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PRELIMINARY DESIGN OF AN AUXILIARY POWER UNIT FOR THE SPACE SHUTTLE — COMPONENT AND SYSTEM CONFIGURATION SCREENING ANALYSIS

by R. L. Binsley, J. P. Maddox, R. D. Marcy, R. S. Siegler, and R. Spies

Prepared by
ROCKETDYNE
Canoga Park, Calif. 91304
for Lewis Research Center

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Project Manager, Harry M. Cameron, Space Power Systems Division, NASA Lewis Research Center, Cleveland, Ohio

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.

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INTRODUCTION

This report presents data on a Phase I study of auxiliary power units to supply hydraulic and electrical power for both the orbiter and booster vehicles of the space shuttle. The program objective is to provide analysis and design information for gaseous oxygen/hydrogen and storable propellant APU's. Input of this information into the APU tradeoff studies (Phase I) and configuration decisions (Phase II) will ensure that APU specifications are realistic and achievable, and will provide a basis for space shuttle APU development programs.

In addition to meeting APU missión requirements, the design reflects pertinent non-operational criteria, including commercial aircraft reliability standards, minimized development cost and risk, minimized dry weight, high performance (low specific fuel consumption), use of state-of-the-art materials and technology wherever possible, and 1000-hr useful life capability. Discrete preplanned steps are arranged to reach the program objectives in an efficient manner consistent with the study time schedule. These steps are on the establishment of system specifications (or characteristics), the creation of applicable concepts, a design analysis of those functions to obtain design requirements, a synthesis of the data resulting from the design analysis, and an evaluation of the synthesized concepts. The evaluation results in the identification of a preliminary design that best satisfies the APU requirements and criteria. The primary evaluation criteria are those consistent with space shuttle vehicle criteria: (1) a minimum of new technology required, (2) inherent simplicity, (3) low weight, (4) flexibility to accommodate mission changes, and (5) minimal and easy system maintenance and refurbishment.

The Phase I program consists of system definition, operational analysis, and optimization of various system configurations with respect to weight, reliability, and development risk. For purposes of analysis, the APU was separated into three subsystems: The propellant feed system, the turbo-power unit, and the power control system. A total of 18 H_2/O_2 systems and 4 storable propellant systems were evaluated in Phase IB utilizing a digital computer program for weight optimization and analog modeling to determine dynamic operational characteristics and control requirements. Propellant sources considered included pumped and pressure (supercritical) fed propellant and high/low pressure gas supply from a vehicle source. Pressure compounded and velocity compounded two stage turbines were considered for the turbo-power unit. Three power control methods consisting of pressure modulation, pulse modulation, and a hybrid control (area modulation) were evaluated with various combinations of TPU designs and propellant sources.

SUMMARY AND CONCLUSIONS

The first phase of a design study for the Space Shuttle Auxiliary Power System (APU) has been completed. The Phase IA evaluation emphasized comparison of various options for the propellant system, turbo power unit, and power control techniques as illustrated in Fig. 1. As a result of the evaluations conducted in Phase IA, the 22 specific systems shown in Fig. 2 were evaluated in Phase IB. It was recommended that the baseline system for Phase II should use hydrogen and oxygen conditioned gaseous propellants with hydrogen supplied from vehicle integrated tanks. The turbo power unit (TPU) is designed with a two-stage turbine using an inlet pressure of 150 to 600 psi.

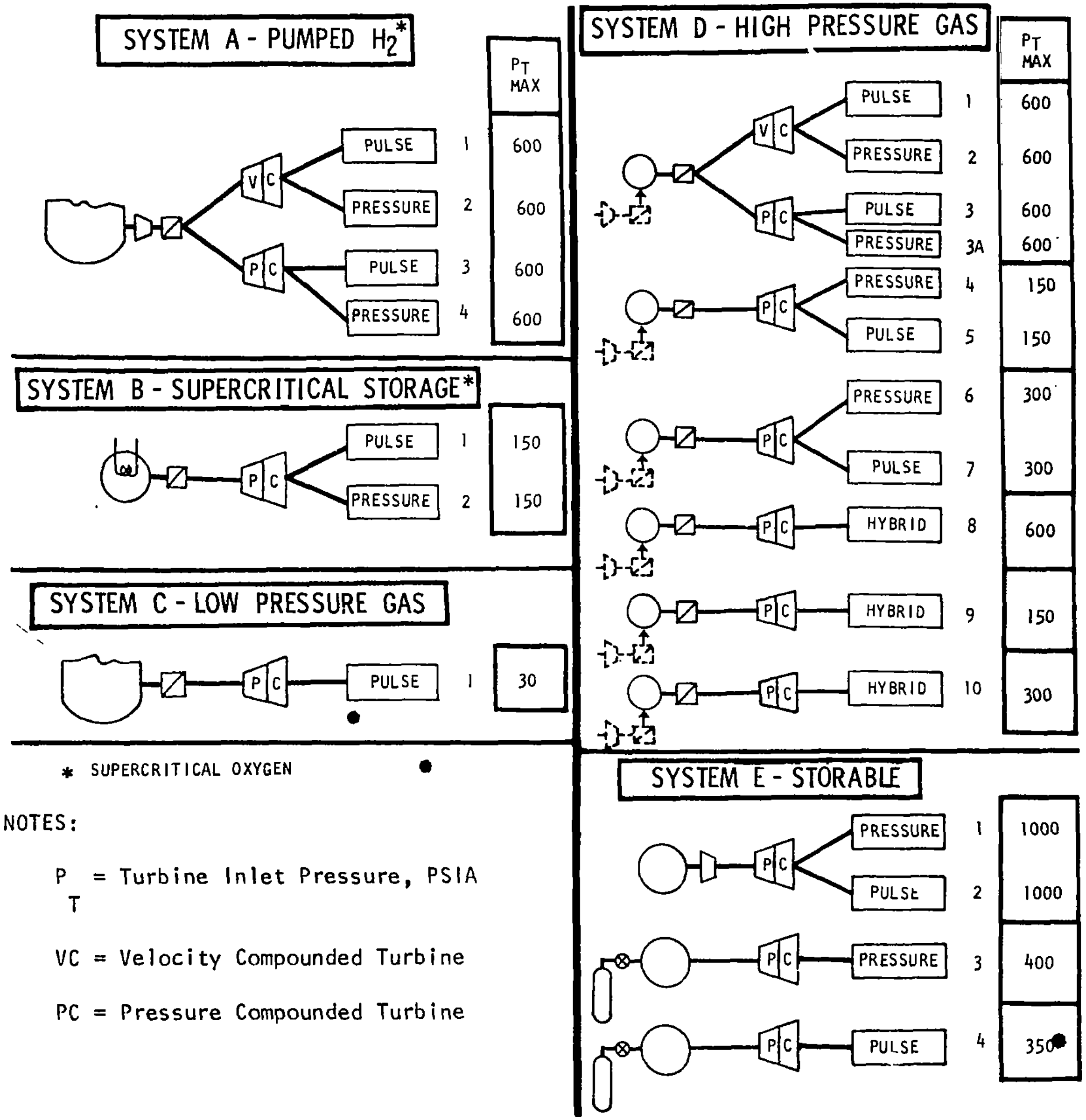
The Phase IB specified APU power profile is shown for the booster and orbiter in Fig. 3 and 4 and the APU power flow needed to meet the requirements is illustrated in Fig. 5. In addition, the entire APU system, including TPU and propellant conditioning system, was designed to provide steady full power, if demanded, for sustained periods. The propellant conditioning system utilizes hydrogen to perform the hydraulic cooling function for the vehicle in all H_2/O_2 systems evaluated. For storables, separate cooling is supplied by a water boiler.

During the initial Phase IB study the pressure compound turbine was compared with and chosen over the velocity compound turbine due to its performance advantage and known level of technology for this particular high turbine Mach number application. Three power control systems (pulse, pressure modulation, and hybrid), were investigated. The first two were analyzed in depth.

SYSTEMS EVALUATED – PHASE IA STUDY

	PROPELLANT SYSTEM	TURBO-POWER UNIT	POWER CONTROL
SYSTEM A	• PUMP FED LIQUID HYDROGEN • PRESSURE FED SUPERCRITICAL OXYGEN • SHARED VEHICLE TANKAGE OR SELF-CONTAINED APU TANKAGE	• 2 STAGE VELOCITY COMPOUNDED, SUPERSONIC • 2 STAGE, PRESSURE COMPOUNDED, SUBSONIC	• PULSE CONTROL • PRESSURE MODULATION
SYSTEM B	• PRESSURE FED SUPERCRITICAL HYDROGEN • PRESSURE FED SUPERCRITICAL OXYGEN • SHARED VEHICLE TANKAGE OR SELF-CONTAINED APU TANKAGE	• 2 STAGE, VELOCITY COMPOUNDED, SUPERSONIC • 4 STAGE, PRESSURE COMPOUNDED, SUBSONIC	• PULSE CONTROL • PRESSURE MODULATION
SYSTEM C	• PRESSURE FED GH_2 (20→35 PSIA) • PRESSURE FED GOX (20→35 PSIA) 20 PSIA SOURCE WITH COMPRESSOR • VEHICLE TANKAGE (ULLAGE)	• 2 STAGE, PRESSURE COMPOUNDED, SUBSONIC • SINGLE STAGE	• PULSE CONTROL • PRESSURE MODULATION • VARIABLE NOZZLE AREA • HYBRID

FIGURE 1



* SUPERCRITICAL OXYGEN

NOTES:

P_T = Turbine Inlet Pressure, PSIA

VC = Velocity Compounded Turbine

PC = Pressure Compounded Turbine

PHASE IB

SS/APU SYSTEM OPTIMIZATION SUMMARY

FIGURE 2

APU POWER PROFILE BOOSTER
GEARBOX OUTPUT SHAFT

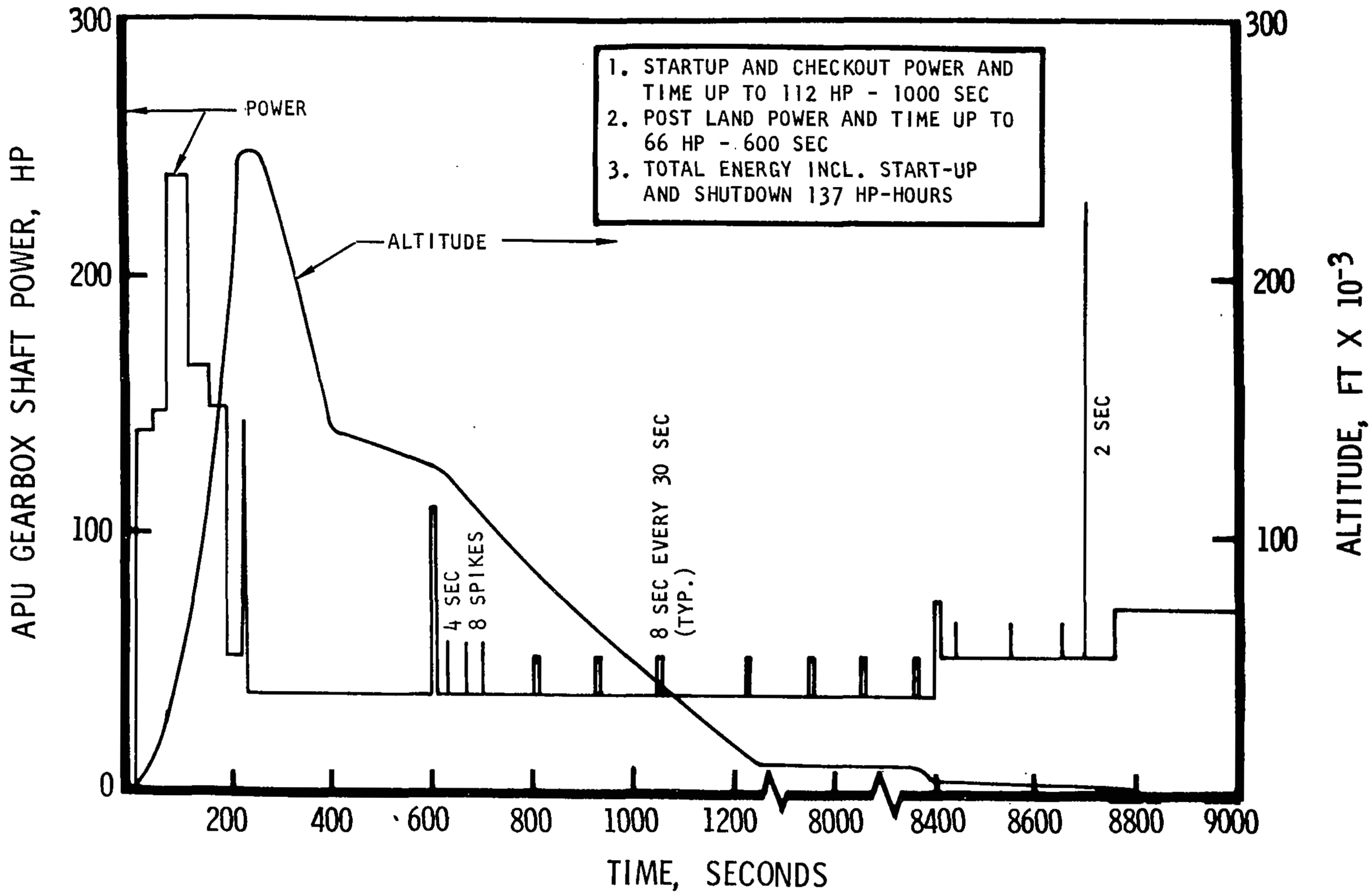
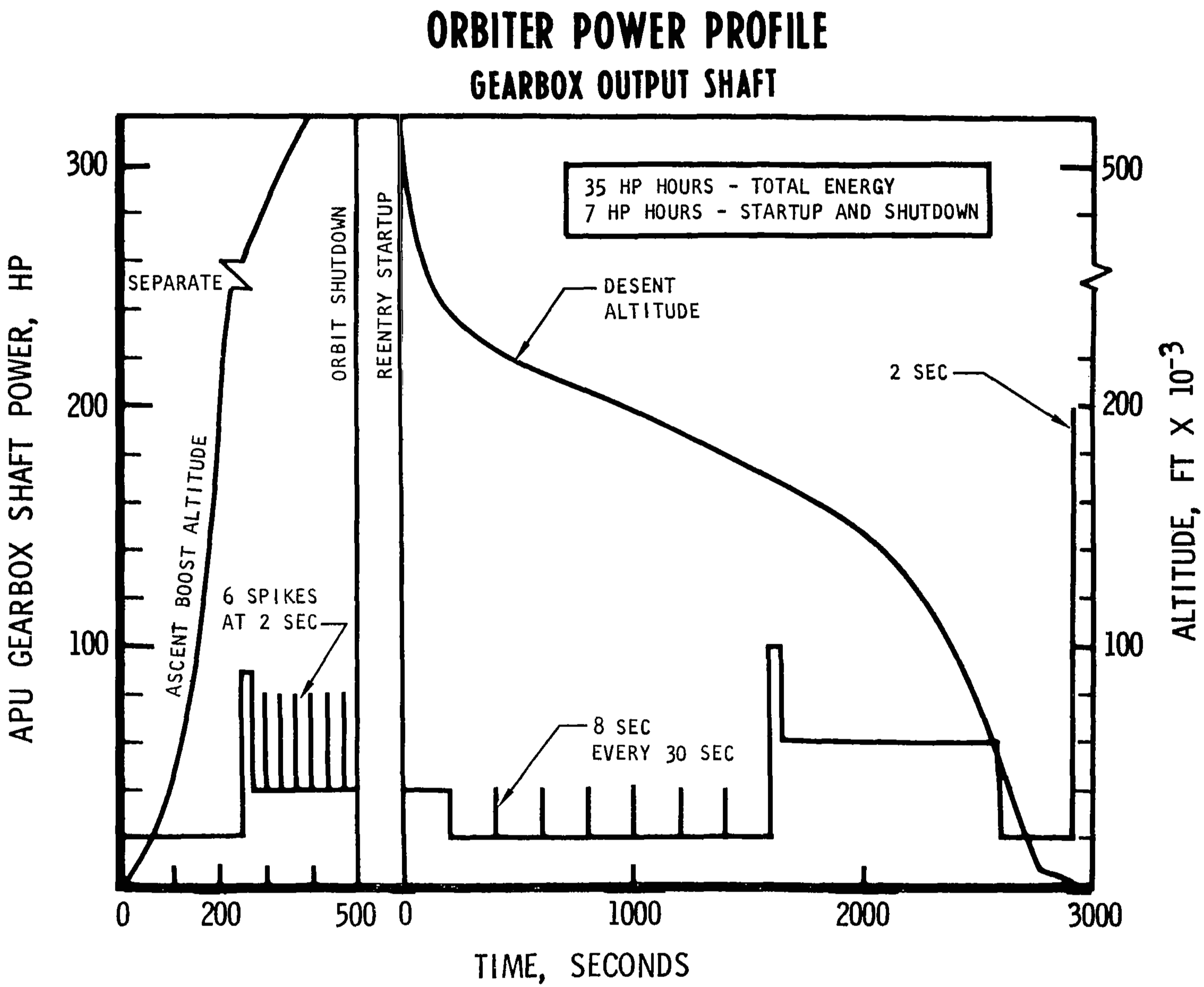


FIGURE 3



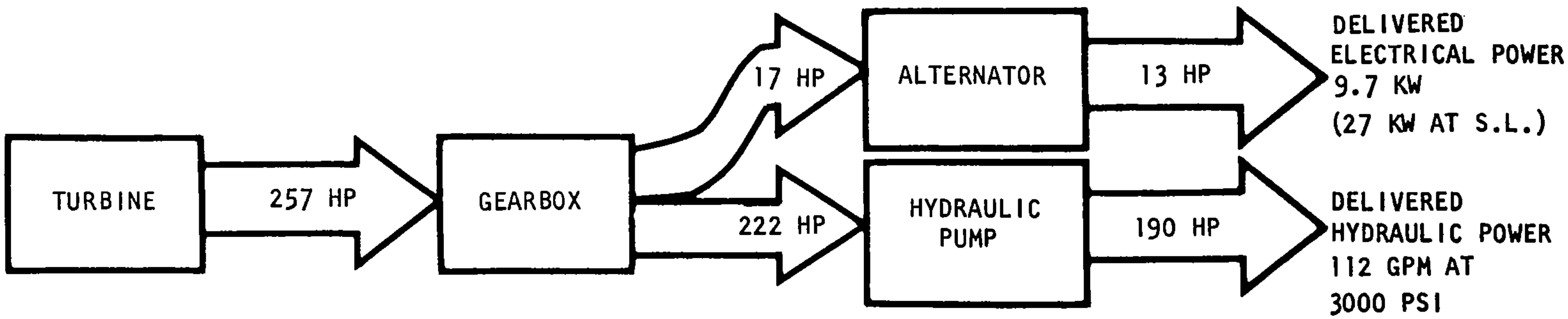
ORBITER POWER PROFILE
GEARBOX OUTPUT SHAFT
35 HP HOURS - TOTAL ENERGY
7 HP HOURS - STARTUP AND SHUTDOWN
SEPARATE
ASCENT BOOST ALTITUDE
ORBIT SHUTDOWN
REENTRY STARTUP
DESENT
ALTITUDE
2 SEC
6 SPIKES
AT 2 SEC
8 SEC
EVERY 30 SEC
APU GEARBOX SHAFT POWER, HP
ALTITUDE, FT X 10^{-3}
300
200
100
500
200
100
0
200
500
0
1000
2000
3000
TIME, SECONDS

FIGURE 4

TYPICAL APU POWER FLOW-BOOSTER

PEAK

20,000 ⟶ 70,000 FT



SUSTAINED IDLE

100,000 ⟶ 3000 FT

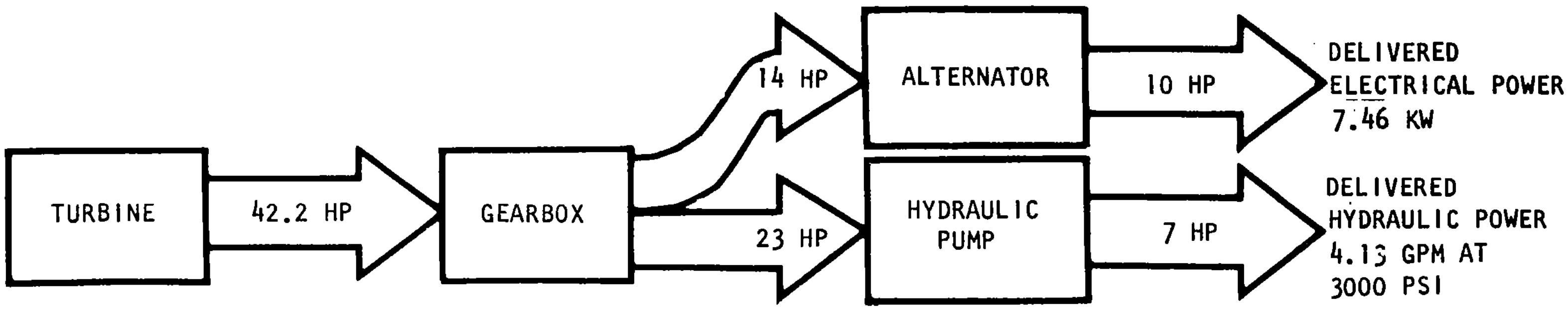


FIGURE 5

A general APU schematic as synthesized is illustrated in Fig. 6. The system is comprised of three subsystems with their associated primary functions as noted:

1. Propellant Conditioning System - acquires and conditions the propellant as necessary to provide controlled mixture of gaseous propellants, at the predetermined temperature and pressure to the TPU. In addition it performs the necessary cooling of the hydraulic and lubricating oil.
2. Turbo Power Unit - utilizes the propellants to provide necessary hydraulic and electrical power to the vehicle in the specified flight profile.
3. Power Control System - provides speed control to ensure that the TPU operates within specified limits providing the required output.

Utilizing configurations as illustrated in Fig. 6 for the H_2/O_2 systems and a configuration for the storable system as shown in Fig. 7 the 22 systems shown in Fig. 2 were evaluated. Five general system types (A, B, C, D, and E) were defined, each typifying a different propellant supply system. The evaluation was performed for each APU system optimized for a booster vehicle and for an orbiter vehicle, as well as for a common system utilizing a booster APU in the orbiter. These results are illustrated in Fig. 8. The "best" system was then chosen from each of the five general system types and compared with each other in Fig. 9 for the booster vehicle and in Fig. 10 for the orbiter vehicle using a common booster APU.

APU SYSTEM SCHEMATIC

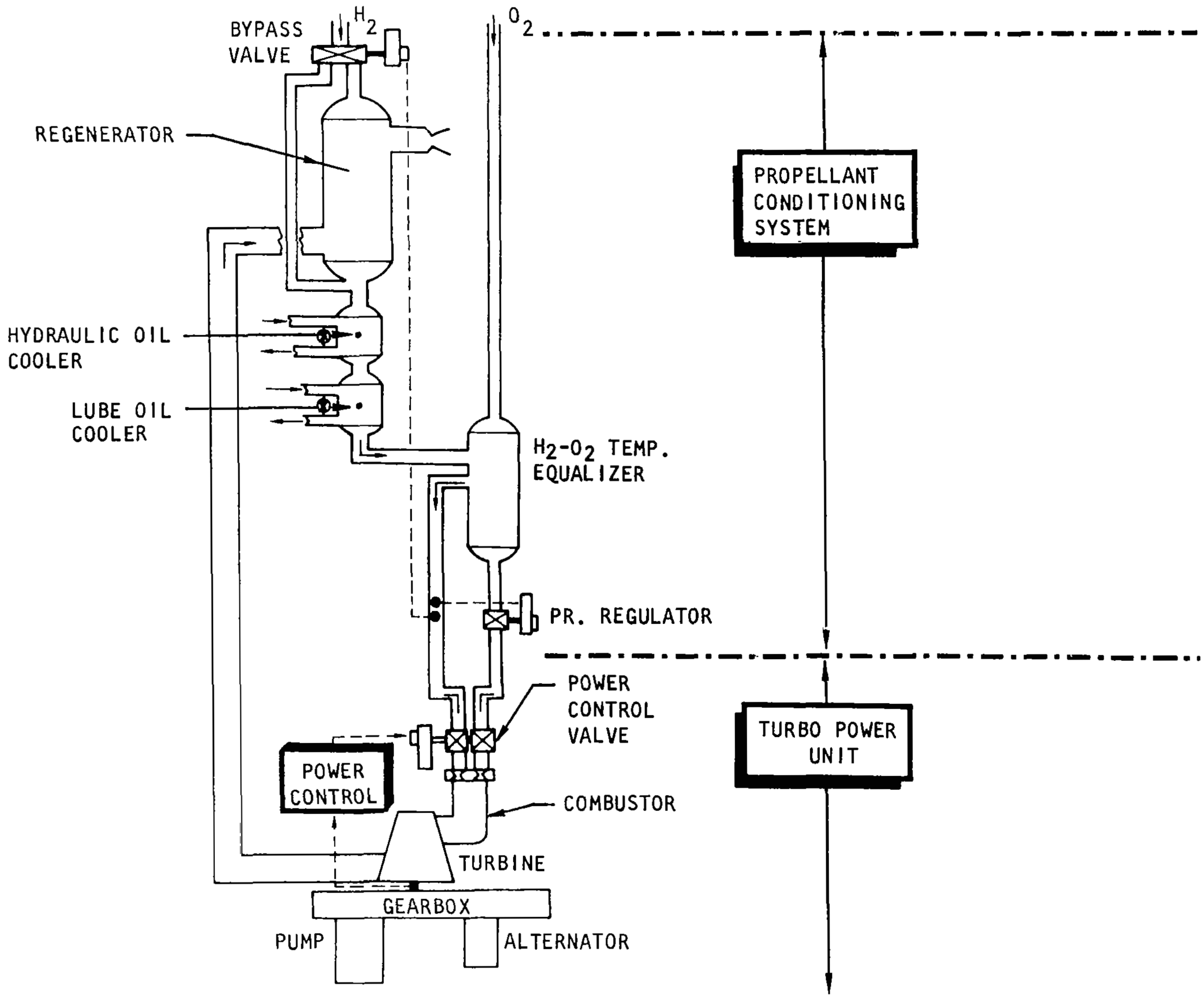


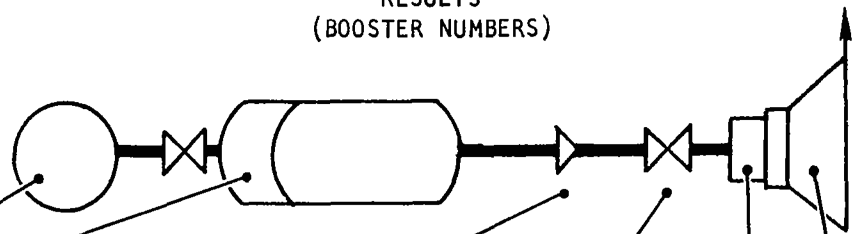
FIGURE 6

STORABLE PROPELLANT STUDIES

OBJECTIVES
- RELIABILITY
- MINIMUM DEVELOPMENT COST
- MINIMUM DEVELOPMENT RISK
- MINIMIZED WEIGHT
- STATE-OF-ART MATERIALS
- EXISTING TECHNOLOGY
- LONG LIFE

CONSIDERATION
- PROPELLANT SELECTION
- IGNITION/DISSOCIATION CONTROL
- GAS TEMPERATURE
- CARBON FORMATION
- FREEZING POINT
- PERFORMANCE

RESULTS
(BOOSTER NUMBERS)



TANKAGE
- TITANIUM TANK FOR PRESSURANT
  - 4500 PSI
  - He
  - 6.0 POUNDS
- ALUMINUM TANK FOR HYDRAZINE
  - MINIMUM WALL
  - 11 POUNDS
  - RUBBER BLADDER
  - LOW PRESSURE (<150 PSIA)

PUMP
- GEARBOX MOUNTED
  - MK 36 CENTRIFUGAL
  - 80,000 RPM

VALVE
- PRESSURE MODULATION

TURBINE
- TWO-STAGE, PRESSURE COMPOUND
  - 1700 F GAS
  - 60,000 RPM
  - 1550 FT/SEC TIP SPEED
  - EFFICIENCY = 67%

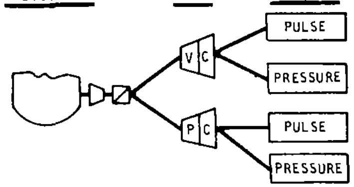
PROPELLANT
- HYDRAZINE
  - MOST DEVELOPED
  - NO CARBON IN EXHAUST
  - LOWER DISSOCIATION REQUIRED
  - LOWER PERFORMANCE THAN BLEND
    (10-15% ≈ 100 POUNDS)
    SPC = 5.46 LB/HP-HR
  - REQUIRES HEATING (~30W IN ORBIT)

GAS GENERATOR
- ELECTRICALLY HEATED THERMAL BED
- DEVELOPED SYSTEM
- NO CONTAMINATION PROBLEM
- SLOWER INITIAL RESPONSE
- 1700 F - REQUIRES 60% DISSOCIATION

Figure 7

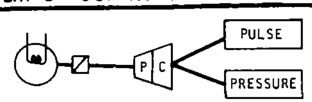
PHASE IB **SS/APU SYSTEM OPTIMIZATION SUMMARY**

SYSTEM A - PUMPED H_2^*



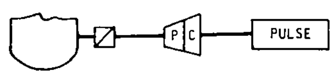
No.	Power Control	Booster Optimum P_T Max	Booster Optimum System Weight LBS	Orbiter Optimum P_T Max	Orbiter Optimum System Weight LBS	Booster APU in Orbiter P_T Max	Booster APU in Orbiter System Weight LBS
1	PULSE	600	643	600	348	600	356
2	PRESSURE	600	773	600	365	510	392
3	PULSE	600	608	600	338	600	347
4	PRESSURE	600	726	600	360	507	388

SYSTEM B - SUPERCRITICAL STORAGE



No.	Power Control	Booster Optimum P_T Max	Booster Optimum System Weight LBS	Orbiter Optimum P_T Max	Orbiter Optimum System Weight LBS	Booster APU in Orbiter P_T Max	Booster APU in Orbiter System Weight LBS
1	PULSE	150	775	150	377	150	387
2	PRESSURE	150	1056	150	461	130	474

SYSTEM C - LOW PRESSURE GAS



No.	Power Control	Booster Optimum P_T Max	Booster Optimum System Weight LBS	Orbiter Optimum P_T Max	Orbiter Optimum System Weight LBS	Booster APU in Orbiter P_T Max	Booster APU in Orbiter System Weight LBS
1	PULSE	30	1534	30	723	30	784

* SUPERCRITICAL OXYGEN

NOTES: P_T = Turbine Inlet Pressure, PSIA

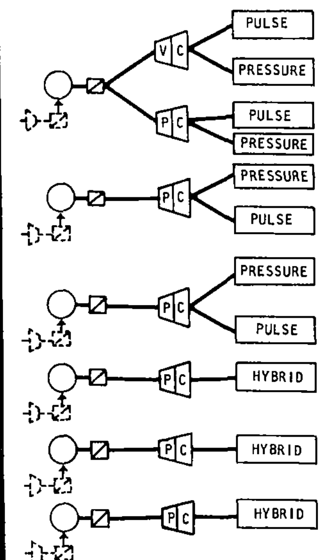
PC = Pressure Compound Turbine

VC = Velocity Compound Turbine

Storage Pressure = $1.39\ P_T + 20$

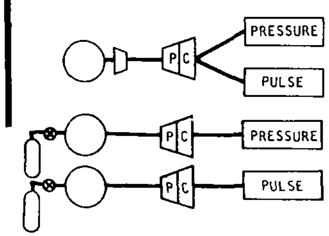
System Weight = Dry Weight + Propellant Burned + 5% Reserve + Tankage w/5% Ullage + 10% Installation Penalty

SYSTEM D - HIGH PRESSURE GAS



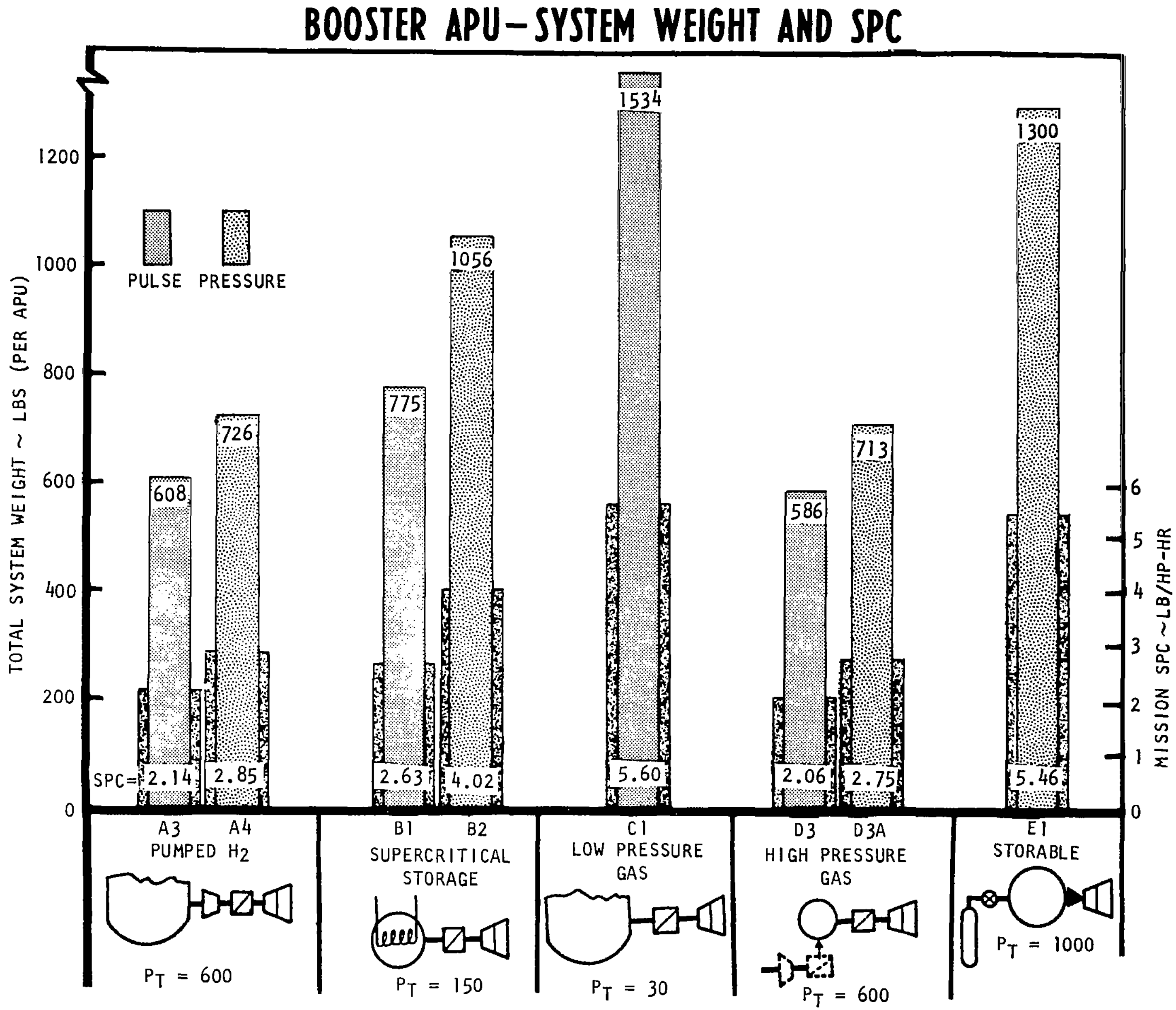
No.	Power Control	Booster Optimum P_T Max	Booster Optimum System Weight LBS	Orbiter Optimum P_T Max	Orbiter Optimum System Weight LBS	Booster APU in Orbiter P_T Max	Booster APU in Orbiter System Weight LBS
1	PULSE	600	620	600	326	600	333
2	PRESSURE	600	758	600	347	510	372
3	PULSE	600	586	600	318	600	324
3A	PRESSURE	600	713	600	342	507	368
4	PRESSURE	150	910	150	406	130	418
5	PULSE	150	675	150	335	150	344
6	PRESSURE	300	787	300	377	259	390
7	PULSE	300	630	300	327	300	337
8	HYBRID	600	637	600	333	600	339
9	HYBRID	150	743	150	354	150	363
10	HYBRID	300	686	300	344	300	355

SYSTEM E - STORABLE



No.	Power Control	Booster Optimum P_T Max	Booster Optimum System Weight LBS	Orbiter Optimum P_T Max	Orbiter Optimum System Weight LBS	Booster APU in Orbiter P_T Max	Booster APU in Orbiter System Weight LBS
1	PRESSURE	1000	1300	1000	527	850	535
2	PULSE	1000	1154	1000	463	1000	471
3	PRESSURE	400	1658	250	573	300	583
4	PULSE	350	1296	200	492	350	500

FIGURE 8



BOOSTER APU—SYSTEM WEIGHT AND SPC
TOTAL SYSTEM WEIGHT ~ LBS (PER APU)
MISSION SPC ~LB/HP-HR
PULSE
PRESSURE
1200
1000
800
600
400
200
0
6
5
4
3
2
1
0
608
726
775
1056
1534
586
713
1300
SPC=
2.14
2.85
2.63
4.02
5.60
2.06
2.75
5.46
A3 A4
PUMPED H_2
P_T = 600
B1 B2
SUPERCRITICAL STORAGE
P_T = 150
C1
LOW PRESSURE GAS
P_T = 30
D3 D3A
HIGH PRESSURE GAS
P_T = 600
E1
STORABLE
P_T = 1000

FIGURE 9

ORBITER/BOOSTER APU-SYSTEM WEIGHT AND SPC

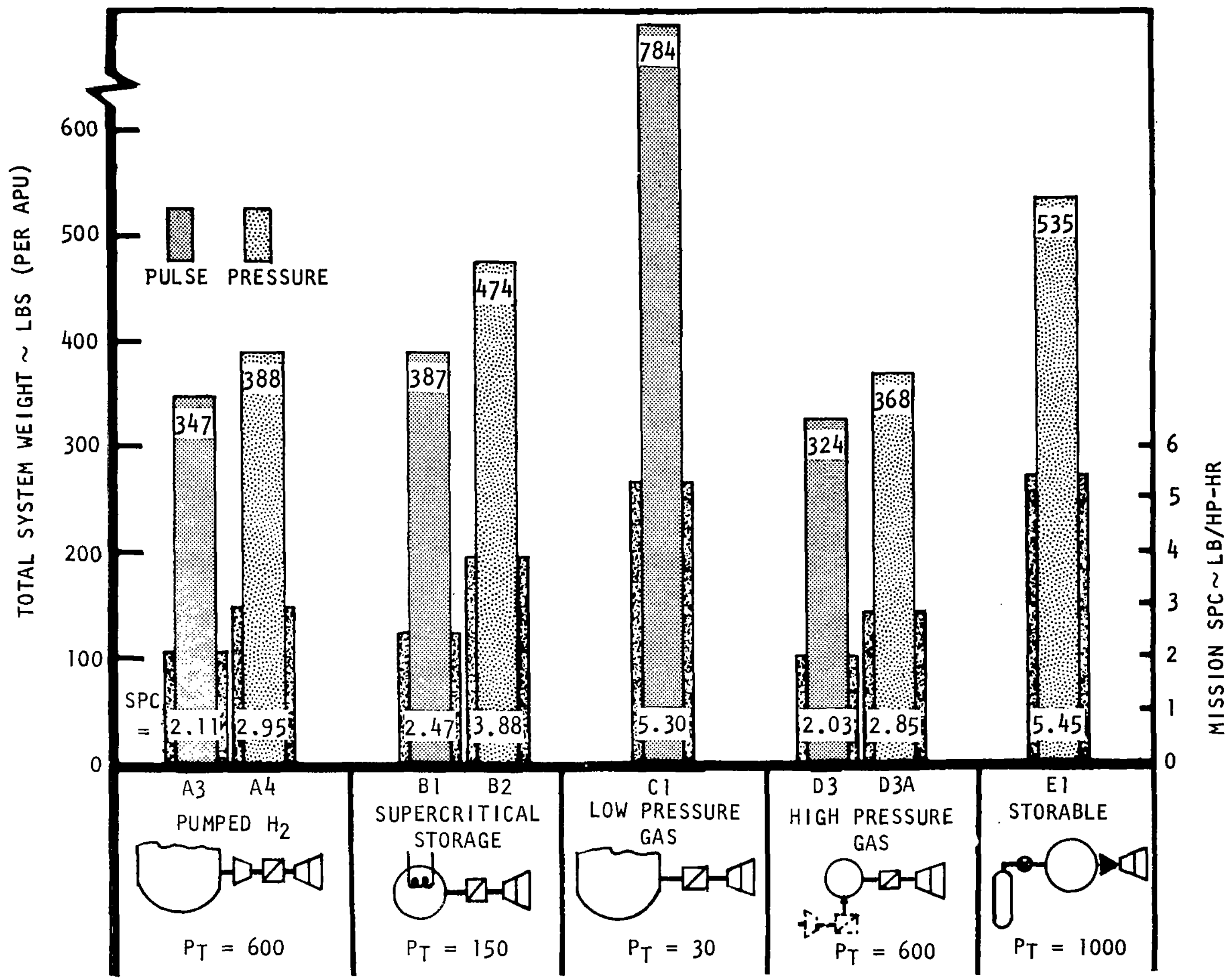


FIGURE 10

The High Pressure Vehicle Supplied H_2/O_2 Gas System (D) and Pumped Hydrogen/ Supercritical Stored Oxygen System (A) are clearly lowest in weight for both booster and orbiter. For all systems the pulse modulated power control option is lower in weight. Detailed weight breakdowns are shown in Tables 1 and 2 for the booster and orbiter systems.

Figure 11 illustrates the advantages to be gained by considering booster/ orbiter commonality. The weight penalty per APU and the vehicle related cost of orbiting payload assuming three APU's per orbiter vehicle proves to be small. The various APU system weights are compared in Fig. 12 in terms of equivalent orbiter system weight* and vehicle related cost. The baseline chosen was common APU's with six APU's per booster and three APU's per orbiter vehicle.

Estimates for the development costs of each of the major APU systems are shown in Fig. 13. Research, Technology, Development and Engineering (RTD&E) costs in $\triangle$ dollar form added to the $\triangle$ dollar vehicle related costs of Fig. 12 are compared. The Storable Propellant System (E-1) indicates the lowest RTD&E cost while the High Pressure Supply Gas H_2/O_2 Pulse Control System (D-3) had the lowest vehicle related costs. The total costs shown in Fig. 13 show a definite advantage for the High Pressure Vehicle Supply Gas System over all but the Pumped Hydrogen Gas System. The latter, while somewhat more expensive, is felt to be cost competitive.

The ground support equipment necessary is not included in the cost figures and since more GSE is required for the storable system, the H_2/O_2 system would appear even more favorable from a cost viewpoint.

---

*Equivalent orbiter weight = $\frac{\text{booster related weight}}{5}$ + orbiter related weight.

SSAPU WEIGHT SUMMARY — BOOSTER

COMPONENTS \ SYSTEM	A1	A2	A3	B1	B2	D1	D2	D3	D4	D5	D6	D7	D8	D9	A4	D3A	C1	D10	E1	E2	E3	E4
TURBINE, GEAR BOX, CONTAINMENT COMBUSTOR	73	74	72	76	76	73	74	72	76	76	74	74	72	76	74	72	113	74	67.4	67.4	65.7	65.7
40 KVA ALTERNATOR	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33
HYDRAULIC PUMP (2) PLUS VALVE PVC3- 300	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
HYDROGEN BURNED PLUS 5% RESERVE	185	248	168	207	317	178 21**	239 29**	162 19**	306 37**	200 24**	250 30**	180 22**	185 22**	230 28**	226	218 26**	810	206 25**	796.4***	653.0***	956.7***	714.0***
OXYGEN BURNED PLUS 5% RESERVE	154	207	141	173	264	148 18**	200 24**	136 16**	254 30**	166 20**	210 25**	150 18**	156 18**	191 23**	188	181 22**		172 21**	0.3✡	0.3✡	12.0✡	5.0✡
H_2 HEAT EXCHANGER	11	11	10	12	14	5	5	5	7	6	6	6	6	7	12	5	75	7	NA	NA	NA	NA
O_2 HEAT EXCHANGER	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	30	2	NA	NA	NA	NA
LUBE COOLER	6	6	6	5	5	6	6	6	5	5	5	5	6	5	6	6	42	6	10	10	10	10
HYDRAULIC COOLER	8	8	8	7	7	8	8	8	7	7	7	7	8	7	8	8	240	8	240	240	240	240
HYDROGEN PUMP PLUS DRIVE	12	12	12	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	12	NA	NA	NA	3.5	3.5	NA	NA
SUPERCRITICAL HEATER, FAN	5 (O_2)	5 (O_2)	5 (O_2)	10	10	NA	NA	NA	NA	NA	NA	NA	NA	NA	5 (O_2)	NA	NA	NA	NA	NA	NA	NA
VALVES, FILTERS, INSTRUMENTATION, LINES, CONTROLS	25	25	25	25	25	20	20	20	20	20	20	20	20	20	25	20	65	20	25	25	20	25
INSTALLATION, SUPPORT STRUCTURE	24	24	24	23	23	21	21	21	21	21	21	21	21	21	24	21	66	21	46.5	46.2	63.7	52.9
TOTAL	598	715	566	633	836	593	721	560	858	640	743	598	609	703	675	674		655	1282.1	1138.4	1461.1	1205.6
H_2 TANK	24 (SSE)	30 (SSE)	23 (SSE)	120 (RCS)	187 (RCS)	26 (SSE)	35 (SSE)	24 (SSE)	48 (SSE)	32 (SSE)	40 (SSE)	29 (SSE)	26	37	27 (SSE)	36 (SSE)	NA	28	11.0***	10.0***	65.0***	32.0***
O_2 TANK	21 (RCS)	28 (RCS)	19 (RCS)	22 (RCS)	33 (RCS)	2 (SSE)	2 (SSE)	2 (SSE)	4 (SSE)	3 (SSE)	4 (SSE)	3 (SSE)	2	3	24 (RCS)	3 (SSE)	NA	3	5.7	5.3	131.4	58.4
TOTAL	643	773	608	775	1056	621	758	586	910	675	787	630	637	743	726	713	1534	686	1298.8	1153.7	1657.5	1296.0
MISSION SPC* LB/HP-HR (HP-HR = 137.4)	2.35	3.15	2.14	2.63	4.02	2.26	3.04	2.06	3.88	2.54	3.18	2.28	2.37	2.92	2.85	2.75	5.60	2.62	5.46	4.48	6.66	4.95

* BASED ON BURNED PROPELLANT

** PUMPING POWER PLUS PROPELLANT CONDITIONING

*** HYDRAZINE PLUS H_2O COOLANT SYSTEM

✡ PRESSURANT

TABLE 1

SSAPU WEIGHT SUMMARY – ORBITER

COMPONENTS \ SYSTEM	A1	A2	A3	B1	B2	D1	D2	D3	D4	D5	D6	D7	D8	D9	A4	D3A	C1	D10	E1	E2	E3	E4
TURBINE, GEAR BOX, CONTAINMENT COMBUSTOR	67	68	66	70	70	67	68	66	70	70	68	68	66	70	68	68	110	68	67.4	67.4	65.7	65.7
20 KVA ALTERNATOR	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26
HYDRAULIC PUMP (2) PLUS VALVES PVC3– 30G	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
HYDROGEN BURNED PLUS 5% RESERVE	54	62	50	58	91	52 6**	60 7**	48 6**	88 11**	56 7**	76 9**	52 6**	55 6**	65 8**	61	60 7**	235	60 7**	231.6***	168.1***	261.5***	189.5***
OXYGEN BURNED PLUS 5% RESERVE	45	51	42	49	76	43 5**	49 6**	40 5**	73 9**	47 6**	63 7**	43 5**	46 6**	55 7**	50	48 6**		49 6**	0.1✱	0.1✱	1.6✱	1.1✱
H_2 HEAT EXCHANGER	8	8	7	9	10	4	4	4	5	4	5	4	4	4	7	4	60	4	NA	NA	NA	NA
O_2 HEAT EXCHANGER	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	30	2	NA	NA	NA	NA
LUBE OIL COOLER	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	20	3	5.0	5.0	5.0	5.0
HYDRAULIC COOLER	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	4	72*	4	72.2*	72.2*	72.2*	72.2*
HYDROGEN PUMP PLUS DRIVE	12	12	12	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	NA	12	NA	NA	NA	3.5	3.5	NA	NA
SUPERCRITICAL HEATERS, FAN	5 (O_2)	5 (O_2)	5 (O_2)	10	10	NA	NA	NA	NA	NA	NA	NA	NA	NA	5 (O_2)	NA	NA	NA	NA	NA	NA	NA
VALVES, FILTERS, INSTRUMENTATION, LINES, CONTROLS	25	25	25	25	25	20	20	20	20	20	20	20	20	20	25	20	65	20	25	25	20	20
INSTALLATION, SUPPORT STRUCTURE	21	21	21	21	21	18	18	18	19	19	18	18	18	18	21	18	45	18	27.1	26.8	28.3	27.5
TOTAL	332	347	323	337	398	310	327	302	390	324	361	311	316	342	344	326	–	327	517.9	454.1	540.3	467.0
H_2 TANK	10 (OMS)	11 (OMS)	9 (OMS)	33 (OMS)	53 (OMS)	10 (OMS)	12 (OMS)	10 (OMS)	14 (OMS)	10	14 (OMS)	10 (OMS)	11	11	10 (OMS)	10 (OMS)	–	11	5.9***	6.4***	11.5***	8.3***
O_2 TANK	6 (OMS)	7 (OMS)	6 (OMS)	7 (OMS)	10 (OMS)	6 (OMS)	8 (OMS)	6 (OMS)	2 (OMS)	1	2 (OMS)	6 (OMS)	6	1	6 (OMS)	6 (OMS)	–	6	3.7	3.3	21.4	16.9
TOTAL	348	365	338	377	461	326	347	318	406	335	377	327	333	354	360	342	723	344	527.5	463.8	573.2	492.2
MISSION SPC* LB/HP-HR (HP-HR = 42)	2.24	2.56	2.08	2.42	3.79	2.15	2.47	2.00	3.65	2.53	3.15	2.15	2.28	2.64	2.52	2.46	5.30	2.47	5.26	3.81	5.94	4.30

