	-.0	-.0	-.0	131795.0	75.3	.0	1.70	.00	1.08	.00
10	195.0	210.0	20.0	-.0	-.0	-.0	187204.9	5.0	.0	.18	.00	.10	.00
11	210.0	230.0	20.0	-.0	-.0	-.0	233500.0	6.7	.0	.24	.00	.14	.00
12	230.0	240.0	113.0	-.0	-.0	-.0	248000.0	18.8	.0	.43	.00	.27	.00
13	240.0	300.0	17.0	-.0	-.0	-.0	234500.0	17.0	.0	.67	.00	.38	.00
14	300.0	390.0	17.0	-.0	-.0	-.0	162073.2	25.5	.0	1.00	.00	.57	.00
15	390.0	410.0	17.0	-.0	-.0	-.0	143000.0	5.7	.0	.22	.00	.13	.00
16	410.0	600.0	17.0	-.0	-.0	-.0	132705.9	53.8	.0	2.13	.00	1.20	.00
17	600.0	610.0	90.0	-.0	-.0	-.0	125058.8	15.0	.0	.32	.00	.20	.00
18	610.0	700.0	17.0	37.0	4.0	8.0	115852.9	16.4	19.7	.65	.54	.37	.32
19	700.0	900.0	17.0	25.0	8.0	6.0	78931.8	43.1	20.0	1.75	.66	.98	.38
20	900.0	1250.0	17.0	25.0	8.0	12.0	25168.8	72.0	40.0	4.15	1.78	2.03	.91
21	1250.0	8360.0	17.0	25.0	8.0	237.0	10000.0	1477.3	790.0	108.55	43.63	49.21	20.81
22	8360.0	8400.0	17.0	25.0	8.0	1.0	6910.0	9.1	3.3	.71	.19	.31	.09
23	8400.0	8410.0	55.0	-.0	-.0	-.0	3973.7	9.2	.0	.34	.00	.18	.00
24	8410.0	8700.0	17.0	30.0	2.0	10.0	3184.2	76.5	10.0	6.40	.54	2.80	.26
25	8700.0	8702.0	190.0	-.0	-.0	-.0	2415.8	6.3	.0	.15	.00	.09	.00
26	8702.0	8760.0	17.0	-.0	-.0	-.0	2257.9	16.4	.0	1.40	.00	.61	.00
27	8760.0	8800.0	35.0	-.0	-.0	-.0	1052.6	23.3	.0	1.18	.00	.57	.00
28	8800.0	9000.0	35.0	-.0	-.0	-.0	.0	116.7	.0	6.02	.00	2.88	.00
29	9000.0	9600.0	30.0	-.0	-.0	-.0	.0	300.0	.0	17.26	.00	8.04	.00

TOTAL ENERGY OUTPUT	4319.8
TOTAL AVERAGE POWER OUTPUT	24.5
TOTAL HYDROGEN REQUIRED	242.84
TOTAL OXYGEN REQUIRED	115.63

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC. PROPELLANT CONSUMED IN LB.