* BASED ON BURNED PROPELLANT

** PUMPING POWER PLUS PROPELLANT CONDITIONING

*** HYDRAZINE PLUS H_2O COOLANT SYSTEM

✱ PRESSURANT

TABLE 2

COMMONALITY SYSTEM WEIGHT AND COST IMPACT

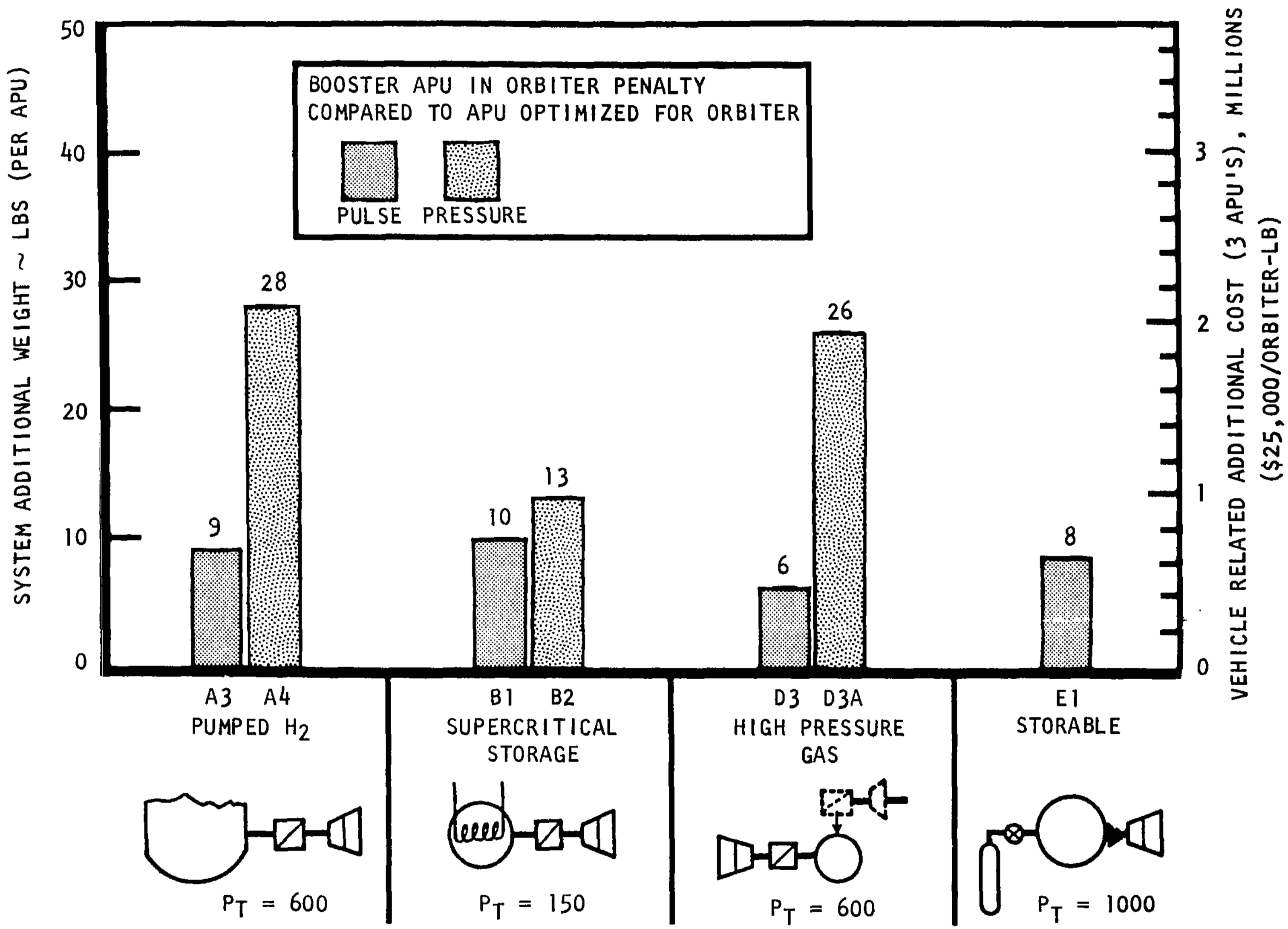


FIGURE 11

VEHICLE EQUIVALENT ORBITER APU WEIGHT AND RELATED COST PENALTY

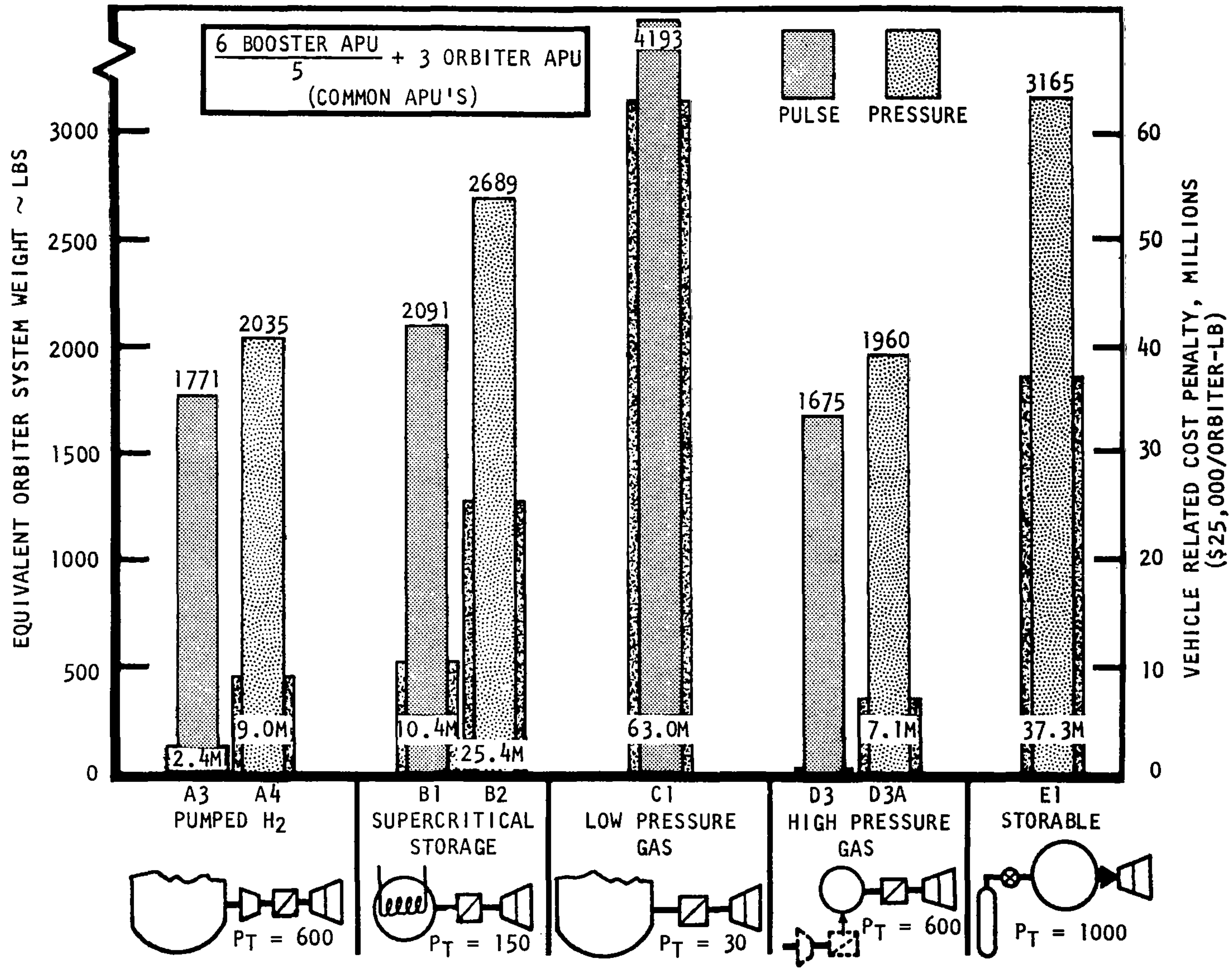


FIGURE 12

SYSTEM COST COMPARISON

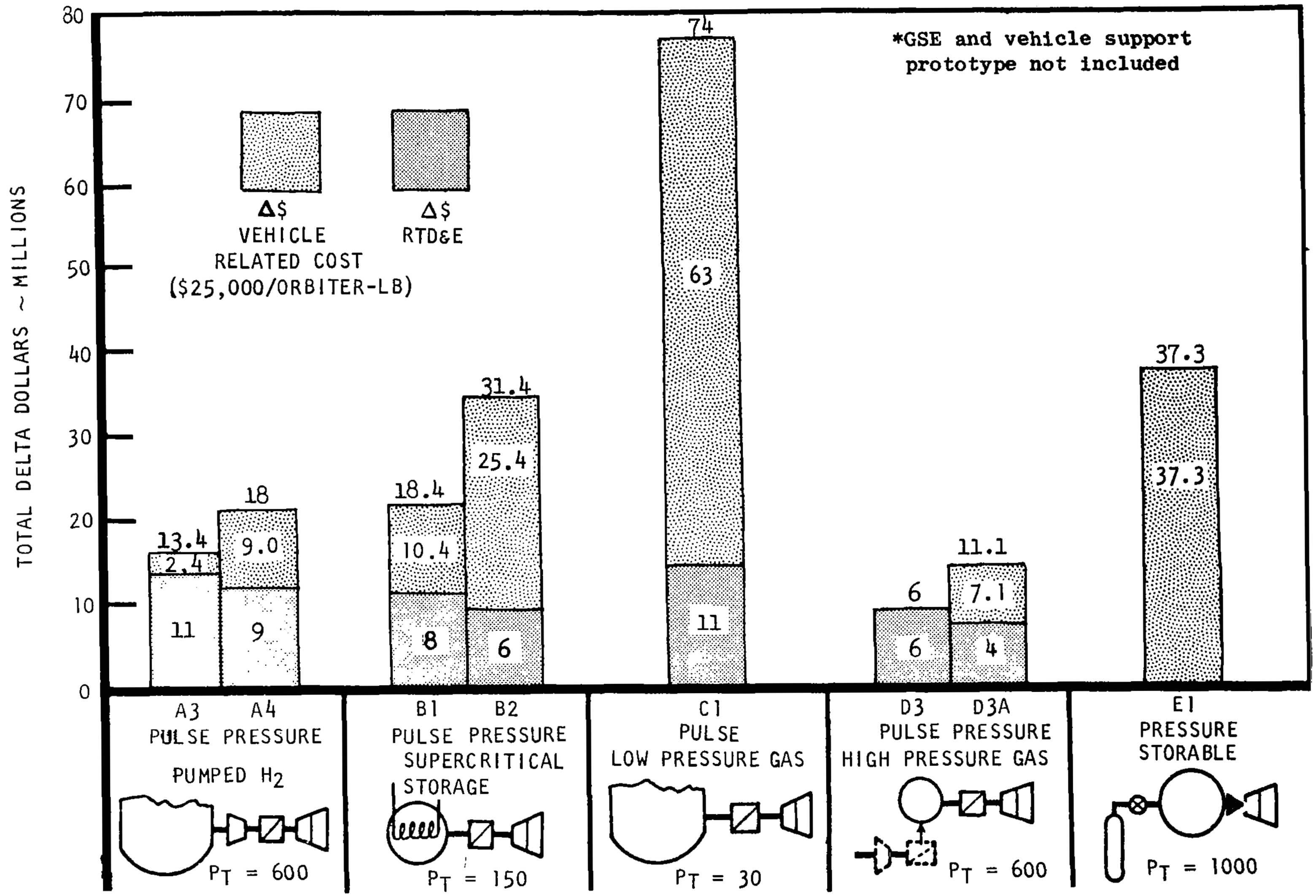


FIGURE 13

The various systems were rated for Reliability, Development Risk, Maintainability, and Flexibility as shown in Fig. 14. The weighting given to each factor is shown on the left. A high rating number is favorable. From this it is seen that the high pressure gas system is most reliable (as summarized below), the storable system most developed, the low- and high-pressure gas system easiest to maintain, and the pumped hydrogen system most flexible. Overall, the storable and high-pressure gas systems rate highest. The low maintainability rating of the hydrazine system must be emphasized. "In-flight" reliability is significantly degraded on all hydrazine components requiring close tolerances, tight clearances, and sliding fits due to detrimental effects of propellant residuals. Hydrazine in extended use leaves a residue which can reduce valve response times or in the extreme prevent valve movement. Achieving high reliability levels therefore is dependent on frequent maintenance with its attendant costs.

In systems selection criteria, these ratings must be considered together with weight and cost factors. Figure 14 also summarizes the cost data from Figure 13.

The APU systems studied were also evaluated for reliability. Acceptable reliable is obtained for all of the various options, The most significant influences on relative system reliability are due to pressure system elements (valves, regulators), propellant modulation control complexity, and tankage. Therefore, the low-pressure system having the fewest valves is better than the supercritical system which has more valves, and the pumped system which requires a pump. Pulse modulation is more reliable than pressure modulation because a single bipropellant valve replaces the dual

SYSTEM RATING
(COMMON APU)

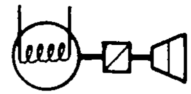
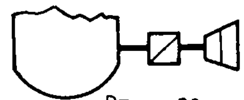
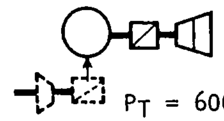
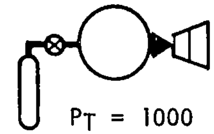
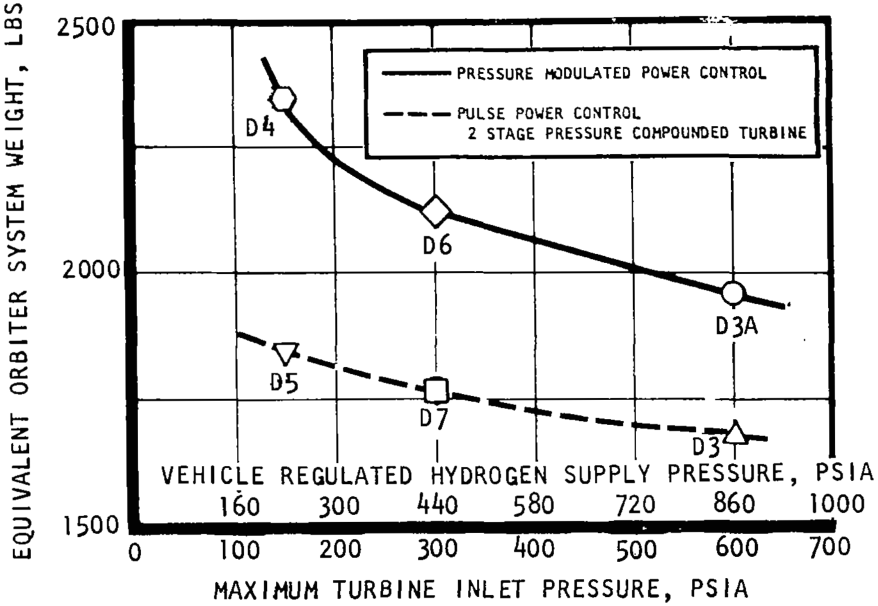
		A		B		C	D		E
		PUMPED H_2  P_T = 600		SUPERCRITICAL STORAGE  P_T = 150		LOW PRESSURE GAS  P_T = 30	HIGH PRESSURE GAS  P_T = 600		STORABLE  P_T = 1000
		A3	A4	B1	B2	C1	D3	D3A	E1
RATING									
RELIABILITY	35	11	11	16	16	18	19	19	16
DEVELOPMENT RISK	10	4	6	6	8	6	7	9	10
MAINTAIN-ABILITY	5	3	3	4	4	5	5	5	1
FLEXIBILITY	20	17	12	15	11	4	14	10	20
TOTAL	70	35	32	41	39	33	45	43	47
COST-DELTA MILLION DOLLARS									
VEHICLE RELATED + RTD&E		13.4	18	18.4	31.4	74	6	11.1	37.3

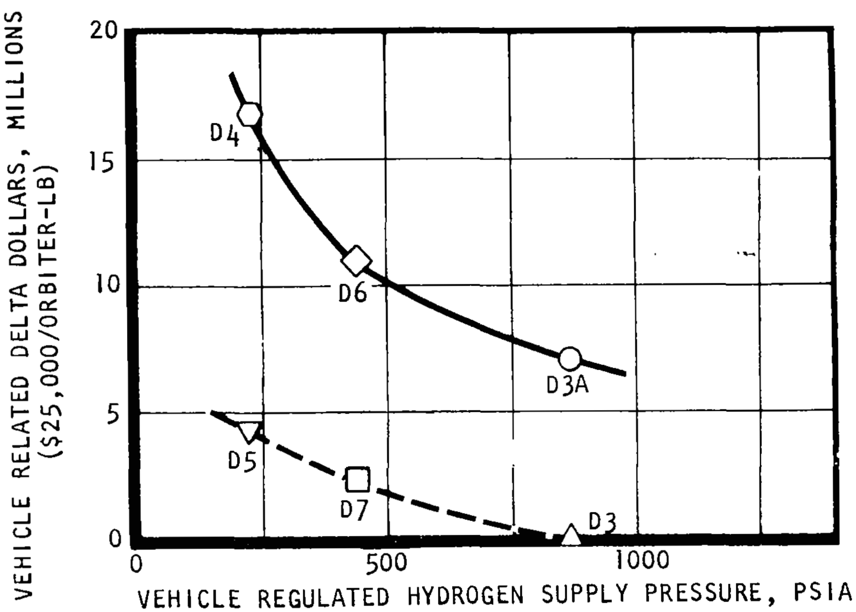
FIGURE 14

modulation valves and also provides shutoff capability without separate valving. Area modulation is least reliable because of required complexity and development risk. If vehicle tankage can be used instead of separate APU related tankage, reliability is enhanced because of the elimination of fill, drain, and relief controls.

The choice between pulse and pressure modulation for the power control is a function of propellant pressure available from the vehicle. Figure 15 illustrates the vehicle related cost and weight differences between a pulse and pressure modulated system as vehicle propellant supplied design pressure is varied. As the vehicle supply pressure level is reduced the pulse control system weight advantage increases. The vehicle supply pressure must be large enough to accommodate control and injector pressure drops as well as line losses.

As a result of the study, a vehicle integrated, high-pressure gas supply system was selected. The conclusions are summarized in Fig. 16. The H_2/O_2 would be supplied to the APU by propellant accumulators sized to supply the attitude control system(s). Figure 16 also indicates the region of overlap where pulse and/or hybrid as well as pressure modulated power control systems are applicable. The overlap region comprises a grey area where the systems are competitive and no clear cut overall advantage for either exists.





SYSTEM D-VEHICLE EQUIVALENT ORBITER WEIGHT AND DELTA DOLLARS

$\frac{6 \text{ BOOSTER APU}}{5}$ + 3 ORBITER APU

(COMMON APU'S)

HIGH PRESSURE GAS

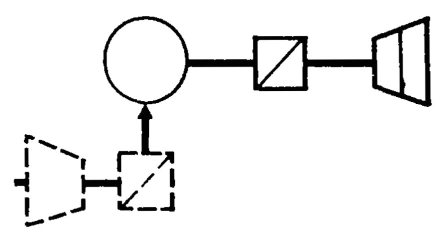
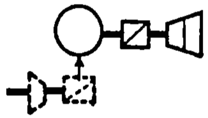


FIGURE 15

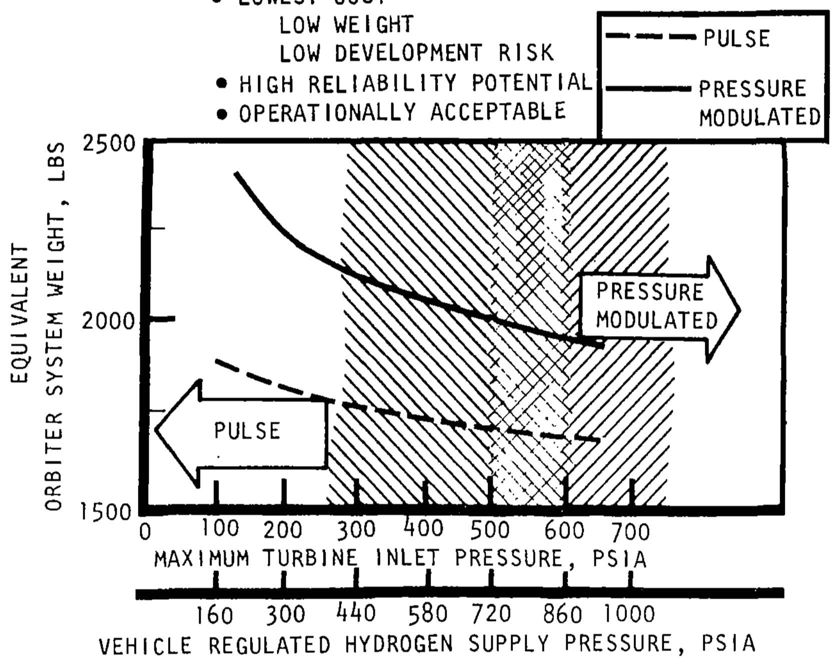
SUMMARY RECOMMENDATIONS

INTEGRATED

VEHICLE HIGH PRESSURE GAS SUPPLY
SYSTEM D



- LOWEST COST
  LOW WEIGHT
  LOW DEVELOPMENT RISK
- HIGH RELIABILITY POTENTIAL
- OPERATIONALLY ACCEPTABLE



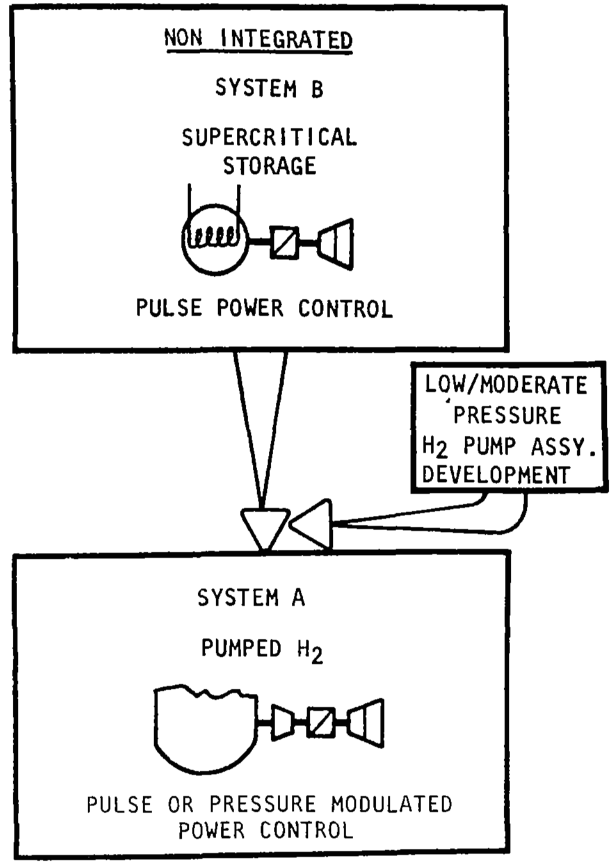


FIGURE 16

In the event that propellant is not made available from the vehicle and a nonintegrated system must be selected, either the supercritical storage system (B-1) with pulse or hybrid power control or the pumped hydrogen system appear to be the best compromise.

The nonintegrated pumped system saves considerable weight over the supercritical system, but a reliable hydrogen pump with the required flow and modulation capacity is not available. It is, therefore, recommended that applicable pump technology be developed if a nonintegrated system is contemplated.

In comparing various systems, it is also necessary to summarize the status of the applicable technology. Figures 17 and 18 identify the technology status for the H_2/O_2 and the storable propellant systems. Those components and functions of the H_2/O_2 APU requiring an advancement in technology are shown in Fig. 17. An asterisk is placed next to those items which are deemed critical to the APU development, and the extent of the technology effort required is estimated.

If a vehicle gas supply system is developed, the hydrogen pump assembly, drive and propellant acquisition are not applicable. The deep throttling requirement of the combustor assembly is not applicable to a pulsed system, and is substantially reduced for a hybrid system. If a pulsed power control is not selected, the valve and combustor cycles life requirement as well as the turbine blade thermal cycling are not applicable.

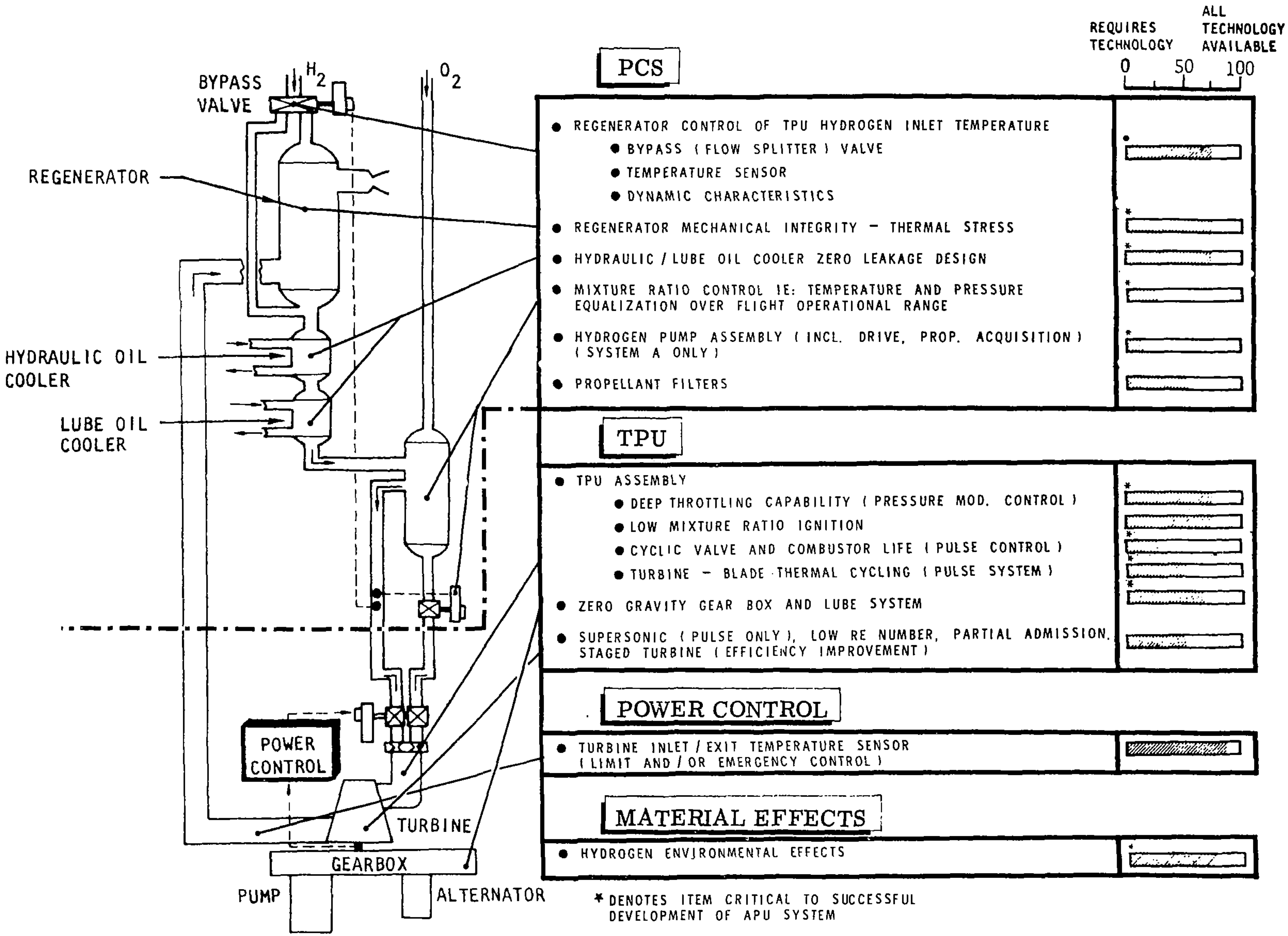
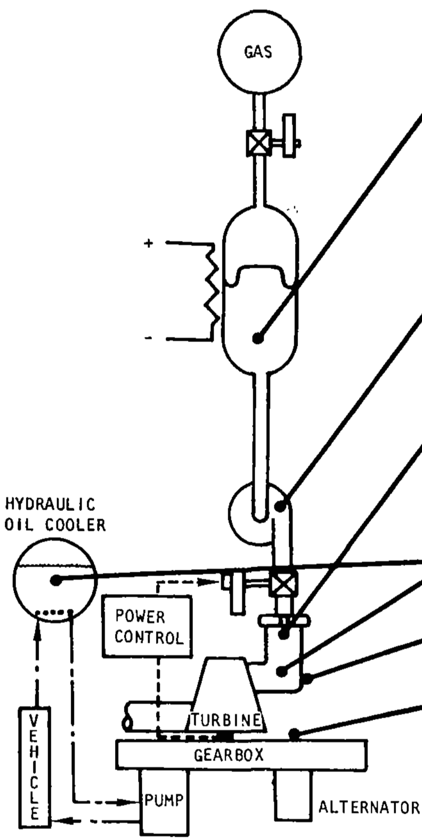


Figure 17. Technology Requirements

TECHNOLOGY STATUS

REQUIRES DEVELOPMENT — 0 — 50 — 100 — DEVELOPED



PROPELLANT HYDRAZINE	CONSIDERABLE EXPERIENCE (FLIGHT DATA AVAILABLE) 10^6 CYCLES DEMONSTRATED REQUIRED SIZE TESTED REQUIRES EXTRA GROUND SUPPORT EQUIPMENT FOR APU
HYDRAZINE BLENDS	SOME EXPERIENCE CARBON FORMATION IN TURBINES EXPERIENCED REQUIRES EXTRA GROUND SUPPORT EQUIPMENT FOR APU
PROPELLANT FEED PUMPED	PUMP DEVELOPMENT REQUIRED MK 36 AVAILABLE DESIGN SEAL REQUIRES GAS PURGE BEARINGS DEVELOPED (DN $\sim 10^6$)
POSITIVE EXPULSION	BLADDERS AND BELLOWS AVAILABLE (100 TO 200 CYCLES) TITANIUM TANKS DEVELOPED
IGNITION HEATED THERMAL BED	DEVELOPED FOR THRUSTERS RELIABLE OPERATION REQUIRES APU DEVELOPMENT IN PROPER SIZE
CATALYTIC	HIGHLY DEVELOPED (FLIGHT EXPERIENCE) LIMITED LIFE RESTART PROBLEM IF SHUTDOWN IN ATMOSPHERE
HYPERGOLIC	BI-PROPELLANT EXPERIENCE MUST BE ADAPTED CONTINUOUS DECOMPOSITION (TEMPERATURE) CONTROL DIFFICULT
COMBUSTOR	WELL DEVELOPED REQUIRES SUFFICIENT L^* TO CONTROL DECOMPOSITION
HYDRAULIC OIL COOLER — POOL BOILER	REQUIRES DEVELOPMENT REQUIRES "BURNOUT" PROTECTION IN ZERO-G
MATERIALS	COMPATIBILITY WITH HYDRAZINE OF MOST METALS DEMONSTRATED (Al, CRES, Ti, PLASTICS)
TURBOPOWER UNIT	TECHNOLOGY DEVELOPED REQUIRES DEMONSTRATION IN SIZE RANGE SIMILAR TO DEVELOPMENT H_2/O_2 TPU EXCEPT SEALS - DEMONSTRATED COMPATIBLE WITH OIL TEMPERATURE - HIGHER BUT TIP SPEED LOWER
CONTROLS	WELL DEVELOPED FROM ROCKET TECHNOLOGY MULTIPLE IGNITION DEMONSTRATED ADEQUATE DECOMPOSITION DEMONSTRATED

FIGURE 18

It is recommended that the major technology effort be concentrated on the reference propellant conditioning system. Developmentof this system is mandatory for the successful development of the APU system. Furthermore, it is useable with any power-control selection, pulse, pressure, and hybrid controls.

From Fig. 18 it is seen that most aspects of the hydrazine system are well developed but that some developmental effort will be required before a system is usable.

PRELIMINARY ANALYSIS - PHASE IA

At the outset of the program, a preliminary analysis was performed to identify the necessary component, subsystem, and system information to provide a comparison between candidate systems. A more refined analysis was then performed in Phase IB. Figure 1 describes the various component-oriented combinations evaluated in Phase IA for each of the three major subsystems, i.e., Propellant Feed, Turbo Power Unit and Power Controls. The Propellant Feed Subsystem includes the necessary tankage and pressurizing or pumping equipment as well as the propellant conditioning subsystem. The subsystems were chosen and combined into various system combinations to allow comparative evaluation for selection of the "best" combination of components and subsystems. The propellant systems chosen are representative of the wide range of vehicle influenced propellant supplies which may be available.

The results of this study were presented in the First Monthly Technical Progress Narrative covering work done in September 1970. Key elements of this report are reprinted here as Appendix A for completeness and reference.

APU TRADE OFF STUDIES - PHASE IB

APU SYSTEM OPTIMIZATION

The booster and orbiter mission profiles require APU operation over a wide range of power at altitudes varying from sea level to vacuum ambient conditions. Consequently, turbopower unit operation can be achieved at the maximum efficiency design point for only a part of the total mission duration. Total APU system weight, of which the propellant is a major portion, is then strongly influenced by turbine off-design performance. Selection of the turbine design point resulting in a minimum weight system must then be accomplished by parametric analysis.

During Phase IA, two digital computer programs were written to calculate total system weight based upon a given turbine design and mission profile. Program SSAPU utilized a pressure modulation power control and SSAPU2 a pulse modulation power control. Inputs to the programs consist of fixed weights, such as the turbine, gearbox, hydraulic pumps, alternator, combustor, valves, etc, turbine design/off-design characteristics, and mission power, altitude, and duration requirements. Total propellant required and its associated tankage weight for various types of sources (supercritical, low pressure, vehicle integrated or self-contained) is combined with fixed weight, heat exchanger, hydrogen pumps, and supercritical heater weights, with 10% of fixed weight for support structure, to determine APU total weight.

Turbine performance degradation occurs for pressure ratios less than design, which can result from throttling and back pressure increases for pressure modulation control, and back pressure increases for pulse control. In the pressure modulated system a degradation in injector combustion efficiency due to throttling occurs. In the pulse system degradation occurs due to the time lag required to build up to peak (design) turbine inlet pressure after opening the bipropellant valves. These degradations are accounted for in the programs, Additional assumptions include allowance for 5% propellant reserve at the end of the mission, 95% expulsion efficiency, and 5% volume ullage at the end of propellant fill. Tankage material is 2219-T87 aluminum with a safety factor of 2.0 on the ultimate for pressurized designs or minimum wall of 0.020 inch for low pressure (40 psia) tankage. An allowance of 20% of shell weight was made for flanges and welds.

Phase IA system weight analysis was performed for an idealized mission profile and several turbine designs ranging from 2 stage velocity and pressure compounded supersonic machines to a 4 stage pressure compounded subsonic machine.

Results of the Phase IA analysis (see Preliminary Analysis-Phase IA-Appendix) were reviewed by NASA and the systems to be carried into the Phase IB study were selected.

For Phase IB a new profile was provided for the booster, with an additional analysis requirement to determine optimum orbiter APU system weight based on an orbiter mission profile. The effect on system weight of using the booster APU in the orbiter vehicle was also to be explored. In addition, analysis to determine the sensitivity of system weight to variations in mission duration and power level was requested.

The power and altitude profiles were approximated by discrete slices for computer input with 18 intervals for the booster (Fig. 19) and 15 intervals for the orbiter (Fig. 20). A total of 18 different H_2/O_2 system types were evaluated assuming vehicle integrated tankage for a high and low pressure gas source, supercritical storage, and pump fed hydrogen and four storable systems were evaluated. Three types of power control were considered: pressure modulation, pulse modulation, which were investigated in detail, and hybrid on which a preliminary analysis was conducted. The hybrid control utilizes two combustors each feeding a separate nozzle block to provide close to design operation at both mode and peak power. The low power combustor is controlled by pressure modulation and the high powered combustor utilizes either pulsing or pressure modulating control. Both velocity and pressure compounded two stage turbines were considered in various combinations with propellant sources and power controls, as shown in Fig. 2 .

Computer results are presented in Figures 21 and 22 for the booster and orbiter APUs showing the effect of turbine design pressure ratio on total system weight for fixed maximum inlet pressure as indicated in Fig. 2. Results showing optimum system weights and booster and orbiter APU weight is summarized in Fig. 8. The supercritical and low pressure gas systems are not competitive on a weight basis with either the pump fed or high pressure gas systems. The lightest weight system utilizes 600 psia gaseous propellant with a pulse controlled pressure compounded turbine (D3). The optimum booster APU weighs 586 pounds and the optimum orbiter 318 pounds. The weight penalty for using the optimum booster APU in the orbiter is only 6 pounds (2%). The use of a pulse power control compared to a pressure modulated control saves 127 pounds on the booster APU (18%) and 24 pounds on

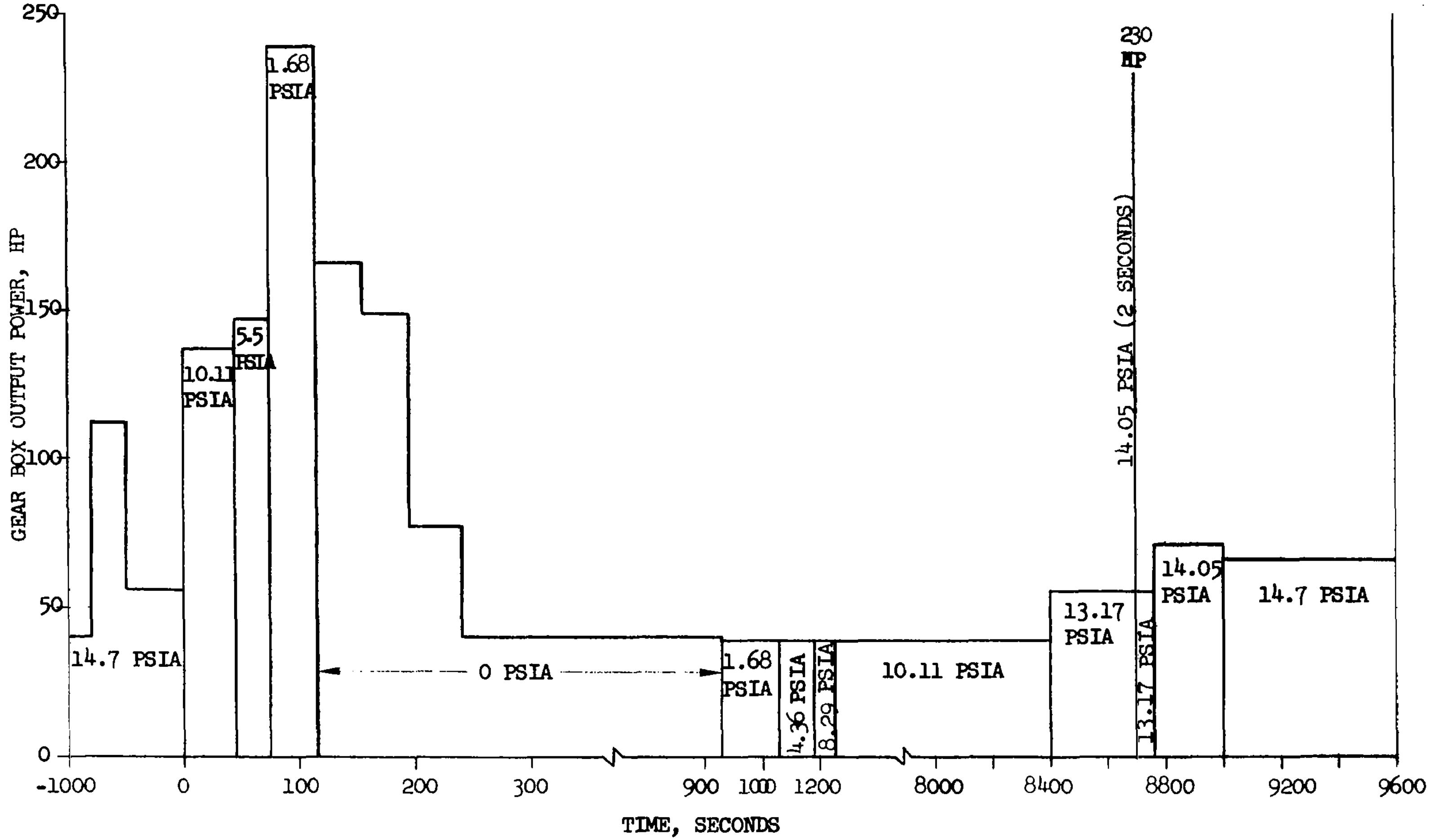


Figure 19 . Booster Mission Profile, Computer Input

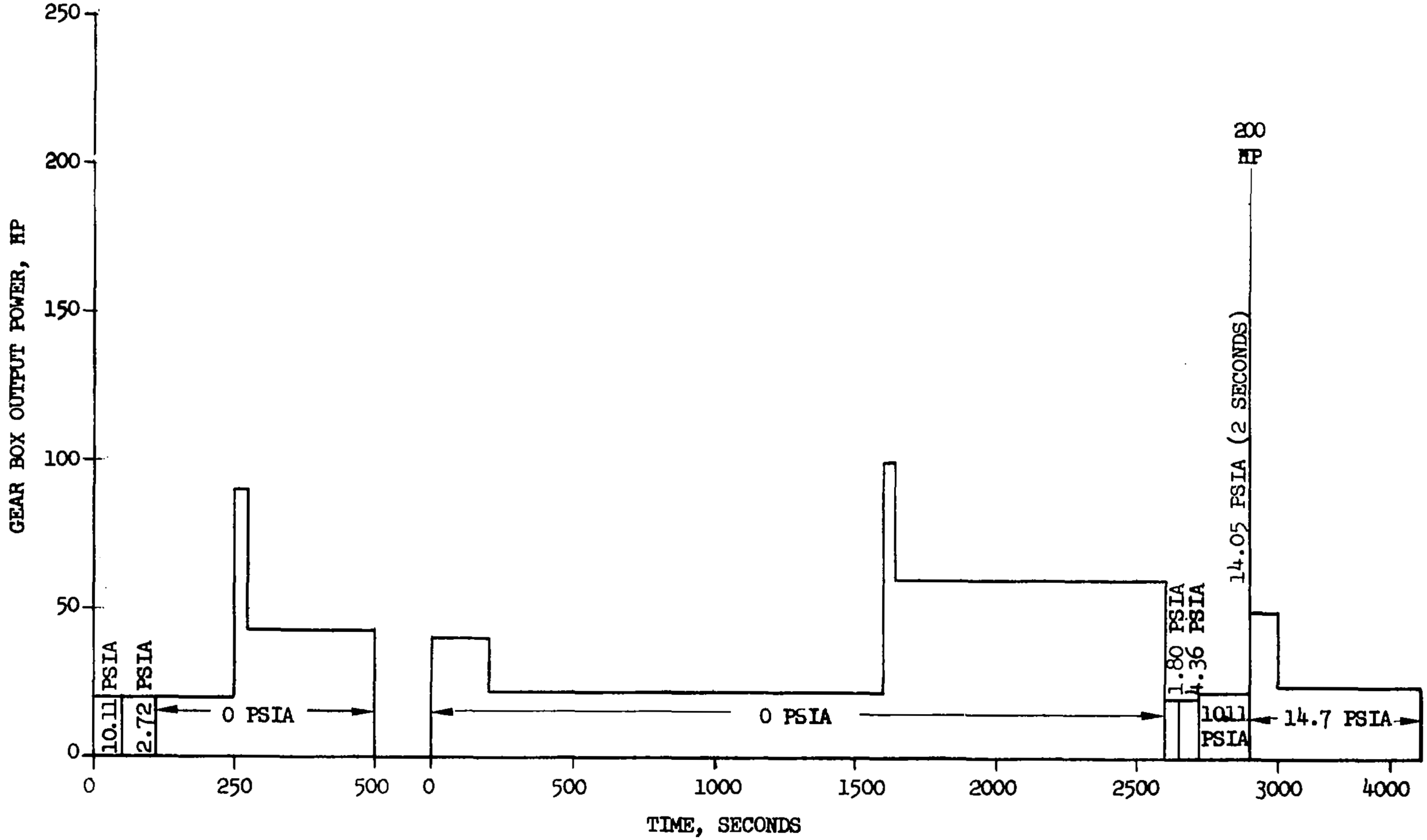


Figure 20. Orbiter Mission Profile, Computer Input

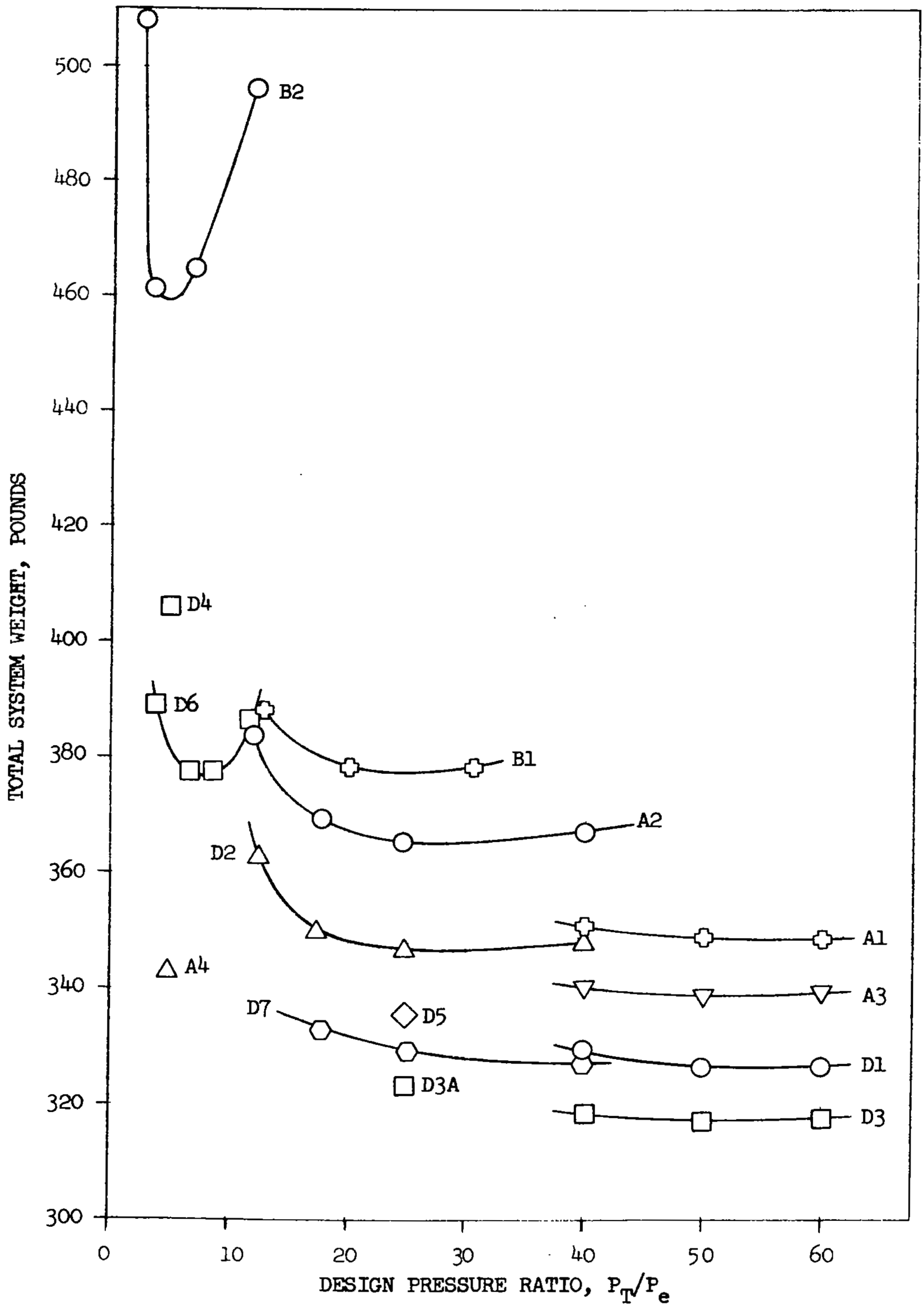


Figure 21. Orbiter APU - System Weight Optimization

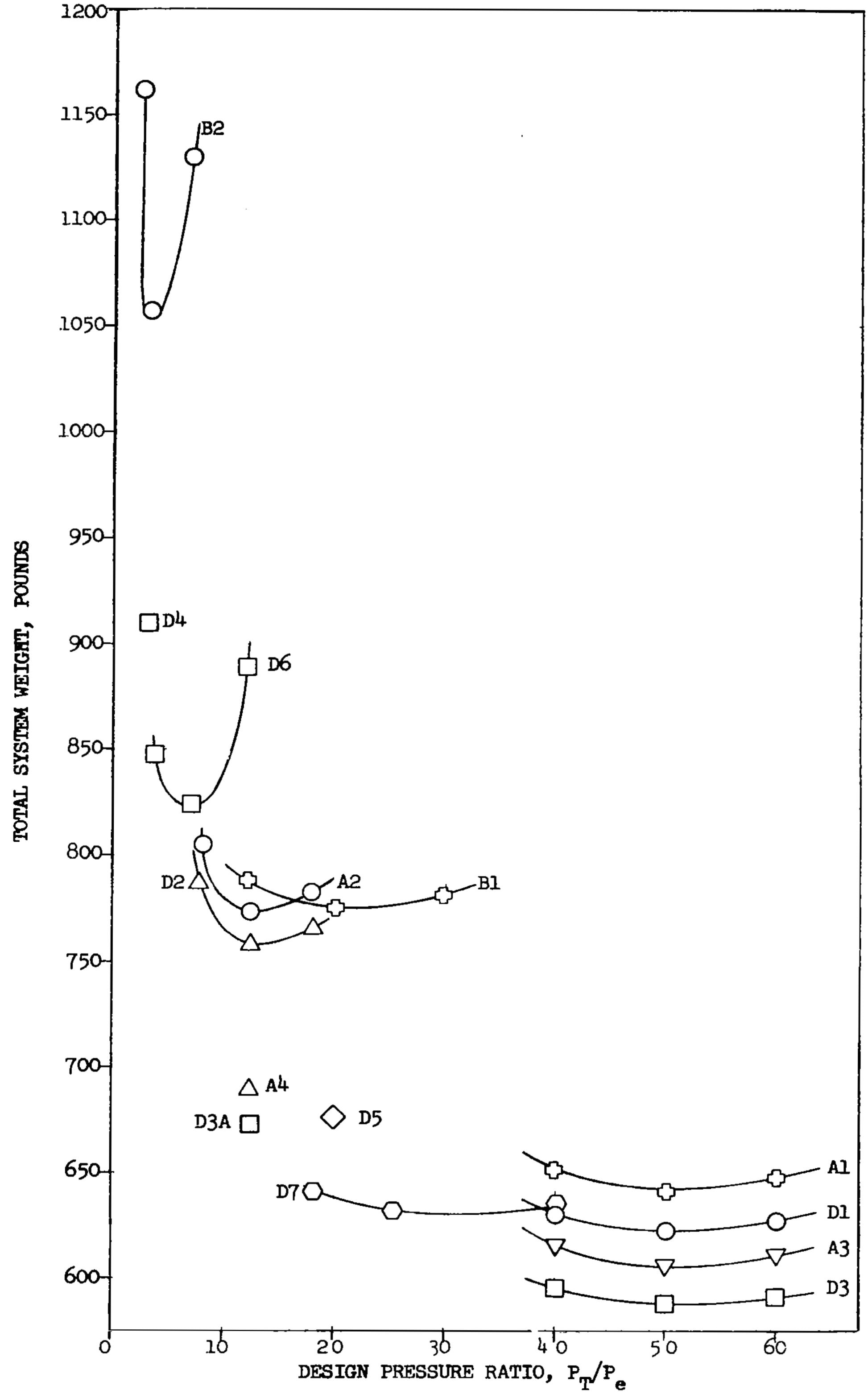


Figure 22. Booster APU - System Weight Optimization

the orbiter APU (7%). This is due primarily to the effect of ambient pressure and power level on SPC, which is presented in Fig. 23 . The pulse controlled system (D3) SPC shows modest sensitivity (10% increase) to reduction in power and increased back pressure operation while the pressure modulated system (D3A) is very sensitive (30% increase). Changing the propellant source to a pumped hydrogen - supercritical oxygen system incurs a penalty of 22 pounds for the booster and 20 pounds for the orbiter (A3). A hybrid power control with a 600 psia gas source weighs 637 pounds for the booster and 333 pounds for the orbiter (D8), an increase of about 9% and 5% respectively, compared with a pulse power control. A breakdown of each system with respect to individual component weight was presented in Tables 1 and 2 for the booster and orbiter APU, respectively. The mission SPC based upon gearbox output and burned propellant is indicated for each system. System D3 has the lowest SPC, 2.06 lb/HP-HR for the booster and 2.00 lb/HP-HR for the orbiter.

Differences in total system weight and SPC as affected by power control and system type are presented in bar chart form in Figures 24 , 25 , and 10 for the optimum booster, orbiter, and booster APU in the orbiter, respectively. The sensitivity of system weight and SPC with maximum turbine inlet pressure was determined for a gaseous propellant source pressure ranging from 230 psia (P_T = 150) to 860 psia (P_T = 600) for pressure, pulse, and hybrid control. Results are presented in Figures 26 , 27 , and 28. A reduction in source pressure from 860 psia to 230 psia increases the booster APU pressure controlled system weight penalty from 127 pounds to 235 pounds. The heaviest pulse system (D5, P_T = 150 psia) is 40 pounds lighter than the lightest pressure modulated system (D3A, P_T = 600 psia).

EFFECT OF AMBIENT PRESSURE AND POWER LEVEL ON SPC
(GEARBOX OUTPUT)

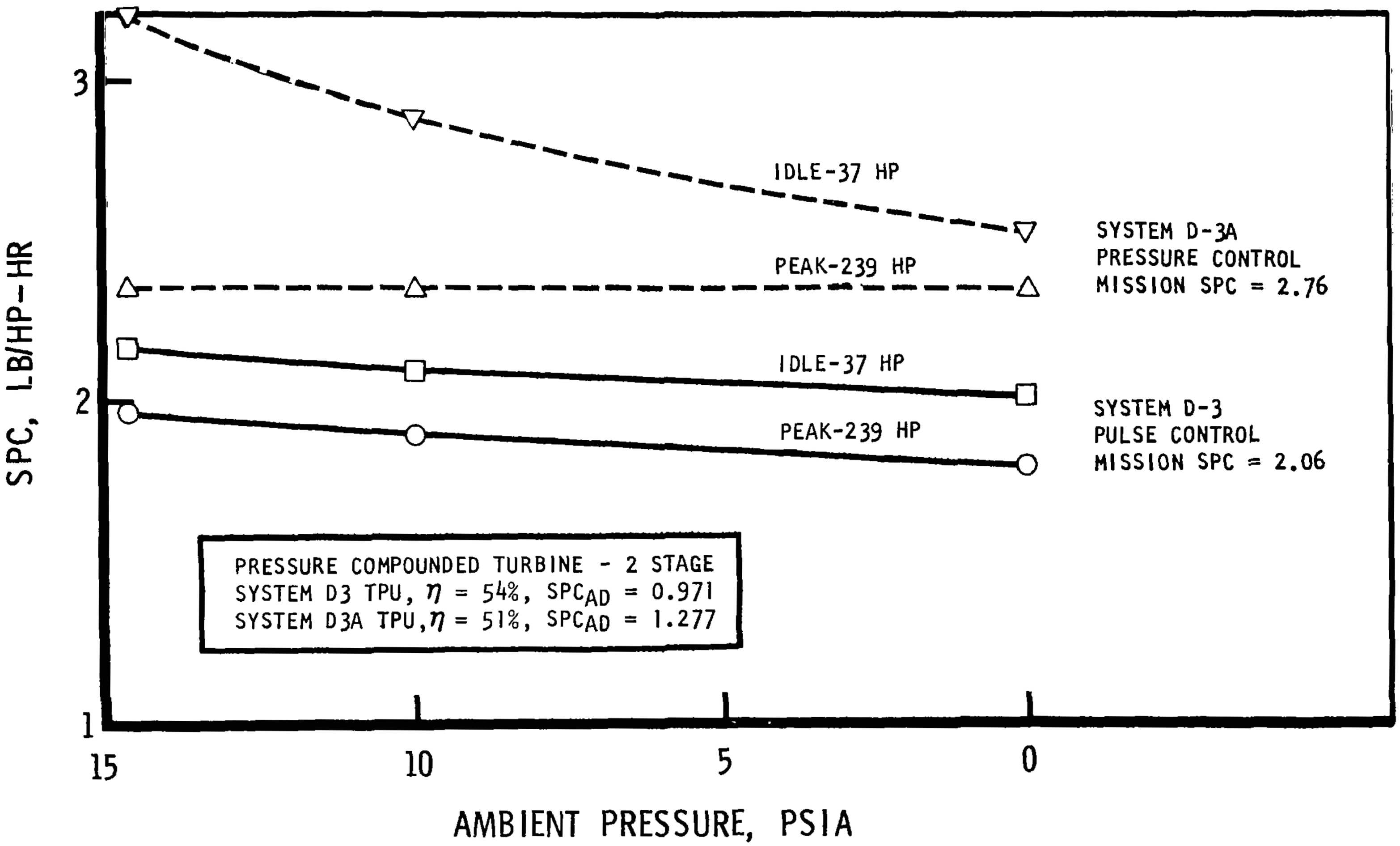


FIGURE 23

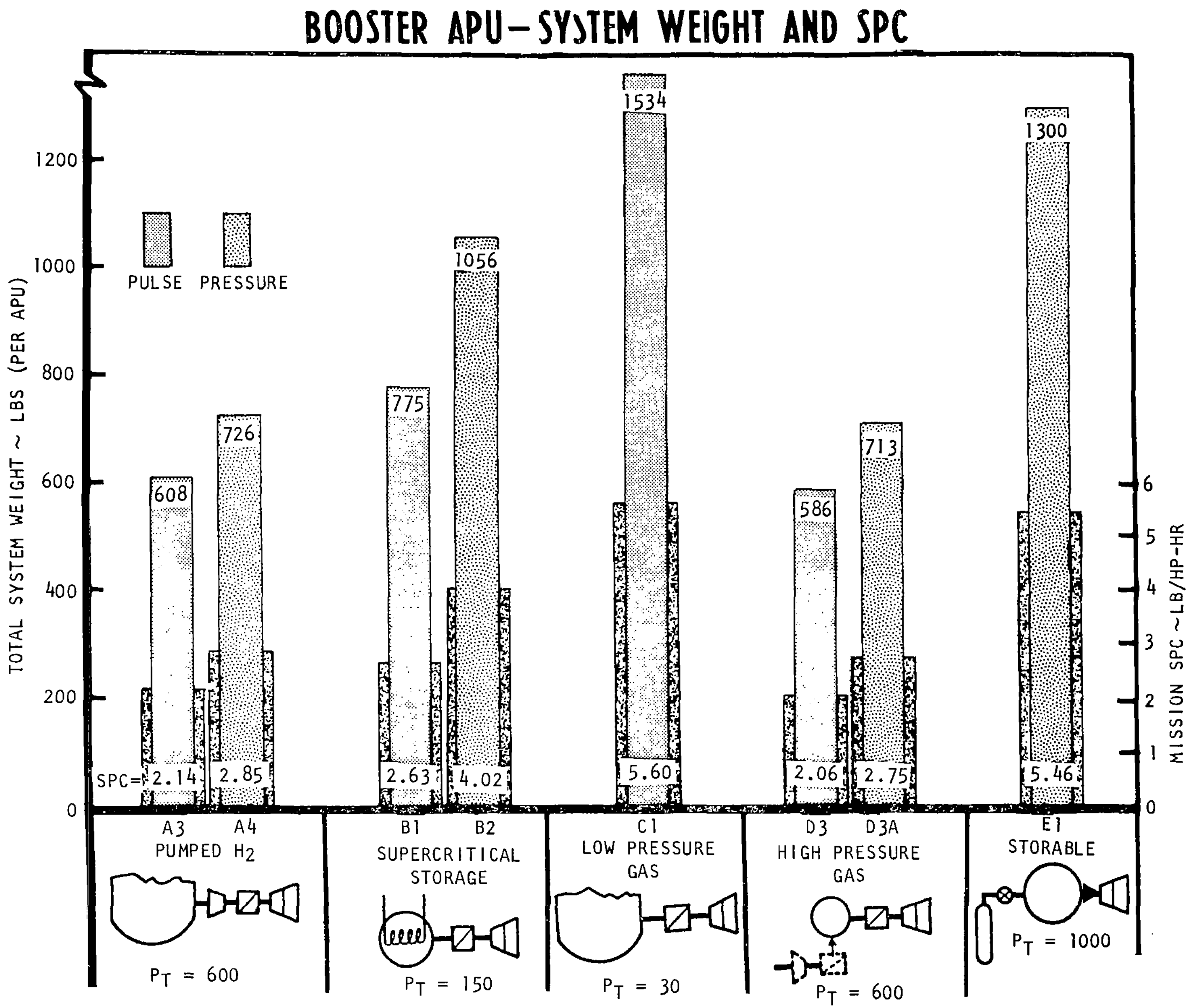

BOOSTER APU-SYSTEM WEIGHT AND SPC
TOTAL SYSTEM WEIGHT ~ LBS (PER APU)
MISSION SPC ~LB/HP-HR
PULSE
PRESSURE
608
726
775
1056
1534
586
713
1300
SPC=
2.14
2.85
2.63
4.02
5.60
2.06
2.75
5.46
A3
A4
PUMPED H2
B1
B2
SUPERCRITICAL STORAGE
C1
LOW PRESSURE GAS
D3
D3A
HIGH PRESSURE GAS
E1
STORABLE
PT = 600
PT = 150
PT = 30
PT = 600
PT = 1000

FIGURE 24

ORBITER/ORBITER APU–SYSTEM WEIGHT AND SPC

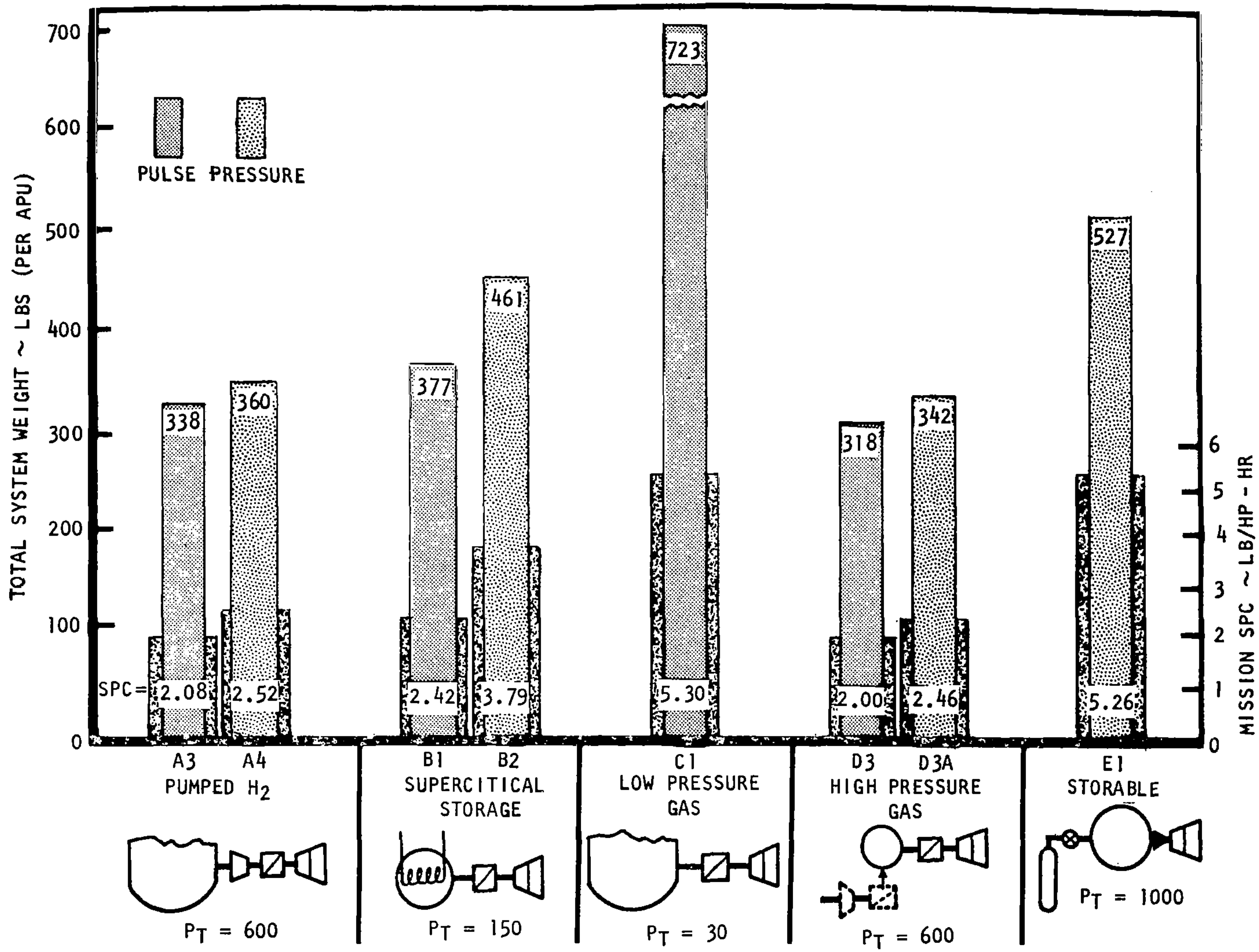
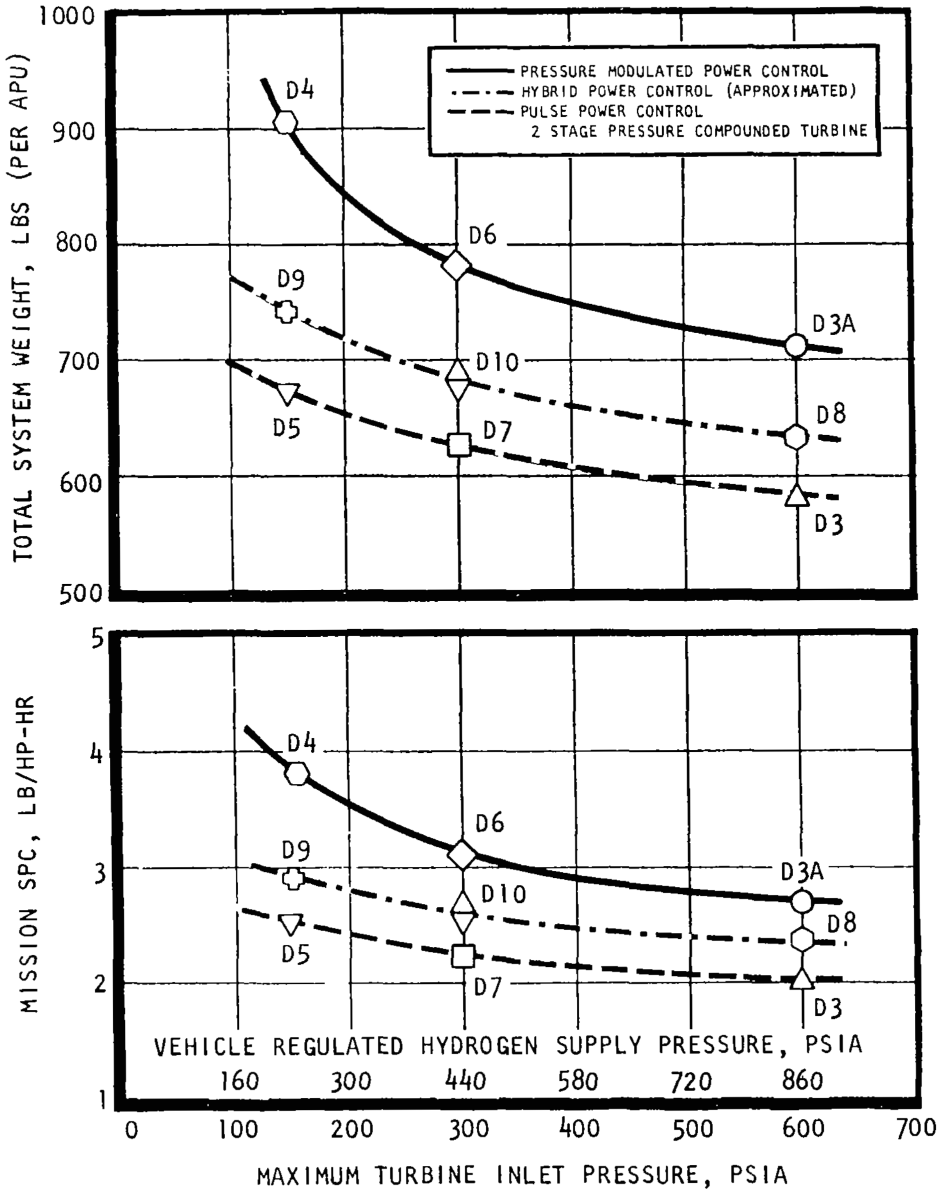


FIGURE 25

SYSTEM D-BOOSTER APU-SYSTEM WEIGHT AND SPC



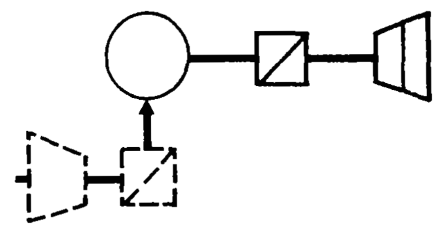
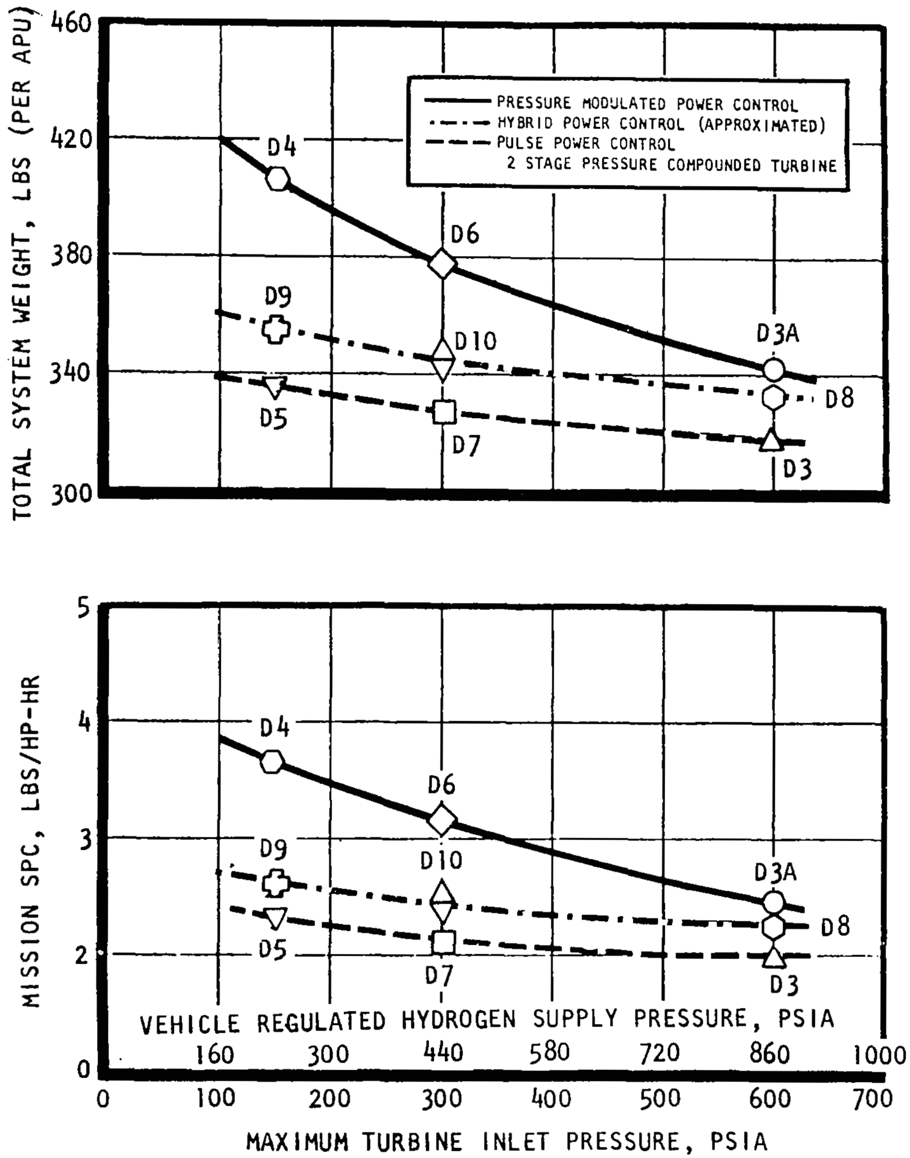


FIGURE 26

SYSTEM D-ORBITER/ORBITER APU
SYSTEM WEIGHT AND SPC



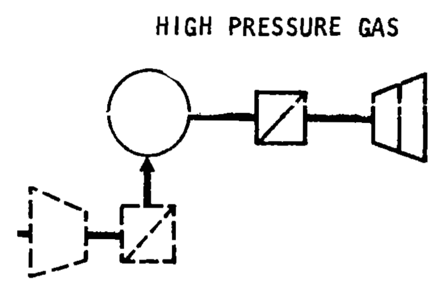
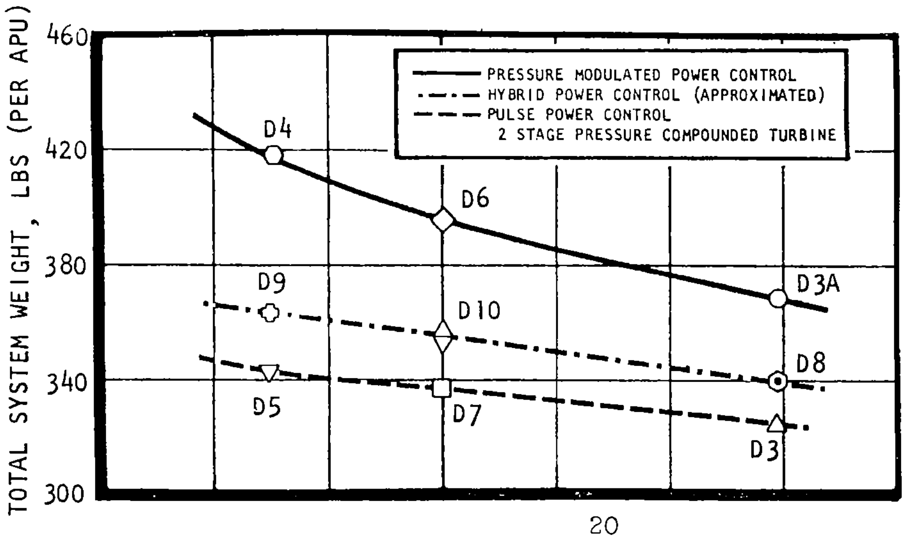
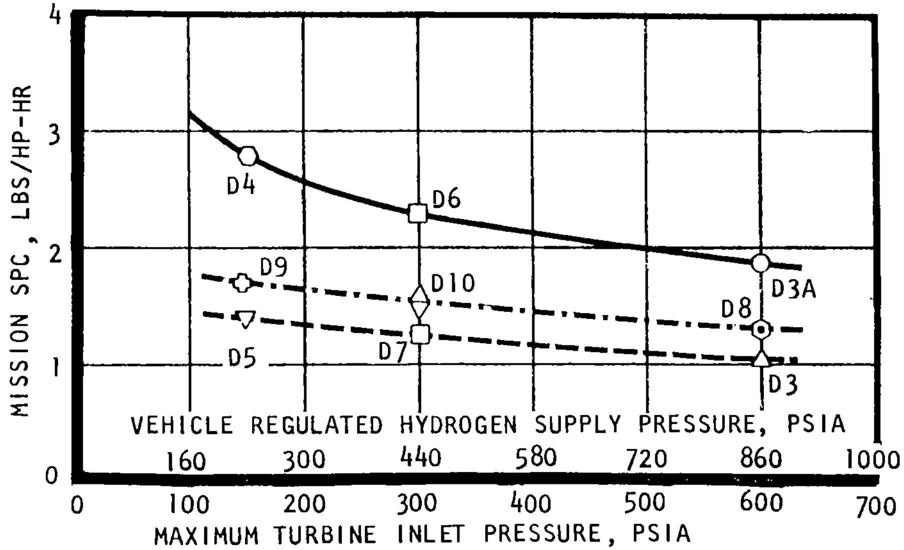


FIGURE 27

SYSTEM D—ORBITER/BOOSTER APU SYSTEM WEIGHT AND SPC





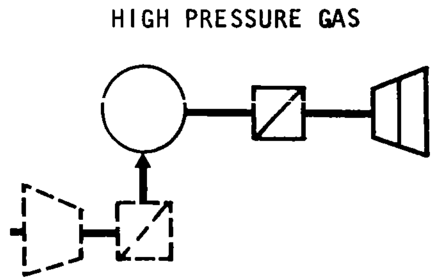
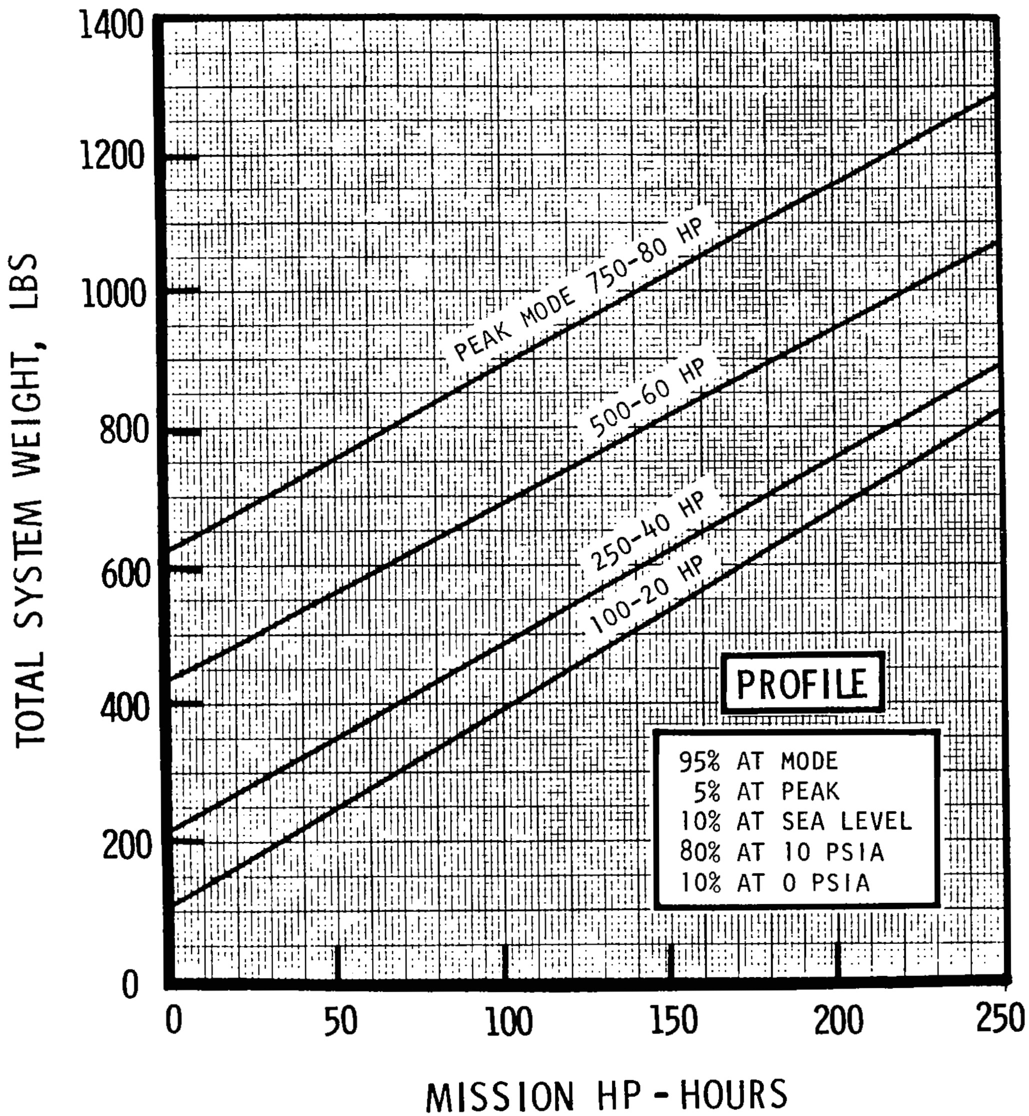


FIGURE 28

Effect of Energy Requirement

A special study was initiated during Phase IB to determine the sensitivity of total H_2/O_2 APU system weight to variations in mission energy requirements. The study was necessitated to provide input data for total vehicle system weight studies, recognizing the present uncertainty in APU power and duration estimates. An APU system utilizing high pressure (860 psia) vehicle supplied gaseous propellants at 200R was selected for evaluation. Turbine performance was calculated at several peak power levels for a two stage pressure compounded machine assuming a 600 psi inlet pressure at a design pressure ratio of 50 with a pulse modulated power control system. The effect of gearbox power level on turbine, combustor, gearbox, alternator, and hydraulic pump(s) weight was estimated from Rocketdyne and vendor technology data to provide fixed weight input to the APU system weight analysis computer program (SSAPU2). The mission booster profile was simplified by assuming operation at only two power levels (peak-mode) with 95% of mission duration at mode and 5% at peak. The altitude profile was assumed to be 10% operation at sea level, 80% at 10 psia, and 10% at vacuum. Four peak-mode power levels were analyzed over a range of mission energy requirement from 50 to 250 HP-HR.

Results of the study shown in Fig. 29 indicate that system weight increases at a rate of about 270 pounds per 100 HP-HR, approximately independent of peak-mode power level. The optimum booster APU total weight for system D3 for the Phase IB profile (137 HP-HR) falls on the 250-40 HP curve, so that this curve should be fairly representative of the effect of mission duration changes on the high pressure gas booster APU system weight (D3).



EFFECT OF POWER LEVEL AND MISSION ENERGY REQUIREMENTS ON TOTAL APU WEIGHT

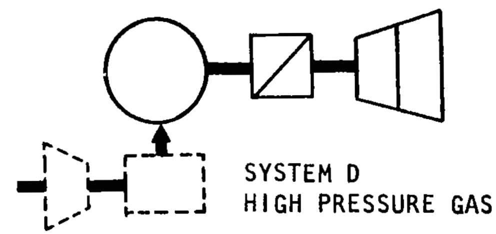


FIGURE 29

Effect of Source Pressure

The effect of hydrogen source pressure on total system weight for system A (pumped H_2, supercritical O_2) and B (supercritical H_2/O_2) is shown in Fig. 30 . The pressure modulated supercritical system (B) weight is essentially independent of hydrogen source pressure for levels below 600 psia. The improved SPC and corresponding reduction in propellant weight with increasing pressure is off-set by increased tankage weight. The pulse controlled supercritical system weight increases almost linearly with source pressure at a rate of about 0.2 lb/psi, with the increased tank weight dominating a slight SPC improvement with pressure. In the pump-fed system, low pressure tankage is used so that total system weight is reduced with increasing pressure. The pressure modulated system is more sensitive than the pulse controlled system because of the relatively large off-design performance degradation of the former at low reduced turbine inlet pressure. Present limitation on pump discharge pressure for low specific speed hydrogen pump (based on recent tests by Pesco, Div. of Borg Warner) is indicated on the figure. A 100-pound weight saving results with a pumped versus supercritical system based upon present technology (P_{DIS} = 260 psia). As pump discharge pressure increases,the weight savings increases significantly, approaching 200 pounds for a 600 psia source pressure, representing close to 5 million dollars in vehicle related and RTD & E costs. Pump development to achieve higher discharge pressures is thought feasible by Pesco for only a fraction of the cost savings, indicating strong incentive for selection of a pumped system over a supercritical system.

EFFECT OF HYDROGEN SOURCE PRESSURE ON SYSTEM WEIGHT
BOOSTER

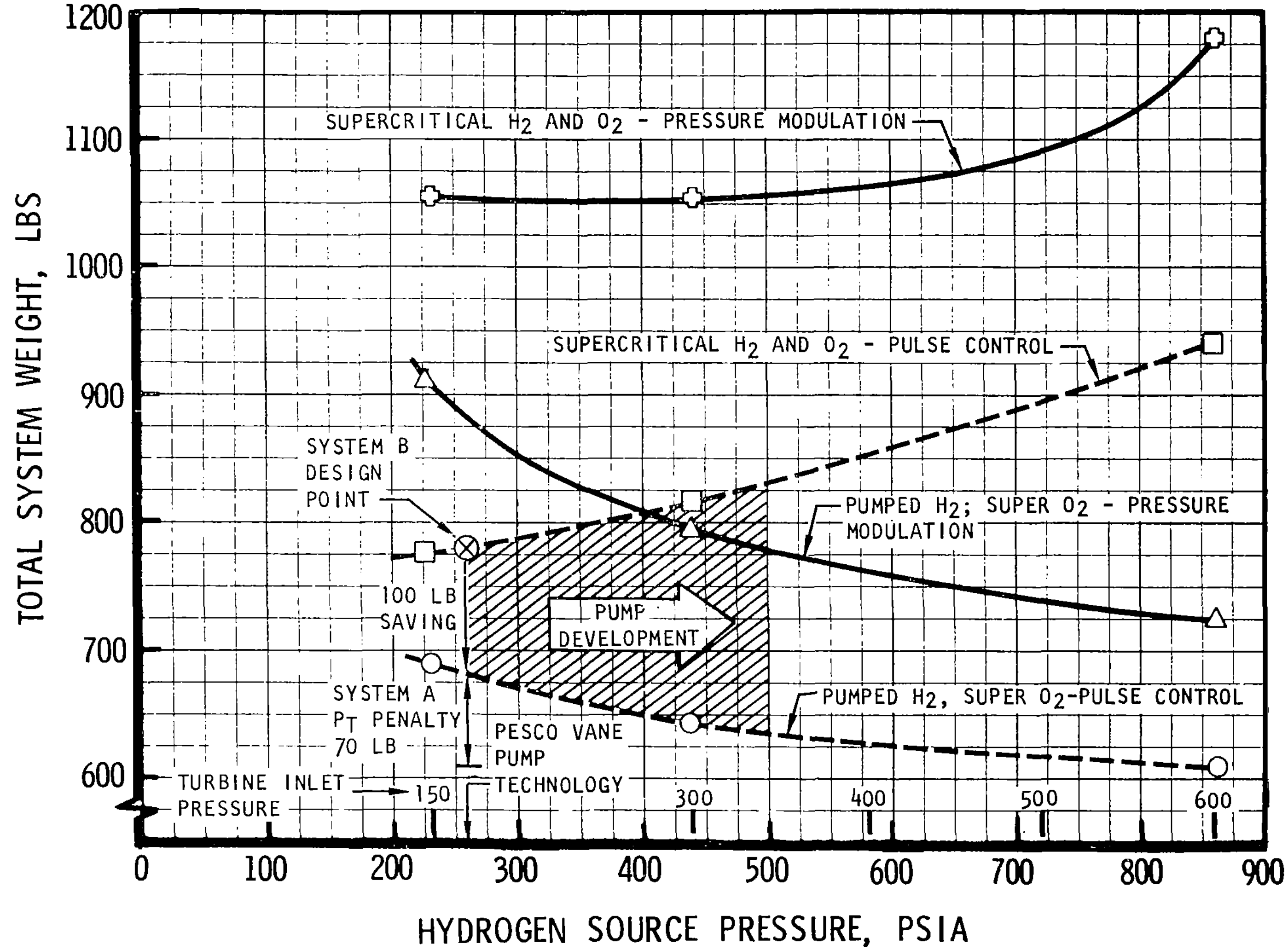


FIGURE 30

Effect of Combustor Inlet Temperature

The Phase 1B SS/APU system weight optimization analysis was based, in part, on an assumed combustor inlet temperature of 600 R, combustion temperature of 2015 R, mixture ratio of 0.835, and C-star efficiency of 98%. Selection of this operating condition was predicated on turbine stress considerations and analysis to evaluate the effect of combustor inlet temperature on propellant related system weight (propellant plus tankage plus regenerator). To determine the effect of other values of inlet temperature on system weight, a more detailed analysis was made.

System D3, utilizing high pressure (860 psia source) gaseous propellants at a supply temperature of 200 R, was selected to determine the optimum combustor inlet temperature for a fixed combustion temperature of 2015 R. Variations of specific impulse, mixture ratio, and spouting velocity with inlet temperature ranging from 200 R to 1000 R were evaluated using a digital computer program for the TPU optimum design inlet pressure (600 psia) and pressure ratio (50). At the preselected design point of 600 R, a total propellant weight of 298 pounds is required for the Phase 1B booster mission profile. This weight was recalculated for new inlet temperatures, taking the effect of changing specific impulse, turbine efficiency, and tank weight into account. Because the adiabatic head increases with increasing temperature, more energy is available. But, this also increases spouting velocity, decreasing u/C_o, and decreasing turbine efficiency. For purposes of analysis, two approaches were taken: turbine efficiency was assumed constant (optimistic) and turbine efficiency was assumed proportional to u/C_o (pessimistic). For the latter case, efficiency varies as 1/Isp for constant u because C_o is proportional to Isp.

Results of the study are shown in Fig. 31 indicating that there is little incentive to designing for higher than 600R inlet temperature. The total weight curves optimize at about 900R but indicate a weight saving of only 4 pounds for the variable η_T and about 12 pounds for the constant η_T case. The potential for improved bipropellant valve life for the 600R system over the 900R system could be traded against the small weight savings. If very large quantities of propellant were required higher inlet temperatures would be justified.

Effect of Cooling Requirements on Hydrogen Control

The contractual requirement that the APU be thermally self contained, i.e., that no heat be transferred to the vehicle, makes it necessary to use the hydrogen to cool the hydraulic oil and the lubricating oil. In both of these cases the heat load is fixed, so that the temperature rise in the hydrogen is determined.

It is, of course, also necessary to control the temperature of both the hydrogen and the oxygen entering the combustor chamber. Analysis has shown that there is little to be gained by allowing this temperature to exceed 600R. The design point has been chosen with inlet gas temperature at 600R. A temperature equalizer is positioned ahead of the combustor chamber to ensure equal temperature level in the two gases.

COMBUSTOR INLET TEMPERATURE OPTIMIZATION-SYSTEM D-3

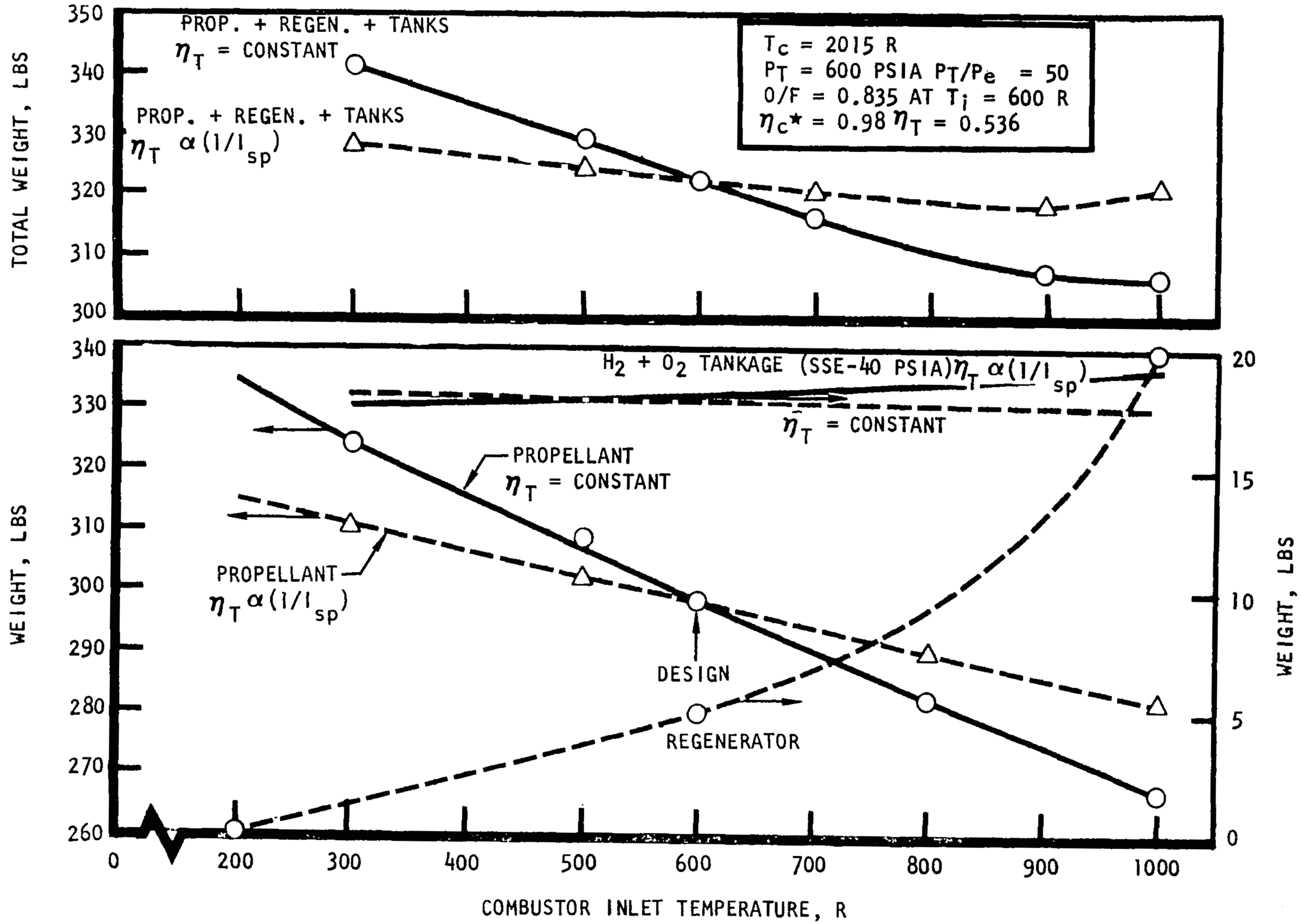


FIGURE31

Adjustment of gas temperature is accomplished in the regenerator where heat is acquired from the exhaust gases. By using a bypass around the regenerator, control is maintained.

The arrangement of the regenerator, hydraulic and lube-oil coolers has been studied and is shown in Fig. 32 . Because exposure of the oils to cryogenic temperatures may result in freezing, especially of "dead" fluid at startup, it is desirable to regenerate first. This ensures the supply of warm hydrogen. To maintain 600R (140F) at the inlet to the combustor, the minimum hydrogen temperature into the lube-oil cooler is 66F which is controlled by the regenerator bypass. The required hydraulic-oil cooler hydrogen inlet temperature varies between -305 (full hydraulic cooling) and +66F (no hydraulic cooling). The corresponding tube-wall temperatures vary between -100 and +150F which is above the pour point of the hydraulic fluid. If the relative location of the cooler is reversed, and hydraulic cooling is required without lube cooling, the "dead" lube oil would see a -200F hydrogen temperature which could plug the cooler depending upon the duration of this out-of-phase cooling condition.

For purposes of analysis the effectiveness of the coolers was assumed constant ($\epsilon_H = 69\%$, $\epsilon_L = 85\%$), with maximum hydraulic and lube cooling loads of 25 Btu/sec and 7 Btu/sec respectively, being accommodated by a hydrogen flow of .0177 lb/sec (SPC = 2.1 @ 56 HP). Since the lube cooler is a counterflow type the lube oil inlet temperatures determine hydrogen

MAXIMUM HYDROGEN SUPPLY TEMPERATURE TO PROVIDE HYDRAULIC AND LUBE SYSTEM COOLING AT MAXIMUM HEAT LOAD CONDITION

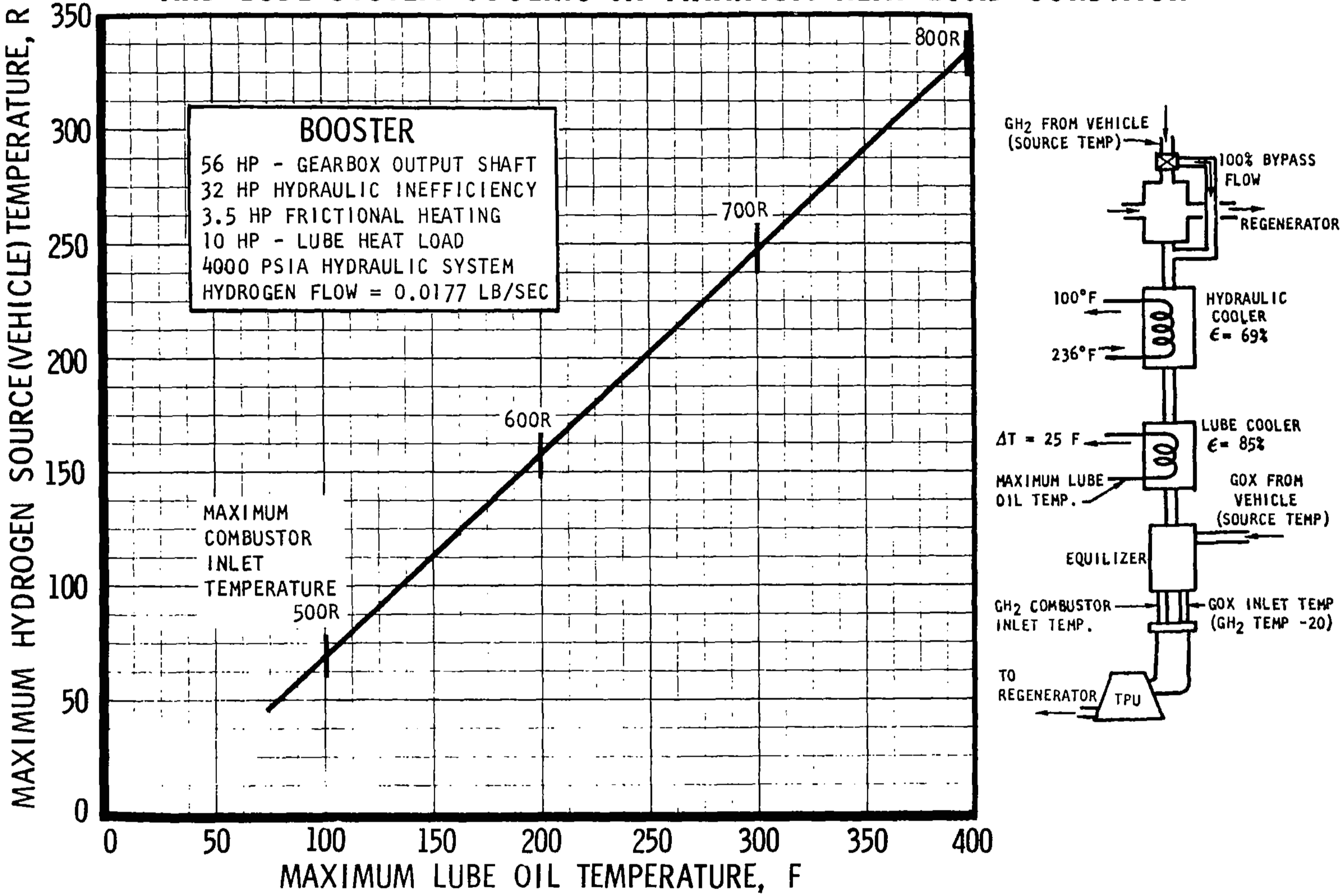


FIGURE 32

exit temperature (equalizer inlet). The regenerator was assumed to be in the 100% bypass mode providing hydrogen into the hydraulic cooler at the vehicle source temperature. As vehicle hydrogen temperature increases the lube oil inlet temperature must increase for the hydrogen to serve as the heat sink for the 7 Btu/sec load. A maximum combustor inlet temperature is then uniquely determined by the lube cooler and equalizer effectiveness. Results of the study are presented in Fig. 32.. The region below the curve represents conditions for which adequate cooling is provided by the hydrogen. Auxiliary cooling must be provided for points above the curve. If the source temperature is less than maximum for a given lube-oil temperature,the regenerator will provide the additional heat required in order to achieve the maximum combustor inlet temperature. The maximum recommended continuous operating temperature of MIL-L-23699 oil is 400F so that a vehicle source temperature of about 370R is acceptable resulting in a combustor inlet temperature of 800R for the 85% effectiveness lube cooler. Reduced cooler effectiveness results in lower combustor inlet temperature.

APU SYSTEM DEFINITION

The Space Shuttle APU consists of three major subsystems as shown in Fig. 33.

1. The propellant conditioning system
2. The turbo power unit
3. The power control

Although the power control is an integral part of the turbo power unit, its design has major implications on the rest of the APU and for that reason is considered as a separate subsystem.

Propellant Conditioning

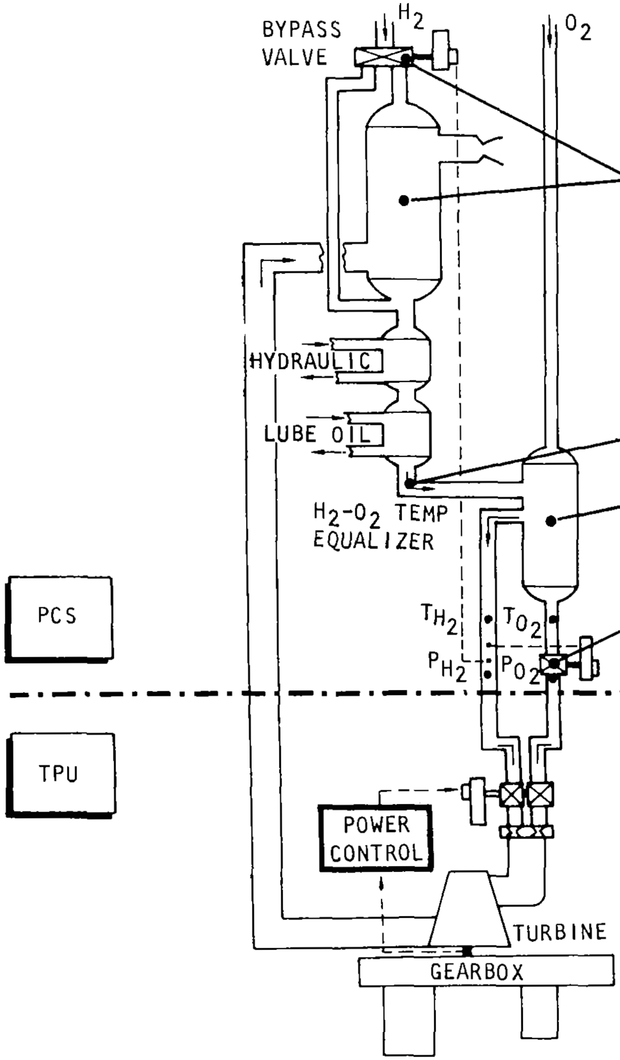
The major tasks of the propellant conditioning subsystem are:

1. to control mixture ratio to the TPU
2. to establish the proper level of propellant pressure and temperature to the TPU inlet
3. to accommodate variable hydraulic and lube oil cooling loads.

Mixture ratio is controlled to the TPU by three separate functions:

1. the relative value of hydrogen and oxygen pressure is maintained nearly constant at the main propellant valve (MPI) inlet. This is accomplished by a differential pressure regulator located in the oxidizer line just upstream of the TPU. The regulator senses hydrogen pressure and throttles the oxidizer to equalize pressures.

MAJOR SUBSYSTEM TASKS



PROPELLANT CONDITIONING SYSTEM

MAJOR TASK:

- MR CONTROL TO TPU
- MAINTAIN TPU PROPELLANT INLET TEMP (T_{H_2})
- PROVIDE HYDRAULIC/OIL COOLING

METHOD:

- REGENERATOR AND BYPASS VALVE CONTROLS T_{H_2} = PRESET VALUE
- PROVIDE CONTROLLED TEMP. SINK FOR HYDRAULIC/OIL COOLER
- EQUALIZER BRINGS $T_{O_2} = T_{H_2}$
- PRESSURE REGULATOR CONTROLS $P_{O_2} = P_{H_2}$ + BIAS

TURBO POWER UNIT

MAJOR TASK:

- EFFICIENT ENERGY CONVERSION

CHEM. ENERGY ⇨ MECH. ENERGY ⇨ HYDR/ELEC ENERGY

POWER CONTROL

MAJOR TASK:

- SPEED CONTROL OVER POWER/ ALTITUDE OPERATIONAL ENVELOPE

METHODS (3):

- PRESSURE MODULATION OF TPU
- PULSE MODULATION OF TPU POWER
- HYBRID (PRESSURE + AREA MODULATION)

FIGURE 33.

2. the relative value of hydrogen and oxygen temperature is maintained nearly constant at the MPV inlet. This is accomplished by a passive H_2-O_2 temperature equalizer-heat exchanger.
3. the main propellant valves are mechanically linked and provide a constant effective area ratio independent of valve position thus, mixture ratio is maintained constant at all power levels.

The accuracy of this control is of course dependent upon the accuracy with which the above functions can be accomplished during both steady state and transient performance. Propellant pressure level is established by a pressure regulator located in the hydrogen supply line to the APU. It should be noted that propellant pressure level at the TPU inlet may vary over a moderate range as a result (for example) of a variation in PCS pressure drop or regulator inaccuracy, without any detrimental effect to performance. Propellant temperature level delivered to the TPU is controlled by the regenerator and flow splitter control. TPU hydrogen inlet temperature is maintained nearly constant independent of supply temperature variations, hydraulic and/or lube oil cooling load variations, or APU power modulation. The bypass control senses hydrogen temperature at the TPU inlet. Any error with respect to a reference temperature results in activation of the bypass flow splitter valve which diverts more or less flow through the bypass line, varying regenerator exit temperature to restore the proper control temperature.

The propellant conditioning system accommodates varying hydraulic and lube oil cooling loads. The regenerator and flow splitter control maintains a nearly constant hydrogen sink temperature for the lube oil cooler consistent with maximum allowable lube oil temperatures.

Turbo Power Unit

The major task of the turbo power unit is to efficiently convert chemical energy to hydraulic and electric energy. This is accomplished by optimizing the specific propellant consumption over the flight-power profile.

Power Control

The major task of the power control is to accomodate the complete range of power demands over the altitude profile and maintain TPU speed within an acceptable range. This is accomplished by:

1. pressure modulation of turbine inlet pressure
2. pulse width modulations of turbine inlet pressure
3. hybrid control - combined pressure modulation plus a variable (two step) turbine area.