PROPELLANT AND POWER STUDY - -ORBITER MISSION DESCENDING - 300 PSI PUMP TURB 9

SEG	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE E GEN'D	SPIKE E GEN'D	HYDROGEN CONSUMED BASE HP	HYDROGEN CONSUMED SPIKE HP	OXYGEN CONSUMED BASE HP	OXYGEN CONSUMED SPIKE HP
1	.0	100.0	20.0	-.0	-.0	-.0	250000.0	33.3	.0	1.20	.00	.69	.00
2	100.0	200.0	20.0	-.0	-.0	-.0	244000.0	33.3	.0	1.20	.00	.69	.00
3	200.0	300.0	.0	20.0	8.0	9.0	234000.0	.0	24.0	.18	.86	.09	.49
4	300.0	900.0	.0	20.0	8.0	20.0	216000.0	.0	53.3	2.80	1.92	1.38	1.10
5	900.0	1600.0	.0	20.0	8.0	18.0	184280.5	.0	48.0	3.54	1.72	1.75	.99
6	1600.0	1610.0	.0	80.0	10.0	1.0	167725.0	.0	13.3	.00	.29	.00	.18
7	1610.0	1800.0	40.0	-.0	-.0	-.0	162225.0	126.7	.0	3.34	.00	2.01	.00
8	1800.0	2000.0	40.0	-.0	-.0	-.0	148445.0	133.3	.0	3.52	.00	2.12	.00
9	2000.0	2200.0	40.0	-.0	-.0	-.0	136000.0	133.3	.0	3.52	.00	2.12	.00
10	2200.0	2400.0	40.0	-.0	-.0	-.0	142069.8	133.3	.0	3.52	.00	2.12	.00
11	2400.0	2600.0	40.0	-.0	-.0	-.0	67551.0	133.3	.0	3.65	.00	2.17	.00
12	2600.0	2780.0	.0	-.0	-.0	-.0	20359.4	.0	.0	2.30	.00	.92	.00
13	2780.0	2870.0	.0	-.0	-.0	-.0	4923.7	.0	.0	1.64	.00	.61	.00
14	2870.0	2900.0	.0	-.0	-.0	-.0	3100.0	.0	.0	.57	.00	.21	.00
15	2900.0	2902.0	164.0	-.0	-.0	-.0	2140.0	5.5	.0	.13	.00	.08	.00
16	2902.0	3000.0	14.0	-.0	-.0	-.0	1040.0	22.9	.0	2.34	.00	.99	.00

TOTAL ENERGY OUTPUT	893.7
TOTAL AVERAGE POWER OUTPUT	17.9
TOTAL HYDROGEN REQUIRED	38.27
TOTAL OXYGEN REQUIRED	20.71

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC. PROPELLANT CONSUMED IN LB.

PROPELLANT AND POWER STUDY - -ORBITER MISSION ASCENDING - 300 PSI PUMP TURB 9

SEG	TIME START	TIME ENDING	BASE HP OUT	SPIKE HP OUT	SPIKE DURATN	NO OF SPIKES	AVERAGE ALTITUDE	BASE E GEN'D	SPIKE E GEN'D	HYDROGEN CONSUMED BASE HP	HYDROGEN CONSUMED SPIKE HP	OXYGEN CONSUMED BASE HP	OXYGEN CONSUMED SPIKE HP
1	-630.0	.0	20.0	-.0	-.0	-.0	.0	210.0	.0	16.41	.00	7.21	.00
2	.0	100.0	.0	-.0	-.0	-.0	15350.3	.0	.0	1.43	.00	.56	.00
3	100.0	250.0	.0	-.0	-.0	-.0	65102.0	.0	.0	1.08	.00	.51	.00
4	250.0	260.0	70.0	-.0	-.0	-.0	250000.0	11.7	.0	.26	.00	.16	.00
5	260.0	500.0	20.0	60.0	2.0	6.0	250000.0	74.0	12.0	2.73	.28	1.56	.17

TOTAL ENERGY OUTPUT	309.7
TOTAL AVERAGE POWER OUTPUT	16.4
TOTAL HYDROGEN REQUIRED	22.18
TOTAL OXYGEN REQUIRED	10.18

ALTITUDE IN FEET, ENERGY IN HP-MIN, POWER IN HORSEPOWER, TIME IN SEC. PROPELLANT CONSUMED IN LB.

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Figure 3. Typical Booster and Orbiter Missions Propellant Requirements

APPENDIX G

COMBUSTOR DESIGN CONCEPTS

Two different types of combustor designs have been considered; these are:

- Diffusion-type combustor (Figure 1)
- Can-type combustor (Figure 2)

The can combustor has a smaller chamber volume than does the diffusion combustor, and hence the can combustor will offer quicker response capability. However, previous AiResearch testing has demonstrated that the diffusion combustor will operate over a wide range of pressures with a high combustion efficiency. Figure 3 shows a plot of combustor efficiency vs chamber pressure for a hydrogen-oxygen diffusion combustor (AiResearch drawing PA-34525). Based on this experience, the diffusion combustor is recommended. The data of Figure 4 show that the blowdown time of either combustor is quite rapid. Figure 4 assumes a 1250-psia combustor chamber pressure. Thus, the blowdown times shown are considerably larger than those that would occur for lower pressure combustors. In the lower pressure combustors, the turbine nozzle (to develop the same output power) would be considerably larger than that for the 1250-psia combustor, so that the blowdown time would be reduced.