Pressure modulation of turbine inlet pressure is provided by a mechanically linked bipropellant throttle valve which is actuated when the control system senses a speed error and attempts to maintain the TPU at a fixed reference design speed. Pulse width modulation of turbine inlet pressure is provided by an "on-off" mechanically linked bipropellant valve which is actuated when the control system senses a speed outside a predetermined band. TPU speed varies continuously with an amplitude as set by the speed band and at a frequency which is dependent primarily on TPU inertia. The hybrid power control is composed of a dual combustor-nozzle assembly, each of which contains a mechanically linked bipropellant valve. Each combustor-nozzle assembly covers a discrete portion of the turbine arc. The "sustainer" power level is approximately 25 to 35 percent. The high-power combustor is activated when the power demand exceeds the capacity of the sustainer assembly. The high-power combustor can either be operated in a pressure modulated or pulse mode.

TURBO POWER UNIT

Figure 34 summarizes the turbine design conditions and characteristics which were evaluated. For example, the first line in the chart represents a velocity-compound turbine with pulse modulating power control for use in systems A1 or D1. This turbine was designed for a maximum turbine inlet pressure of 600 psi. Because it is a pulsed machine, that is also the design pressure level. From previous studies (Phase IA) a pressure ratio of 50 has been established as approximately optimum for this inlet pressure level. The efficiency predicted at this design point condition is 0.491. The corresponding first and second stage blade heights and admission fractions are also given in the table. The shaded areas in the following columns indicate that the off-design performance of this turbine was calculated and, in addition,

TURBINE SUMMARY
CONDITIONS AND CHARACTERISTICS STUDIED

2 STAGES	DIA. = 6.68 IN.
N = 60,000 RPM	U = 1750 FT/SEC
T_T = 1556 F	TIP CLEARANCE = 0.0156 IN.

SYSTEM	CONTROL TYPE	TURBINE TYPE	MAX. PRESSURE PSIA	DESIGN PRESSURE	DESIGN PRESSURE RATIO	BLADE HEIGHT, IN.		ADMISSION		DESIGN EFFICIENCY	OFF-DESIGN PERFORMANCE	THERMAL ANALYSIS	STRESS	CLEARANCE SENSITIVITY	POWER* VARIATION 100 HP-750 HP
						FIRST STAGE	SECOND STAGE	FIRST	SECOND						
D1 A1	PULSE	VELOCITY	600	600	50	.368	.461	.30	.60	.491					
D2 A2	PRESSURE	VELOCITY	600	126	12.45	.189	.407	.20	.350	.426					
D3 A3	PULSE	PRESSURE	600	600	50	.168	.388	.14	.305	.540					
D3A A4	PRESSURE	PRESSURE	600	120	12					.510 (EST)	EST.	EST.			
B1 D5	PULSE	PRESSURE	150	150	20	.310	.481	.28	.62	.550					
B2 D4	PRESSURE	PRESSURE	150	32.4	3.2	.316	.407	.285	.350	.578					
B2 D4	PRESSURE	PRESSURE	150	22.25	12	.357	.470	.30	.62	.445					
B2 D4	PRESSURE	PRESSURE	150	36	12	.296	.454	.28	.52	.475					

*ROTATIVE SPEED ALLOWED TO VARY

ANALYSIS COMPLETE

FIGURE 34

steady state temperature distributions and corresponding thermal and centrifugal stresses were determined.

The remainder of the table is essentially self-explanatory. For systems A4 and D3A the efficiency was estimated by interpolation. On systems A3 and D3, which represent pressure-staged turbines with pulse-modulating power control, the effects of tip-clearance variation and design power level were also investigated. Power level was varied from 100 to 750 HP. The clearance and power level effects are described later in this section.

One of the significant results shown in Fig.34 is the clear superiority of the pressure-staged machine over the velocity-compound machine at a pressure ratio of 50. It would be expected that at lower design pressure ratios the difference would be even more marked.

Figure 35 summarizes the thermal and stress analyses performed in Phase IB. It shows schematically the manner in which allowable stresses were established for use with the calculated temperature distributions. Safety factors are applied to the minimum (3σ) strength of the material to determine the allowable stress at rated operating speed and temperature. This is done both for short-time and long-time strengths.

PHASE IB ANALYSIS

STRESS

- STEADY STATE TEMPERATURE DISTRIBUTIONS
- CONSTANT STRESS DISKS INCLUDING THERMAL STRESSES

SAFETY FACTORS

- AT RATED SPEED
- COMPOSITION:

	SHORT TIME		LONG TIME	
	YIELD	ULTIMATE	CREEP	RUPTURE
BASIC SAFETY FACTOR	1.1	1.5	1.1	1.5
THERMAL TRANSIENT ALLOWANCE	1.1	1.1	1.0	1.0
SPEED CONTROL BAND	$(1.05)^2$	$(1.05)^2$	1.0	1.0
OVERSPEED TEST PER TSO-C77	$(1.15)^2$	$(1.15)^2$	1.0	1.0
OVERALL FACTOR (PRODUCT)	1.77	2.41	1.1	1.5

STRENGTH

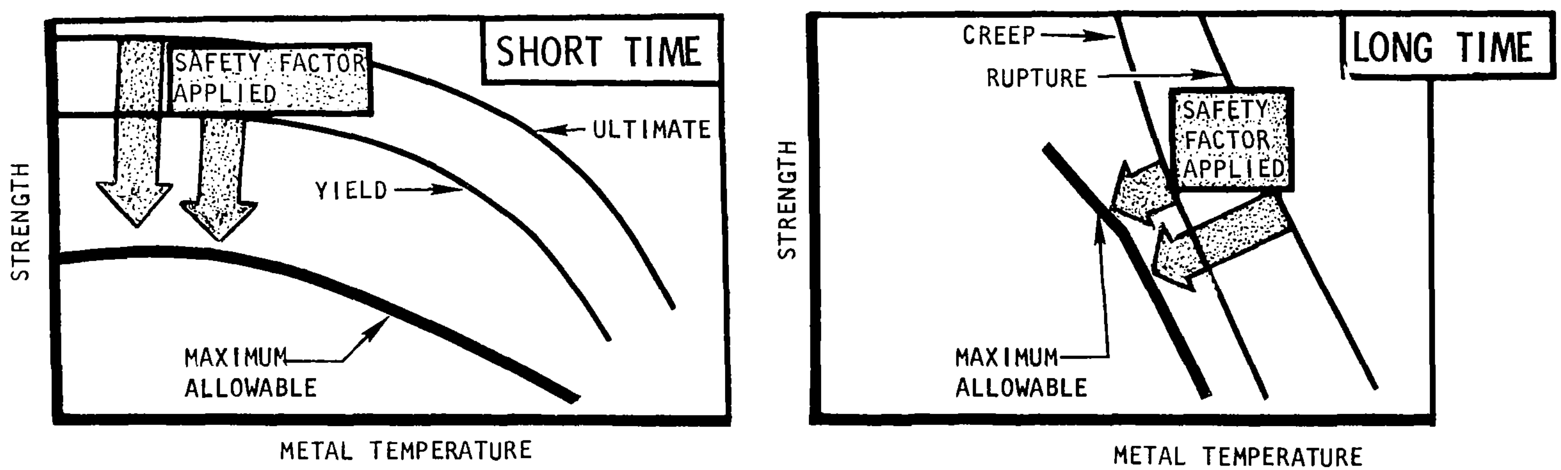


FIGURE 35

An overall safety factor is constructed for each failure mechanism and duration. For example, the short-time yield overall safety factor is constructed as follows. The basic material safety factor of 1.1 is applied to account for unknowns in the basic applied stress. An additional allowance of 1.1 is applied to represent the thermal transient stresses which are known to be significant but have not yet been estimated. A factor of $(1.05)^2$ is applied to represent the upper limit of the normal speed control range. Finally, a factor of $(1.15)^2$ is applied to represent a turbine overspeed test such as that required by TSO-C77 for auxiliary power units for aircraft. These last two factors are squared because the centrifugal stresses are a function of the square of the rotative speed. The overall safety factor applied to the material yield strength to determine an allowable stress based on yield is 1.77. This represents the allowable stress at rated rotative speed and temperature conditions which will prevent yielding at the stated overspeed conditions. The same process is applied to the ultimate, creep, and rupture strengths.

Figure 36 presents the actual plots of allowable stresses for Astroloy using factors of safety constructed as described in Fig. 35. The short-time stresses are assumed to apply at the full power condition which represents the maximum disk and blade temperatures. Because 5 percent of the design life will be spent at full power conditions, creep and rupture data for 50 hours are also shown on this curve. The region of safe operation is below

ALLOWABLE TURBINE STRESS LEVELS

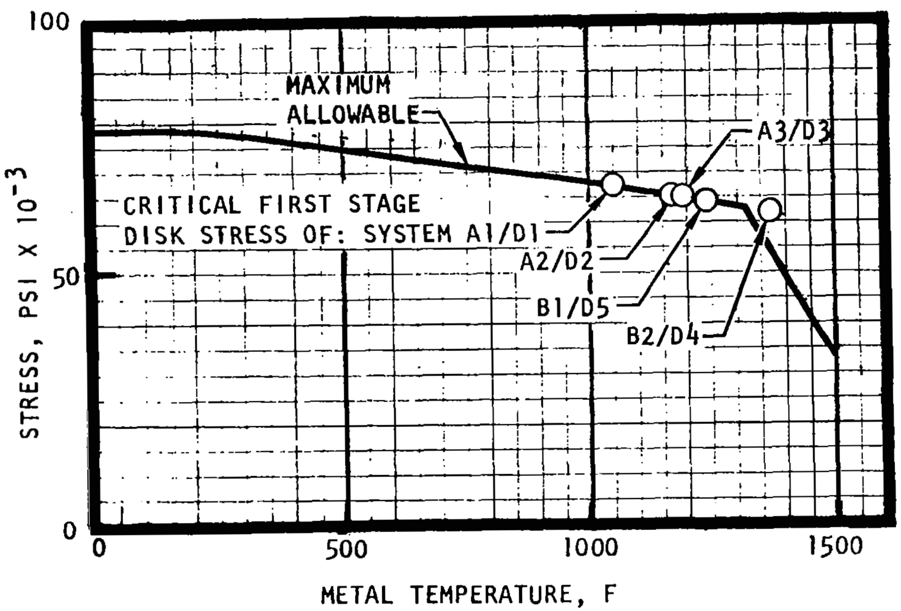
MATERIAL - ASTROLOY
FACTORS OF SAFETY ON:

YIELD	1.77
ULTIMATE	2.41
CREEP	1.1
RUPTURE	1.5

SYSTEM	CONTROL TYPE	TURBINE TYPE	MAX. PRESSURE PSIA
A1, D1	PULSE	VELOCITY	600
A2, D2	PRESSURE	VELOCITY	600
A3, D3	PULSE	PRESSURE	600
B1, D5	PULSE	PRESSURE	150
B2, D4	PRESSURE	PRESSURE	150

FIRST STAGE DISK DESIGNS
T_t = 1550 F
TIP SPEED ≃ 1700 FT/SEC
STEADY STATE TEMPERATURES
CRITICAL TEMPERATURE AT DISK NECK

SHORT TIME
(FULL POWER)



LONG TIME
(MINIMUM POWER)

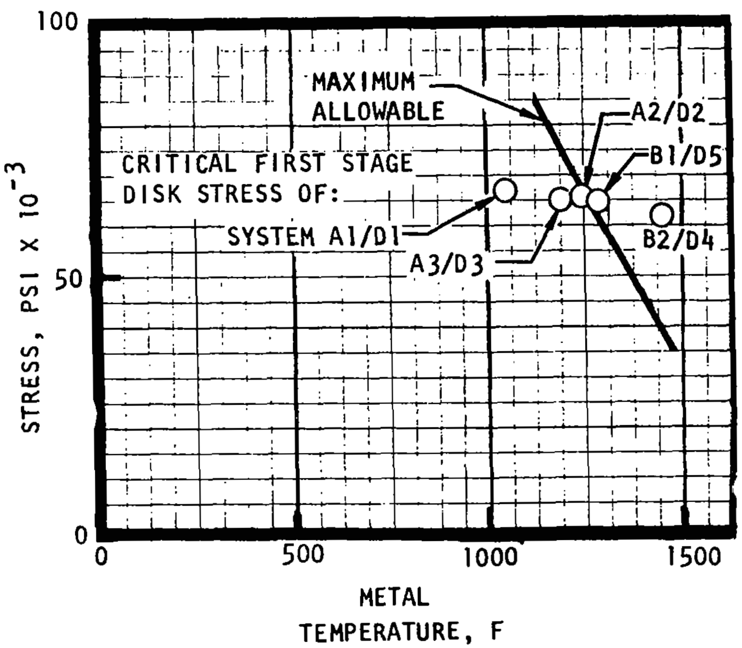


FIGURE 36

the curves. The critical stress and metal temperature which applies at the full power condition to each of the five first stage disks studied is also shown on the short-time portion of the figure. All (except one) are within the allowable range.

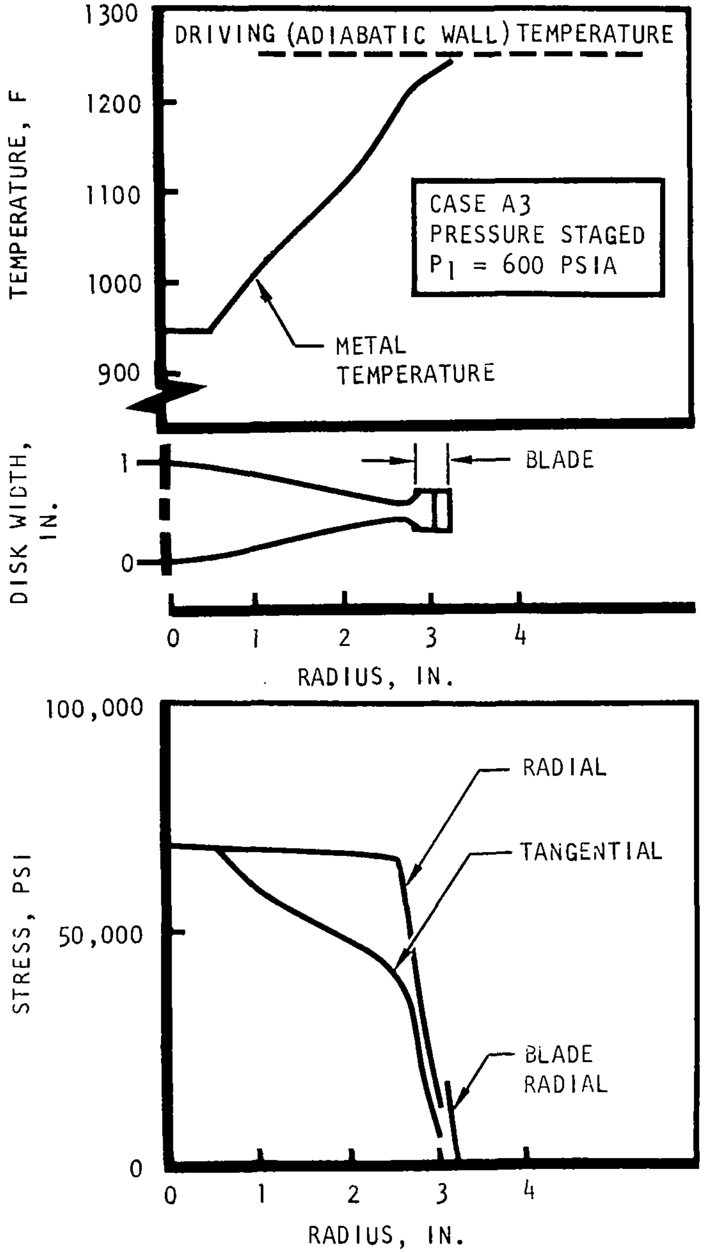
The right-hand side of the figure shows the allowable stresses corresponding to the long-time application of stresses associated with the minimum power condition and its somewhat lower turbine metal temperatures. The region of safe operation again is below the curve. Data points for the first stage disk stress-temperature combinations under the minimum power condition are shown on the curve. Two of the five turbines are in the unsafe region. Additional adjustments to disk geometry, inlet temperature, or tip speed will be necessary to bring those machines into the safe region.

Careful study of Fig. 36 shows some important trends which should be noted. Comparison of Systems A1/D1 and A3/D3, for example, shows that the pressure-compound machine is significantly higher in disk temperature than the velocity-compound machine for the same conditions. This results from the larger expansion ratio (to a lower temperature) of the velocity-compound first-stage nozzles. When the pressure and pulse modulated systems are compared (A2/D2 vs A1/D1 and B2/D4 vs B1/D5), the pulse modulated is seen to be lower in disk temperature. Again this is caused by the higher design expansion pressure ratio of the pulse modulated machine. Finally, comparison of the 600 psia A3/D3 system with the 150 psia B1/D5 system shows that the higher pressure system runs lower in disk temperature, again because of higher design pressure ratio.

Figure 37 shows an example of the design for the first stage of systems A3/D3. It is seen that the blade temperature is very close to the adiabatic wall temperature which causes the heat transfer to occur. This is caused by a very high heat transfer coefficient between the gas and the blades which in turn is caused by very high thermal conductivity of the gas (primarily hydrogen) and by the small dimensions of the blades. This results in small thermal time constants which produce large transient thermal stresses. The lower portions of the figure show the disk shape and the corresponding radial and tangential stresses. The disk has been designed as a constant stress disk from the hub up to the neck or critical region of the disk. It should be noted that the blade root stress on a steady state basis is significantly below the allowable stress, thus leaving room for variations in blade height, if desired.

Figure 38 presents a summary statement of the kinds of mechanical analyses which must be done on the selected turbine during Phase II of the present study based on analysis done under Phase I of the contract. The heat transfer analysis done so far indicates that it is necessary to consider not only the steady state temperature distributions in the machinery, but also the transient distributions. Important transients will occur at startup when the machine is cold and suddently subjected to a high temperature, at load changes when the effective heat transfer level changes, and at shutdown when the driving temperature is suddenly removed from the turbine.

Under each of these steady state and transient temperature conditions, it is necessary to examine the critical stresses. For the steady-state temperature



TYPICAL FIRST STAGE DISK DESIGN

SYSTEMS A3, D3
PULSE MODULATED
PRESSURE STAGED
P = 600 PSIA
T_T = 1556 F

- BLADE TEMPERATURE ≃ GAS DRIVING TEMPERATURE
  - CAUSED BY HIGH HEAT TRANSFER COEFFICIENT
    - HIGH GAS (H_2) THERMAL CONDUCTIVITY
    - SMALL BLADE DIMENSIONS
  - RESULTS IN
    - SMALL THERMAL TIME CONSTANTS
    - SIGNIFICANT TRANSIENT THERMAL STRESSES
- CONSTANT STRESS DISK
- BLADES NOT AT LIMITING STRESS

FIGURE 37

PHASE II TURBINE MECHANICAL ANALYSIS

THERMAL ANALYSIS

- STEADY STATE
  - VARYING LOAD LEVELS
- TRANSIENT
  - STARTUP
  - LOAD CHANGE
  - SHUTDOWN

STRESS ANALYSIS

- INDIVIDUAL STRESSES (ULTIMATE, YIELD, RUPTURE, CREEP, HIGH AND LOW CYCLE FATIGUE)
- ACCUMULATIVE DAMAGE
  - FRACTION OF TOTAL MATERIAL CAPABILITY UTILIZED =

$$\frac{\text{OPERATING TIME}}{\text{RUPTURE LIFE}} + \frac{\text{OPERATING CYCLES}}{\text{MECHANICAL (HIGH CYCLE FATIGUE) CYCLICAL LIFE}} + \frac{\text{PLASTIC STRAIN CYCLES}}{\text{LOW CYCLE FATIGUE CYCLICAL LIFE}} \leq 1.0$$

  - FACTORS OF SAFETY APPLIED EITHER TO STRESS OR LIFE

TRADEOFF

- COOLING - TEMPERATURE
- TEMPERATURE - TIP SPEED

FIGURE 38

distributions, all the individual stresses such as ultimate, yield, etc. must be examined with their appropriate safety factors. This is similar to the analysis which was performed in Phase I, and it is intended that it will be done for more conditions and on a more refined basis.

Transient thermal stresses require that the contribution of each potential failure mechanism with respect to the total material capability be assessed. For example, the fraction of the total material capability utilized at a particular stress level can be represented as the ratio of the operating time at that stress level to the stress rupture life at that stress level. Similarily, the number of plastic stress cycles experienced as a fraction of the low-cycle fatigue cyclical life at this strain level represents another portion of the total material capability utilized. The sum of all these fractions (with appropriate safety factors applied to either the applied stress or to the required life) must be less than unity.

The thermal and stress analysis to be performed in Phase II will allow additional tradeoff studies to be made. Specifically, the relationships between combustion temperature and cooling penalties can be assessed as well as the relationship between combustion temperature and turbine tip speed.

Figure 39 shows the typical effect of turbine tip clearances on system performance. The design value of tip clearance is 0.0156 inch at a tip diameter of 6.87 inches (in the second stage). Figure 39 shows that if the clearance is cut in half from the design value the reduction in required propellant flow will reduce the propellant system weight approximately 6 percent. Similar reductions in propellant system weight would be expected under

TYPICAL TIP CLEARANCE SENSITIVITY

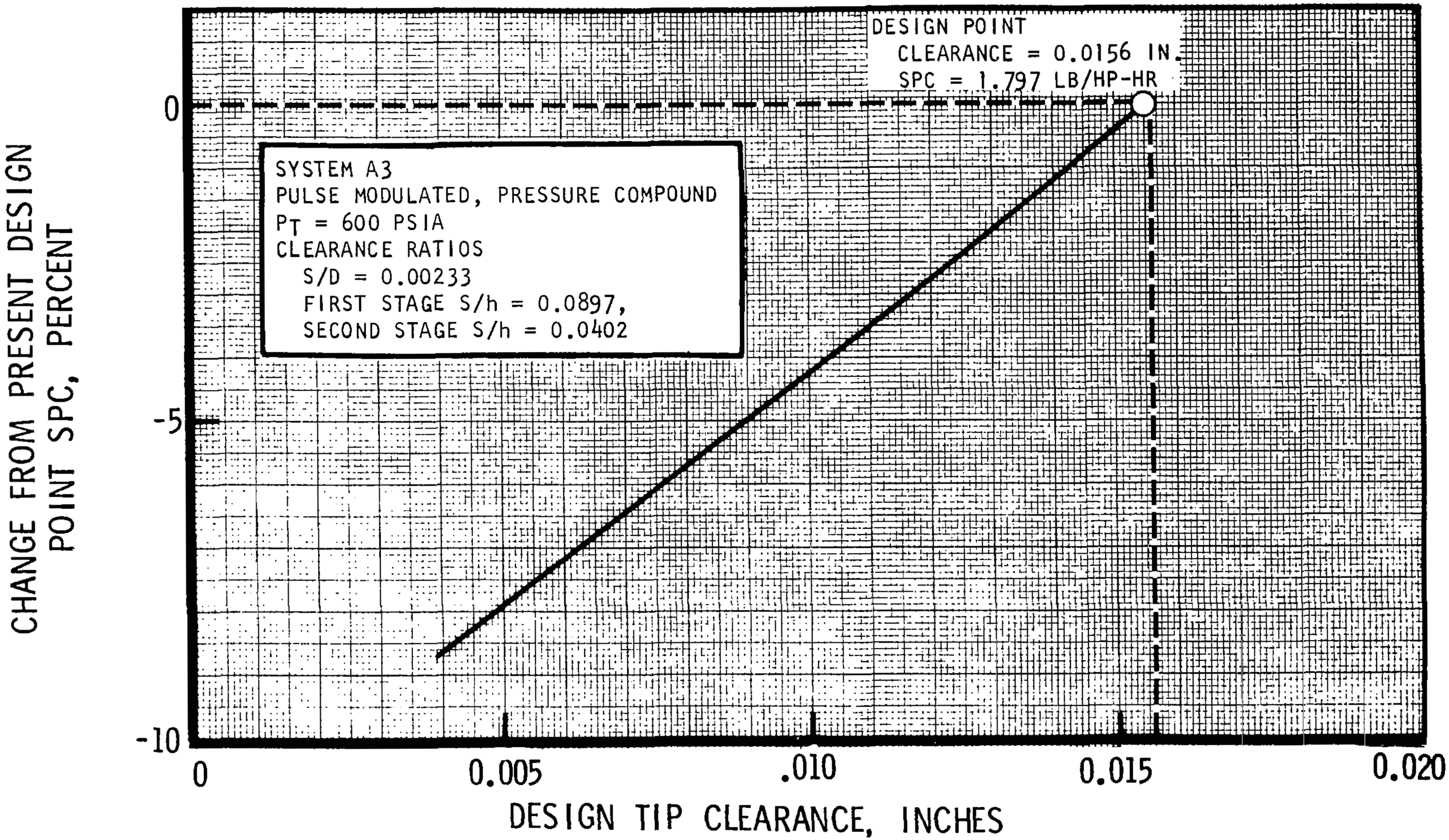
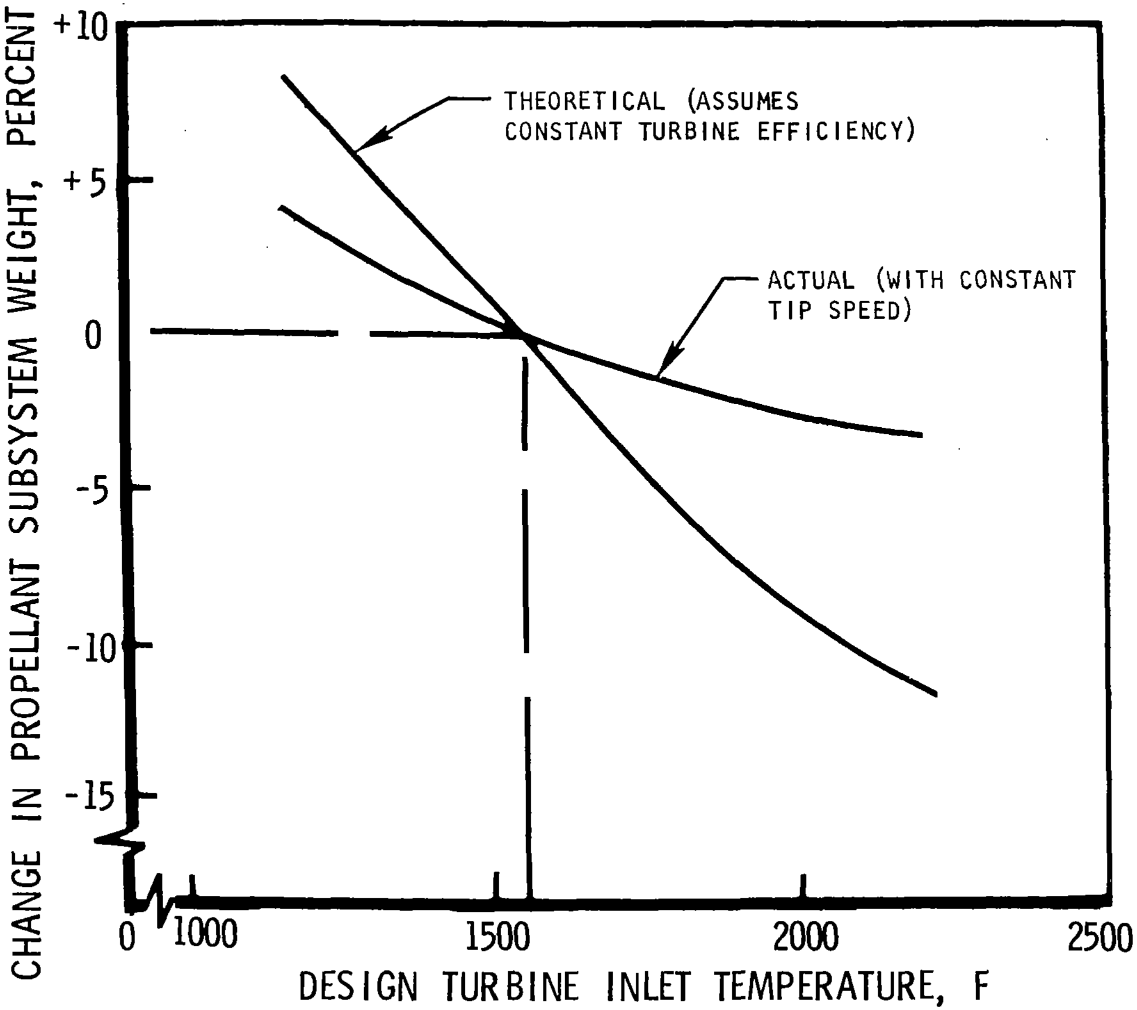


FIGURE 39

other design conditions. This improvement in performance must be traded against the necessity to reduce the total allowable creep of the turbine material in order to prevent rubbing of the turbine during the design life of the machine. The creep is reduced either by reducing the inlet gas temperature or the turbine tip speed or both. These reductions will reduce the system performance and increase the propellant system weight. The potential 6 percent weight reduction mentioned above, however, implies that additional study is worthwhile.

Figure40 shows the effect of turbine inlet temperature on propellant system weight. If only the propellant properties are considered, i.e., the turbine efficiency is assumed constant at all design temperatures, increasing inlet temperature produces fair benefits. If the turbine efficiency variation is considered, the beneficial effects are greatly reduced. The increased temperature reduces turbine efficiency by increasing spouting velocity (reducing velocity ratio, $\frac{U}{C_O}$), and by reducing the Reynolds numbers in the turbine due to reduced gas density and increased gas viscosity. Thus, there appears to be a minimal real benefit to increased turbine inlet temperature.

Figure 41 shows a typical turbopower unit (TPU) assembly. The hydraulic power is produced by two 3,000 PSI hydraulic pumps and the electrical power by one 12,000 RPM alternator. The corresponding gearbox, turbine assembly, and combustor assembly are also shown in Fig. 41. The total TPU weight as pictured is 165 pounds. Of this weight, 93 pounds are contributed by the pumps and alternator and only 72 pounds by the gearbox, turbine assembly, and combustor assembly.

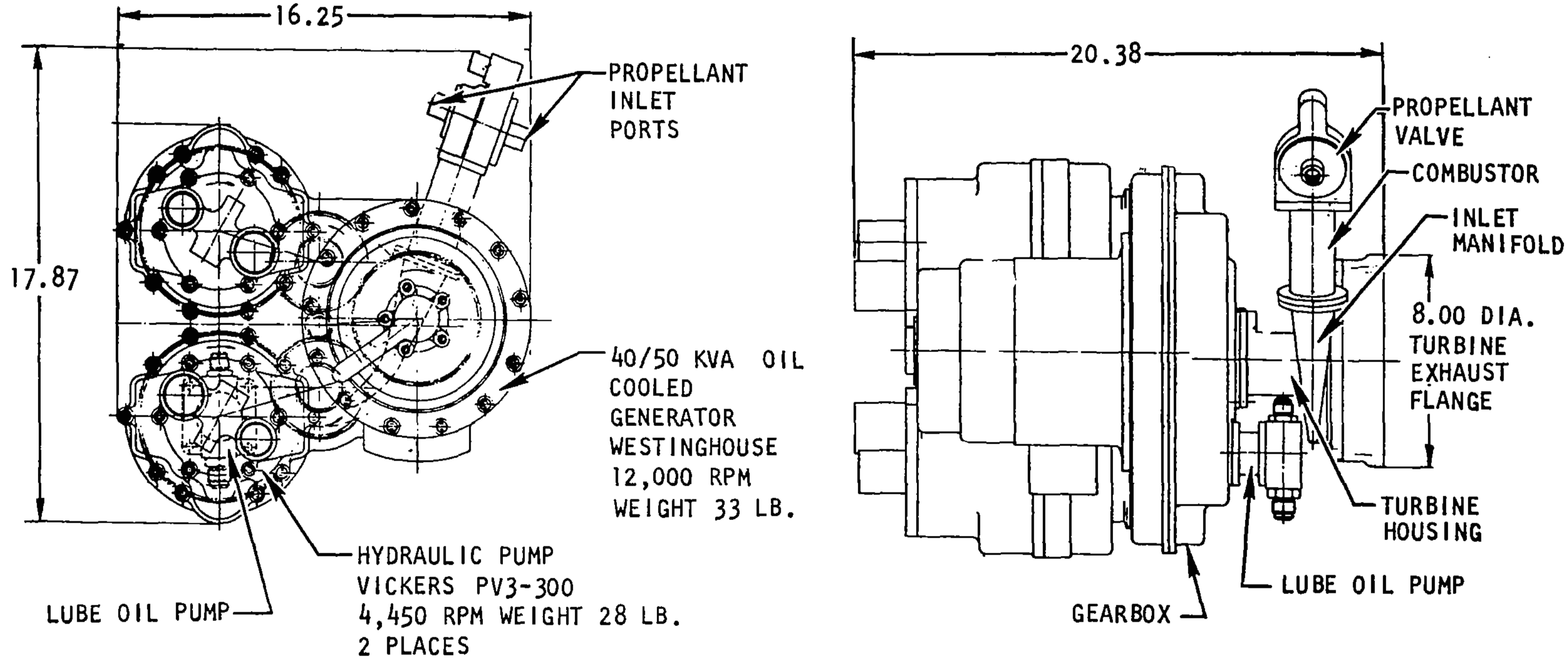


TYPICAL EFFECT OF TURBINE INLET TEMPERATURE ON PROPELLANT SUBSYSTEM WEIGHT

FIGURE 40

TYPICAL TPU ASSEMBLY

TWO PRESSURE STAGES, PULSE MODULATED, SYSTEM A3/D3



DESIGN CONDITIONS

- P_T = 600 PSIA
- T_T = 1550 F ±100
- PRESSURE RATIO = 50
- SPEED = 60,000 RPM
- MAXIMUM TIP SPEED = 1750 FT/SEC
- 3000 PSI HYDRAULIC SYSTEM

TPU WEIGHTS

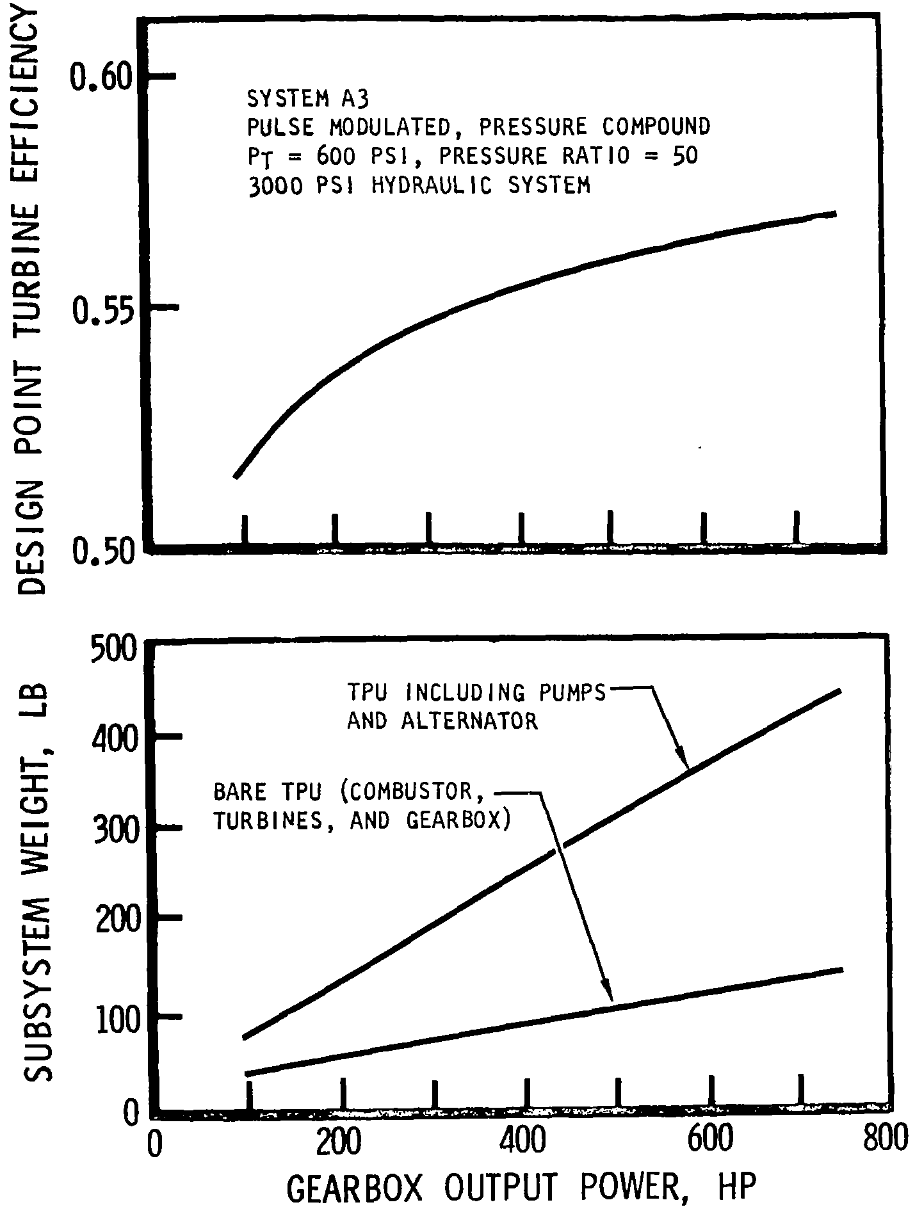
• TURBINE, COMBUSTOR, GEARBOX	72
• ALTERNATOR	33
• PUMPS AND VALVES	60
TOTAL	165 LB

FIGURE 41

The results of a preliminary study of the effect of output power level on turbine efficiency and TPU weight are shown in Fig. 42. Output power was varied from 100 to 750 HP. Turbine efficiency varied by about 10 percent over this power range. The weight of the bare TPU varied by a factor of approximately 3 while the complete TPU varied by about 5. Thus the specific weight (lb/hp) of the TPU, as expected, decreased as power level increased. It should be noted that part of this weight variation is caused by rotative speed being reduced at the higher power levels. Tip speed was held constant. Tip diameter therefore increased with power level.

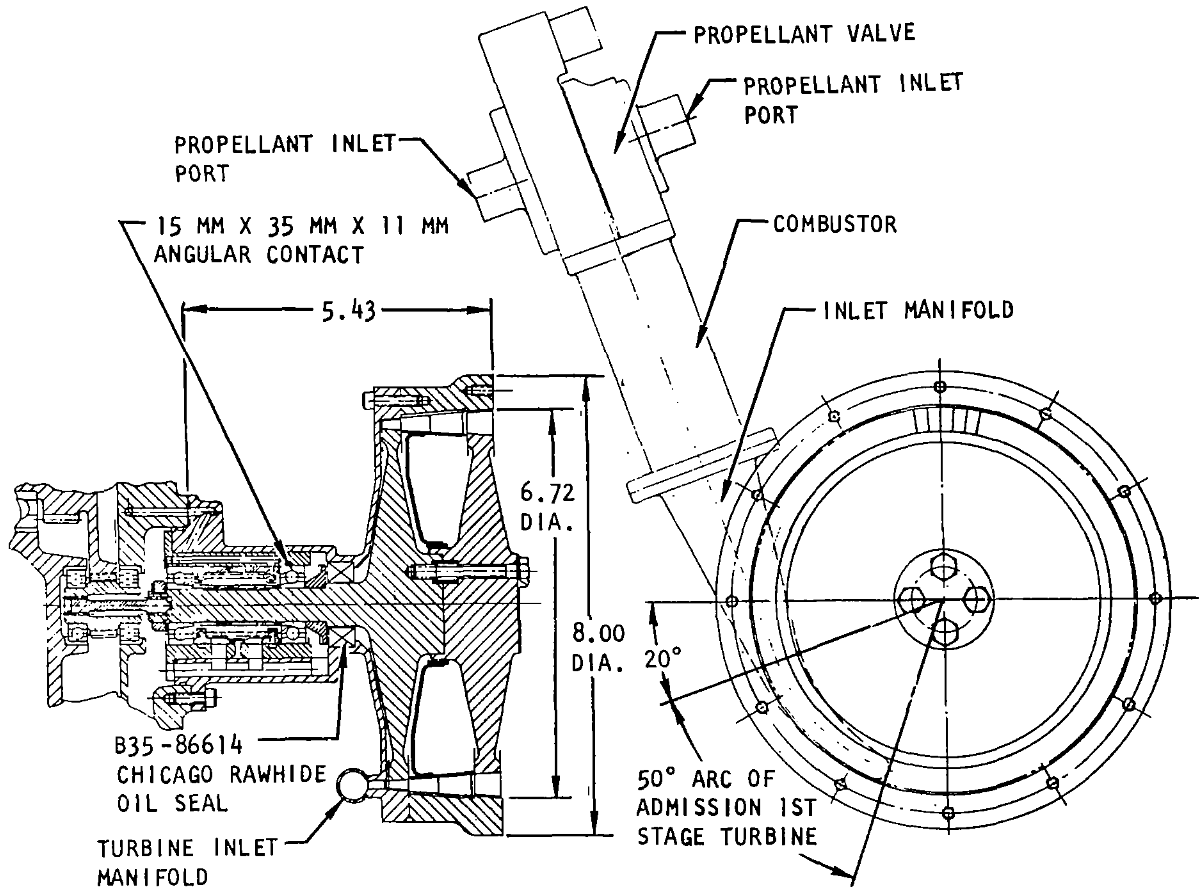
Figure 43 shows the cutaway view of the typical turbine assembly. Among the additional details which may be seen in this figure are the inlet arc of the first stage. The second stage inlet arc is approximately twice the size of the first stage, but is not shown explicitly. The interstage seal shown is a simple labyrinth. Performance calculations as reported here have been based on this seal configuration. Also shown in Fig.43 is additional armor material around the turbine disks to provide burst protection. The amount of material shown is based on using a weight equal to the weight of the rotating disks. This approximation has been inferred from the limited data so far available from the NASA-Sponsored Rotor Burst Protection Program being conducted at the Naval Air Propulsion Test Center in Philadelphia.

Table 3 presents more detailed information on the five preliminary turbine designs studied in Phase IB. All these turbines were two stage machines designed for an inlet temperature of 1550 F with a maximum second stage tip speed of 1750 ft/sec (first stage maximum tip speed is 1700 ft/sec) and



EFFECT OF POWER LEVEL ON TPU WEIGHT AND PERFORMANCE

FIGURE 42



TYPICAL TURBINE ASSEMBLY

TWO PRESSURE STAGES, PULSE MODULATED

DESIGN CONDITIONS

P_T = 600 PSIA
T_T = 1550 F ± 100
PRESSURE RATIO = 50
SHAFT POWER = 276 HP
SPEED = 60,000 RPM
TIP SPEED = 1750 FT/SEC

STAGE DATA

	FIRST	SECOND
BLADE HEIGHT, IN.	.168	.388
DEGREE OF ADMISSION	.14	.305
TIP CLEARANCE, IN.	.0150	.0156
INTERSTAGE SEAL CLEARANCE, IN.	.0042	
RELATIVE MACH NUMBER	1.463	1.717

FIGURE 43

SYSTEM	A 1 / D 1		A 2 / D 1		A 3 / D 3		B 1 / D 5		B 2 / D 4	
CONTROL TYPE	PULSE		PRESSURE		PULSE		PULSE		PRESSURE	
MACHINE TYPE	VELOCITY		VELOCITY		PRESSURE		PRESSURE		PRESSURE	
MAXIMUM INLET PRESSURE, PSIA	600		600		600		150		150	
DESIGN INLET PRESSURE, PSIA	600		126		600		150		32.4	
DESIGN EXIT PRESSURE, PSIA	12		10.11		12		7.5		10.11	
DESIGN PRESSURE RATIO	50		12.45		50		20		3.20	
DESIGN POWER LEVEL, HP	276		42.2		276		320		42.2	
DESIGN POINT TIP VELOCITY RATIO U/C_0	0.1528		0.1753		0.1528		0.1656		0.238	
DESIGN POINT EFFICIENCY	0.491		0.426		0.540		0.550		0.578	
HEAT FLOW INTO (400 F) BEARING BTU / SEC	0.1024		0.124		0.104		0.1132		0.1537	
STAGE	FIRST	SECOND	FIRST	SECOND	FIRST	SECOND	FIRST	SECOND	FIRST	SECOND
TIP DIAMETER, IN.	6.45	6.68	6.45	6.68	6.45	6.68	6.45	6.68	6.45	6.68
BLADE HEIGHT, h, IN.	0.368	0.461	0.189	0.407	0.1675	0.388	0.31	0.481	0.316	0.407
DEGREE OF ADMISSION	0.30	0.60	0.20	0.350	0.14	0.305	0.28	0.62	0.285	0.350
CLEARANCE RATIO, S/h	0.0409	0.0338	0.079	0.0362	0.0897	0.0402	0.0485	0.0324	0.0476	0.0382
BLADE NUMBER	74	88	107	98	107	77	80	78	89	88
RELATIVE MACH NUMBER, M_{W2}	2.37	0.723	1.756	0.452	1.463	1.717	1.21	1.367	0.642	1.670
REYNOLDS NUMBER, $Re_2^* = \frac{UD}{\nu_2}$	5.19×10^5	2.34×10^5	2.42×10^5	1.462×10^5	1.602×10^5	4.55×10^5	5.15×10^5	1.94×10^5	1.804×10^5	1.358×10^5
DISK WIDTH AT HUB, b_0, IN.	1.063	1.186	1.037	1.426	1.102	1.083	1.390	1.232	1.474	1.350
DISK HUB / NECK WIDTH RATIO b_0 / b_n	4.34	3.87	5.44	4.32	6.07	3.92	5.25	3.29	5.22	4.58
DRIVING (ADIABATIC WALL) TEMPERATURE, F	1085	943	1215	1125	1247	878	1295	976	1415.5	1215
MAXIMUM POWER AVERAGE BLADE TEMPERATURE, F	1080	942.6	1211	1123	1239	879	1288	978.6	1409	1214
MAXIMUM POWER NECK, TEMPERATURE, F	1053	939	1185	1113	1190	888	1236	999	1372	1213
MAXIMUM POWER HUB, TEMPERATURE, F	920	928	1048	1079	944	921	1004.5	1060	1207.1	1210
MINIMUM POWER NECK, TEMPERATURE, F	1031.9	934.8	1175.7	1108	1156.4	889.9	1219.3	990.2	1359	1212

TABLE 3. PRELIMINARY TURBINE DESIGNS

rotative speed of 60,000 rpm. (During Phase II of this present study additional values of these parameters will be studied.) This chart presents several types of key parameters: geometric--tip diameter, blade height, degree of admission; performance--efficiency, velocity ratio, relative Mach number, and Reynolds number; thermal--heat transfer driving (adiabatic wall) temperature, average blade, neck and hub temperatures at maximum power, and neck temperature at minimum power. These latter metal temperatures are based on holding the bearing temperature at a constant 400 F. One important factor shown clearly in the chart is the effect of reduced design pressure ratio. This may be seen by comparing the metal temperatures for systems A3/D3, B1/D5, and B2/D4. The lower pressure ratio increases the driving temperature; the metal temperature closely follows the driving temperature.

Figure44 shows the off-design characteristics of the five turbines studied. Figure 44A shows the basic energy available from the propellants as a function of expansion pressure ratio. Figure 44B presents the same expansion energy data in a different form. For each of several design pressure ratios, the off-design ideal work output as a function of off-design pressure ratio is shown. It will be seen that low pressure machines are much more sensitive to the same ratio of off-design pressure ratio.

Figure 44C presents the variation of turbine efficiency with off-design pressure ratios. These data were constructed based on the estimated stage off-design torque and flow rate as functions of stage velocity ratio and

TURBINE OFF-DESIGN PERFORMANCE SUMMARY

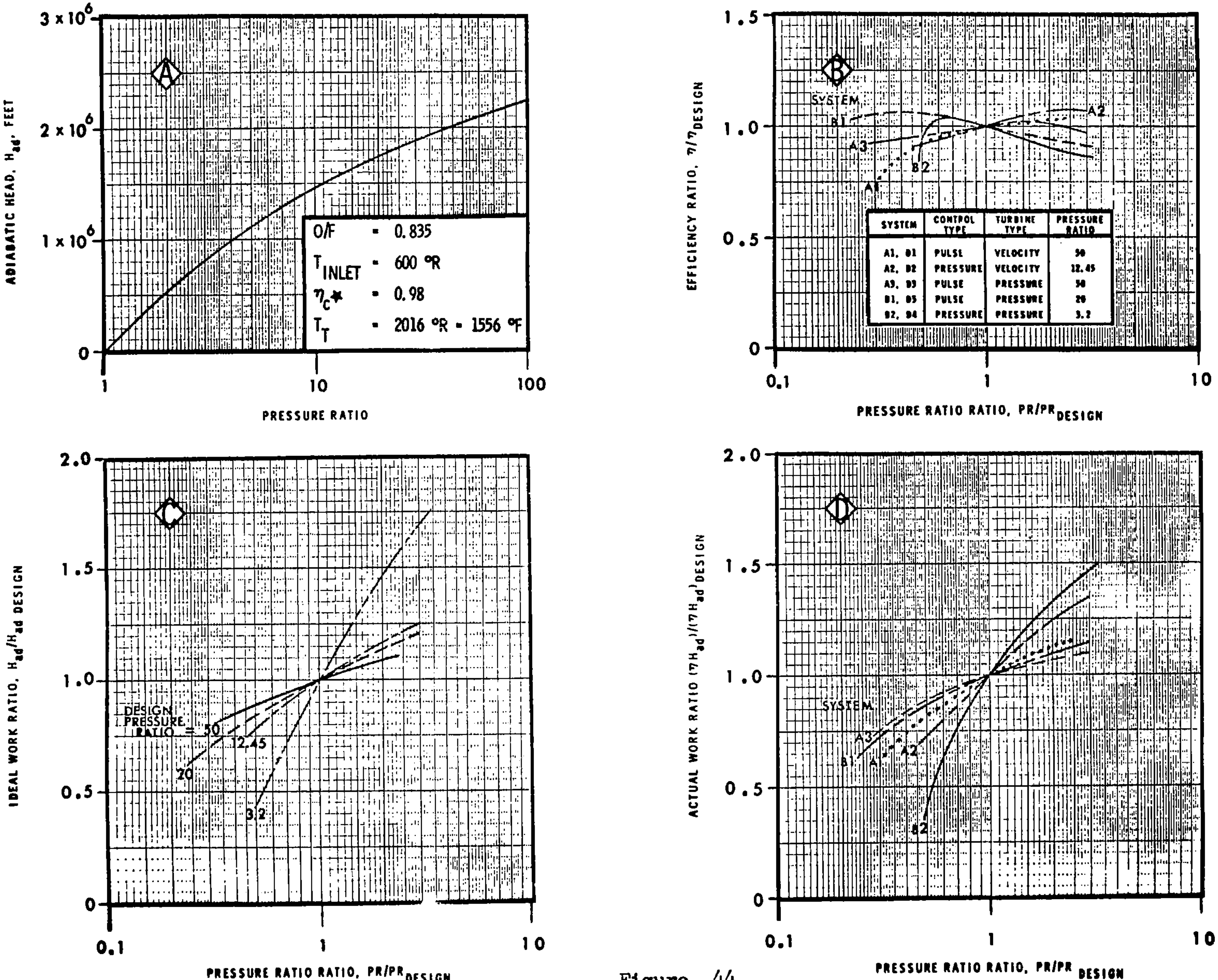


Figure 44

stage pressure ratio. Stage data were combined to determine stage pressure ratio for each overall pressure ratio. The efficiencies greater than the design value shown in Fig. 44C at pressure ratios below the design value result from the combined effects of pressure staging, the work ratio effect of Fig. 44B, and the pressure ratio effect on stage performance.

When the off-design efficiency and adiabatic head ratios are combined, the off-design work output is obtained as shown in Fig. 44D. These curves are generally similar to those of Fig. 44B, but somewhat modified. Low design pressure ratio machines still seem to be most sensitive to off-design pressure ratios. This should not be especially surprising. For example, Case B2 which has a design pressure ratio of 3.2 would have a zero adiabatic head at a pressure ratio ratio of 0.288. Cases A1 and A3 on the other hand with design pressure ratios of 50 would have zero output head at a pressure ratio ratio of 0.02.

The data of Fig. 44D were used in the system optimization studies described earlier. At each power level and pressure condition in the power profile the off-design data of Fig. 44D were used to determine the SPC at that condition. Total propellant usage is the sum of these individual values.

PRELIMINARY COMPONENT DESIGNS

A review of component availability and a preliminary design was performed. In general, these designs formed the basis for the operational analysis conducted with the analog model which will be discussed in the following section. However, pump designs were not utilized in the model because the analog model simulated supercritical propellant storage.