Figure 5 shows the diffusion combustor heat loss and insulation outside surface temperature for various combustor insulating schemes. The data show that about 1 in. of insulation will provide a low heat loss and maintain the insulation surface temperature at acceptable levels (340°F for 0.9 emissivity).

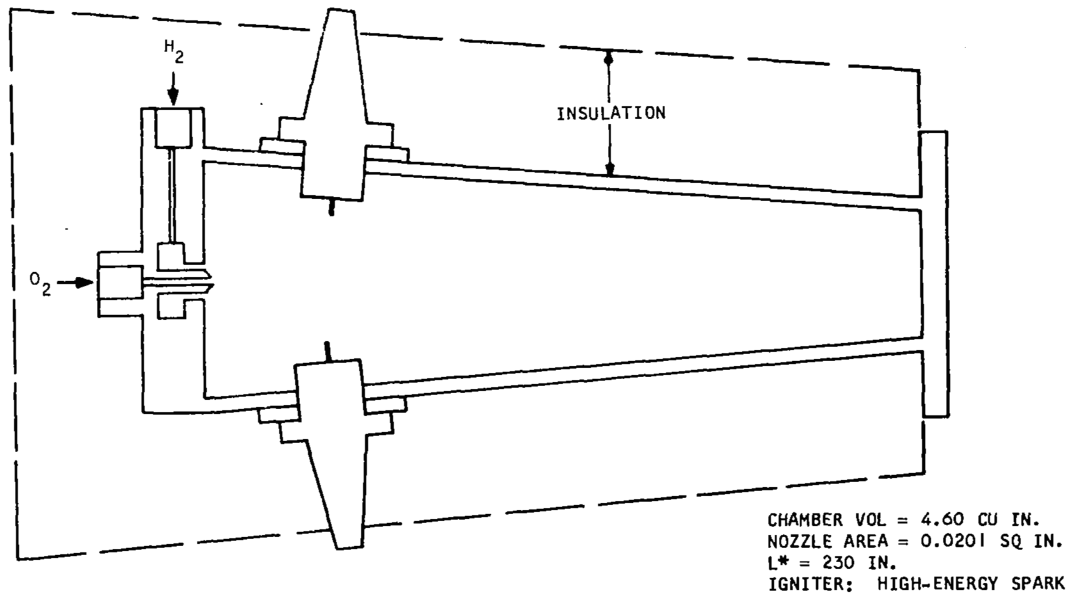