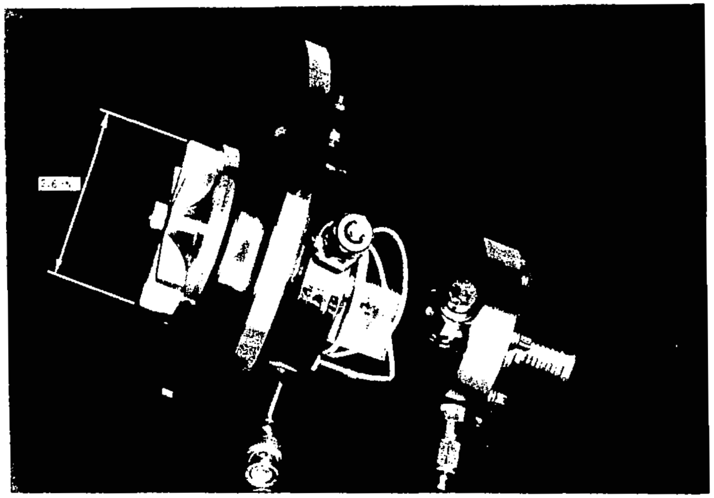
Pumps

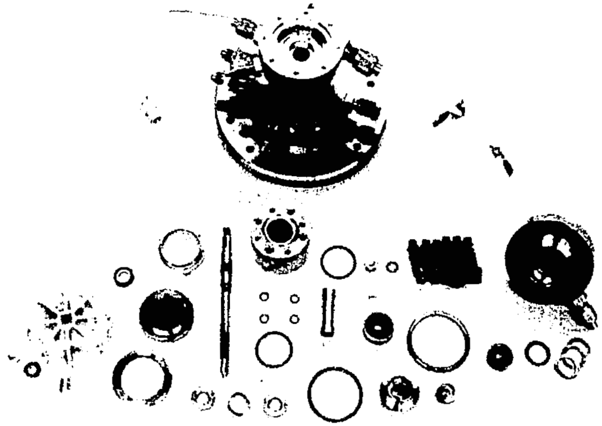
Pump-fed systems will require the delivery of hydrogen under pressure. For System A, this pressure is 860 psia, corresponding to a head of 28,500 feet, at a peak flow of 8 gpm. A number of pump types have been evaluated for applicability and availability.

Centrifugal Pump. A small, single-stage centrifugal pump of the type required here has been designed, fabricated, and tested at Rocketdyne under NASA Contract NAS3-12022. This pump (designated Mark 36 and depicted in Fig. 45) has been run for over one hour with Freon 12 at 75,000 rpm, 1500 psia (2800 ft head), and flows up to 13 gpm. It was designed for liquid fluorine service.

As discussed in the section on storable propellants, this pump is applicable to hydrazine service where a slight adjustment to 81,000 rpm will provide the required head and flow. For hydrogen a redesign is necessary to produce the vastly increased head required (10 times) by either increasing the speed or staging the pump, or both. As presently designed, the head variation over the required delivery range is small (2 percent). If this characteristic was maintained for the multistage design the TPU gearbox would be utilized as a constant speed drive. However, this would require the APU to be located close to the hydrogen tank to assure adequate NPSH to the pump. If the APU was remote from the tank, a separate drive using a gas turbine with a hydrogen gas bleed

CRYOGENIC CENTRIFUGAL PUMP-MARK 36





DESCRIPTION

TYPE: SINGLE STAGE CENTRIFUGAL-
SHROUDED IMPELLER
MATERIAL: INCO 718
TIP DIAMETER: 1.3"
EXTERNAL LEAKAGE: ZERO
WEIGHT*: 11.5 LBS
VOLUME: 25 CU. IN.
*LESS DRIVE

PREDICTED PERFORMANCE IN HYDROGEN

SPEED: 75,000 RPM
DN: 600,000
H_2 FLOW: .075 LB/SEC (PEAK)
EXIT PRESSURE: 125 PSIA
INLET PRESSURE: 40 PSIA
INLET TEMP: 37R
SHAFT POWER: 0.85 HP
OVERALL EFFICIENCY: 45%

DEVELOPMENT STATUS

OVER 1 HOUR IN FREON 12
AT 75,000 RPM AND
FLOW FROM 0 TO 13 GPM

DRIVES

H_2 GAS TURBINE
TPU GEARBOX

DESIGN MODIFICATION FOR SYSTEM A1 - 840 PSIA

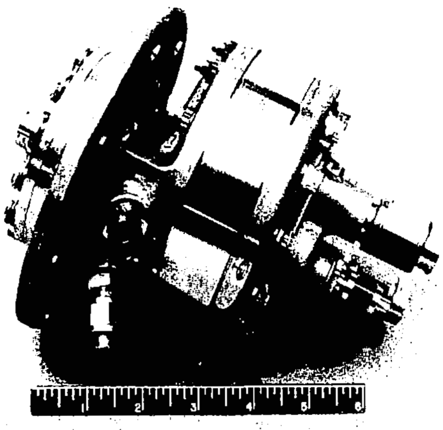
- UPRATE SPEED TO 150,000 RPM
  DN = 1.2 X 10^6
- ADD 2 STAGES
- SHAFT POWER: 7.6 HP

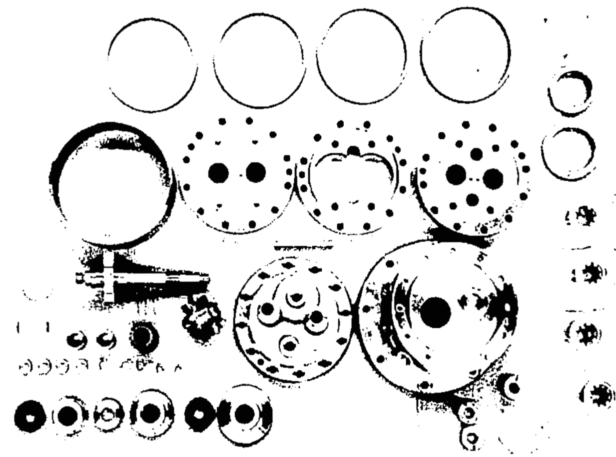
source and closed loop speed control would be required.

Gear Pump. Rocketdyne has also designed and fabricated a low specific speed gear pump (Mark 37) for a liquid fluorine application, which is directly applicable to System A1. Unfortunately, no test data for the pump has been obtained to date due to funding limitations. However, performance analysis with liquid hydrogen as the pumped fluid indicates that the required pressure of 860 PSIA should be obtained with reasonable radial and axial clearances at speeds of 5750 RPM (peak flow) to 3750 RPM (idle flow). Flow control could be obtained by means of a hydraulic or electrical motor which would modulate speed accordingly to maintain a fixed (reference) discharge pressure. Use of the electrical motor would allow in-tank mounting of the pump to assure positive liquid acquisition at the inlet. A photograph of the Mark 37 and summary of predicted performance in hydrogen at peak and idle power level is presented in Fig. 46.

Liquid Hydrogen Vane Pump. Pesco Products, Division of Borg Warner, has designed and tested a vane pump (under NASA Contract) for a liquid methane system (SST application) to provide 12 GPM at 900 PSIA and 4000 RPM. Initial testing is being conducted with liquid hydrogen to determine material compatibility and clearance effects. A total of 12 hours of operation at flows to 40 GPM and discharge pressure to 260 PSIA at the rated speed of 4000 RPM has been accumulated in liquid hydrogen. Further development is required

CRYOGENIC GEAR PUMP-MARK 37





DESCRIPTION

- TYPE: SPUR GEAR
- MATERIAL: INCO 718
- GEAR DIAMETER: 1.9" O.D.
- RADIAL CLEARANCE: 1×10^{-3}
- AXIAL CLEARANCE: 2×10^{-3}
- EXTERNAL LEAKAGE: ZERO
- WEIGHT : 18.5 LBS
- VOLUME: 88.0 CU IN.

PREDICTED PERFORMANCE IN HYDROGEN - SYSTEM A1

	PEAK	IDLE
•SPEED:	5750	3750
•DN:	86,000	56,000
•H_2 FLOW:	.075	.0126
•EXIT PRESSURE, PSIA:	860	860
•INLET PRESSURE, PSIA:	20	20
•INLET TEMP, R:	37	37
•MINIMUM NPSH, PSI:	1	1
•VOLUMETRIC EFFICIENCY:	42%	11%
•SHAFT POWER:	8 HP	5.2 HP

DEVELOPMENT STATUS

- DESIGN COMPLETED
- HARDWARE PROCUREMENT COMPLETED
- ASSEMBLY COMPLETED
- TESTING NOT SUPPORTED DUE TO FUNDING LIMITATIONS

DRIVES

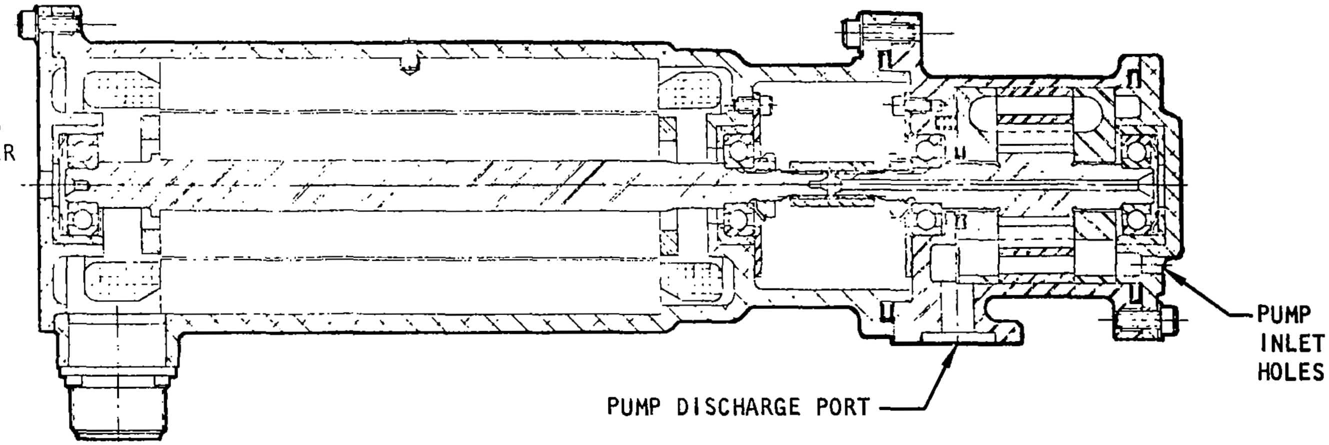
- HYDRAULIC MOTOR
- ELECTRICAL MOTOR

to obtain the higher discharge pressures for System A. Preferred location for the pump is in the tank to maintain NPSH, necessitating an electric motor drive. The pump utilizes six vanes in a double lobe configuration to balance the pumping element and minimize radial loads on the bearings and minimize discharge pressure oscillations. A schematic of the pump integrated with an electric motor drive (Pesco-Brushless DC) is shown in Fig.47 , including a summary of predicted performance in hydrogen for System A1.

It is evident that considerable development of low specific speed cryogenic pumps is required to meet the technology required for a pump fed hydrogen source. The small clearance requirements of the vane and gear pump to achieve acceptable volumetric efficiency are incompatible with the long life and high reliability required of APU components. The high speed required for the centrifugal pump (DN 10^6) with complex dirve and controls also implies questionable life and reliability. Development risk for the pump component must be carefully evaluated in terms of time and cost with respect to development requirements of alternate schemes.

LIQUID HYDROGEN VANE PUMP

MODEL S-5659
PESCO PRODUCTS
DIV. BORG WARNER
BEDFORD,OHIO



DESCRIPTION

- TYPE: DOUBLE LOBE, SIX VANES .38 IN.^{3}/REV
- MATERIAL: M-2 TOOL STEEL BRONZE VANES
- AXIAL CLEARANCE: .0002" TOTAL
  PREFERRED LOCATION: IN TANK
- WEIGHT: 4 LBS
- VOLUME: 20 IN.^{3}

DRIVES

- HYDRAULIC MOTOR-IN LINE
- ELECTRIC MOTOR - IN TANK
  - INDUCTION, 6 POLE
  - 28 VDC INPUT ("BRUSHLESS DC" PESCO PRODUCTS)
  - η = 86%

PREDICTED PERFORMANCE IN HYDROGEN-SYSTEM A1

	PEAK	MODE
SPEED:	8000	3500 RPM
DN:	80,000	35,000
H_2 FLOW:	.075	.0126 LB/SEC
EXIT PRESSURE:	860	860 PSIA
INLET PRESSURE:	35	35 PSIA
INLET TEMP.:	37	37R
OVERALL EFFICIENCY:	60	21%
SHAFT POWER:	6	3 HP
ELECTRICAL POWER:	5.5	2.75 KW
MTBO:	300 HOURS	

DEVELOPMENT STATUS

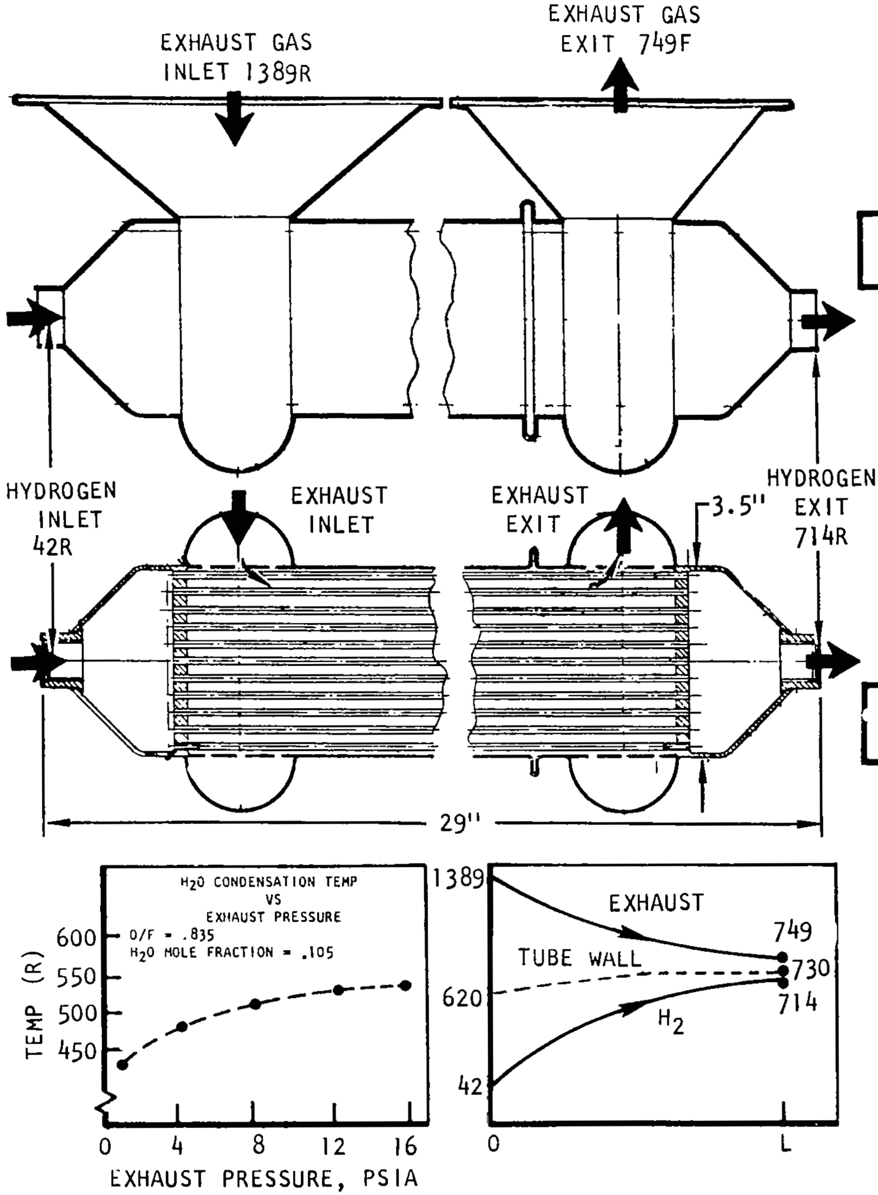
- LARGER PROTOTYPE DESIGNED, FABRICATED, TESTING INITIATED
- 12 HOURS IN LIQUID HYDROGEN AT FLOWS TO 40 GPM, DISCHARGE PRESSURE TO 260 PSIA AT 4000 RPM
- MATERIAL COMPATIBILITY DEMONSTRATED
  - ACCEPTABLE VANE WEAR
  - EFFECT OF TEMPERATURE GRADIENTS ON CLEARANCES SMALL
- TECHNOLOGY ADVANCEMENT REQUIRED FOR HIGHER PRESSURES

FIGURE 47

Regenerator

The regenerator was designed to provide full propellant conditioning at peak power (170 Btu/sec) for a 600 R injector with 10 percent bypass flow and no heat input from either cooler. The bypass is included to provide some control margin.

Several configurations were evaluated, resulting in selection of a single pass, parallel flow, shell and tube design. A typical preliminary design applicable to a system A1 type booster APU is shown in Fig. 48. At peak power, 90 percent of the total hydrogen flow passes through the tube bundle (283 1/8-in-dia tubes) in parallel flow with the exhaust gas on the shell side. The parallel flow exchanger resulted in a 5 pound weight penalty compared to a counterflow type. This is considered acceptable to avoid fouling due to condensation and freezing of exhaust gas water vapor. The tube wall temperature for the parallel flow design remains above the condensation temperature over the entire range of vacuum to sea level operation.



REGENERATOR

BOOSTER-SYSTEM AI

DESCRIPTION

- TYPE: SHELL AND TUBE PARALLEL FLOW
- MATERIAL: 347 STAINLESS STEEL
- CONSTRUCTION:
  - 283 TUBES, 1/8 OD X .010" WALL WITH .060" TUBE TO TUBE SPACING
  - BRAZED TUBE BUNDLE
  - TIG WELDED MANIFOLDS
  - SINGLE RIBBED .020" OUTER SHELL (STRESS RELIEF)
- WEIGHT: 11 LBS
- VOLUME ENVELOPE: 0.16 FT^3
- HEAT TRANSFER AREA: 18.5 FT^2

PERFORMANCE AT PEAK POWER (Q_{MAX})

- HYDROGEN FLOW: .0675 LB/SEC
- EXHAUST FLOW: .138 LB/SEC
- 10% HYDROGEN BYPASS FLOW - 600R COMBUSTOR INLET
- EFFECTIVENESS: 50%
- HEAT FLOW: 170 BTU/SEC
- EXHAUST $\Delta P = 1.2$ PSI AT S.L. 2.6 PSI AT VAC
- HYDROGEN $\Delta P = 0.05$ PSI

FIGURE 48

An exhaust nozzle sized to provide a minimum pressure of 6 psia at the regenerator hot-gas exit at zero ambient pressure is provided in the system. This increases gas density permitting a smaller regenerator size. At peak power the maximum exhaust gas pressure drop through the regenerator is then 2.6 psi, which together with a 1.5 psi duct loss, results in a turbine exhaust pressure of 10 psia. No significant performance degradation results at this condition because the turbine is still operating above design pressure ratio. The contribution of regenerator pressure drop to turbine off-design performance is about 1 percent at pressures greater than 10 psi, representing about 3 pounds of propellant weight.

Detailed stress analysis of the regenerator will be initiated in Phase II of the program after definitive system selection. Preliminary analysis during Phase I indicated that stress due to thermal expansion resulting from an exhaust gas ΔT of 650 F and an $H_2 \Delta T$ of 672 F can be relieved by providing circumferential convolutions (bellows) to accomodate about 0.050 inches of differential expansion of the outer shell relative to the tube bundle. The axial force acting on the bulkheads due to the hydrogen-exhaust pressure differential is assumed to be carried by external structure coupled to the bulkhead or hydrogen header. Brazing experience at Rocketdyne has indicated that a joint engagement length of three times the tube wall thickness results in a shear strength equal to the tube wall tensile strength. To preclude leakage and ensure a high strength joint, a bulkhead thickness of 0.25 inches was selected (25 times wall thickness).

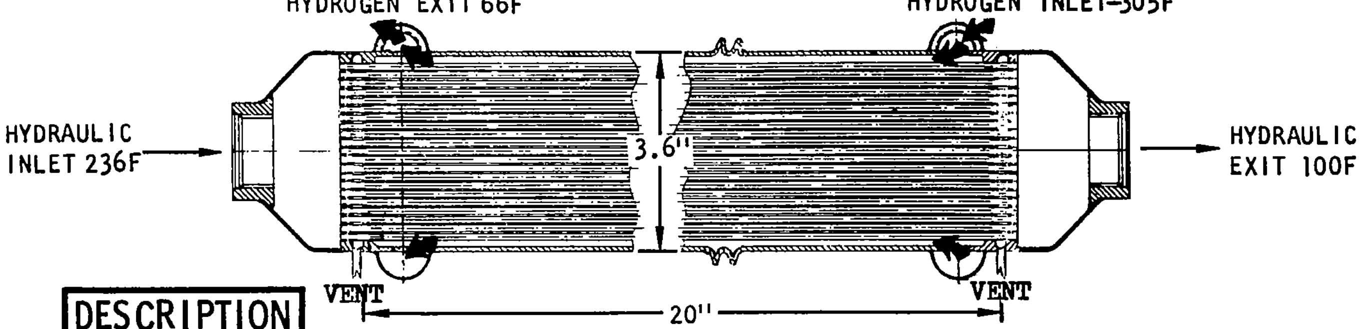
Hydraulic Cooler

Sizing of the hydraulic cooler was based on the assumption that 50 percent of the hydraulic flow power representing servo and actuator leakage and 100 percent of hydraulic pump inefficiency must be accommodated by the cooler. The maximum heat load occurs with one hydraulic pump pressurized and the other depressurized resulting in 32 HP of losses and 3.5 HP (50 percent hydraulic power) in frictional heating. Following aircraft practice, the additional 50 percent is assumed to be dissipated through the lines. The cooler is located downstream of the regenerator in the return side of the system, utilizing the hydrogen gas as the coolant. Analysis was based on the use of a silicate ester base hydraulic fluid (Chevron M2V) with a maximum operating temperature of 250 F in a 4000 PSI system. Reservoir pressure was assumed to be 100 PSI. Corresponding flow at idle is 3.1 GPM and 83 GPM at peak power with 80 GPM of peak flow doing work. The maximum fluid ΔT is 136 F for the maximum 25 BTU/sec heat load which occurs at a 56 HP gearbox output condition due to the low hydraulic flows at that condition (3.1 GPM).

Cooler configuration analysis resulted in selection of a single pass shell and tube counterflow design using 6061-T6 aluminum to minimize weight. A typical design for a system A1 booster APU is shown in Fig. 49. The hydraulic fluid flows on the tube side and hydrogen on the shell side. Flow is in the laminar regime for both fluids throughout the entire power profile so heat transfer coefficients are independent of flowrate. Hydraulic ΔP

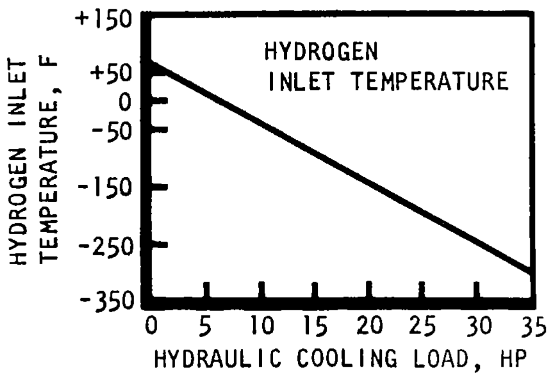
HYDRAULIC COOLER

BOOSTER SYSTEM AI



DESCRIPTION

- TYPE: SHELL AND TUBE COUNTERFLOW
- MATERIAL: 6061 ALUMINUM
- CONSTRUCTION:
  - 417 TUBES, 1/8" OD X .020" WALL WITH .030" SPACING
  - BRAZED TUBE BUNDLE, BUFFERED
  - WELDED MANIFOLDS
- WEIGHT: 8 LBS
- VOLUME: 0.12 FT3
- HEAT TRANSFER AREA: 23 FT^2



PERFORMANCE

- (MAXIMUM COOLING REQUIREMENT, LOW SHAFT POWER (56HP))
- HEAT INPUTS:
  - HYDRAULIC FRICTIONAL HEATING 3.5 HP (50% TOTAL HTG.)
  - PUMPING INEFFICIENCY 32.0 HP

  TOTAL 35.5 HP (25 BTU/SEC)

- HYDRAULIC FLUID: CHEVRON M2V
- EFFECTIVENESS: 69%
- M2V PRESSURE DROP: 2 PSI
- M2V EXIT PRESSURE: 100 PSIA
- H2 INLET PRESSURE: 840 PSIA
- H_2 PRESSURE DROP: .002 PSI

FIGURE 49

ranges from 2 psi at idle to 54 psi at peak power. This will result in a slight increase in pumping power for the short peak-power periods, but will result in a negligible spc penalty. Temperature control is achieved with a three-way thermostatic bypasss valve designed to maintain 100 F hydraulic outlet mixed temperature. The cooler was sized such that the hydrogen and hydraulic side heat transfer coefficients are equal (47 BTU/hr-ft^2-^{o}F) providing a minimum tube wall temperature greater than the pour point of the hydraulic fluid. It should be noted that when hydraulic cooling is not required (i.e., startup) the hydrogen inlet temperature will be high (66 F) since the regenerator provides the heat input normally contributed by hydraulics to achieve the required injector inlet temperature. Consequently, 'freezing' (excessive ΔP) of the hydraulic fluid should not be a problem.

Leakage of the higher pressure hydrogen (800 PSI) into the low pressure hydraulic fluid (100 PSI) due to failure of a brazed tube joint is avoided by use of a double bulkhead at each end with the section between bulkheads vented to low pressure (turbine exhaust).

Preliminary stress analysis of the cooler indicates that the largest differential temperature (bulk metal) between the outer shell and tubes will be about 145 F at the maximum heat load, resulting in a differential contraction of 0.035 inches between the outer shell and tube bundle. A tension force due to the hydrogen-hydraulic ΔP across the bulkhead (700 PSI) reduces the contraction by 0.010 inches. The critical buckling load per tube is 12 pounds indicating an outer shell circumferential convolution with a stiffness of less than 50,000 lb/in. will accommodate the travel without buckling the tubes (S.F. = 4.0). Another possible method of compensating for thermal growth is to prestress the outer shell.

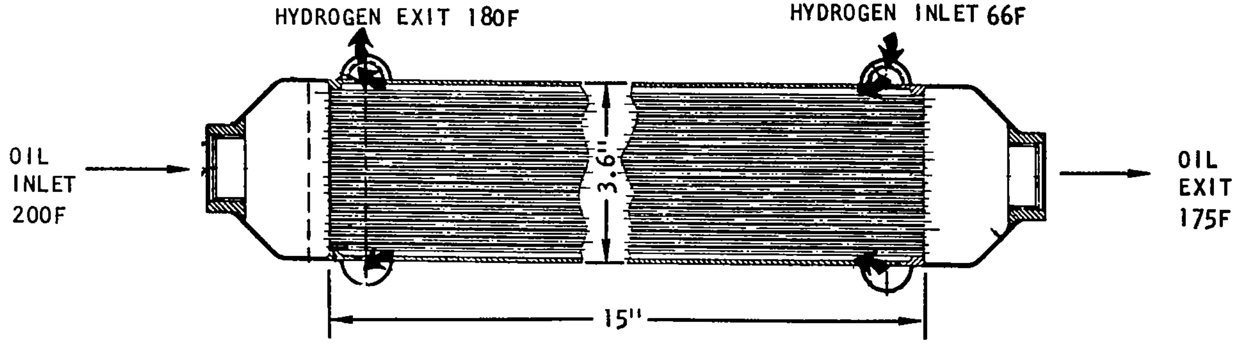
Detailed steady state and transient stress analysis will be conducted during Phase II in support of the specific system(s) selected for evaluation.

Lubricating-Oil Cooler

The lubricating-oil cooler is located downstream of the hydraulic-oil cooler, utilizing the hydrogen gas as the sink for the heat load contributed by the alternator, gearbox, bearings, seals, and lube pump. The maximum cooling load was estimated to be 10 HP for an oil flow of 5 GPM resulting in a temperature rise of 25 F. A counterflow shell and tube design almost identical to the hydraulic cooler (except for tube length) was selected to take advantage of the technological commonality (Fig.50). The counterflow configuration is lighter weight than a parallel flow type and results in higher hydrogen temperature for the same effectiveness. A cross-flow exchanger is more compact but heavier than the counterflow unit due to more complex header

LUBE COOLER

BOOSTER-SYSTEM AI



DESCRIPTION

- TYPE: SHELL AND TUBE COUNTERFLOW
- MATERIAL: 6061 ALUMINUM
- CONSTRUCTION:
  - 417 TUBES, 1/8" OD X .020" WALL WITH .030" SPACING
  - BRAZED TUBE BUNDLE
  - WELDED MANIFOLDS
- WEIGHT: 6 LBS
- VOLUME: 0.09 FT^3
- HEAT TRANSFER AREA: 17.3 FT^2

PERFORMANCE

- (MAXIMUM COOLING REQUIREMENT, LOW SHAFT POWER (56HP))
- HEAT INPUTS:

• LUBE PUMP	1.0 HP
• GEARBOX	4.3 HP
• BEARINGS AND SEALS	0.7 HP
• ALTERNATOR	4.0 HP
TOTAL	10.0 HP (7.07 BTU/SEC)

- LUBE OIL: MIL-L-23699
- EFFECTIVENESS: 85%
- OIL ΔP: 1 PSI
- H_2 ΔP: .002 PSI
- OIL INLET PRESSURE: 100 PSIA
- H_2 INLET PRESSURE: 840 PSIA

FIGURE 50

design to preclude leakage. Hydrogen (high pressure) leakage to lube-oil (low pressure) is avoided with a vented double bulkhead design at each end of the cooler, as in the hydraulic-oil exchanger. Temperature control is provided by a three way bypass valve with passive thermostatic actuation.

Calculations were based on the properties of a synthetic ester base oil (MIL-L-23699), which is widely used in gas turbines and auxiliaries. Flow is in the laminar regime for both the hydrogen and the oil so that the heat transfer coefficients are independent of flow and pressure drop for both fluids is less than 1 PSI.

Preliminary stress analysis indicates that the force acting on the bulkhead free-flow area dominates the stresses induced due to differential thermal contraction of the outer shell relative to the tubes. Differential travel is only 0.005 inches so that a bellows is not required in the outer shell. Both the outer shell and tubes are in tension with a resultant shell axial stress of 3000 PSI and tube stress of 392 PSI. The radial stress in the outer shell is 18,200 PSI for 860 PSIA hydrogen giving a factor of safety of 2.5 on the ultimate and 2.2 on the yield for the 6061-T6 aluminum. Tube radial stress is only 300 PSI assuming 100 PSIA lubricating-oil pressure.

It should be noted that since the lubricating-oil cooler is the last heat source element for the hydrogen prior to entering the H2/O2 temperature equilizer (sink) the maximum hydrogen injector inlet temperature is determined solely by the maximum lube oil temperature and cooler effectiveness. When the lube oil is below the cooling reference temperature (i.e., startup) the hydrogen becomes a heat source for the oil. Details of these operational characteristics are discussed in the Controls Section.

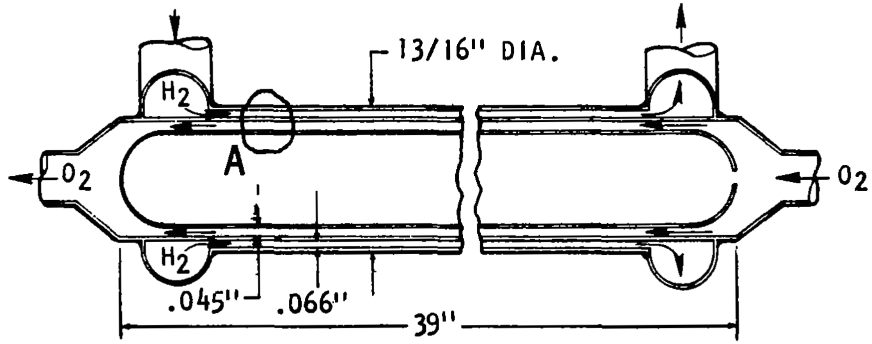
H_2/O_2 Temperature Equalizer

Thermal conditioning of the oxygen is accomplished passively with the use of a counterflow shell and tube heat exchanger using hydrogen as the heat source. The design requires 'virgin metal' construction to eliminate the possibility of a weld or braze joint failure allowing mixing of the hydrogen and oxygen which could result in catastrophic failure. A preliminary design applicable to the system A1 booster APU is shown in Fig. 51. Three concentric tubes are used to form two annular flow passages in which the oxygen flows through the inner annulus counter to the hydrogen flow in the outer annulus. The central tube is used to increase velocity (heat transfer coefficient) of the oxygen. Construction details are given in the figure. No H_2/O_2 communication can occur unless the intermediate tube cracks or develops porosity. A double wall design could be utilized to reduce the probability for this type of failure (see figure).

Heat exchanger effectiveness at peak power is 86 percent and the heat flow is 10.2 BTU/sec. The hydrogen temperature drops only 40 F (640 to 600 R) in bringing the oxygen from 200 R to 580 R. The pressure drops are not excessive, being 12.6 PSI for the GH2 and 2.8 PSI for the oxygen.

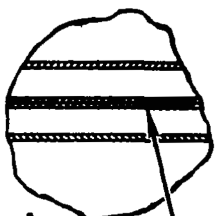
H_2/O_2 EQUALIZER

BOOSTER SYSTEM AI



PERFORMANCE AT PEAK POWER

- HYDROGEN FLOW: .075 LB/SEC
- OXYGEN FLOW: .0625 LB/SEC
- H_2 INLET TEMP: 640R
- H_2 EXIT TEMP: 600R
- O_2 INLET TEMP: 200R
- O_2 EXIT TEMP: 580R
- EFFECTIVENESS: 86%
- HEAT FLOW: 10.2 BTU/SEC
- HYDROGEN ΔP = 12.6 PSI
- OXYGEN ΔP = 2.8 PSI



A

DOUBLE WALL OPTION

- EACH WALL .020", WEIGHT PENALTY 0.5 LBS
- EACH WALL .010", NO WEIGHT PENALTY

} ASSUMING LOW THERMAL CONTACT RESISTANCE

- ADVANTAGE: REDUCED FAILURE PROBABILITY
- DISADVANTAGE: HEAT TRANSFER SENSITIVE TO CONTACT CONDUCTANCE-DIFFICULT TO PREDICT

DESCRIPTION

- TYPE: SHELL AND TUBE COUNTERFLOW
- MATERIAL: 304 STAINLESS STEEL
- CONSTRUCTION: TIG WELDED
- INNER TUBE: 1/2" X .020" WALL
- MIDDLE TUBE: 5/8" X .020" WALL
- OUTER TUBE: 13/16" X .028" WALL
- WEIGHT: 2 LBS
- VOLUME ENVELOPE: 0.12 FT^3
- HEAT TRANSFER AREA: 0.5 FT^2

FIGURE 51

The heat transfer (middle tube) wall temperature is very close to the hydrogen gas temperature since the oxygen heat transfer coefficient controls (h_O = 442, h_H = 4030 BTU/hr-ft^2-°F) and the outer tube wall will also be close to hydrogen temperature since it will be insulated. Consequently, stresses due to differential expansion are small and no stress-relieving expansion joints are required. Radial stress in the outer tube due to hydrogen pressure of about 800 PSI is 11,600 PSI providing a more than adequate margim of safety in the 304 stainless steel. No pressure differential is present across the inner tube since it sees the oxygen pressure on both sides. The middle tube is very lightly stressed since it sees only differences in H_2/O_2 pressure, which are small.

Propellant Conditioning System-Control Elements

A description of the propellant conditioning system control elements is shown in Fig. 52.

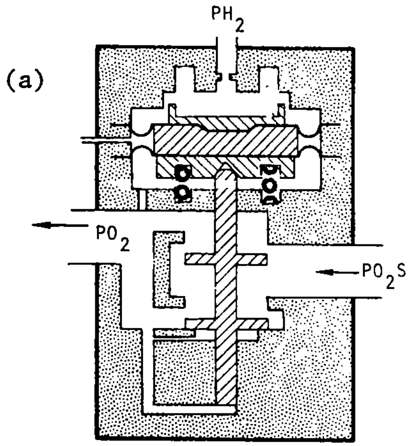
Differential Pressure Regulator - The schematic shown in the upper left of Fig. 52 illustrates a design concept for the differential pressure regulator. The regulator senses hydrogen pressure delivered to the fuel side of the bipropellant throttling valve. Oxidizer flow resistance through the regulator is controlled so that, under flowing conditions, the oxidizer pressure delivered to the bipropellant throttling valve is an increment lower than the fuel pressure.

A linear-displacement valve is positioned by a spring-biased metal-diaphragm actuator. Two opposing single-convolution diaphragms are used, with an intermediate vent cavity, to eliminate the potential for convolution reversal. The normally-closed dual-poppet valve is designed for nominal balancing of oxidizer inlet pressure axial forces applied to the valve so that regulated outlet pressure is not significantly affected by variations in inlet pressure.

Under flowing conditions, the regulator valve and actuator are positioned to maintain an axial force balance in which sensed fuel pressure force is opposed by oxidizer outlet pressure forces and by the helical bias spring and metal diaphragm mechanical spring forces. The spring forces result in a regulated oxidizer pressure that is slightly lower than the sensed fuel pressure. The orifice in the fuel pressure port and the volume of the pressure-sensing cavity are sized to provide dashpot damping for dynamic stability.

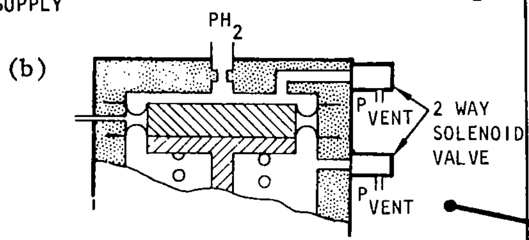
PROPELLANT CONDITIONING SYSTEM - CONTROL ELEMENTS

DIFFERENTIAL PRESSURE REGULATOR



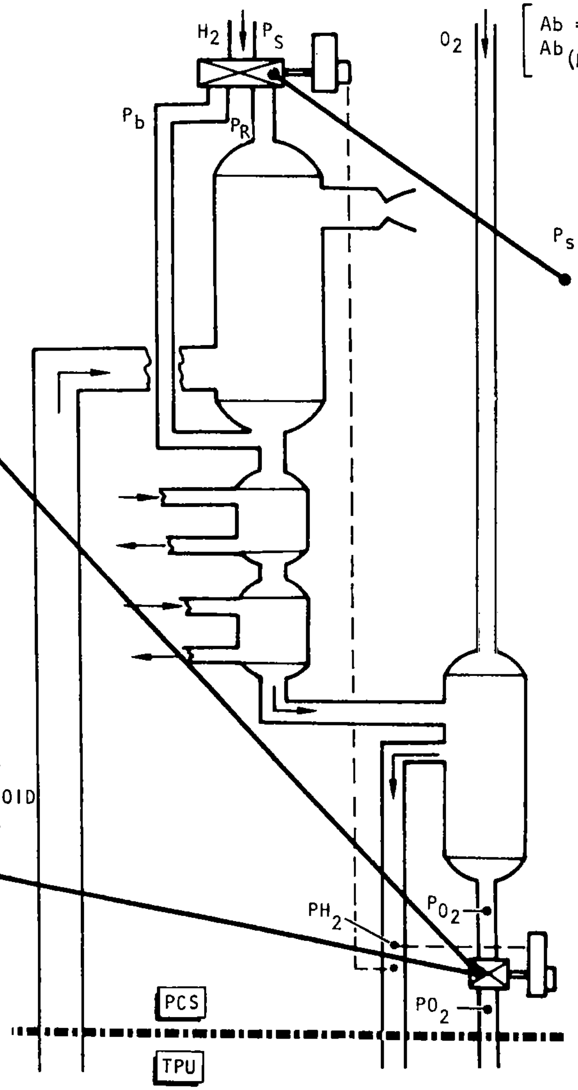
DIFFERENTIAL PRESSURE REGULATOR

- $P_{O_2} = P_{H_2} - P_b$ (FIXED BIAS ADJUSTS FOR A DEVIATION IN O_2 AND H_2 TEMPERATURE)
- TWO OPPOSING SINGLE CONVOLUTION DIAPHRAGMS WITH INTERMEDIATE VENT CAVITY (ELIMINATES CONVOLUTION REVERSAL)
- NORMALLY CLOSED POPPET DESIGNED FOR AXIAL FORCE BALANCE TO REDUCE SENSITIVITY TO P_{O_2} SUPPLY



DIFFERENTIAL PRESSURE REGULATOR

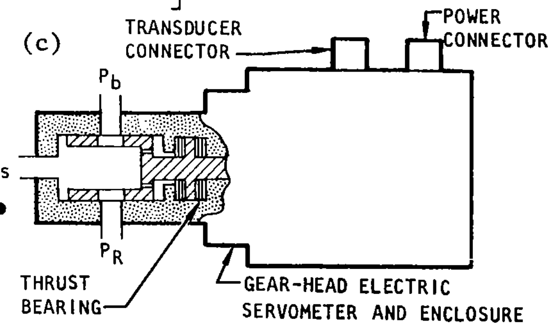
- $P_{O_2} = P_{H_2} - P_b + f(Te)$
  REGULATOR SETTING IS BIASED BY TURBINE EXIT TEMPERATURE SENSOR INDICATING T_T OUTSIDE ACCEPTABLE BAND



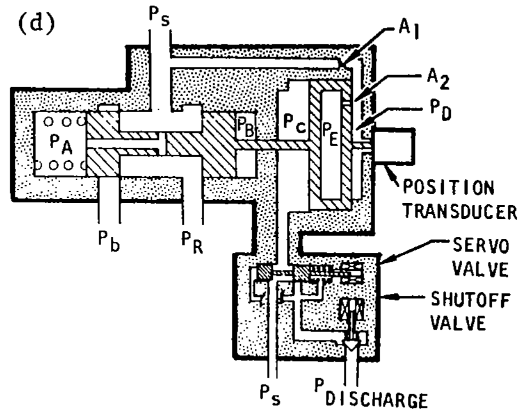
REGENERATOR - BYPASS VALVE

(FLOW SPLITTER)

$\left[\begin{array}{l} Ab = 1 - AR \\ Ab_{(MAX)} = AR_{(MAX)} \end{array}\right]$ MINIMIZES FLOW IMPEDENCE CHANGES WITH VALVE POSITION



- ROTARY DISPLACEMENT FLOW SPLITTER
- INTERNAL H_2 PRESSURIZATION - ELIMINATES DYNAMIC SEALS
- 115V, 400 Hz, 2 Ø SERVO MOTOR (AS USED IN J-2 ENGINE AT 160°R)



- LINEAR DISPLACEMENT SPOOL TYPE
- PNEUMATIC ACTUATION - NORMALLY CLOSED
- NO DYNAMIC SEALS
- WITH SHUTOFF VALVE DE-ENERGIZED - ALL LEAKAGE PATHS ARE BLOCKED

FIGURE 52

The schematic shown in the lower left of Fig. 52 illustrates a differential pressure regulator similar to the one above except for the addition of two solenoid actuated two-way valves. The regulator in the lower left of Fig. 52 is designed for operation in a system in which either turbine inlet temperature or exit temperature is monitored. If the monitored temperature exceeds a preset limit the solenoid valve that is ported to the fuel side of the diaphragm actuator is energized to permit a restricted outflow. Pressure in the sensing cavity is then less than the external fuel pressure, as determined by the ratio of the sensing cavity outflow and inflow effective flow areas. A corresponding decrease in regulated oxidizer outlet pressure results in a lesser oxidizer flowrate and a lower oxidizer/fuel flowrate ratio. A decrease in gas generator combustion temperature results. The solenoid valve remains open until the monitored temperature decreases to a preset lower limit, at which time the solenoid is deenergized, and regulator functioning without an offset in sensed fuel pressure is resumed.

If the monitored temperature is lower than a preset limit, the solenoid valve that is ported to the feedback pressure-sensing side of the diaphragm actuator is energized to permit a restricted outflow. The feedback pressure is then less than the oxidizer regulated outlet pressure, as determined by the ratio of the feedback pressure-sensing cavity outflow and inflow effective flow areas. A corresponding increase in regulated oxidizer outlet pressure is required in maintaining an actuator force balance. A greater oxidizer flowrate and oxidizer/fuel flowrate ratio result, and the gas generator combustion temperature increases. The solenoid valve remains energized until the monitored temperature increases to a preset upper limit, at which time the solenoid is deenergized, and regulator functioning without an offset in sensed feedback pressure is resumed.

Regulator Bypass Valve (Flow Splitter) - Two concepts of a regenerator bypass valve are shown in Fig. 52. As shown in Fig. 52d, the inlet port is supplied with hydrogen at pressure, P_s. One of the outlet ports, at pressure P_c delivers fuel directly to the regenerator. The other outlet port, at pressure P_D delivers fuel to a bypass line. A closed loop control system positions the valve to regulate TPU inlet temperature by controlling the regenerator and bypass flow resistances.

Both design concepts incorporate metering ports designed so that

$$\frac{Ab}{Ab(max)} = 1 - \frac{A_R}{A_R(max)}$$

Ab = bypass flow effective area

A_R = regenerator flow effective area.

as the actuator is displaced from its normal position, i.e., bypass flow area closed. This area relation minimizes flow impedance changes with valve position.

A linear-displacement spool type valve is shown in lower right of Fig. 51 The design includes a torque-motor-actuated three-way servovalve and a solenoid-actuated two-way shutoff valve for control of the cavity pressures in a piston actuator. With the shutoff valve in its normally-closed position when supply pressure is applied, all internal pressures equalize at P_s. The servovalve is spring biased to port supply pressure to the cavity at pressure P_c. Supply pressure is also ported through an orifice with effective flow areas and the piston actuator cavity volumes are sized so that pressure P_c will increase faster than pressure P_D and the spool valve will remain in its normal position when supply pressure is initially applied.

When the shutoff valve is energized to its open position, the servovalve outlet is ported to a discharge pressure that is lower than supply pressure.

The actuator is designed to accommodate operation with a compressible fluid and assumes gaseous inlet condition of the propellant. The actuator piston is hollow and an orifice with effective flow area A_2 interconnects the cavities at pressure P_D and P_E. The orifice area and cavity volume associated with pressure P_E are sized so that the dynamic interaction between pressures P_D and P_E contributes damping in obtaining dynamically stable operation with the compressible operating fluid.

The spring cavity at pressure P_A is ported to supply pressure to maintain P_A nearly equal to P_s during valve motion transients. The cavity at pressure P_B is interconnected with P_s and P_c by valve and stem guide clearances. This cavity volume is small enough to contribute dashpot damping of motion transients.

Under operating conditions, spool valve position is controlled by the modulating servovalve. The cavity at pressure P_c receives inflow through guide clearances from pressures P_B and P_D and servovalve leakage inflow from P_s. The servovalve controls outflow from P_c to $P_{DISCHARGE}$ and thereby controls P_c to maintain a spool valve and piston actuator force balance at any spool valve position. When P_c decreases, P_D also decreases, with flow from P_s to P_D through orifice area A_1 and flow from P_D to P_c through the piston clearance, and pressure P_B has a similar decrease. The helical spring force acts in the direction of permitting P_c to be less than P_D under steady-state conditions with forces balanced.

A position transducer attached to the piston provides a spool valve position feedback electrical signal for closed loop position control.

This design concept requires no dynamic seals other than the solenoid-actuated shutoff-valve poppet-and-seat closure seal. When the shutoff valve is deenergized, all leakage paths are blocked.

The upper right of Fig. 51 illustrates a design concept for a rotary bypass valve with electric motor actuator. The rotary valve flow windows are contoured for the flow area relationship previously described. This design concept includes an anti-friction thrust bearing for axial positioning of the valve in the presence of unbalanced forces. Angular positioning of the valve is controlled by a gear-head electric servomotor with position-limiting mechanical stops in the gear train.

The gear train and the motor are enclosed in a container that is pressurized internally with hydrogen at valve inlet supply pressure. This concept eliminates dynamic seals, and therefore eliminates the potential for dynamic seal leakage.

Because of the low valve effluent temperature, circulation flow between the valve cavity and the motor enclosure is restricted at the valve shaft so that heat transfer will ensure the presence of simi-stagnant gaseous hydrogen in the motor enclosure. It may be necessary to design the enclosure to ensure sufficient heat transfer to maintain the motor temperature within its allowable limits. Some 115 volt, 400 Hz, two-phase servomotors, used as rotary valve actuators, are in use on the J-2 engine with motor operating temperatures as low as 160°R and the gear train will include a valve-position feedback transducer for closed loop control of valve position. A linear-displacement transducer with a rotary-to-linear motion transmission in the gear train can be used. Use of a voltage transformer type of transducer is favored in preference to use of a rotary potentiometer to eliminate the potential for arcing in a hydrogen-filled enclosure.

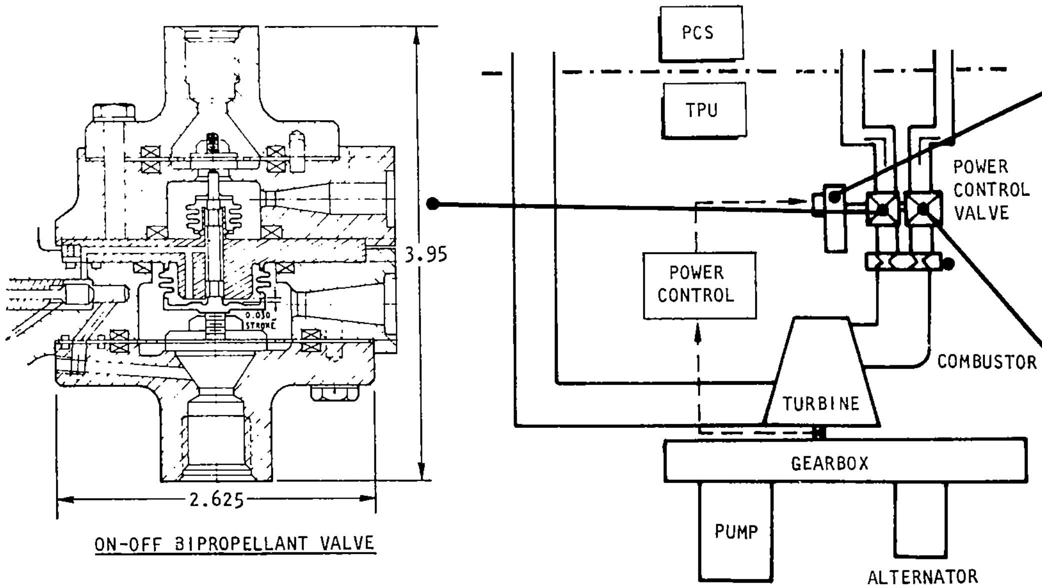
Shaft lip seals can be used to restrict circulation flow between the motor enclosure and the valve cavity and to seal the enclosure against inward migration of air or moisture under storage, handling, and standby conditions. The enclosure will include provisions for purging and filling with a dry inert gas. Tight sealing at low operating temperatures is not required, provided that sealing against moisture is obtained subsequent to each exposure to the operating temperature range.