Figure 1. Diffusion - Type Combustor

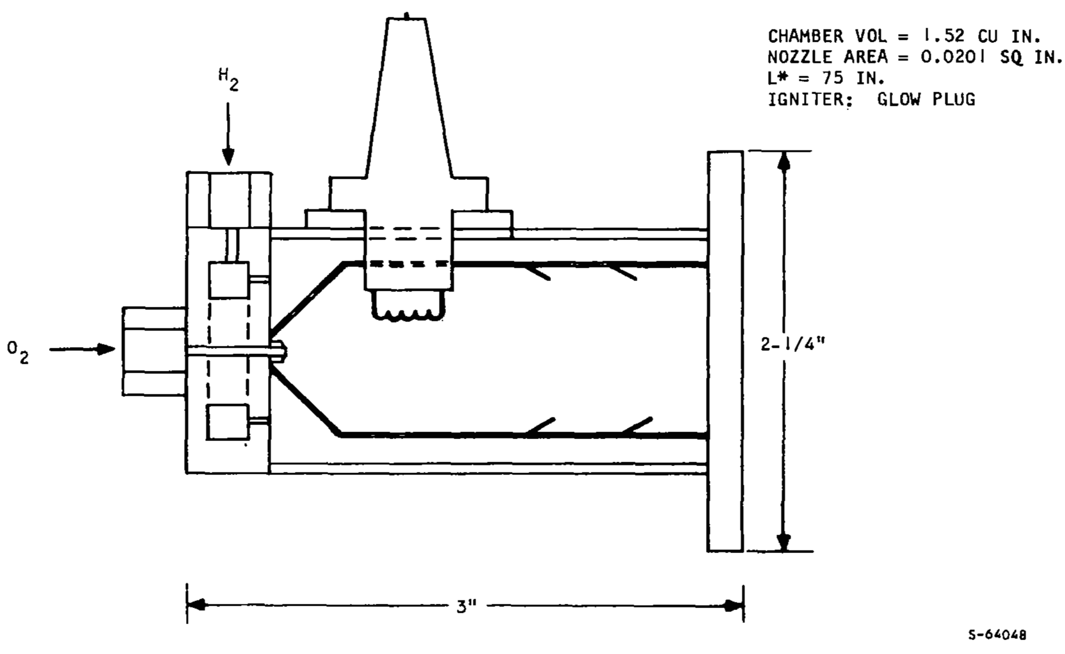

Figure 2. Can-Type Combustor

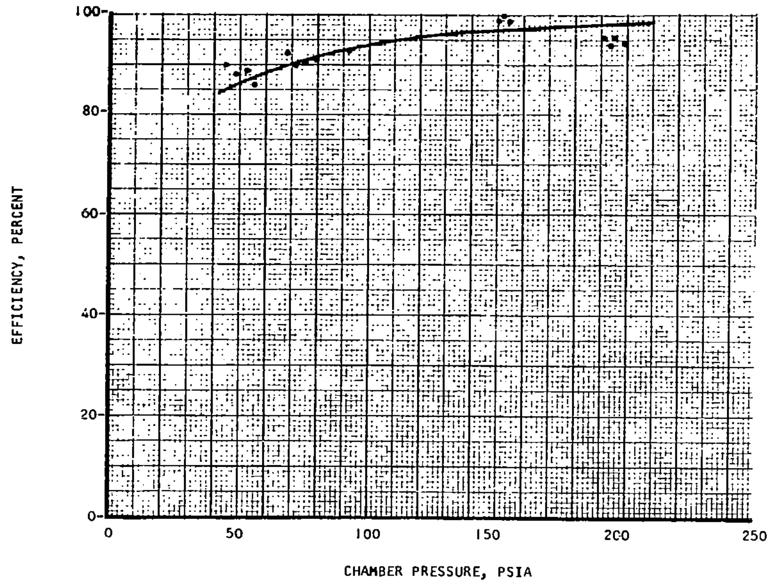

Figure 3. Typical H_2-O_2 Diffusion Combustor Efficiency

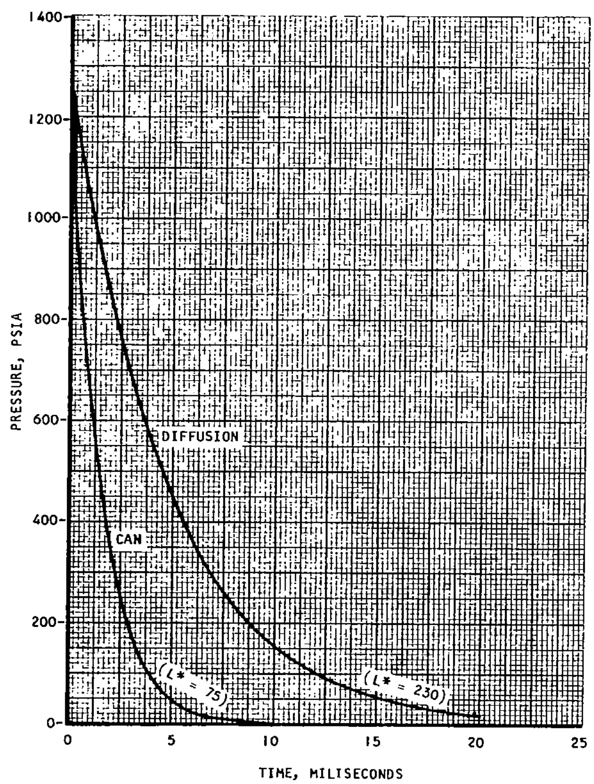

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Figure 4. Combustor Blowdown Time