Power Control Valve Assembly

Pulse Control - **A preliminary design concept for an on-off bipropellant valve, applicable for a pulse speed control is shown in Fig. 53 . Requirements for high cycle reliability is obtained by employing:**

POWER CONTROL VALVE ASSEMBLY

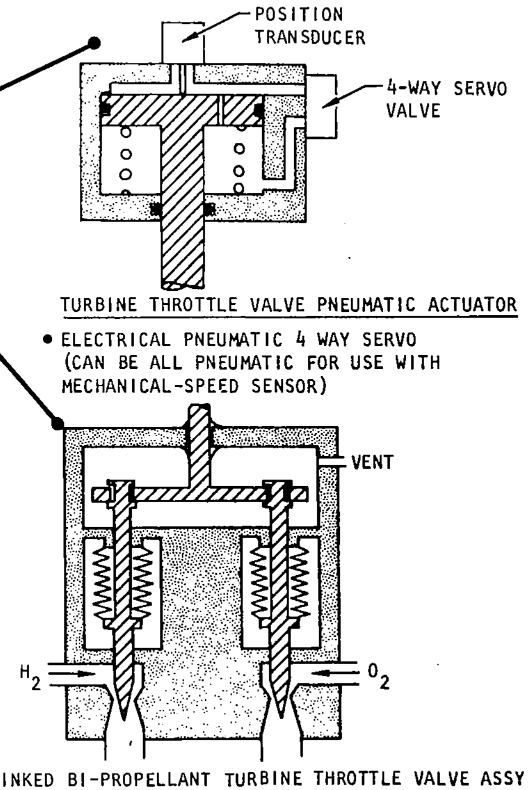
PULSE CONTROL



ON-OFF BIPROPELLANT VALVE

- ELECTRO MAGNETIC - PILOT, PRESSURE ACTUATED
- FLAT SEATS; FLEXTURE MOUNTED MINIMIZES SCRUBBING ACTION
- NO METAL TO METAL BEARING SURFACES
- NO DYNAMIC SEALS
- O_2 AND H_2 ACTUATES FAIL SAFE (CLOSURE)
- TIGHT SHUT OFF PROVIDED BY NET LOAD ON SEATS WHICH INCREASE WITH INLET PRESSURE
- 10 MS RESPONSE (SHORT STROKE, LOW INERTIA)
- REINFORCED TEFLON STEM GUIDES
- HYDROFORM BELLOWS STEM SEALS

PRESSURE MODULATED CONTROL



TURBINE THROTTLE VALVE PNEUMATIC ACTUATOR

- ELECTRICAL PNEUMATIC 4 WAY SERVO (CAN BE ALL PNEUMATIC FOR USE WITH MECHANICAL-SPEED SENSOR)

MECHANICALLY LINKED BI-PROPELLANT TURBINE THROTTLE VALVE ASSY.

- VALVE DESIGNED TO PROVIDE SONIC VELOCITY AT FLOW THROAT TO MINIMIZE SENSITIVITY TO COMBUSTOR PRESSURE VARIATIONS
- METAL BELLOWS USED AS LEAK TIGHT VALVE STEM DYNAMIC SEALS
- BELLOWS PROTECTED FROM DIRECT IMPINGEMENT OF PROPELLANTS
- FLUID PRESSURE FORCES ON BELLOWS PREVENT BACK-LASH OF LINKAGE CLEARANCES
- ZERO EXTERNAL LEAKAGE

FIGURE 53

1. No metal-to-metal bearing surfaces

2. No sliding sealing surfaces

3. Minimum linkage components

The design approach included the following factors:

1. Both oxygen and hydrogen sides actuate fail-safe and provide tight shutoff because net load on seats increases with an increase in inlet pressure.

2. Low differential pressure is achieved by large diameter seats

3. Fast response and low bellows and flexure stresses are achieved by short stroke

4. Positive actuation is achieved by large effective area difference between the oxygen and hydrogen bellows and hydrogen seat

Some detail design features are presented in Fig.52 . Operation of the bipropellant valve is as follows:

<u>Closing</u> - Fuel (gaseous hydrogen at 40 ±30 F) is directed to the inside of the stem bellows where, due to the large effective diameter and spring load of the fuel-side bellows, the bellows loads the fuel poppet to close. Oxidizer (gaseous oxygen at 40 ±30 F) pressure differential across the oxidizer poppet loads it to close. Fuel lag on closing is provided by the clearance between the oxidizer poppet and the stem.

<u>Opening</u> - Electrically energizing the three way solenoid pilot valve vents the inside of the stem bellows and allows line pressure to open the poppets. Fuel lead is provided (if required) by the clearance between the oxidizer poppet and the stem.

Pressure Modulated Control - A conventional pneumatic actuator of the type shown in Fig. 53 can be used in a turbine speed control system that uses an electrical indication of turbine speed. The servovalve controls the actuator cylinder pressures, and the transducer provides a position feedback signal for closed loop control of actuator position. An orifice through the piston permits controlled damping.

The actuator is spring biased in the direction of opening the gas generator bipropellant throttle valve. A normally-open throttle valve permits propellant inflow for starting a turbine when pneumatic supply pressure is not available. As the turbine speed and the corresponding APU hydraulic pump speed increase during a starting transient, pneumatic pressure becomes available for closing the throttle valve to the position at which nominal rated speed is obtained and maintained.

The mechanically linked bipropellant valve assembly is shown in the lower right of Fig. 53. The flow resistance for each gaseous propellant is controlled by the axial position of a movable contoured pintle with respect to a converging-diverging fixed nozzle. Each valve is designed to maintain sonic velocity at the flow throat, so that flowrates are directly proportional to the inlet pressures and independent of outlet pressure variations.

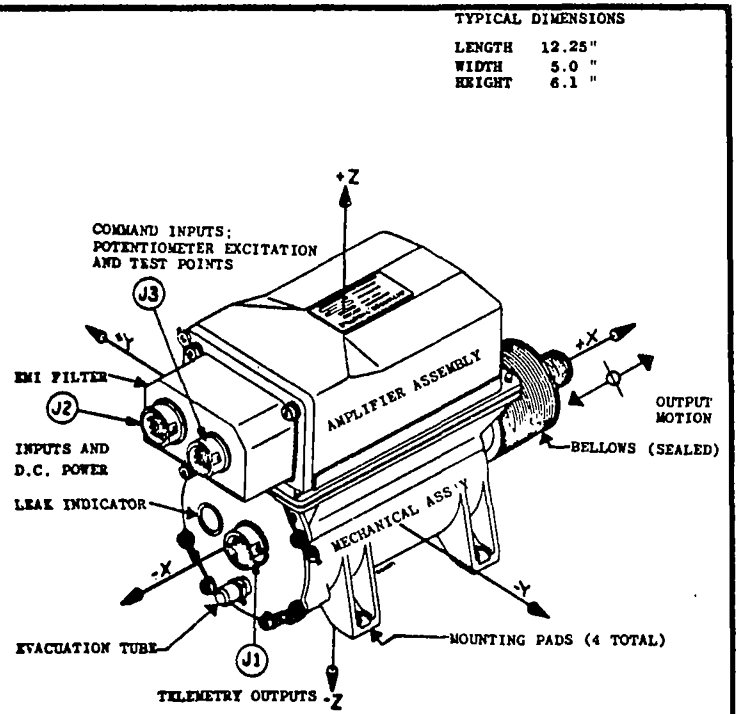
Metal bellows are used as leak-tight valve stem frictionless dynamic seals. Each bellows is enclosed in a cavity that isolates the bellows from direct exposure to propellant flow, so that cycle-life capabilities will not be degraded by direct impingement of fluids or by dynamic driving forces.

The two valves are mechanically linked to a common actuator. Supply pressure forces applied to the bellows provide bias forces that prevent backlash in the linkage clearances that are required in accommodating minor misalignments and fabrication tolerances.

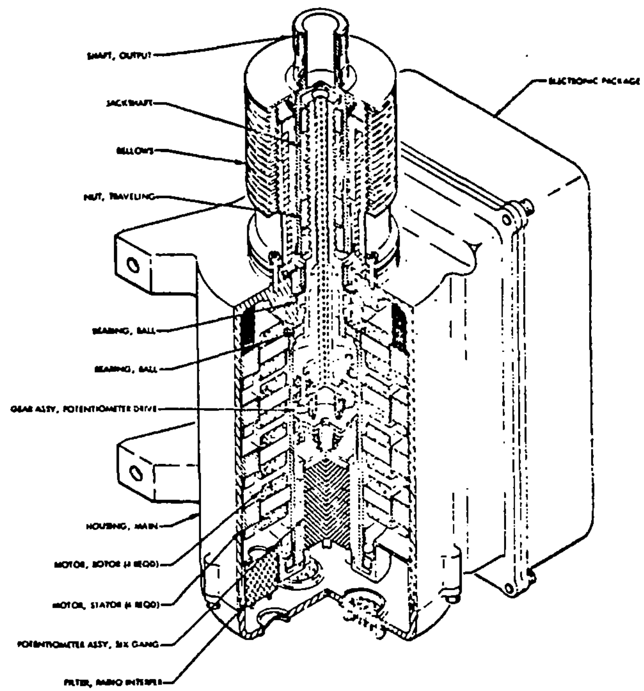
The vented linkage cavity can be protected against inflow of moisture from the surrounding ambient environment by the use of vent port check valves.

Linear displacement of the valves is required. As an alternate to the use of a piston actuator, a rotary electric motor with a ball-screw rotary-to-linear motion transmission could be used. Multiple brushless direct-current motors on a common shaft have been developed to provide redundancy of critical elements in actuators of that type. An example of this type design is shown in Fig.54 depicting a Bendix Corp. redundant electro-mechanical closed loop servo actuator system as used on the Lunar Module Engine.

BENDIX CORPORATION REDUNDANT ELECTRO-MECHANICAL
CLOSED LOOP SERVO ACTUATOR SYSTEM



THROTTLE ACTUATOR (3 MOTOR) SYSTEM
GENERAL CONFIGURATION



ACTUATOR SYSTEM (TYPICAL)
(4 MOTOR)

FIGURE 54

System Considerations - Hydrogen Environmental Effects

It has become well recognized that the properties of metals can be profoundly affected by the environment. Of particular pertinence to hydrogen-exposed systems is the relatively recent discovery that a wide range of metals are susceptible to hydrogen-environment embrittlement. The nature of this embrittlement, the conditions under which it can occur, and the methods available to prevent it have been thoroughly investigated at Rocketdyne during the past years. Based on these studies, sufficient understanding exists at Rocketdyne to ensure that no detrimental affects will be encountered in the APU.

The embrittling effects of hydrogen have long been known but the recognition that there are different types of hydrogen embrittlement is more recent. These different types of embrittlement can be designated as hydrogen-reaction, internal-hydrogen, and hydrogen-environment embrittlement. Hydrogen-reaction embrittlement can result, for example, from the formation of an embrittling hydride (e.g., titanium hydride) or of high-pressure gas pockets as the result of reaction of the hydrogen with oxygen to form water vapor or with carbon to form methane. Internal-hydrogen embrittlement is that due to hydrogen absorbed into and throughout the metal. The best recognized example of this embrittlement is the delayed failure of hydrogen-charged, high-strength steels. Hydrogen reactions and absorption of hydrogen from the gas are accelerated by elevated temperatures and, in most cases, embrittlement requires elevated temperature exposure. Extensive absorption of hydrogen also can occur because of electrolytic and chemical reaction processes. Hydrogen-reaction embrittlement and internal-hydrogen embrittlement have been extensively investigated over many years and methods of preventing these types of embrittlement are well documented.

Hydrogen-environment embrittlement is under intensive study under NASA funding as a result of hydrogen-storage vessel failures encountered in recent years. The most comprehensive of these programs was conducted at Rocketdyne and it was during these investigations that it was found that quite a wide variety of metals are to some degree susceptible to hydrogen-environment embrittlement. These, and ther recent studies, have served to clarify the nature of hydrogen-environment embrittlement and the conditions under which it occurs. This improved understanding has made it possible to develop design approaches and methods of minimizing or preventing embrittlement to ensure that failures do not occur because of hydrogen environment embrittlement. A recent Rocketdyne report* enumerates these considerations.

* Hydrogen Environment Embrittlement of Metals, Rocketdyne Report RSS-8511, Volume 35, April 21, 1971

A thorough review of the conditions which may be encountered in the APU leads to the conclusion that no detrimental effects are to be expected from hydrogen-environment embrittlement. There are five reasons for this:

1. The parts exposed to hydrogen are designed for this service. Materials are selected, based on Rocketdyne's extensive experience, to avoid problems with environmental embrittlement, or stress factors are used when the high strength steel alloys are used. In recent years an increasing knowledge has accumulated regarding hydrogen-environment embrittlement.

2. Turbomachinery and combustion devices for Rocketdyne engines (e.g., J-2), have a long, successful history of performance with hydrogen fuel without a single failure assignable to hydrogen environment embrittlement.

3. Hydrogen-environment embrittlement is most pronounced at high pressures where stress induced grain boundary separation induces the problem. The APU components are operating below 700 psia.

4. Many combustor and turbomachinery components will be exposed to a mixture of hydrogen plus water vapor. It is known that the presence of certain materials (such as oxygen) will inhibit embrittlement by hydrogen. It is anticipated that water vapor will also have an inhibiting effect upon embrittlement.

5. In areas where Rocketdyne's experience indicates that a problem could exist, components will be protected from the environmental effects by providing a barrier that prevents strength or ductility degradation due to that environment. Rocketdyne has demonstrated that copper or gold plating protects the base metal from hydrogen environment embrittlement at hydrogen pressures up to 10,000 psi.

Electro deposited copper from pyrophosphate baths can be employed so that controlled thicknesses can be deposited with a minimum of special tooling. In certain instances, internal anodes will be required due to the throwing power limitations of a copper plating system. It has been shown that as much as 0.012 inch of pyrophosphate copper can be deposited into recesses 1/4 inch deep and 1/4 inch in diameter without special anodes.

In a few components, highly recessed areas exist which do not lend themselves to the use of external anodes. In these cases where still higher throwing power capability is required, gold plating also can be employed. Special gold plating baths have been developed which have excellent throwing power (Sel-Rex BDT bath). Gold plate has been deposited to a depth of 1 inch in a hole 0.030 inch in diameter.

In general, conventional plating procedures are followed while special attention is directed to details that result in high-integrity plated deposits. These include cleaning and strike operations applicable to the alloy being protected and fixture design dictated by the configuration of the part being plated.

Some concern has recently been generated at the loss in strength and ductility for Udimet turbine wheel materials when exposed to water saturated hydrogen, especially at modest temperatures. Tests are currently planned within NASA to determine the extent of potential difficulty.

From data available to Rocketdyne, it has been concluded that the reported effect should abate as temperature increases to the level normally encountered in turbines and that persistance of any loss in strength can be countered by the use of plating to prevent direct exposure of the Udimet. Rocketdyne is, and will continue to follow developments in this area.

APU OPERATIONAL ANALYSIS

Summary

The APU has been separated into three functional subsystems: the Propellant Conditioning System (PCS) the Turbopower Unit (TPU) and the Power Control as shown in Fig. 6. The functions of each subsystem are summarized below:

1. Propellant Conditioning System: Acquire and condition the propellants as necessary, to provide gaseous propellants at controlled pressure and temperature to the TPU. Provide necessary hydraulic and lubricating oil cooling.

2. Turbopower Unit: Efficiently convert the potential chemical energy of the propellants into hydraulic and electrical power to meet the flight power profile.

3. Power Control: Provide speed control to ensure that the TPU operates within specified limits under both steady state and transient conditions during load transmission. Turbine inlet temperature control is accomplished by the relative sizing and mechanical linkage of the power control valve combined with the controlled propellant inlet conditions to the TPU at the combustor.

The PCS controls used to condition the propellants are summarized below:

1. The inlet pressure: Controlled by a GH_2 regulator located at the PCS inlet, and a differential pressure regulator which senses GH_2 pressure and establishes GO_2 pressure at the TPU inlet.

2. TPU inlet temperature: Regulated by a closed loop GH_2 temperature control utilizing a bypass valve around the regenerator. This control maintains a constant GH_2 temperature at the inlet to the TPU under varying hydraulic and lube oil heating loads, as well as GH_2 supply temperature variations. The temperature equalizer passively brings the GO_2 and GH_2 inlet temperature within close proximity.

Two types of power control were investigated, a pressure modulated control and a pulse width modulated control. Each of these power control types is directly applicable to the above described Propellant Conditioning System, and is summarized below:

1. Pressure Modulated Power Control: Function is to throttle TPU inlet pressure to the turbine in order to match turbine power against the load and thereby maintain instant speed. This is accomplished by a closed loop speed control in which any deviation from a reference speed is converted to a driving signal to modulate the valve. The valve itself is mechanically linked, providing a constant area ratio independent of valve position.

2. Pulse-Width Modulated Power Control: Function is to pulse the TPU inlet pressure to the turbine in order to provide turbine power at a fixed level and varying duration and thereby maintain speed within a predetermined band. This is accomplished by a closed loop speed control which signals an "on" pulse, opening the power control valve when TPU speed reduces to the lower band limit, and signals an "off" pulse, closing the valve, when TPU speed increases to the upper band limit. The valve is mechnaically linked, providing a constant area ratio during the "on" pulse.

Steady state and transient operating characteristics of the APU were investigated utilizing an analog model as the main analytical tool. The key problems investigated were:

1. Could the system components function satisfactorily over the required power and altitude profile, and what was the effect on mixture ratio and/or specific propellant consumption (SPC).

2. Could the APU handle varying cooling loads at different power levels without affecting mixture ratio or SPC.

3. Could the APU control system provide stable operation over the flight operational profile.

4. Could the APU control system handle severe steps in load or propellant inlet conditions without exceeding a maximum allowable speed deviation or causing detrimental excursions in turbine inlet temperature.

5. What was the sensitivity of turbine inlet temperature to errors in various components.

A summary of the APU steady-state operational characteristics as modeled on the analog computer is presented in Fig. 55. Tolerance bands for the controlled variables were established and it was shown that at the worst condition, mixture ratio would change by 3.1% causing an associated change in turbine inlet temperature of 68 R. These values are well within the acceptable limits of operation. The steady-state results also show that the pulse-width modulated system control was able to maintain turbine inlet temperature at ±13R, pressure at ±1.2% and mixture ratio at ±0.8% when power output was at 25%. At 100% power, the temperature variation was ±23R. For the pressure modulated system, the variations were essentially zero.

Figures 56 and 57 show the analog output in response to a step change in power. The pressure modulated system (Fig. 56) adjusts within approximately 1 second and the variation in system speed is well within the required ±5%. The turbine inlet temperature variation is short and well within acceptable limits.

TURBINE INLET TEMPERATURE

MAXIMUM PROBABLE STEADY STATE VARIATIONS

TOLERANCE	MIXTURE RATIO VARIATION	TURBINE INLET TEMPERATURE VARIATION
$\Delta T_H = +5\%$ $\Delta P_O = +2\%$ $\Delta A_O = +1\%$	$+3.1\%$	$+68R$

PRESSURE MODULATED POWER CONTROL

TRANSIENT RESULTS

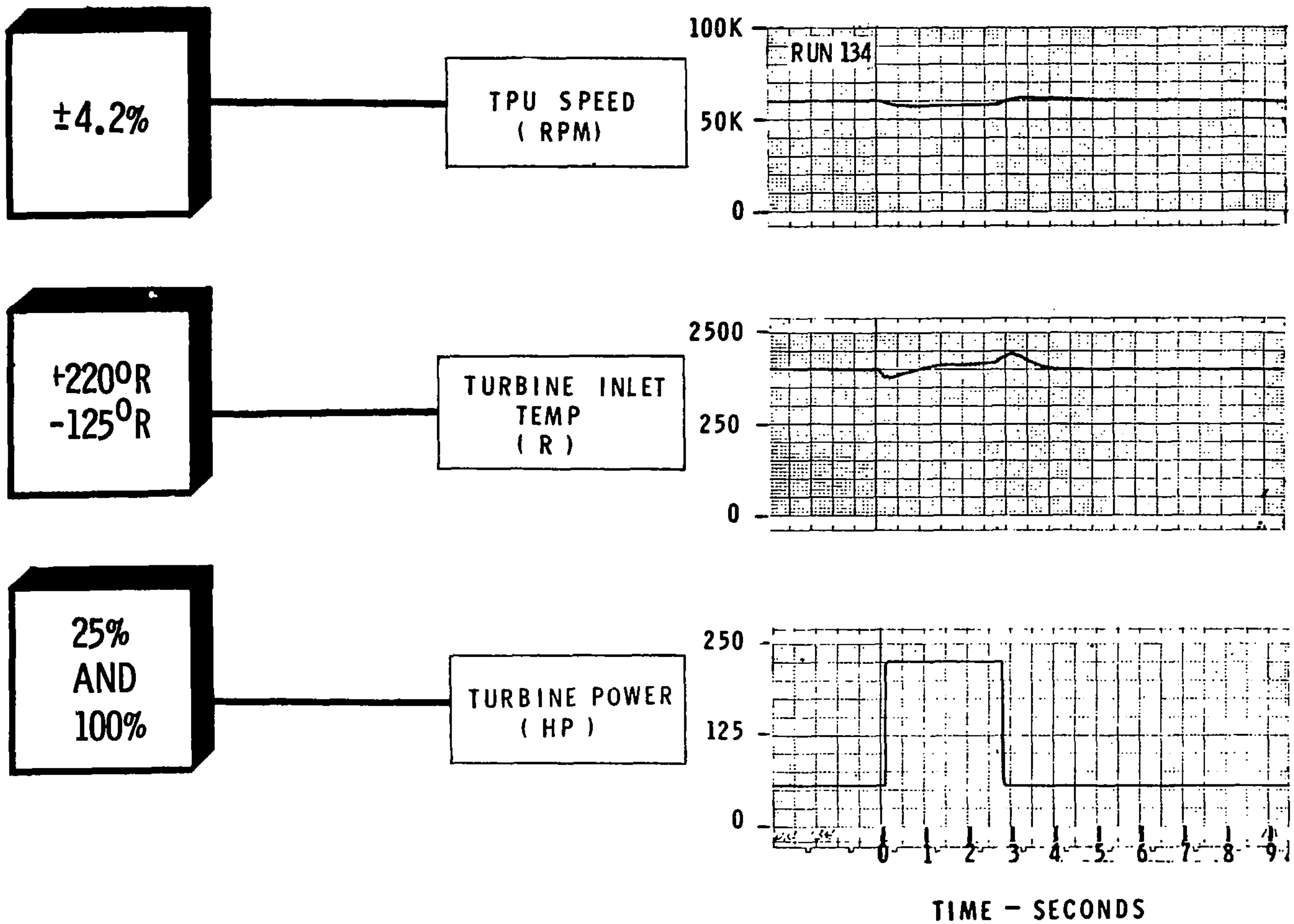


Figure 56

PULSE POWER CONTROL

TRANSIENT POWER

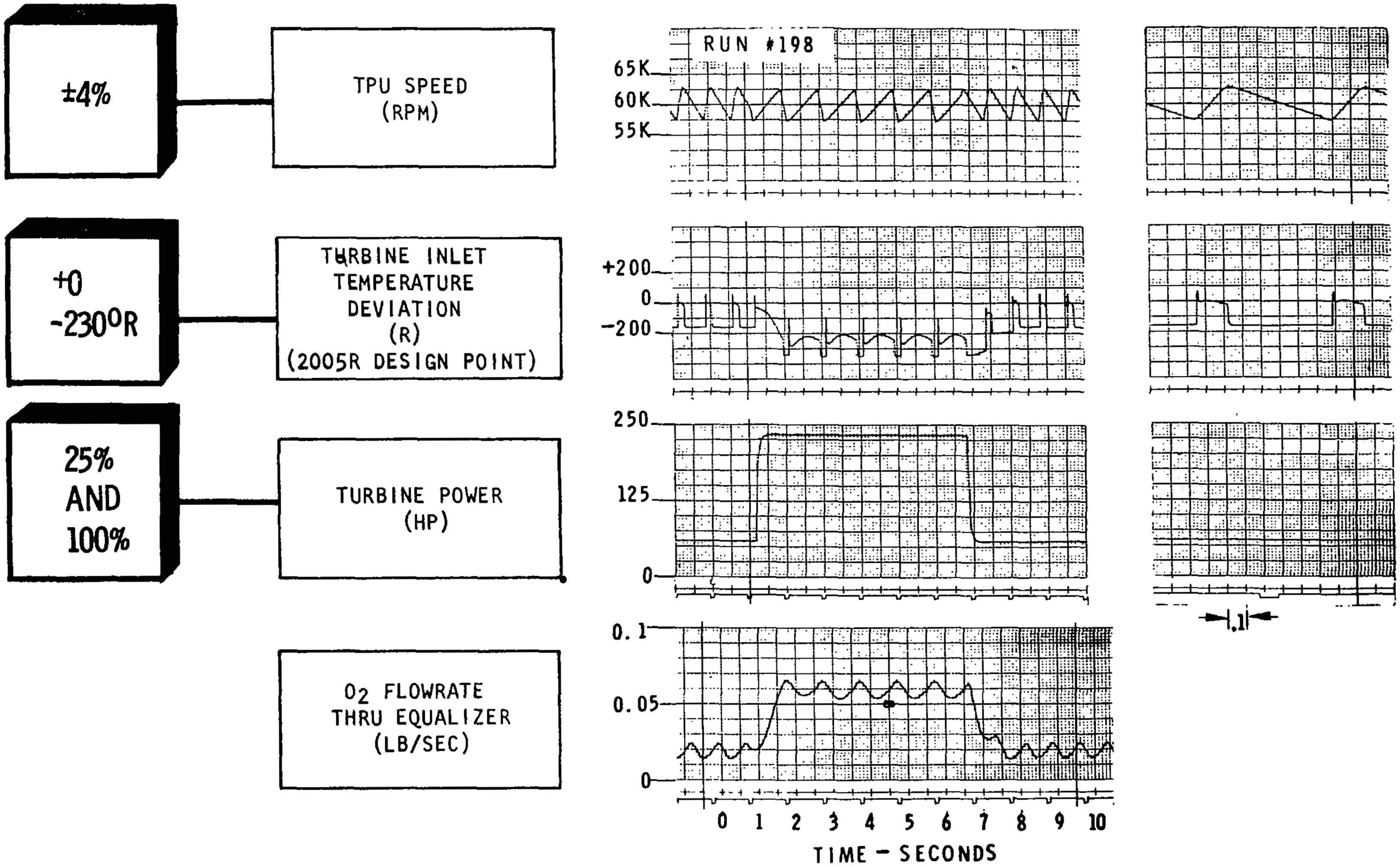


Figure 57

The overtemperature transient of $\pm$220R at the turbine inlet when changing power could be reduced by utilizing a droop type differential pressure regulator. Such a characteristic was incorporated into the analog program and it eliminated any overtemperature condition. The pulse modulated system also accomodates to step power demand within totally acceptable limits (Fig. 57). It should also be noted that while combustor flowrate pulses, the propellant flowrate (e.g., O_2) modulates with only small oscillations. This results because accumulators are used downstream of the pressure equalizer.

A transient condition due to a severe oxidizer supply pressure variation of 200 psi/sec was also investigated. Turbine inlet temperature changed by 15R for the pressure modulated system, and by 60R for pulse-width modulation. Both excursions are felt to be acceptable.

A closed loop turbine inlet temperature control was incorporated in the APU with pressure modulated power control. This control loop senses a deviation in turbine inlet temperature from a "set" value and modifies the differential pressure regulator reference to change TPU GO_2 inlet pressure and thus alter mixture ratio. The effect of this control was to reduce turbine inlet temperature excursions during the power transients to +65 and -45R. Startup and shutdown transients have not yet been investigated on the analog model.

Hydraulic and lube oil cooling loads of up to 28.2 hp at 25% TPU power level and 56.4 hp at 100% TPU power level were satisfactorily accomodated over the Flight Operational Envelope. In addition, a nearly constant sink temperature ($\pm$10R) for the hydraulic and lube oil coolers was maintained over the operating conditions described above. This was made possible by the controlled TPU inlet temperature, T_{H_2}, and the fact that only small variation existed in hydrogen temperature drop across the equalizer.

Two types of APU propellant supply systems were simulated: supercritical storage of hydrogen and oxygen, and a stored gas supply system. The steady state variations in propellant conditions associated with each supply system were accomodated by the propellant conditioning system, and turbine inlet temperature was maintained within an acceptable range of $\pm$60R.

During constant power operation with fixed propellant supply conditions, a pressure modulated APU maintains TPU speed at both 25 and 100% power level. Also, turbine inlet temperature and other system parameters remain virtually constant, the only deviation resulting from normal modulating control action. Under the same operating conditions, the analog model showed that the APU with a pulse power control would hold the critical parameters of TPU speed and turbine inlet temperature within an acceptable band. The deviation in turbine inlet temperature of $\pm$23R occurs during an "on" pulse and is due primarily to the transient variation in relative pressure of the GH_2 and GO_2 alternator tanks which are located at the TPU inlet. At 100% power level, a small downward shift in mixture ratio has been incorporated, by utilizing a differential pressure regulator with standard "droop" characteristics. The extent of this shift is controllable by the gain built into the regulator. The steady state regulator characteristics can be described by the relation

$$X_R = K_G (P_{H_2} - P_{O_2})$$

where

$X_R =$ Regulator valve position

$K_G =$ Regulator gain

$P_{H_2} =$ TPU GH_2 inlet pressure

$P_{O_2} =$ TPU GO_2 inlet pressure

At increased power level, the valve position, X_R, increases, and, depending upon the value of the gain, K_G, results in an increased deviation in TPU inlet pressures, causing a downward shift in mixture ratio. This operating characteristic is desirable in reducing overtemperature excursions during power transients.

It has been concluded that satisfactory steady state operating characteristics was demonstrated over the flight operational range by the dynamic analog of the APU with both a pressure-modulated and pulse-width modulated power control.

Operational Analysis Procedure

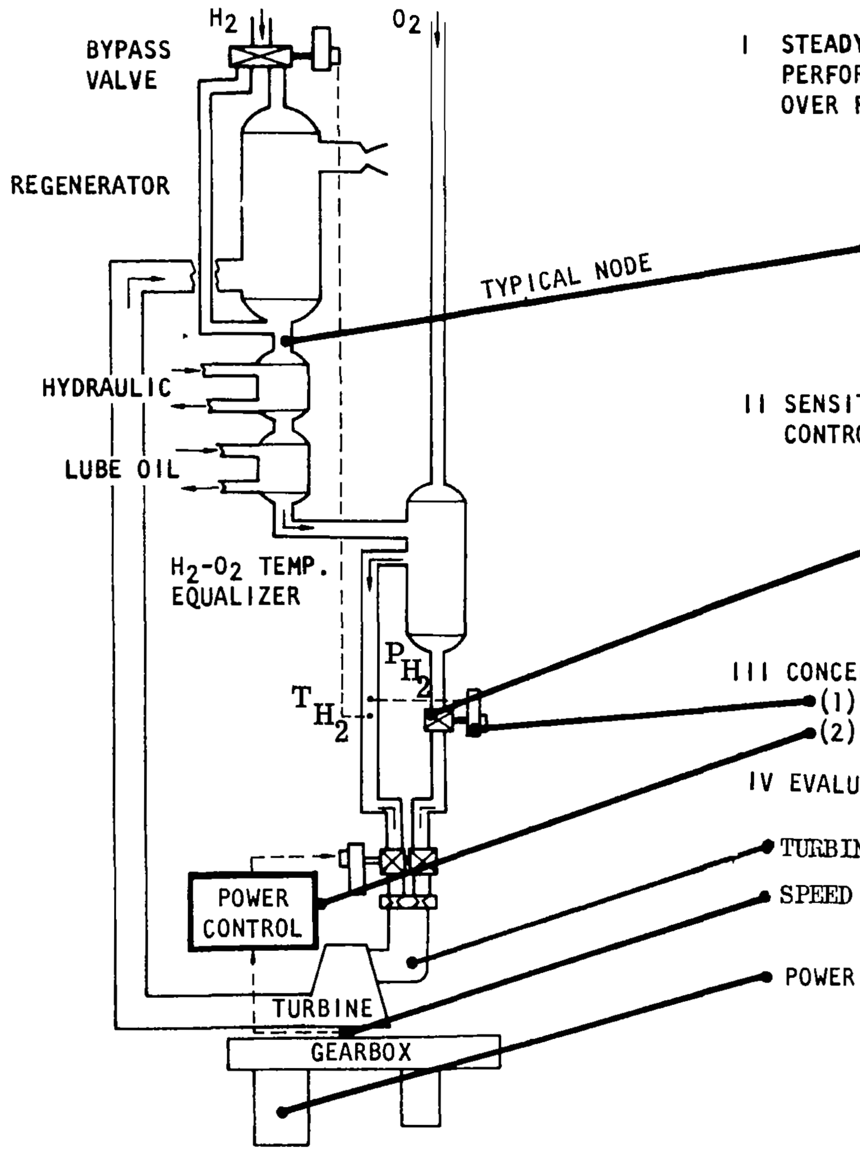
An operational analysis of the APU system was conducted using an analog computer to evaluate steady state and dynamic performance. The analysis procedure is summarized in Fig. 58.

Each of the major components was "mapped" to evaluate such parameters as flowrate, mixture ratio, pressures, and temperature under steady state conditions over the power-altitude flight profile. In addition the effect of hydraulic and lubricating oil cooling loads and APU hydrogen and oxygen inlet conditions was also investigated.

Steady state analysis was performed with the APU control system active to evaluate the sensitivity of mixture ratio and turbine inlet temperature to variations in propellant inlet conditions and errors in control components. During transient studies, the sensitivity of the system to attenuator tank volumes (for the pulse power control), differential pressure regulator characteristics (gain and response), and TPU inertia were also investigated.

Conceptual design of the propellant conditioning and power controls was performed through the use of conventional control synthesis techniques, however control optimization was not carried out during this phase of the contract.

OPERATIONAL ANALYSIS PROCEDURE



I STEADY STATE SYSTEM AND COMPONENT PERFORMANCE (COMPONENT MAPPING) OVER F.O.E.

MEASURE	VARY
PRESSURE TEMPERATURE FLOWRATE	• P_{AMB} • COOLING LOAD • APU INLET CONDITIONS • POWER LEVEL

II SENSITIVITY ANALYSIS TO EVALUATE CONTROL REQUIREMENTS

III CONCEPTUAL DESIGN OF:
(1) PCS CONTROLS
(2) POWER CONTROLS

IV EVALUATE DYNAMIC PERFORMANCE

- TURBINE INLET PRESSURE, TEMPERATURE
- SPEED
- POWER

FIGURE 58

Dynamic performance of the APU was evaluated during simulated operational conditions, e.g., large step changes in power demand, and variations in hydraulic and lubricating oil cooling loads and propellant inlet conditions to the APU.

Detailed characteristics of the APU system components were evaluated at their reference design condition using the analog computer. For example, a distributed parameter analysis of the heat exchangers provided an axial temperature and pressure distribution of the propellants as well as wall-temperature gradients. Using the same basic heat transfer and momentum relatives, one and two node lumped-parameter dynamic equations were written for the heat exchanger and mechanized on the analog model. Similarly for the turbine, detailed off-design characteristics were computed as a function of inlet and exit pressure levels, turbine inlet temperature, and speed. These characteristics were duplicated on dialed function generators for use with the analog model.

Control valves were sized and simulated based upon compressible flow relations with an assumed linear area-travel relation. Since no detailed component designs existed, reasonable response characteristics were assumed. Controllers were either of the Type zero (proportional) or Type I (proportional plus integral).

Development of General Equations

The APU system is divided into "sections" and "nodes" as depicted in Fig. 59. The circled numbers in Fig. 59 indicate node locations where the equation of state and continuity relations are used to calculate propellant pressure and temperature, accounting for fluid capacitance effects. The uncircled numbers represent sections between the nodes where thermodynamic and momentum relations are employed to calculate heat transfer rates, wall temperatures, friction losses and fluid inertia effects.

A summary of the general equations used in the model is shown in Fig. 60. In each model "section," heat transfer relations are calculated as follows where applicable:

$$hA_i = f\,(Re_i,\ Pr_i)$$

hA = product of heat transfer coefficient and area (Btu/sec R)

Re = Reynolds number

Pr = Prandl number

$$\dot{q} = (hA)_i\,(T_i^w - T_i^b)$$

$\dot{q}$ = heat flux, (Btu/sec)

T_i^w = average wall temp. in i^{th} Section, (°R)

T_i^b = Avg. propellant bulk temp. in i^{th} Section (°R)

APU SCHEMATIC ANALOG MODEL MECHANIZATION

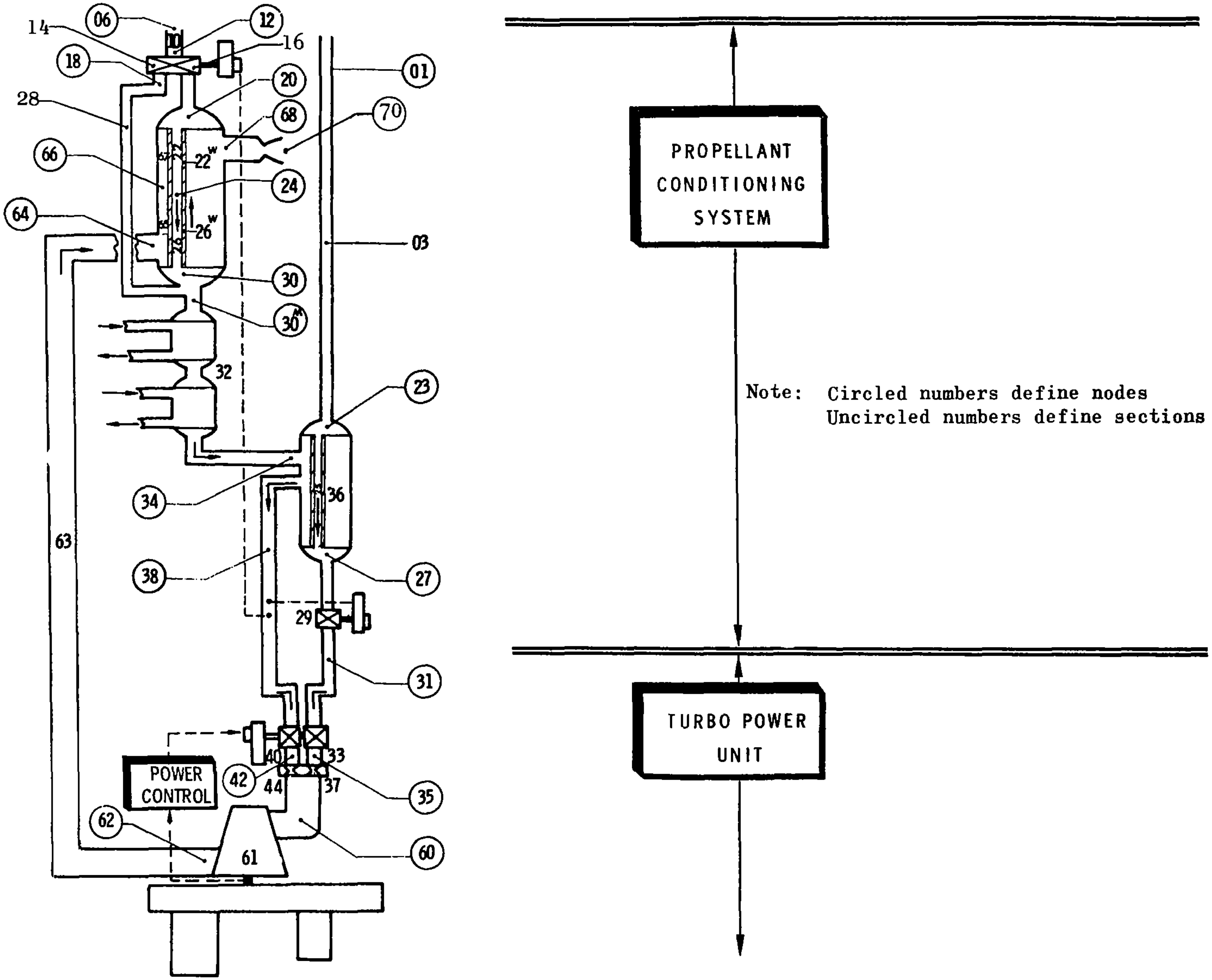


FIGURE 59

ANALOG MODEL GENERATION

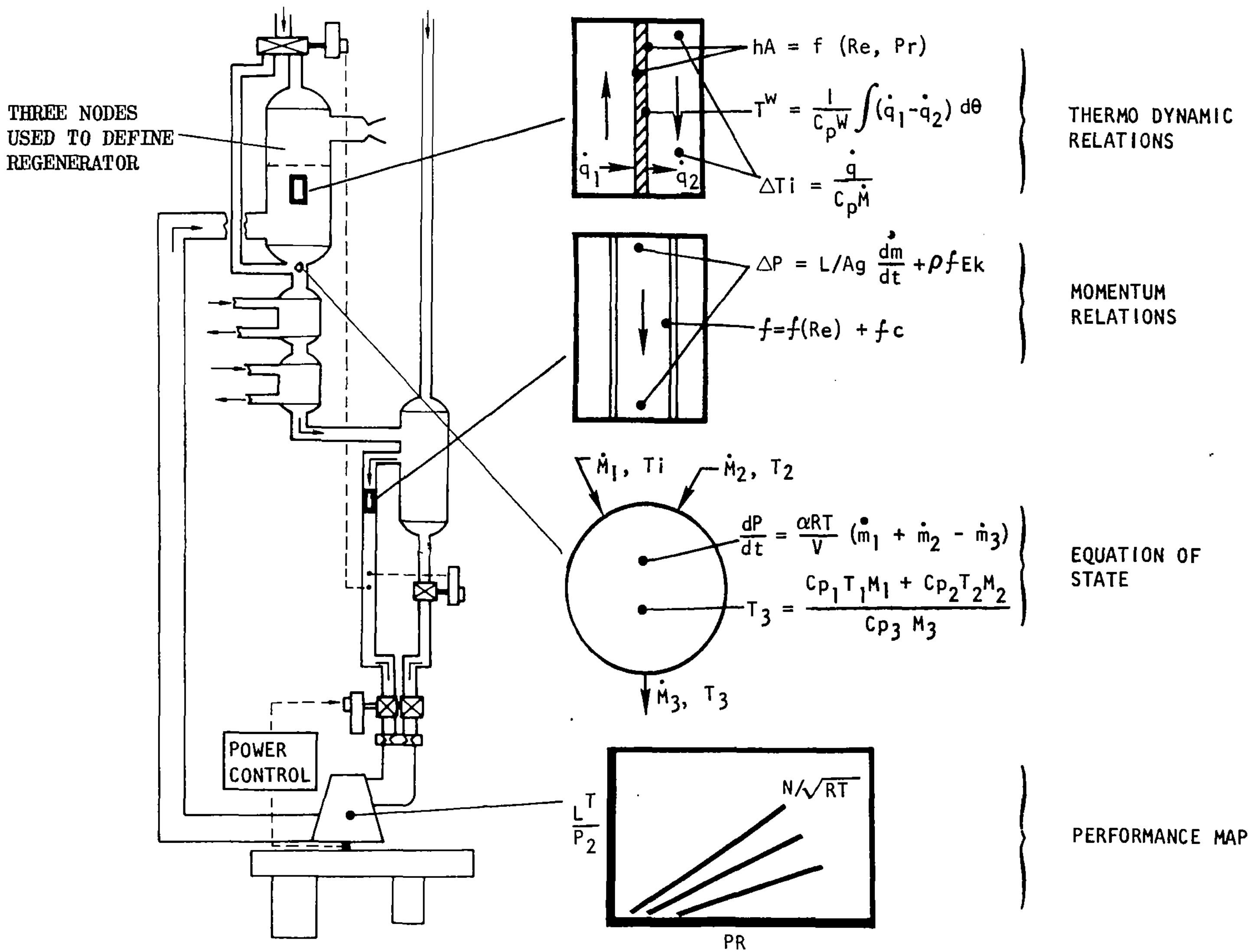


FIGURE 60

$$T_i^W = \frac{1}{(CpW)} \int (\dot{q}_1 - q_o) dt$$

(CpW) = product of wall specific heat and weight (Btu/°R)

$$\Delta T_i = \frac{\dot{q}}{Cp\dot{m}}$$

T_i = propellant temperature change in i^{th} Section, (°R)

Cp = propellant specific heat, (Btu/R-lb)

$\dot{m}$ = mass flowrate through i^{th} Section (lb/sec)

Momentum relations in each of the model sections are calculated as follows where applicable.

$$\Delta P_i = \frac{L}{Ag} \quad \frac{d\dot{m}}{dt} \quad + \rho f_T \; E^k$$

$$f_T = f_d + f_c$$

P_i = = pressure drop in i^{th} Section, (psi)

L = Length of Section, (in)

g = acceleration of gravity (lb/in^3)

f_T = total friction factor

E^k = $\frac{\nu^2}{2g}$

f_d = friction drag, a function of Re

f_c = contraction, expansion and turning loss.

ρ = propellant density (lb/in^3)

= propellant velocity

At the "node" locations the following continuity relations apply:

$$\frac{dP}{dt} = \frac{\gamma RT}{V} (\dot{m}_1 + \dot{m}_2 - \dot{m}_3)$$

P = pressure at the node, (psia)

γ = ratio of specific heat of the propellant

R = propellant gas constant, (in/°R)

V = volume occupied by propellant

$$T_3 = \frac{Cp_1 T_1 \dot{m}_1 + Cp_2 T_2 \dot{m}_2}{Cp_3 \dot{m}_3}$$

In addition to the above relations, turbine and pump performance maps are utilized to describe off-design and transient characteristics. System lines, ducting and valves were sized and incorporated in the analog model as shown in Fig. 61.

The model was mechanized on an AD256 Computer utilizing the following equipment:

140 amplifiers

52 multipliers

In addition, three function generators and digital logic equipment was used. A PDP-8 digital computer is "tied in" to the analog AD256 to perform automatic set-up and read-out functions.

APU SYSTEM LINE SIZES

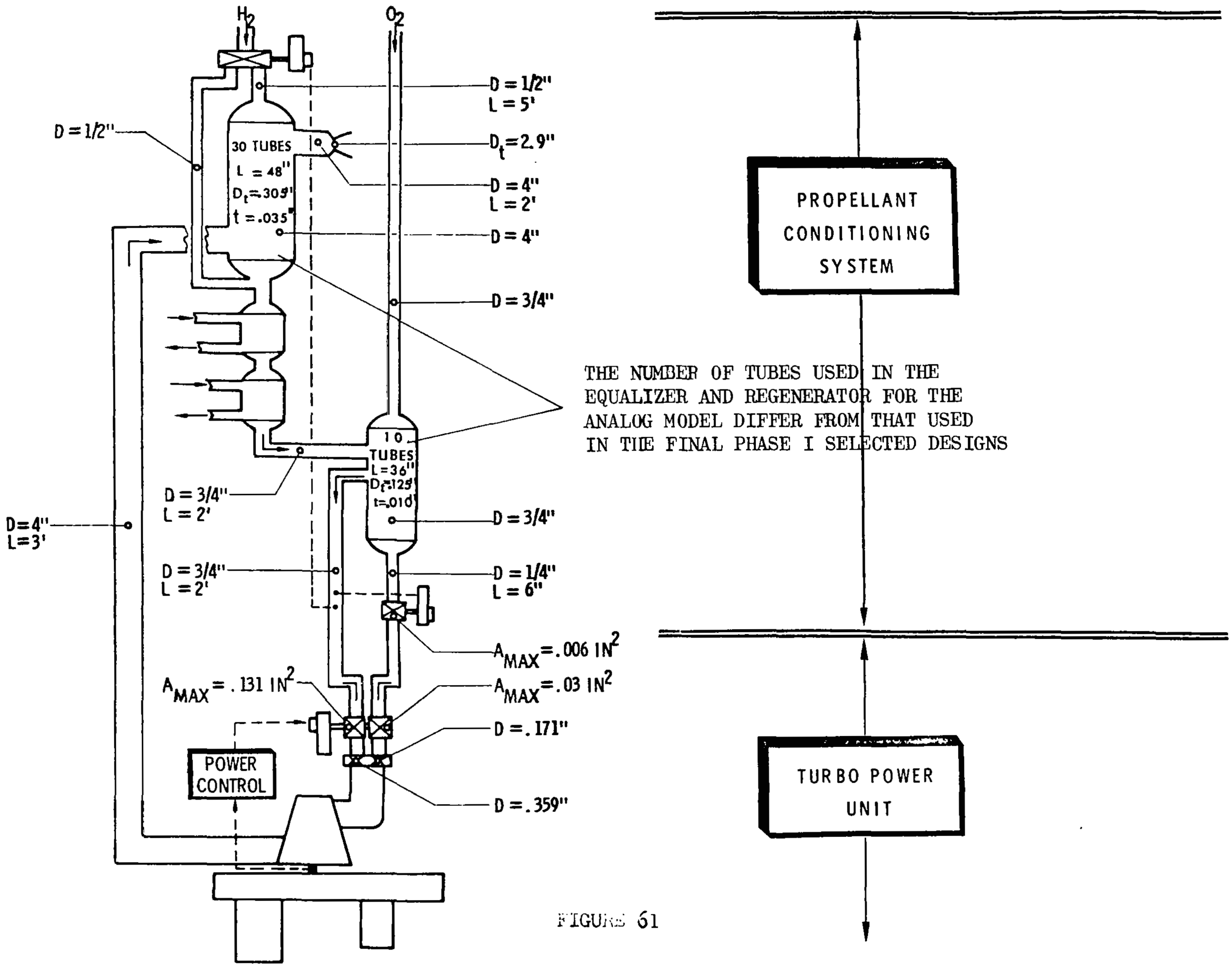


FIGURE 61

Steady State Performance and Component Mapping

The analog model was first used to determine steady state system parameter variations over a Power-Altitutde flight profile matrix utilizing an APU with a pressure modulated control. The pulse control was evaluated later during transient analysis studies.

The bipropellant conditioning system is applicable to both a pressure modulated and pulse power control, with, at most, some minor differences in the two systems. While the efficiency of the TPU design is highly dependent upon the type of power control with regard to selected design pressure, ratio, power level, tip speed, turbine inlet temperature, etc., the operational characteristics are not. A discussion of the TPU design details and optimization for each of several different system concepts is presented elsewhere in this report. For use with the analog model, a single turbine design, optimized for a pulse-power control, was mechanized and used also with the pressure modulated power control. Because the objective of the analog model was to evaluate operational characteristics rather than to make a comparison of SPC, this was considered a reasonable approach.

Utilizing a pressure modulated control, with a supercritical propellant storage system, the major parameters of the APU were evaluated at 100% power level and are shown in Figs. 62 and 63 for both zero and a 39.8 Btu/sec hydraulic/lube oil cooling load. Ambient pressure is 10 psia.

APU SYSTEM PERFORMANCE

100 % POWER LEVEL

NO COOLING LOAD - PRESSURE MODULATED

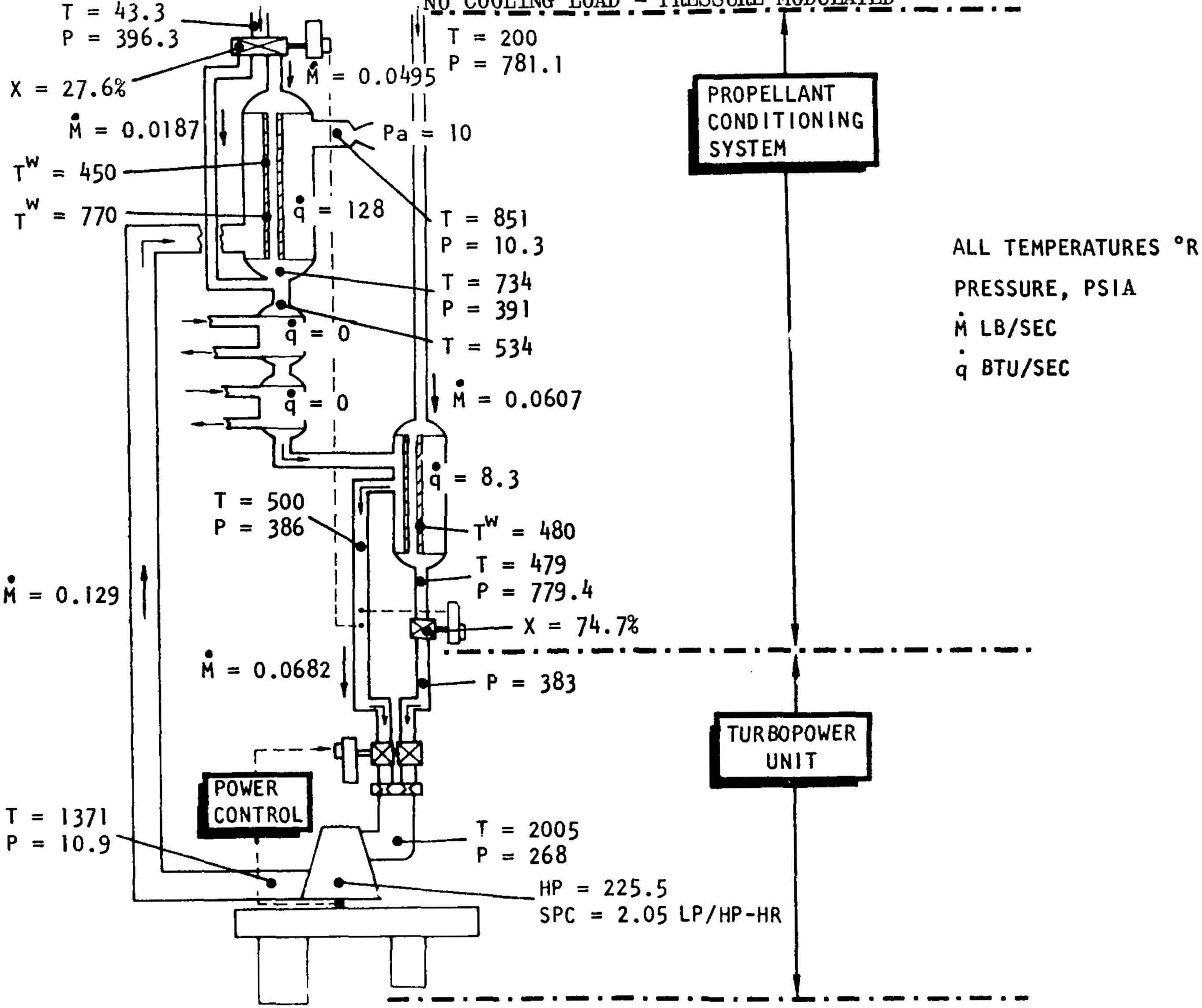


FIGURE 62

APU SYSTEM PERFORMANCE

100 % POWER LEVEL

WITH COOLING LOAD - PRESSURE MODULATED

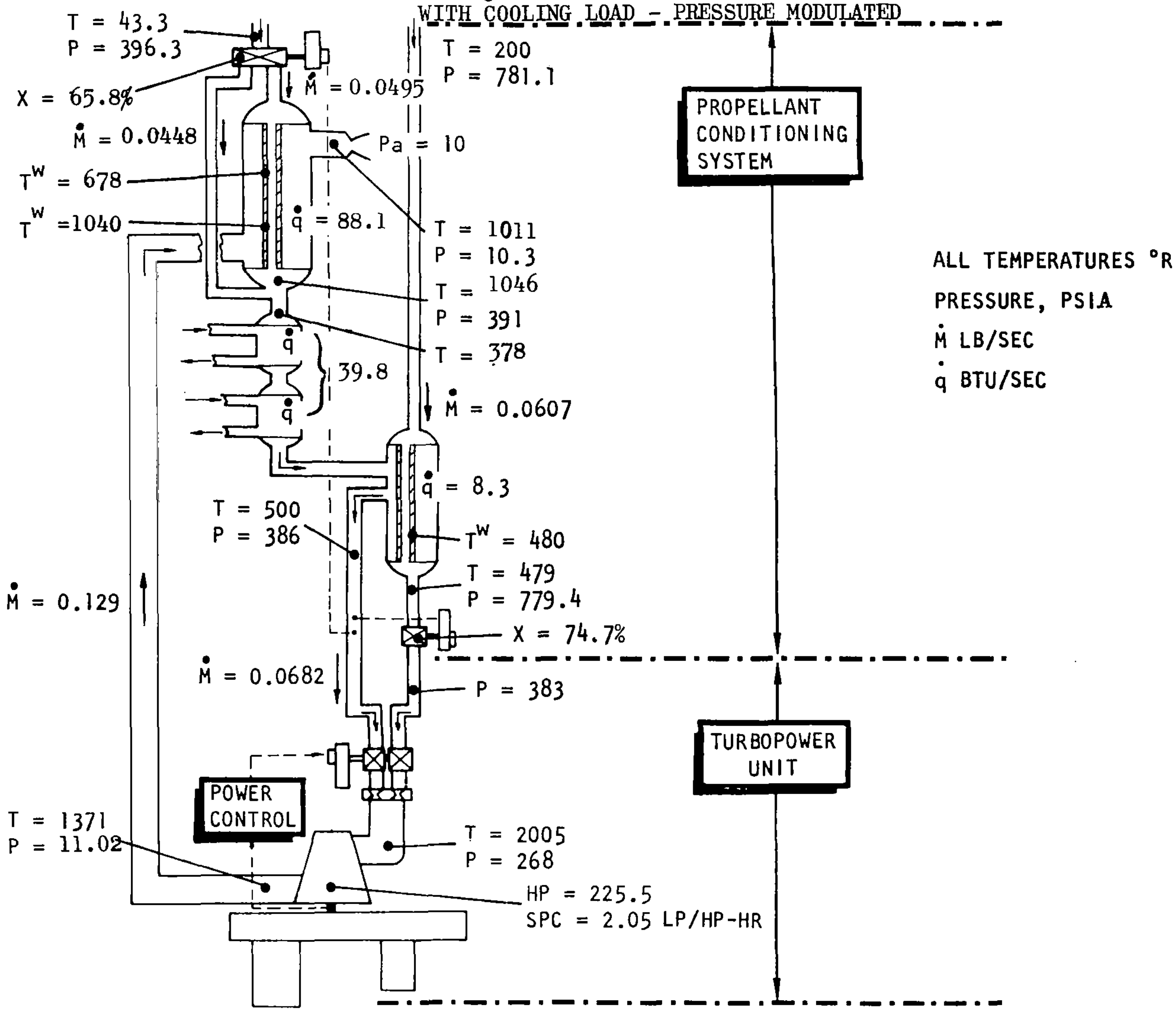


FIGURE 63

Figures 64 and 65 show the same system at 25% power level with zero and a 19.9 Btu/sec hydraulic/lube oil working load.

Turbine performance over the power altitude flight profile is illustrated in Fig. 66. There is no degradation in SPC (constant at 2.05 lb/hp-hr) over the power range at an ambient pressure of 0 psia. At 25% power, SPC is degraded when the ambient pressure (P_a) exceeds 2.6 psia. At 100% power level SPC degradation occurs for $P_a > 10.1$ psia. Compared with the SPC at 0 psia ambient pressure, a 26.7% degradation occurs at 25% power and 10 psia ambient. At P_a=0 psia, a pressure ratio of 51.3 is established across the turbine by sizing of the exit nozzle. This pressure ratio remains constant independent of power level as a result of the choked exit nozzle. At high ambient pressures, turbine pressure ratio drops below the design level, and results in the SPC degradation.

Turbine exit temperature is constant at 1367 R independent of power level for P_a=0. When performance is degraded, turbine exit temperature increases, reaching a maximum of 1567 R at sea level and 25% power level.

Performance of the H_2/O_2 equalizer is illustrated in Fig. 67. Total heat flux rates are linear with power level at P_a=0 psia, reflecting a constant SPC. The average wall temperature varies slightly ($<$ 10R) over the power-altitude flight profile. Hydrogen temperature level is controlled to 500R

APU SYSTEM PERFORMANCE
25% POWER LEVEL

NO COOLING LOAD - PRESSURE MODULATED

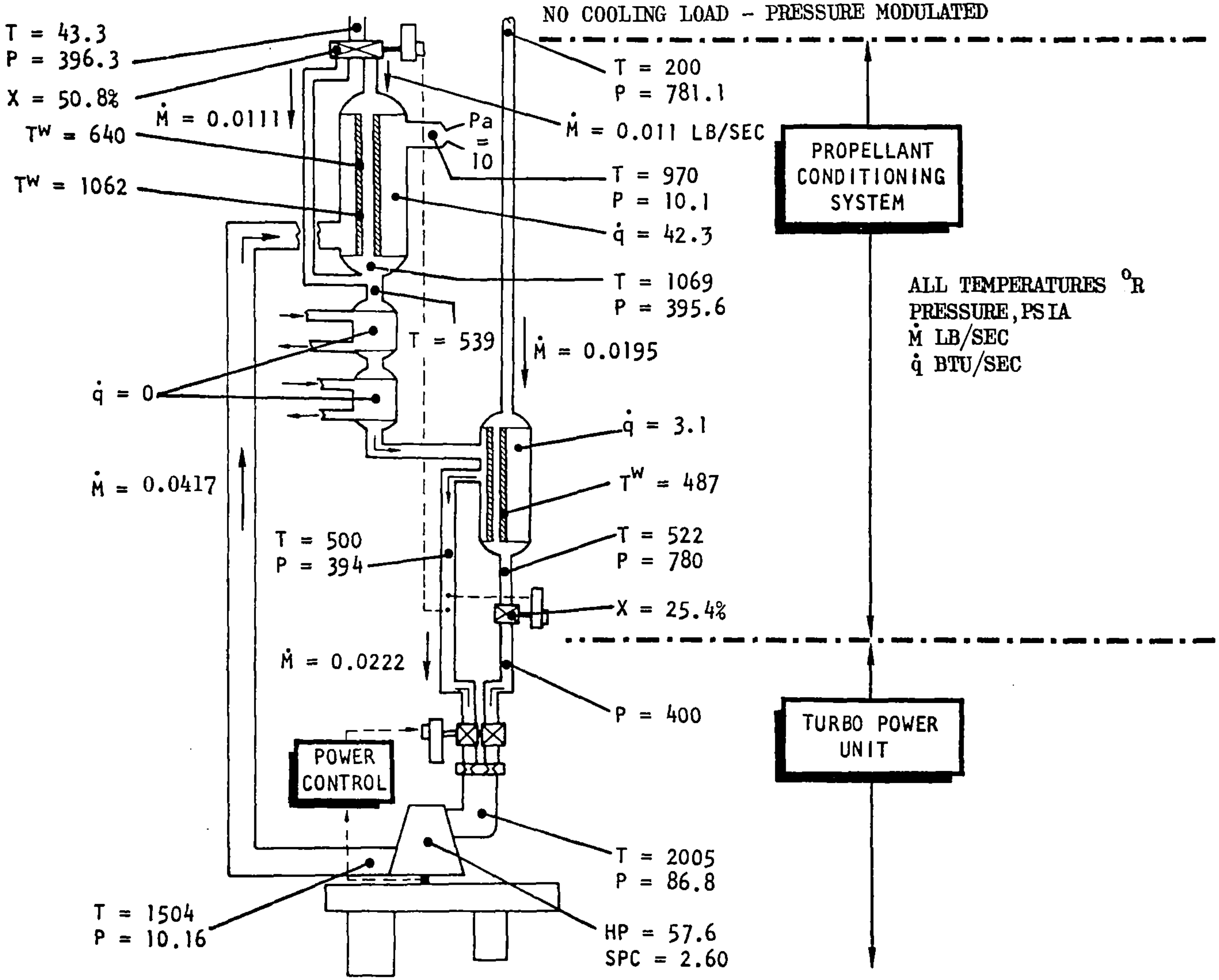


Figure 64

APU SYSTEM PERFORMANCE
25% POWER LEVEL

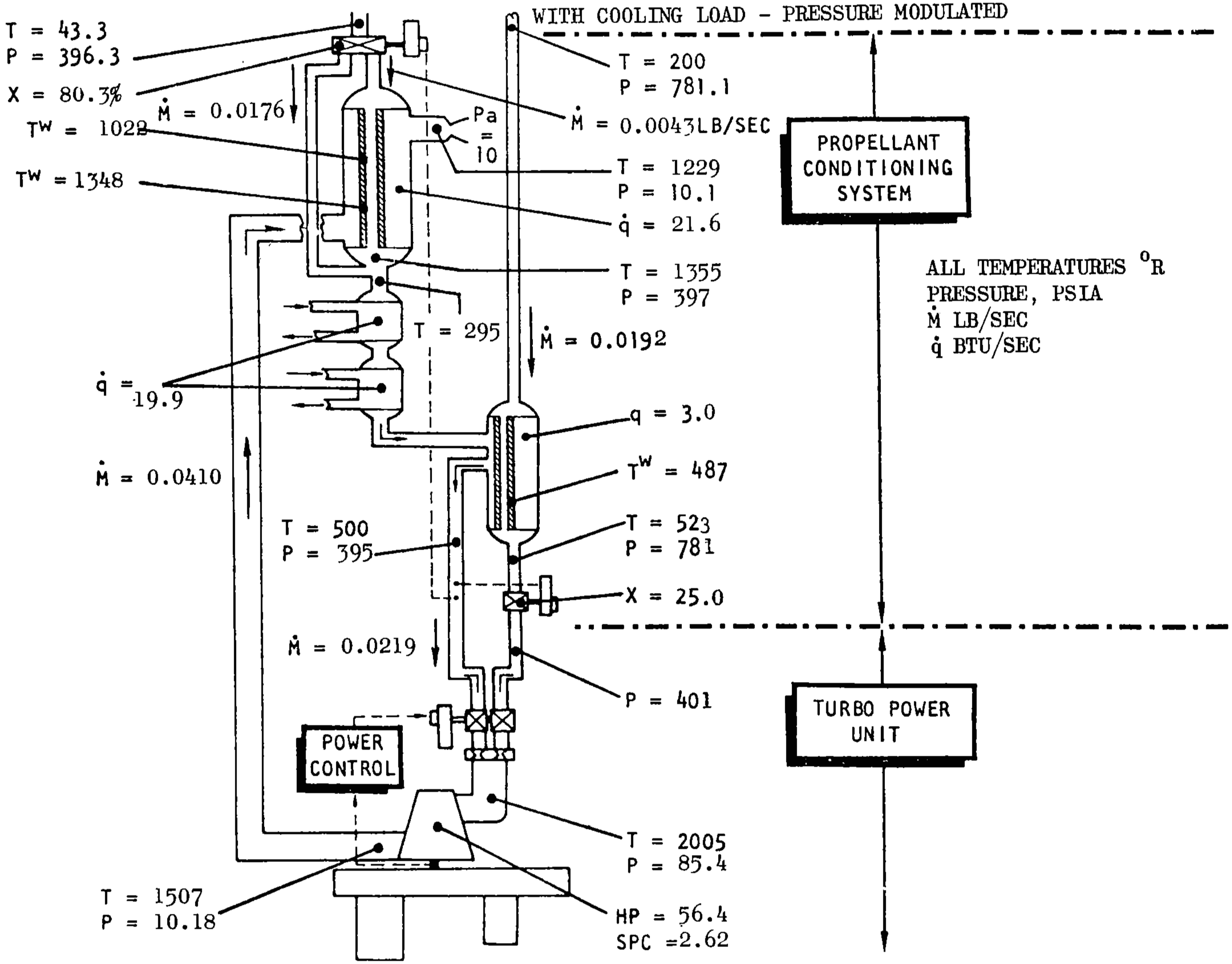


Figure 65

TURBINE PERFORMANCE

PRESSURE MODULATED

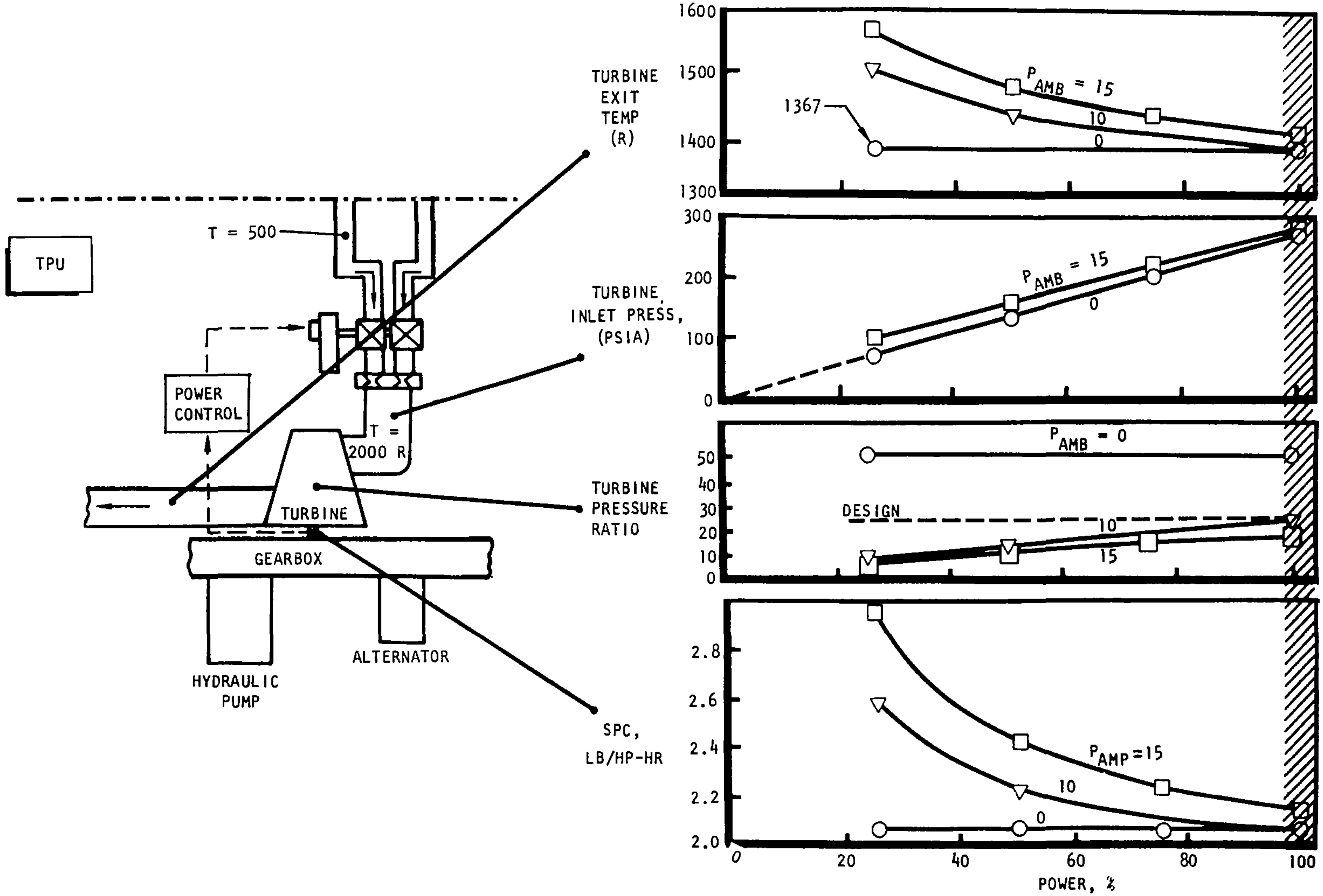


Figure 66

PROPELLANT CONDITIONING SYSTEM
TEMPERATURE EQUALIZER

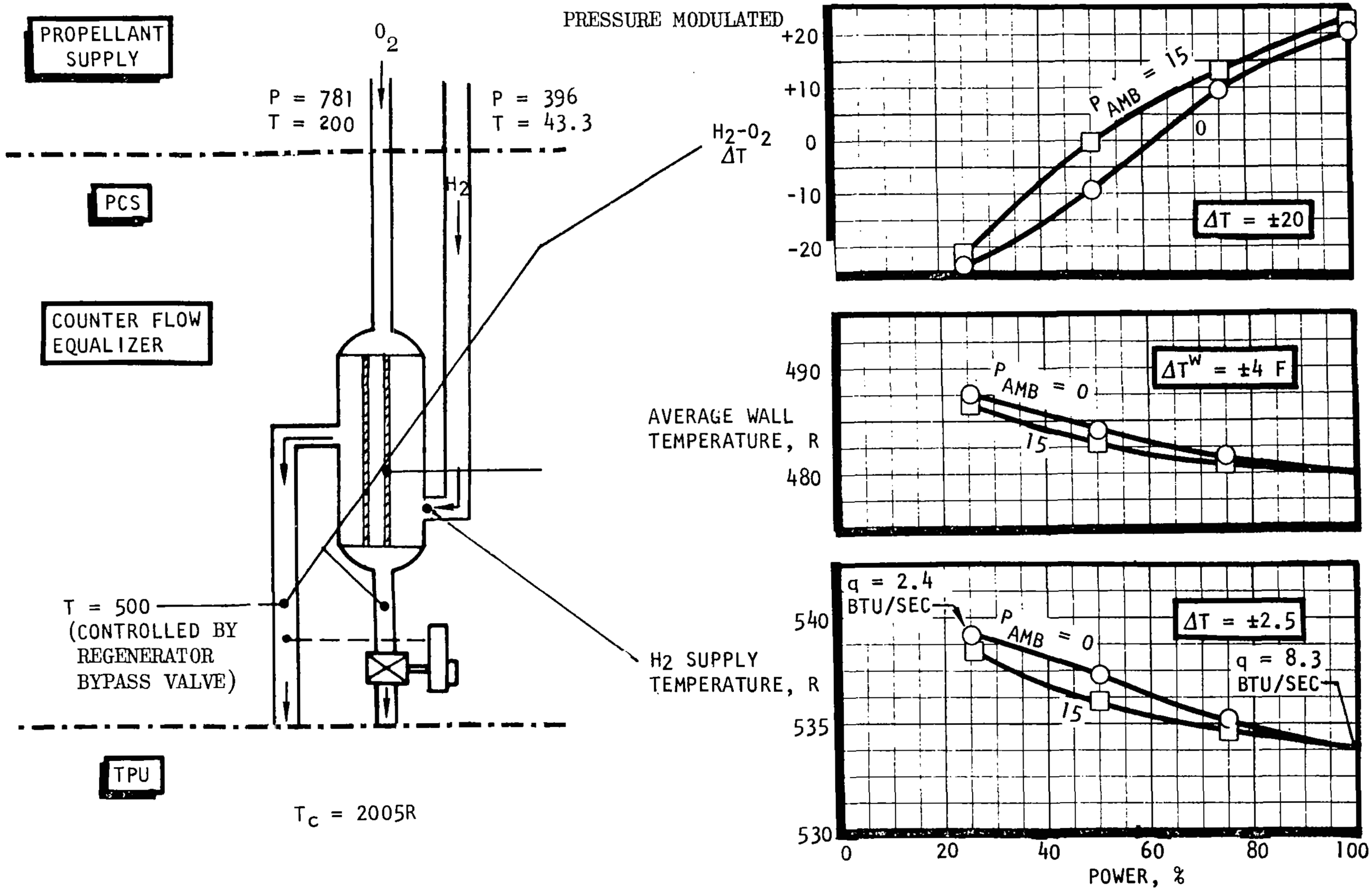


Figure 67

at the APU inlet by the regenerator by-pass control. The H_2/O_2 equalizer maintains an oxidizer-hydrogen temperature differential of approximately $\pm$21R between 100% and 25% power level. The hydrogen temperature drop across the equalizer is very small (a maximum of 39 R at 25% power and $P_a = 0$ psia) while providing an oxidizer temperature rise of 322 R.

Performance of the regenerator is described in Fig. 68 at 25% power level with hydraulic and lube oil cooling load varied from 0 to 40 hp. As the cooling load increases from zero to 28 hp, the fractional bypass flow increases from 51% to 81% in order to maintain a controlled 500 R TPU inlet temperature. Total heat flux in the regenerator reduces from 42 Btu/sec to 22 Btu/sec as a result of the downstream heat addition, and the mixed H_2 outlet temperature decreases from 540 R to 295 R. The hydrogen discharge temperature from the regenerator, prior to mixing is increased from 1070 R to 1350 R due to the severe reduction in throughflow. At zero cooling load the maximum TPU hydrogen inlet temperature achievable with zero regenerator bypass flow was 660 R. The heat flux to the equalizer under these conditions increases (to raise oxidizer temperature from 200 R to 695 R), resulting in a mixed regenerator outlet temperature of 712 R.

The regenerator design has a strong effect on freezeup in this unit. The results of the analog simulation study indicated that parallel flow should be used in the regenerator to ensure wall temperatures high enough to prevent freezing on the exhaust gas side of the heat exchanger. The wall temperature shown in Fig. 69 is that corresponding to maximum hydrogen regenerator flow (minimum bypass and most severe freezing condition) and it

EXTERNAL REGENERATOR CHARACTERISTICS

AS AFFECTED BY HYDRAULIC/OIL COOLING LOADS (25% TPU POWER)

(PRESSURE MODULATED POWER CONTROL)

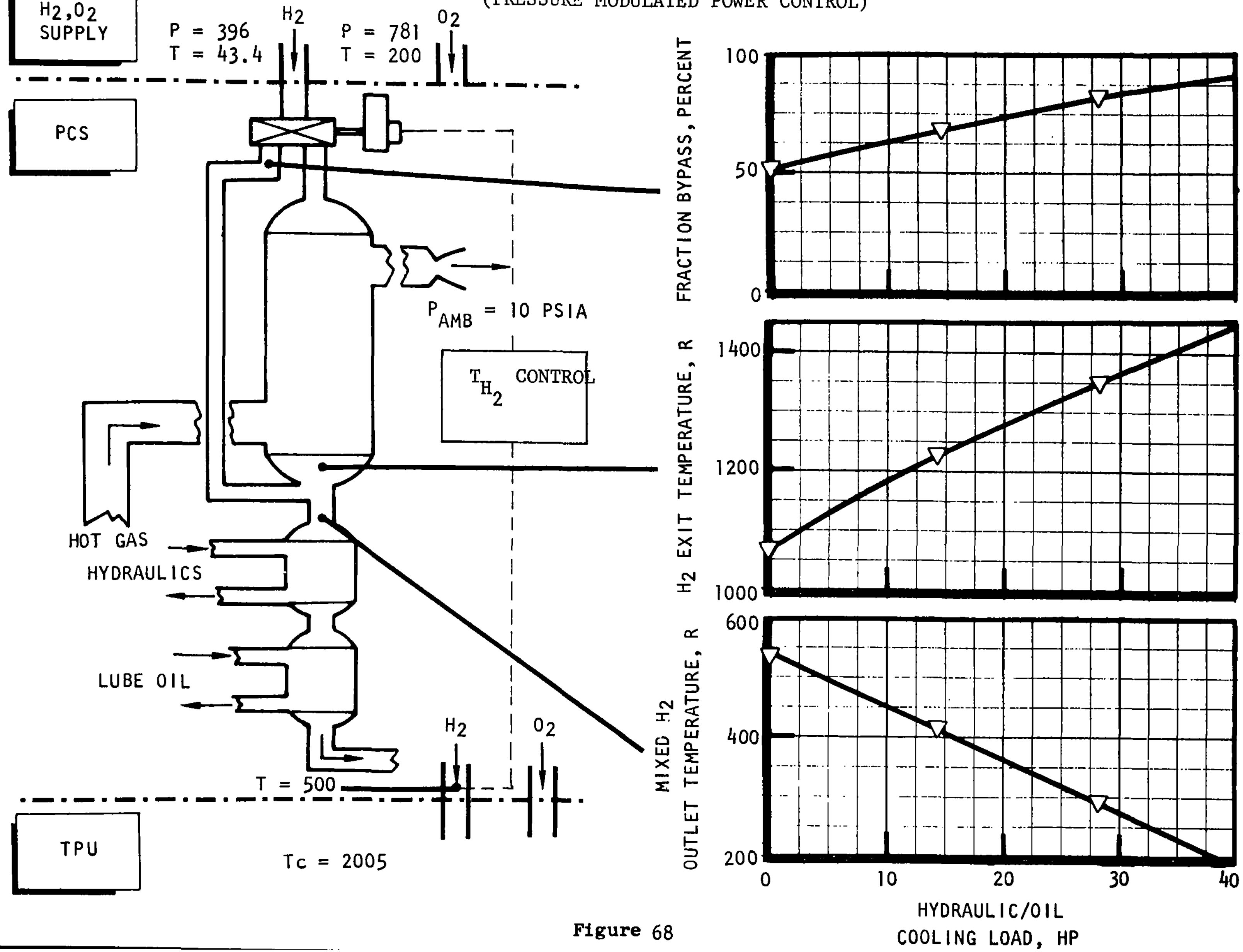


Figure 68

NON FREEZING REGENERATOR

PARALLEL FLOW

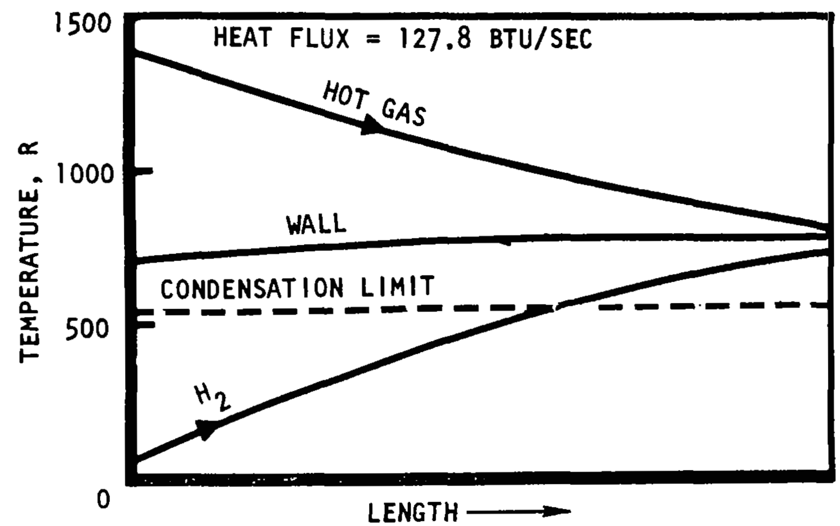


Fig. 69

is well above the condensation limit for the hot exhaust gas. As bypass flow is increased, the wall temperature rises because of lesser regeneration, and condensation will always be avoided. The analog results for counterflow are shown in Fig. 70 for the 25% power level and at zero and 28 hp cooling/lube oil cooling loads, and in Fig. 71 for the 100% power level and at zero and 56.4 hp cooling/lube oil cooling loads. In both cases, temperatures throughout the regenerator increase substantially with increased cooling load because of reduced throughflow. Due to the nature of the counterflow design, wall temperature can drop below the condensation limit. This is unacceptable because it can result in a freezing condition of the exhaust gas condensate on the tube walls. The use of a counterflow unit, therefore, requires a preheater located ahead of the regenerator to eliminate the freezing hazard. Use of a parallel-flow design is evidently to be preferred.

Closed Loop Turbine Inlet Temperature Control. As a result of varying requirements over the power altitude flight profile, the TPU inlet temperature of the oxidizer may vary by approximately 40 R. This results in a change in oxidizer flowrate due to a density variation. To compensate for the effect on mixture ratio, a closed-loop twin control of the differential pressure regulator may be used. This is accomplished by sensing turbine inlet temperature, and comparing the signal to a reference value. Any existing error is used to trim the differential pressure regulator varying oxidizer TPU inlet pressure and thereby eliminating the error. Trimming of the regulator could be mechanized by a variable bleed of the GH_2 reference pressure in order to effect a change in controlled oxidizer TPU inlet pressure.

INTERNAL REGENERATOR CHARACTERISTICS
AS AFFECTED BY HYDRAULIC OIL COOLING LOADS (25% TPU POWER)

(PRESSURE MODULATED POWER CONTROL)

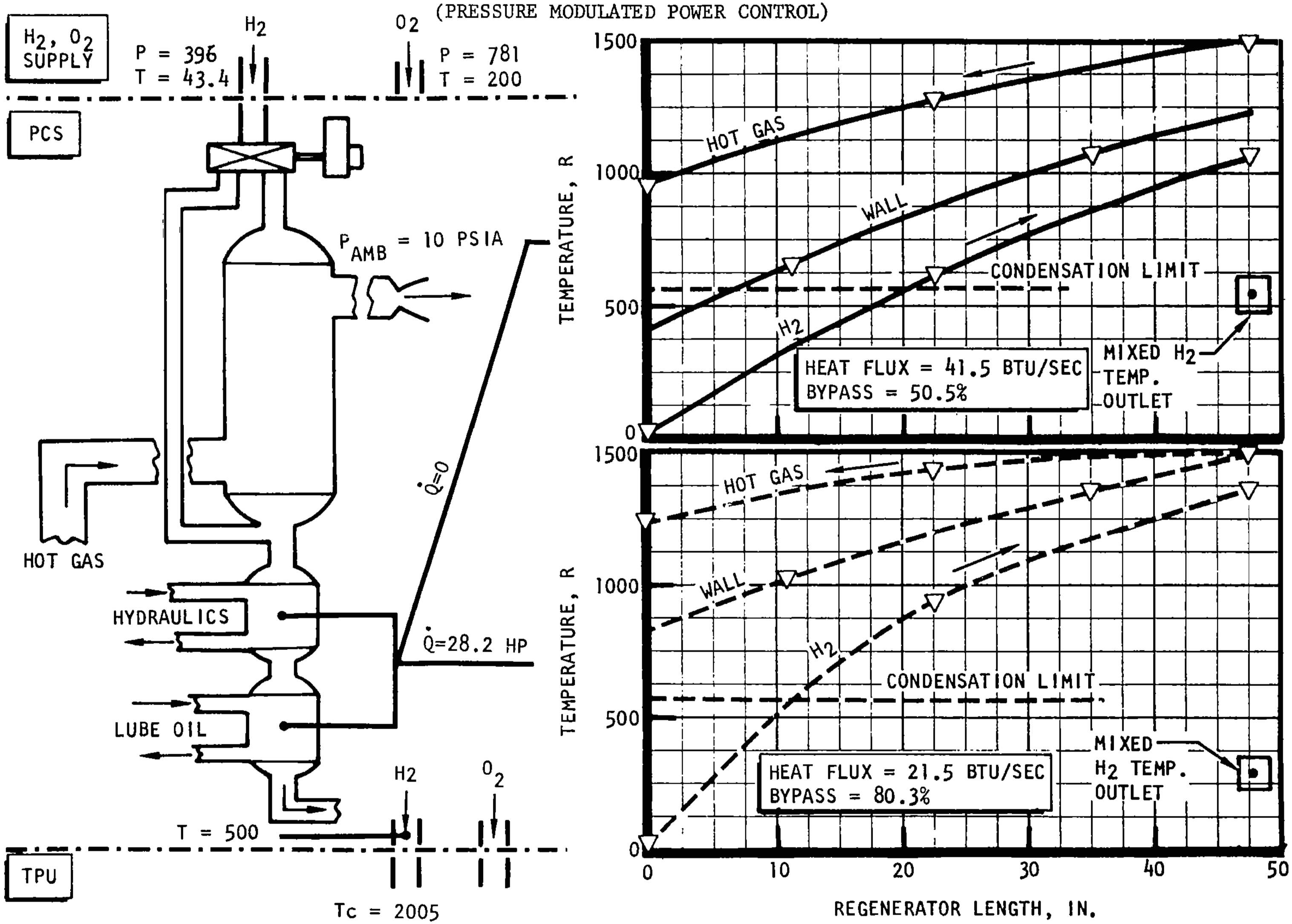


Figure 70

INTERNAL REGENERATOR CHARACTERISTICS
AS AFFECTED BY HYDRAULIC OIL COOLING LOADS (100% TPU POWER)
(PRESSURE MODULATED POWER CONTROL

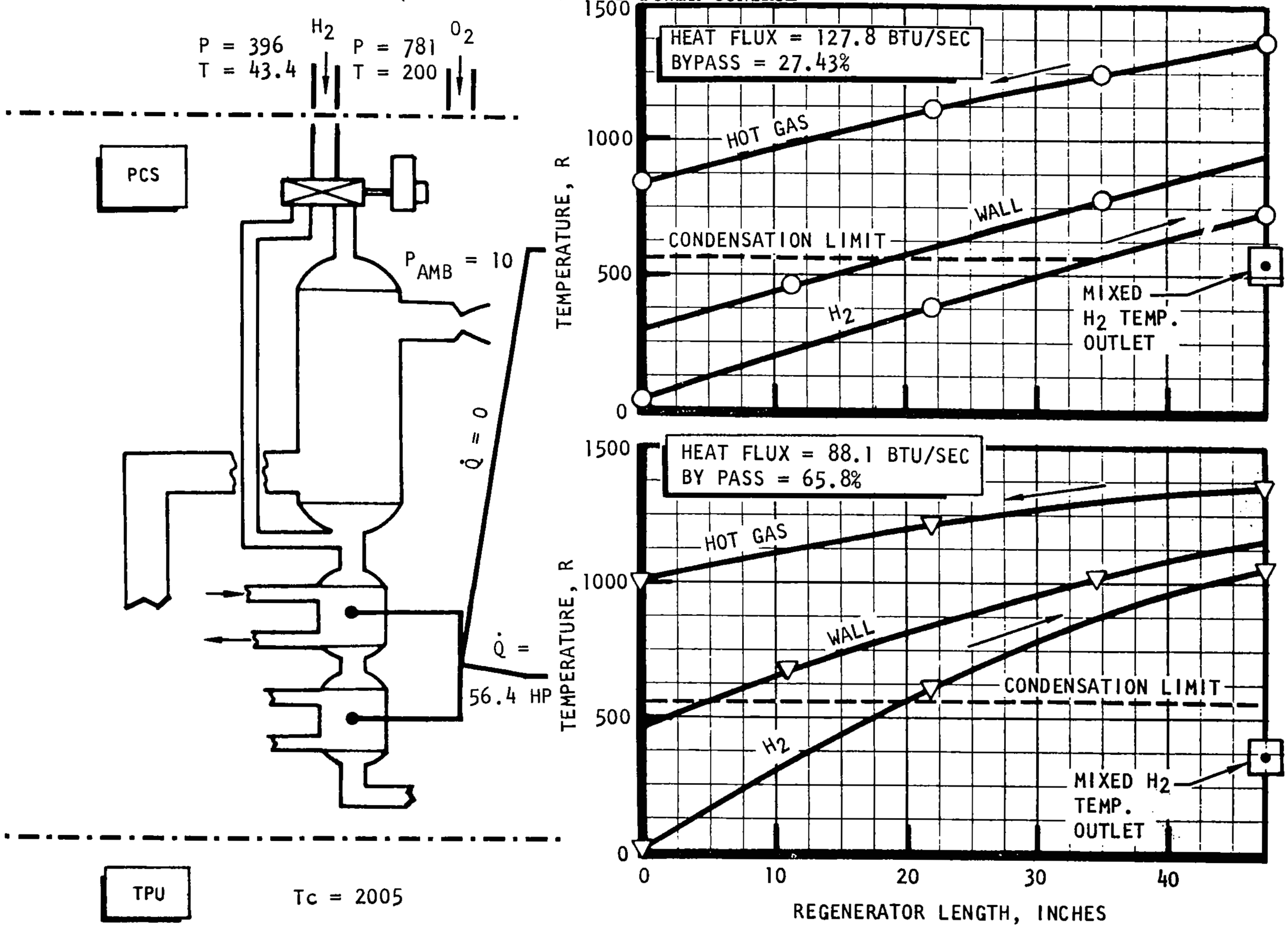


Figure 71

Steady-state performance of the TPU utilizing closed-loop turbine inlet temperature control is illustrated in Fig. 72. The required amount of trim control, i.e., the H_2/O_2 differential TPU inlet pressure has to be varied by 9 psi in order to affect the variation in oxidizer TPU inlet temperature, and maintain a constant turbine inlet temperature of 2005 R over the flight operational envelope. It is important to note here that a continuous closed loop turbine inlet temperature control is not necessary with this APU system. The 40 R variation in oxidizer TPU inlet temperature which occurs over the power profile is not severe. If a mixture ratio correction is required, however, it could be accomplished by utilizing the droop characteristic of the differential pressure regulator to lower H_2/O_2 differential TPU inlet pressure as a function of power level.

Effect of Oxidizer Supply Temperature Variation - A nominal APU oxidizer inlet temperature of 200 R results in a TPU inlet temperature of 522 R, or 22 R higher than the controlled hydrogen temperature of 500 R. This is due to the nature of the counterflow equalizer design. When oxidizer inlet temperature is increased to 500 R, the equalizer heat flux rate is reduced nearly to zero. This results in a reduction in hydrogen inlet temperature to the equalizer and a corresponding increase in regenerator by-pass flow, as illustrated in Fig. 73.

CLOSED LOOP INLET TEMPERATURE CONTROL

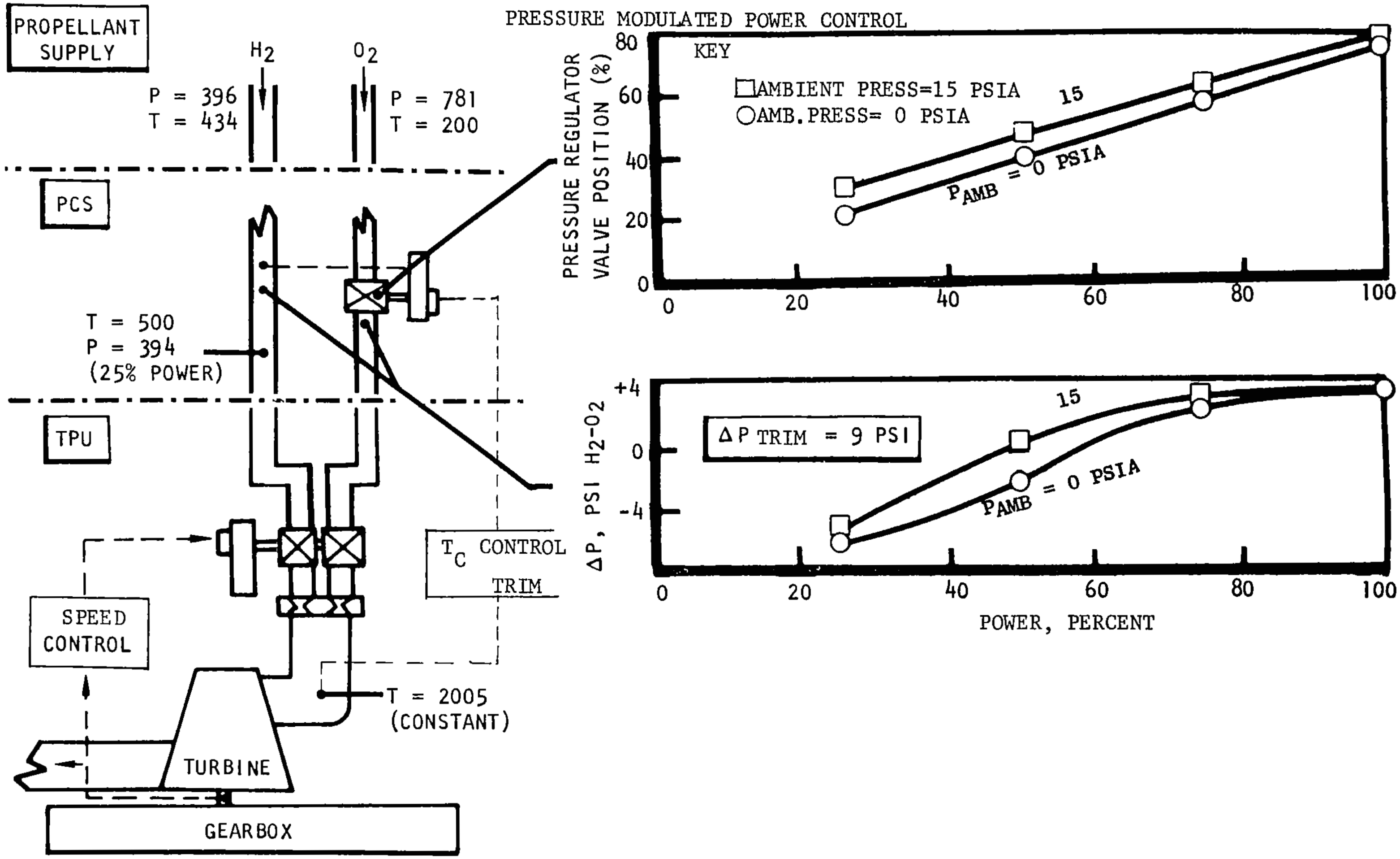


Figure 72

EFFECT OF APU O_2 SUPPLY TEMPERATURE

25 % TPU POWER

PRESSURE MODULATED POWER CONTROL

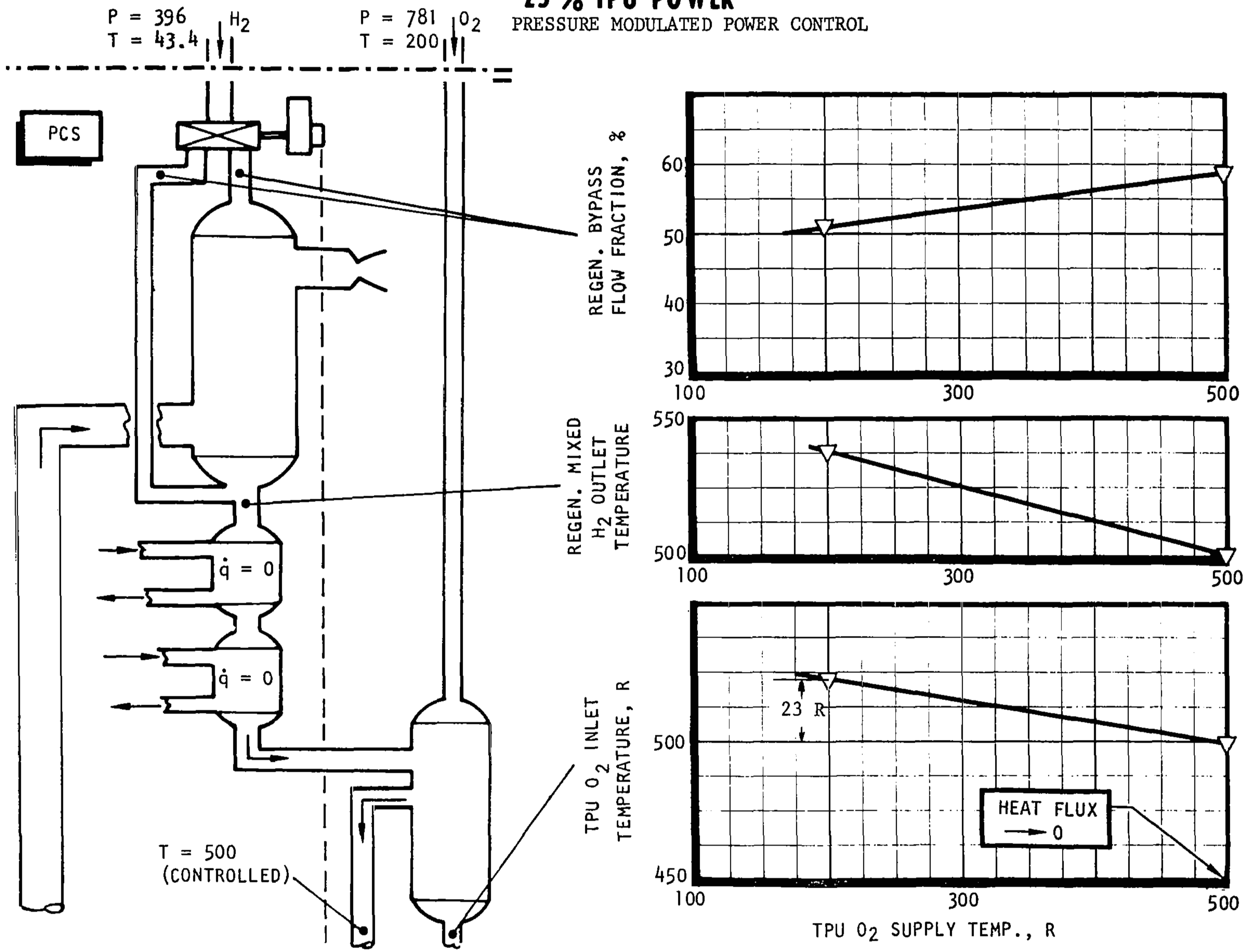


Figure 73

Sensitivity Analysis

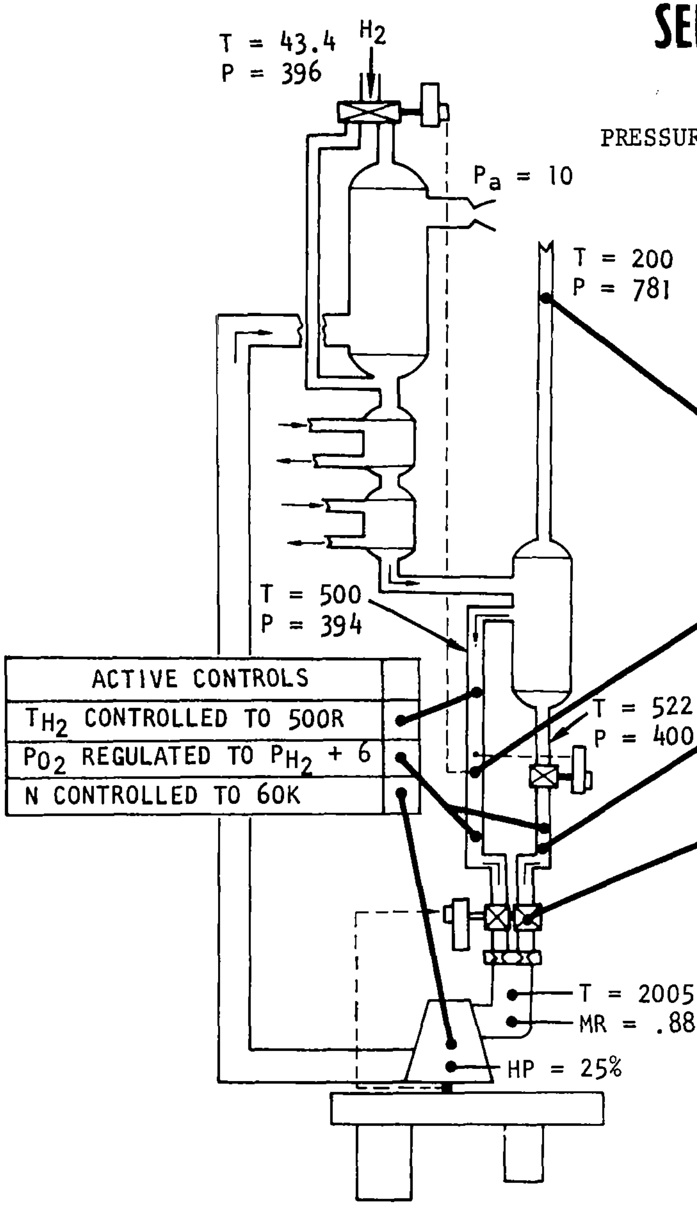
A steady-state sensitivity analysis was performed with the APU control system active in order to evaluate the sensitivity of mixture ratio and turbine inlet temperature to variations in propellant inlet conditions as well as to errors in control components. This analysis was performed with a pressure modulated power control and the results are shown in Fig. 74.

Differential Pressure Regulator Error. A steady state error in the differential pressure regulator was simulated on the analog model. The sensitivity of mixture ratio was approximately 1.0, i.e., a 1.0 percent change in TPU oxidizer inlet pressure resulted in a 1.0 percent change in mixture ratio. The resultant turbine inlet temperature sensitivity was 0.8. This sensitivity of mixture ratio to errors in H_2/O_2 differential pressure can be substantially reduced by incorporating choked nozzles just upstream of the combustor injectors. The effect of these choked nozzles is twofold: first it reduces the sensitivity of flowrate to pressure deviations because flow is proportional to TPU inlet pressure rather than a relatively small differential pressure between the TPU inlet and the combustion chamber; secondly, the nozzles tend to isolate the oxidizer and hydrogen propellants from one another, which reduces mixture ratio excursions.

Oxidizer Supply Temperature Variation. A 300 R increase in oxidizer supply temperature resulted in a 23 R reduction in TPU inlet temperature, and a corresponding increase in oxidizer flow rate producing a 2.3% mixture ratio increase and 1.1% increase in turbine inlet temperature. This characteristic

SENSITIVITY ANALYSIS
25% TPU POWER

PRESSURE MODULATED POWER CONTROL



	Percent Variation	
	MIXTURE RATIO	TURBINE INLET TEMPERATURE, R.
ΔT_o = +300R	+2.27	+1.12
ΔT_H = ±10%	~0	±2.2
ΔP_o = ±5%	±5.34	±3.99
ΔA_o = ±5%	±4.9	±3.62

EFFECT OF PROBABLE MAXIMUM CONTROL COMPONENT ERRORS

ΔA = +1% $+\Delta P$ = +2% $+\Delta T_H$ = +5%	+3.11	+3.43 (+68.6R)

Figure 74

is quite dependent upon the equalizer design. There is some evidence that a parallel flow equalizer would result in less sensitivity to APU oxidizer supply temperature variations.

Regenerator By-Pass Control Error. Simulating a regenerator by-pass control steady state drift, hydrogen TPU inlet temperature was varied $\pm 10\%$. Oxidizer inlet temperature varied virtually the same percentage resulting in a constant mixture ratio. Turbine inlet temperature rose 44 R due to the increased TPU inlet propellant temperatures.

TPU Throttle Valve Area Deviation. A 5% deviation in the TPU throttle valve areas was simulated, and resulted in a 5% error in mixture ratio, and a 73 R variation in turbine inlet temperature as shown in Fig. 74.

Combined Probable Maximum Control Errors. Probable errors were assigned to each of the control components in a manner which would be additive with respect to an increase in turbine inlet temperature. This resulted in a 3.1% mixture ratio increase and 69 R increase in turbine inlet temperature.