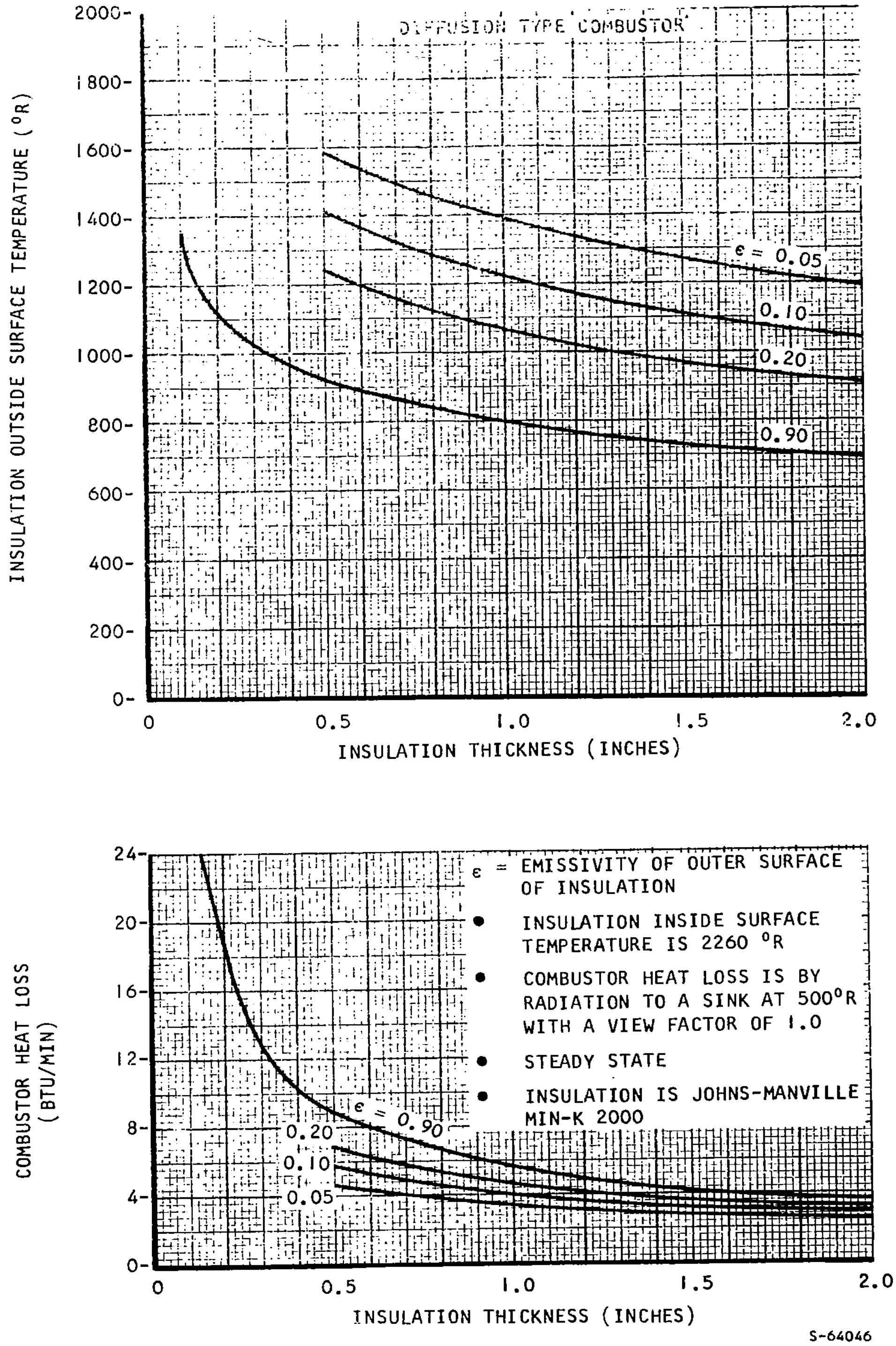

Figure 5. Diffusion Combustor Insulation Temperature and Heat Loss