Dynamic Performance

Stability Characteristics. The variable displacement hydraulic pump with its associated control system, maintains a constant hydraulic discharge pressure (within its "droop characteristic") independent of small speed variations. When the resistive load is constant, a constant discharge pressure implies constant hydraulic power independent of small speed variations. This characterisitic is peculiar to the type of hydraulic pump control used here.

The torque load in the APU is then a hyperbolic function of speed as shown in Fig. 75. The delivered turbine torque is approximately linear with a negative slope in the design region. Under steady state conditions at the design speed, the torques are matched. Speed excursions around the design speed will occur. As indicated by Fig. 75, a small increase (decrease) in TPU speed would result in a continued increase (decrease) in speed due to the nature of the torque functions. This implies that loss of the closed loop speed control would result in a severe change in TPU speed, and would necessitate some emergency control action to prevent damage.

The system is therefore dependent, for stable operation on the speed control. Should the speed control become inoperative, a resulting overspeed or underspeed condition would signal an emergency shutdown of the APU.

Preliminary Control System Design. Standard synthesis techniques were employed to establish preliminary control design characteristics for the APU system. Control characteristics, as used with a pressure modulated power control system, are shown in Fig. 76. All three control loops are a "Type 1" or integrating control which implies a zero steady state error. The integrating rate of the pressure regulator is 2.5%/sec/psi; the regenerator bypass valve is 0.25%/sec/R and the power control valve is 0.004%/sec/rpm. Control characteristics as used with a pulse power control system are shown in Fig. 77. The regenerator bypass control is unchanged. The power control is an on-off type operated within a speed band of $\pm 4.0\%$; the pressure regulator integrating rate is reduced to 0.6%/sec/psi; and the regenerator bypass valve controller is unchanged.

STABILITY CHARACTERISTICS OF APU

PRESSURE MODULATED POWER CONTROL

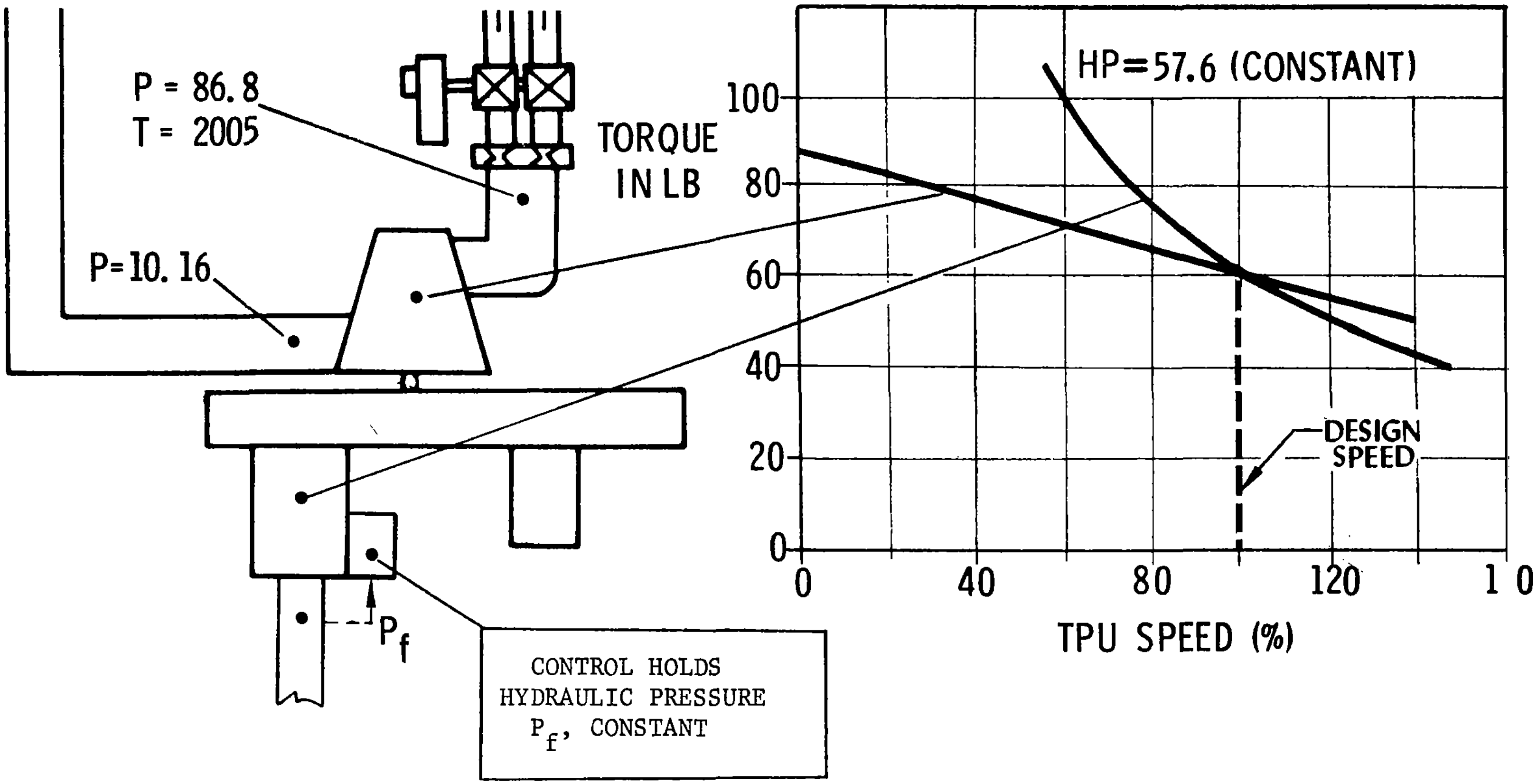


Figure 75

PRELIMINARY CONTROL SYSTEM DESIGN
PRESSURE MODULATED POWER CONTROL

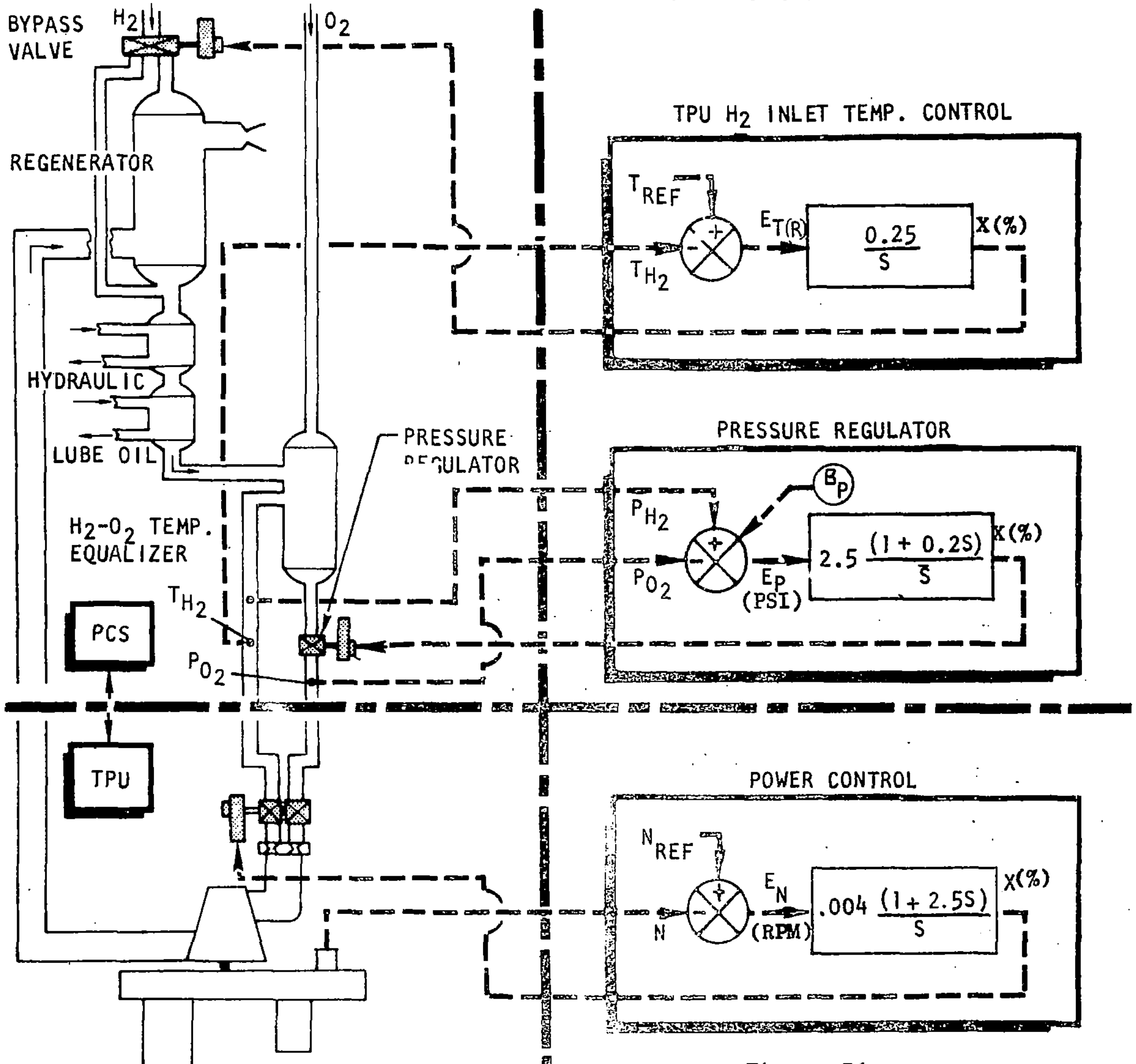


Figure 76

PRELIMINARY CONTROL SYSTEM DESIGN
PULSE POWER CONTROL

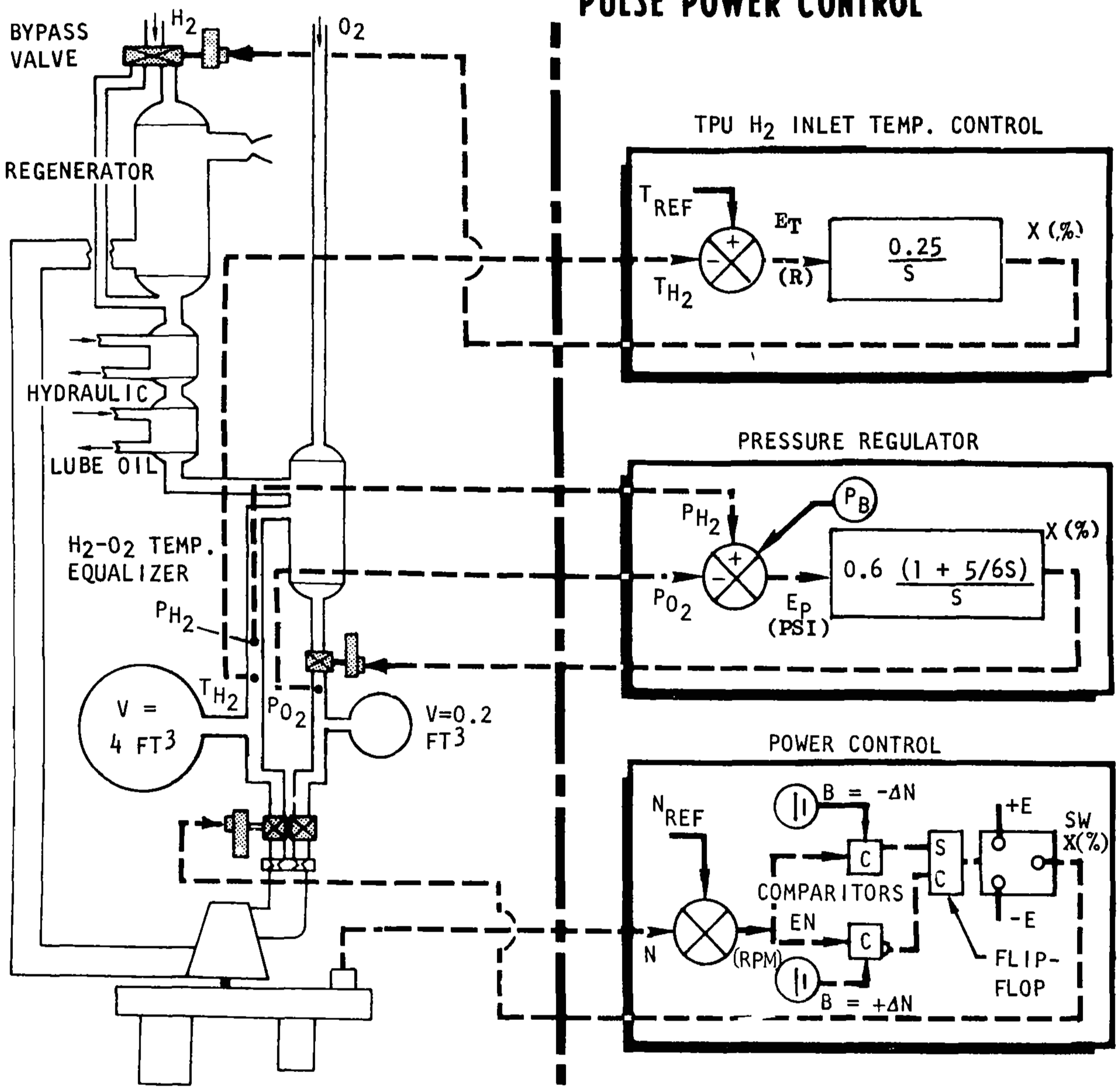


Figure 77

Transient Reponse of APU. The transient response of the APU system was evaluated through use of the analog model. The ability of the system to accommodate power demand steps, hydraulic and lube oil cooling load changes, and propellant inlet condition changes was investigated. Two power control concepts were evaluated: a pressure modulated control, and a pulse control.

PRESSURE MODULATED POWER CONTROL. The pressure modulated power control as shown in Fig. 76 throttles TPU inlet pressure to the turbine in order to match turbine power against the load and thereby maintain constant speed. This is accomplished by a closed loop speed control in which any deviation from a reference speed is converted to an error signal which passes through a controller whose output modulates a mechanically lined throttle valve.

Response of the system to a 2½ second duration power pulse is demonstrated in Fig. 78. Power demand was stepped up from 25 percent (56.4 hp) to 100 percent (225.5 hp) and after 2½ seconds stepped back down. The entire APU control system is shown in Fig. 76. The TPU moment of inertia was 0.0143 $lb\text{-}ft\text{-}sec^2$ which includes all rotating components of the TPU, referenced to turbine speed. The differential pressure regulator was set to provide a nominal turbine inlet temperature of 2005 R at 25% power level. This required an H_2/O_2 differential pressure setting of -7 psi, due to the deviation in TPU inlet propellant temperatures. Supercritical storage propellant inlet conditions to the APU were nominally 396 psia/43.4 R for the hydrogen and 781 psia/200 R for the oxidizer. Following the power step increase, TPU speed dropped 4.2% and did not completely recover to 60,000 rpm by the end of the step change. This is due to the short duration of the pulse compared

3 SECOND DURATION MAXIMUM POWER PULSE - PRESSURE MODULATED POWER CONTROL

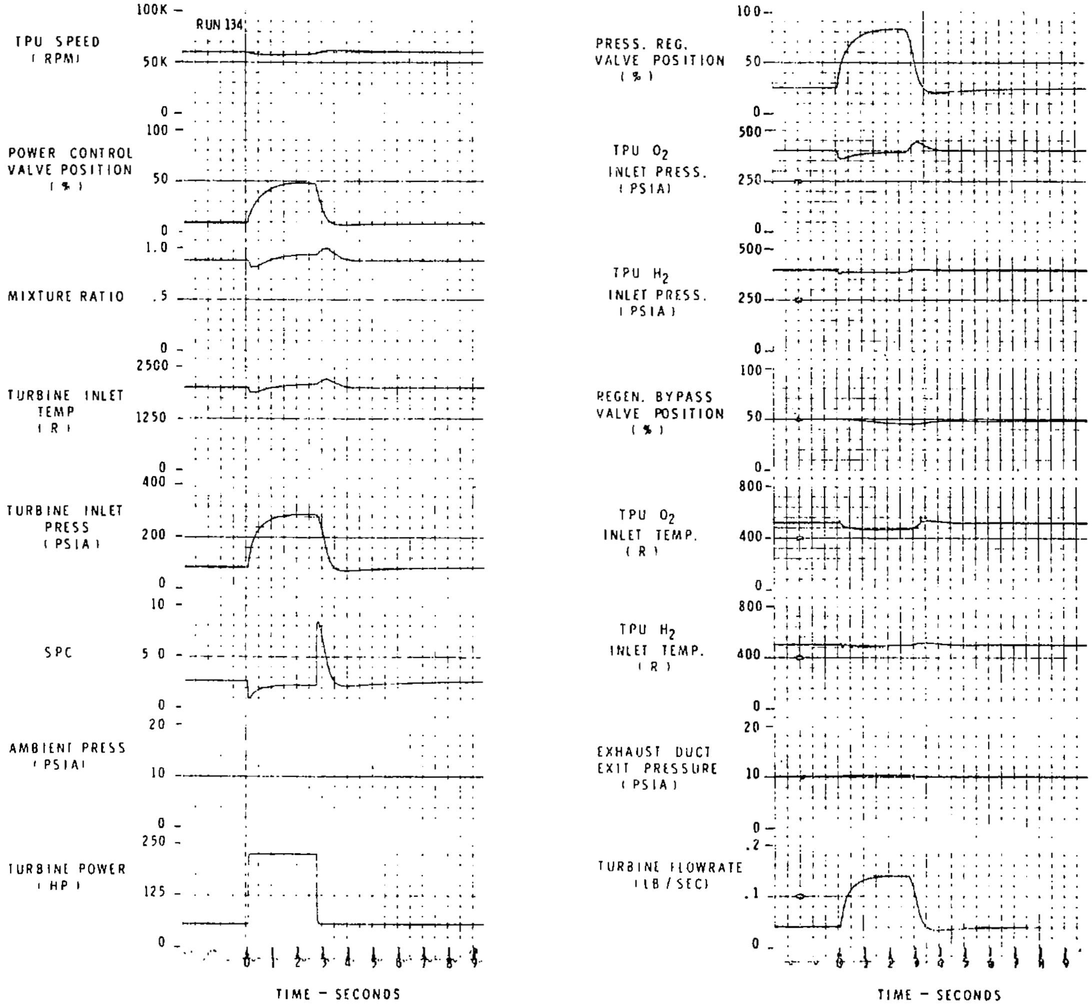


Figure 78

to the dynamics of the speed controller and the TPU inertia. The turbine throttle valve opened from approximately 9 to 48% to accommodate the power increase. The first order time constant of the system with respect to power demand is approximately 0.35 seconds, based upon the rise rate of turbine inlet pressure. Mixture ratio, following the step down to 25%, increased to a maximum of 1.0, resulting in a turbine inlet temperature rise of 220 R. The mixture ratio increase results from the relatively slow response of the differential pressure regulator, which allowed a 40 psi error in TPU oxidizer inlet pressure. At the high power level, TPU oxidizer inlet temperature reduces to a steady state level of approximately 475 R. The transient response of the system as described above is strongly dependent upon the control system design, Fig. 76. Both the turbine inlet temperature overshoot and transient speed error which resulted from the power pulse could probably be improved through optimization of the control design.

The effect of reduced TPU inertia was evaluated and is illustrated in Fig. 79. A 3 second duration power pulse was imposed on the TPU which had 25% of the inertia used for Run #134, Fig. 78. The speed error increased from 4.2 to 6.3%. This resulted in more rapid TPU throttle value actuation which, in turn, caused larger transient differential pressure errors at the TPU inlet. Hence mixture ratio and turbine inlet temperature errors were increased. As in the previous run, the transient response can be improved with control system optimization. However, the speed was able to recover to 60,000 rpm.

3 SECOND DURATION MAXIMUM POWER PULSE – PRESSURE MODULATED POWER CONTROL

(DECREASED TPU INERTIA)

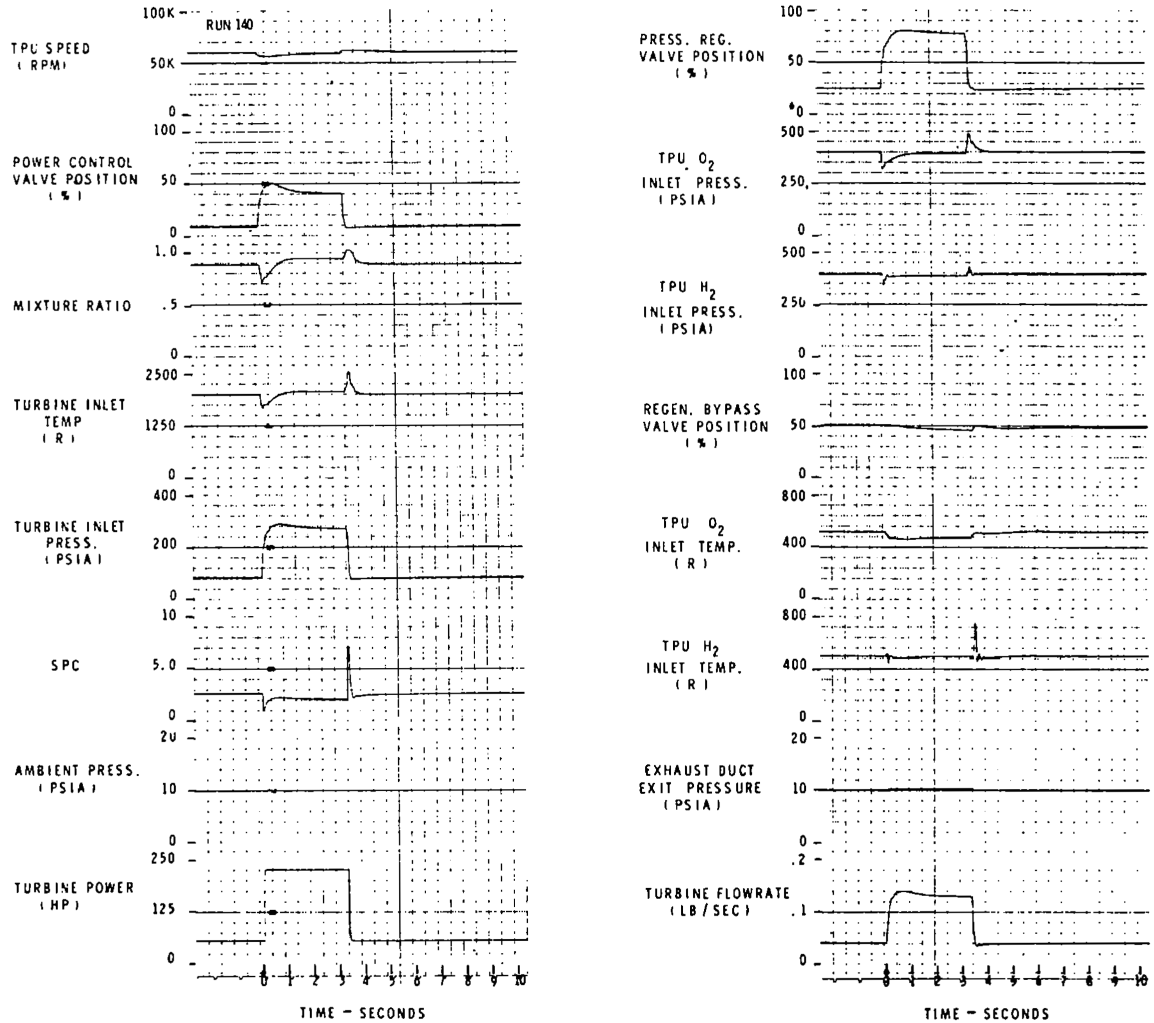


Figure 79

The effect of an increase in TPU inertia is illustrated in Fig. 80. The addition of a flywheel can be used to reduce both speed and turbine inlet temperature errors during a power pulse. Doubling the TPU inertia reduced the speed error to 3.3% and the temperature error to 155 R (compared with 220 R for Run #134).

The addition of various size accumulators in the oxidizer lines at the TPU inlet was also simulated in an attempt to minimize the oxidizer pressure spike and hence reduce mixture ratio error during the power transient. The additional fluid capacitance worsened the system response by extending the duration of the mixture ratio error following the power step, without reducing its amplitude.

A closed loop turbine inlet temperature control was simulated to evaluate operational characteristics during power transients. The control concept, shown in Fig. 81, incorporated a fixed bias to the differential pressure regulator with a gain of 0.5 psi oxidizer pressure reduction per degree increase in turbine inlet temperature. System response to an 8 second duration power pulse is illustrated in Fig. 82. Peak turbine inlet temperature excursion during the power step decrease was held to 45 R, compared with the 220 R temperature excursion shown in Fig. 78.

3 SECOND DURATION MAXIMUM POWER PULSE–PRESSURE MODULATED POWER CONTROL

(INCREASED TPU INERTIA)

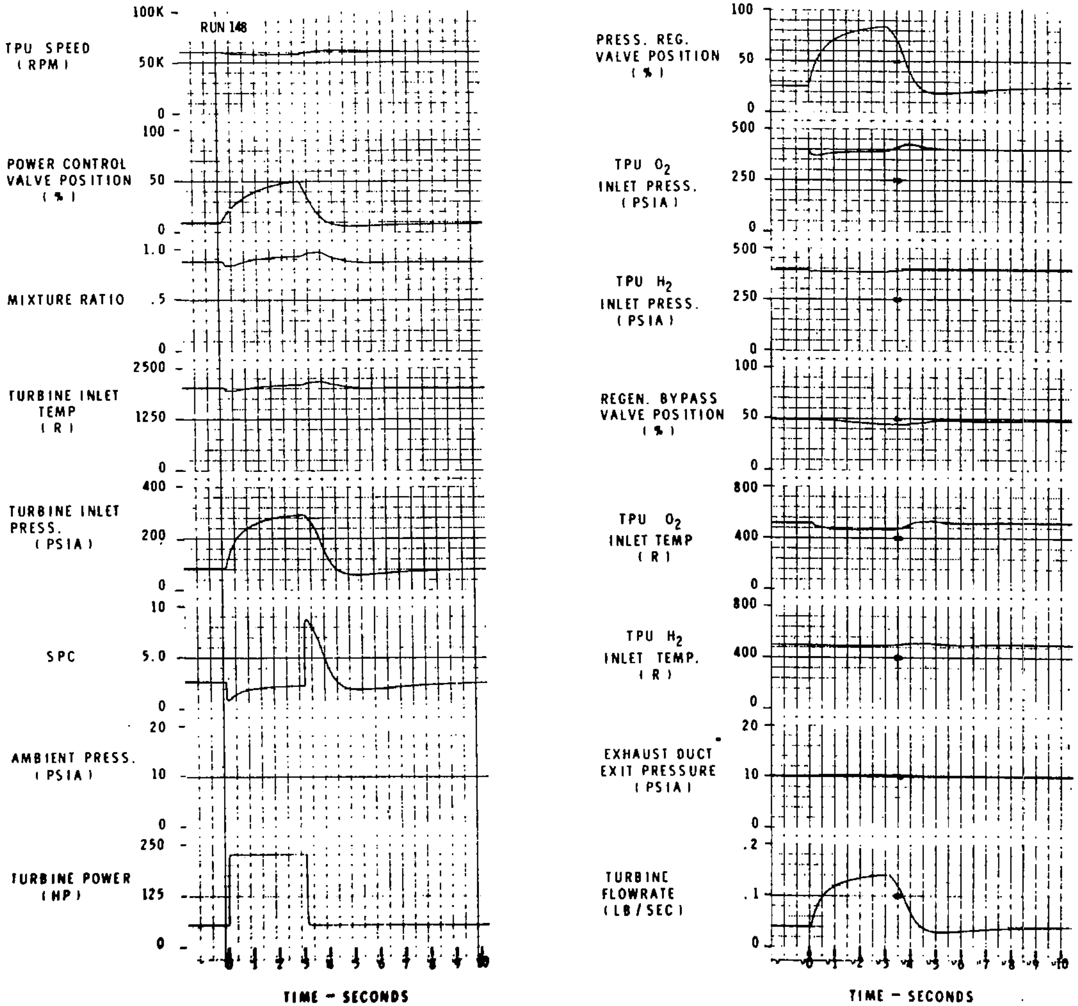
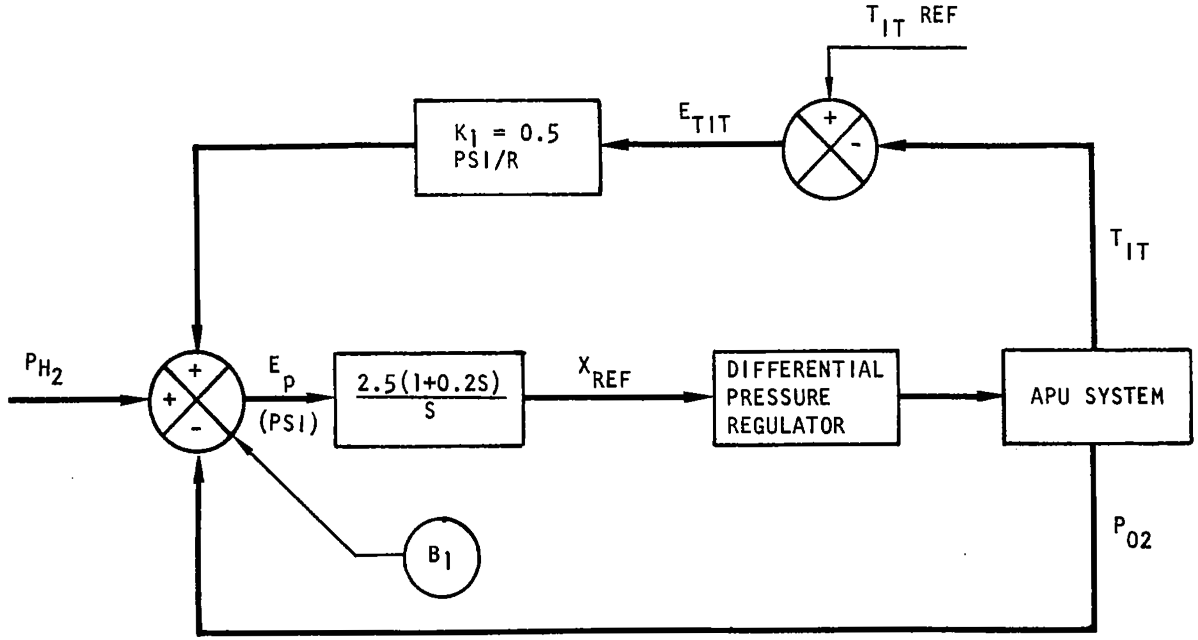


Figure 80

CLOSED LOOP TURBINE INLET TEMPERATURE CONTROL - BLOCK DIAGRAM



E_{TIT} = ERROR SIGNAL, TURBINE INLET TEMPERATURE
S = LAPLACE OPERATOR
E_p = ERROR SIGNAL, HYDROGEN PRESSURE
$X_{REF.}$ = ΔP REG. POSITION REFERENCE
P_{H_2} = TPU HYDROGEN INLET PRESSURE
P_{O_2} = TPU (REGULATED) OXYGEN INLET PRESSURE
B_1 = FIXED BIAS (7 PSI)
T_{IT} = TURBINE INLET TEMPERATURE

FIGURE 81

8 SECOND DURATION MAXIMUM POWER PULSE—PRESSURE MODULATED POWER CONTROL

(CLOSED LOOP T_T CONTROL)

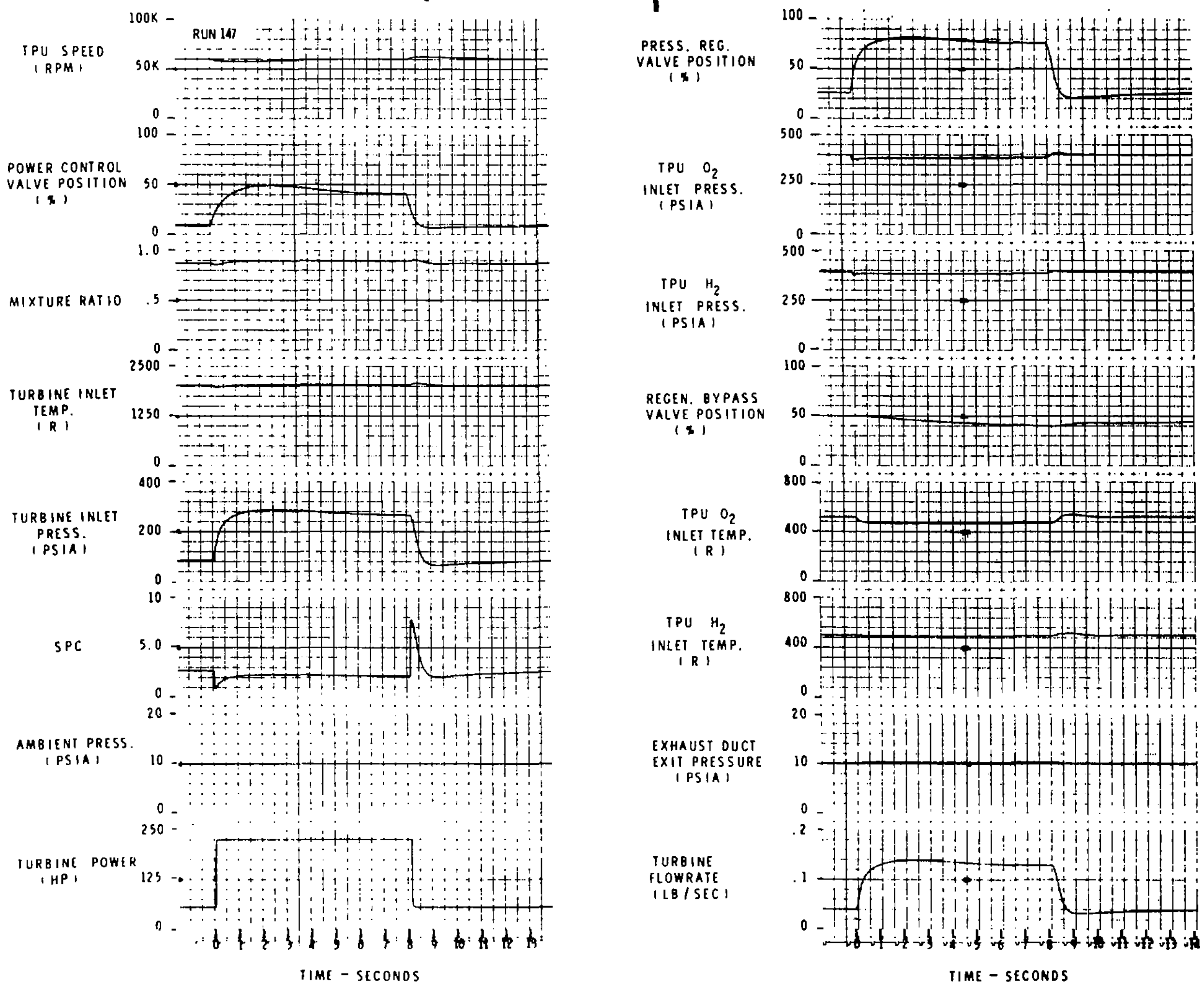


Figure 82

Response of the system to a rapid change in hydraulic/lube oil cooling load (as might occur following a startup) was also demonstrated with the analog model. A cooling load increase of approximately 6 hp/sec was introduced. The regenerator bypass valve opened from approximately 51 to 77% to accommodate the increased heat flux and there was virtually no error in either design speed, mixture ratio or turbine inlet temperature during the transient.

Response of the system to a simulated APU oxidizer supply pressure increase of 200 psi/sec resulted in a minor increase of 15 R in turbine inlet temperature, as the differential pressure regulator closed to maintain TPU oxidizer inlet pressure nearly constant during the transient.

PULSE POWER CONTROL. The pulse control, as shown in Fig. 77, pulses TPU inlet pressures to the turbine in order to provide turbine power at a fixed level and at varying duration, thereby maintaining speed within a predetermined band. This is accomplished by a closed loop speed control which signals an "on" pulse, opening the power control valve, when TPU speed reduces below the lower band limit; and signals an "off" pulse, closing the valve, when TPU speed increases to the upper band limit. Two electronic capacitors and a flip-flop are used to generate the control "on-off" logic. The valve is mechanically linked, providing a constant area ratio during the "on" pulse. The differential pressure regulator integrating rate was set at 0.6%/sec/psi for the pulse power control. An attempt was made to "match" the dynamics of the regulator and the size of the hydrogen and oxidizer accumulators to achieve nearly constant mixture ratio pulses at both 25 and 100% power level, while also accommodating maximum power step demands. The regenerator bypass control was the same as used for the pressure modulated system. A speed band of $\pm$4.0% was used for the on-off TPU throttle valve actuation. Supercritical storage propellant inlet conditions to the APU were 396 psia/43.4 R for the hydrogen and 781 psia/200 R for the oxidizer. TPU inertia was 0.0143 lb-ft-sec^2.

Dynamic characteristics of the system under constant power operation is demonstrated in Fig. 83 (Run #171). The differential pressure regulator was set to provide 2005 R turbine inlet temperature pulses during the 25% constant power operation. During a pulse, the turbine inlet temperature variation was 75 R which resulted primarily from a mixture ratio variation between 0.82 and 0.87. Matching of the hydrogen and oxidizer accumulator sizes achieves approximately equal amplitude oscillations of the TPU inlet propellant pressures so as to minimize the mixture ratio variation. The dynamics of the pressure regulator are set to minimize valve oscillations (8.0% variation during a cycle) and oxidizer flow oscillations in the equalizer. The spikes in hydrogen and oxidizer TPU flowrate at the start of an "on" pulse are due to the low initial chamber pressure--use of choking venturi flow passage in the valve body could be used to prevent these spikes if they are found to be detrimental.

CONSTANT POWER OPERATION, 25 PERCENT HORSEPOWER–PULSE CONTROL

(INTEGRAL-TYPE PRESSURE REGULATOR - 0.6%/SEC/PSI)

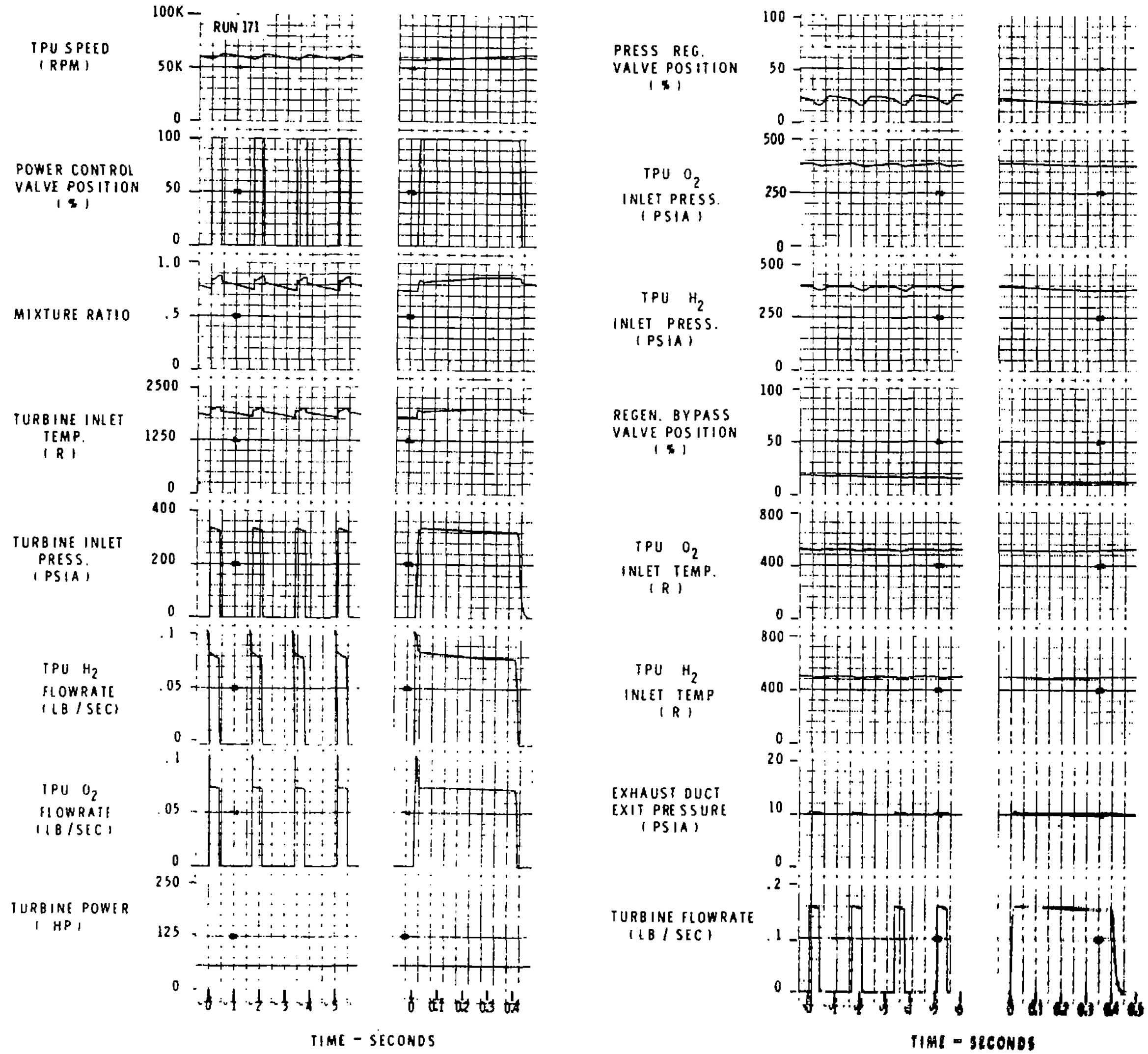


Figure 83

Hydrogen flowrate upstream of the TPU as well as gaseous flow in the exhaust duct pulsed in phase with the TPU throttle valve. The capacitance of the 5 ft long exhaust duct was not large enough to smooth out the flow.

When the system was subjected to changes in the power level from 25 to 100%, turbine inlet temperature excursions proved to be unacceptable. A modified regulator design was, therefore, incorporated in the analog model (which also was modified to simulate a stored gas supply of hydrogen and oxygen). The regulator is a conventional proportional gain type with a simple lag, i.e.:

$$X = \frac{K_g E_p}{1+TS}$$

where

X = Valve position (%)

E_p = H_2/O_2 pressure error

K_g = Regulator gain = 1.73%/psi

T = Lag time constant = 0.086/sec

S = Laplace transform operator

This regulator can be mechanized with a diaphragm, spring and snubber orifice to achieve the desired response lag.

The simulated stored gas supply system has nominal hydrogen inlet conditions of 400 psia regulated pressure at 100 R and oxidizer inlet conditions of 600 psia and 300 R. Accumulators were simulated at the TPU inlet of 4 ft^3* (18 lb) on the hydrogen and 0.25 ft^3 on the oxygen side. A 50 millisecond time lag was assumed for the hydraulic system with respect to power demand transients on the TPU.

* An accumulator volume of 1.0 ft^3 was included in system weight calculations to provide conditioned gas for in-flight restarts.

Response of this system to maximum power demand steps is illustrated in Figure 84. The variation in mixture during a pulse was extremely small: 0.895 to 0.91 at 25% power and 0.76 to 0.79 at 100% power level, with the exception of an initial 50 ms small amplitude spike which, as previously mentioned, can probably be eliminated through valve design. Turbine inlet temperature variation during an "on" pulse is also very small: 25 R at 25% power and 55 R 100% power level. Chamber pressure variation is 8 psi or 2.3% during a 25% power level, which results in a 230 R decrease in the nominal turbine inlet temperature. As shown in Fig. 84, the regulator opens from an average position of 14.0% to 40% when power steps up to the 100% level. Due to the regulator gain of 1.73% psi, this results in a 15 psi differential pressure shiftin the direction to reduce mixture ratio.

Response of the system to a rapid change in oxidizer supply pressure of 200 psi/sec was evaluated. At 25% power level, a 60 R increase in turbine inlet temperature accompanied an oxidizer supply pressure increase. This resulted from closure of the regulator and the associated slight increase in TPU oxygen inlet pressure. At 100% power level, the condition was magnified, however, due to the 230 R steady state downward shift in turbine inlet temperature, no over-temperature condition resulted.

The extent of the downward shift in turbine inlet temperature can be modified by adjustment of pressure regulator gain and possibly modification of the equalizer design. The temperature decrease is inversely proportional to flowrate and results in desirable operational characteristics. The performance penalty associated with the decrease in turbine inlet temperature is minimal and represents a small fraction of the operating time.

POWER DEMAND STEPS PULSE CONTROL
(PROPORTIONAL TYPE PRESSURE REGULATOR)

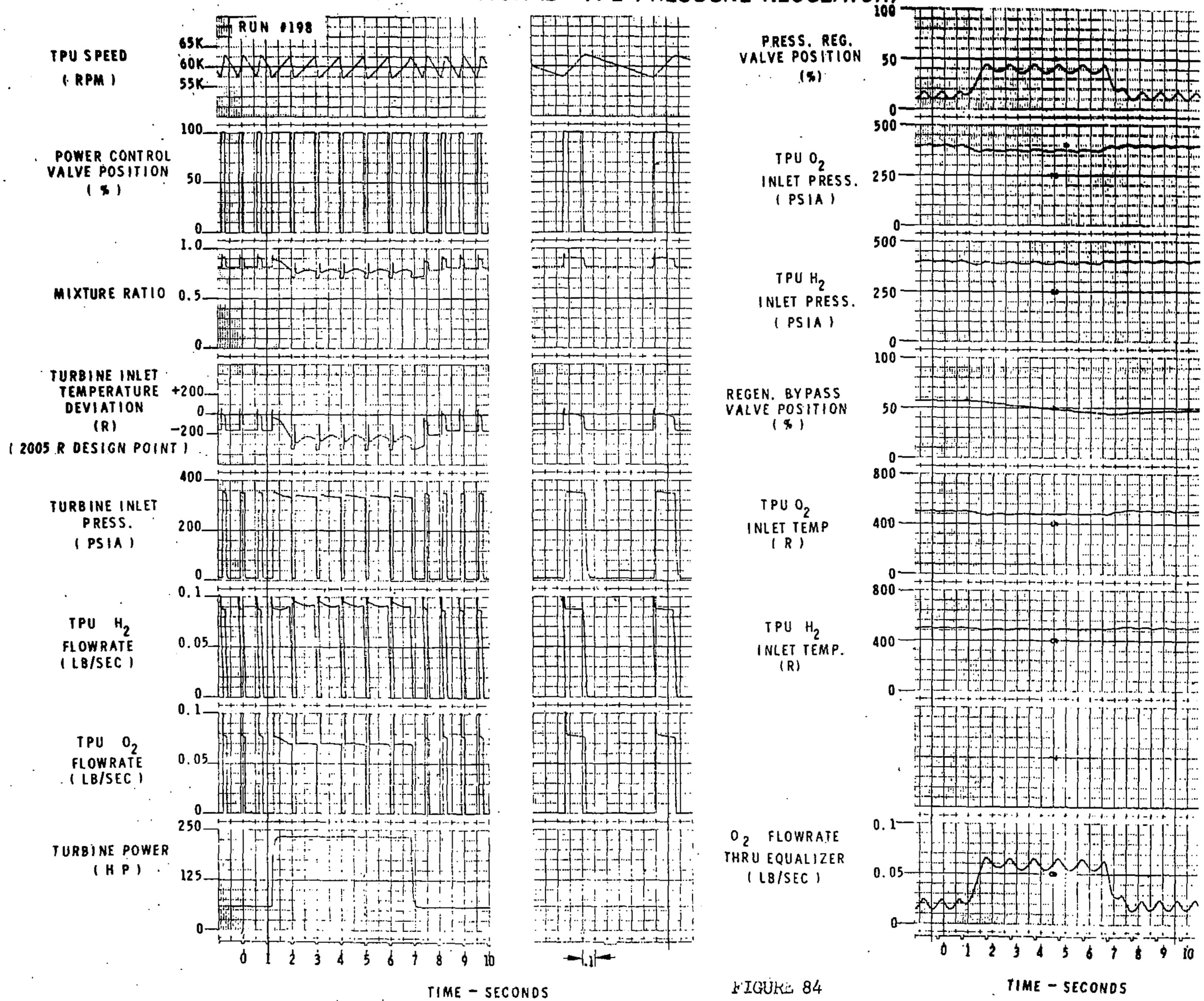


FIGURE 84

In summary, satisfactory APU performance has been achieved with a pulse power control to the extent investigated on this study.

Hybrid Power Control

The pulse power control system maintains a constant inlet temperature high pressure ratio condition across the turbine at all power demand levels. Power demands are matched by control of the pulse width, rather than through modulation of turbine inlet pressure. This operating characteristic minimizes specific propellant consumption, but has some disadvantages:

1. A large number of thermal cycles is required by the combustor and turbine assembly

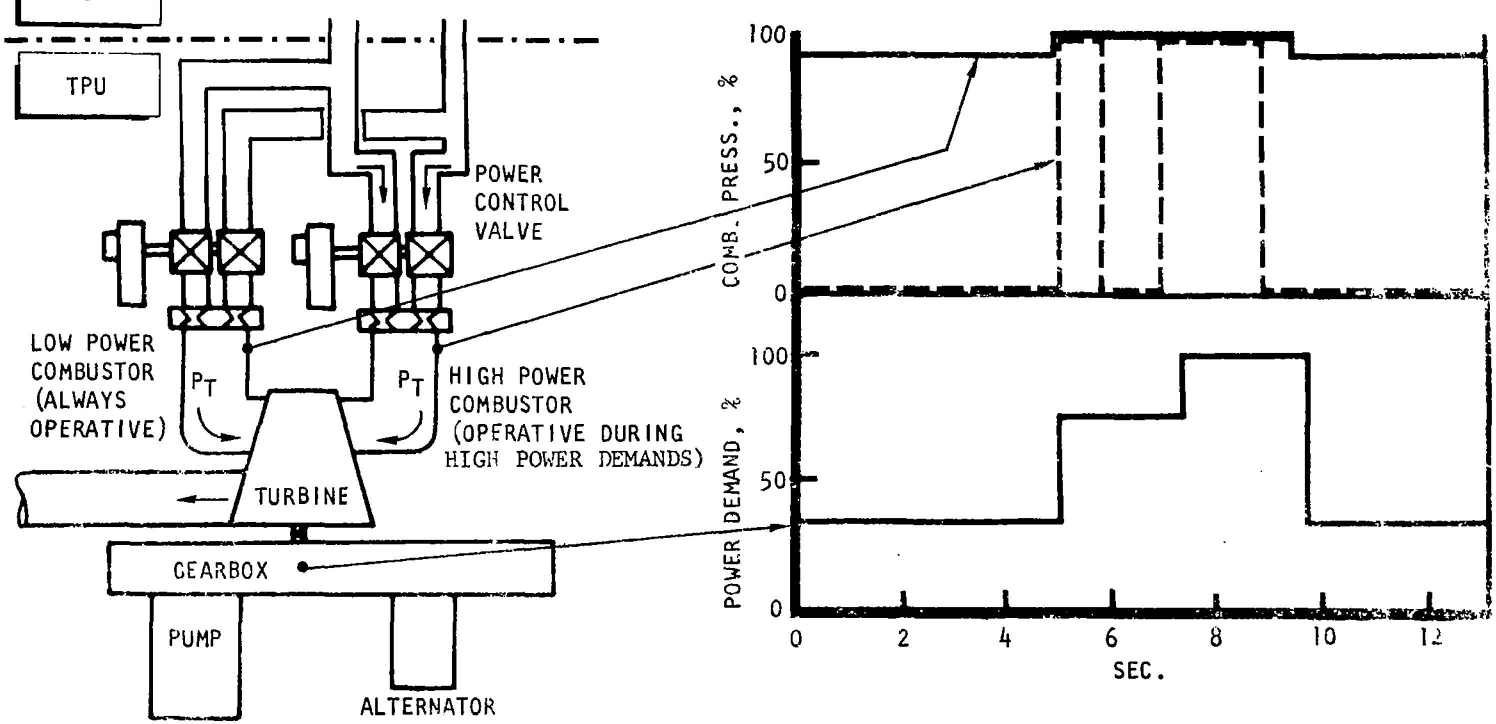
2. The amplitude and rate of change of alternator frequency may be objectionable to the vehicle electrical system

3. The reliable life of a high-response, bipropellant turbine on-off throttle valve must be demonstrated

4. The pulsing nature of hydrogen flow in the propellant conditioning system and gaseous flow in the exhaust duct may introduce undesirable vibrations throughout the system.

The hybrid system was conceived in an attempt to approach the specific propellant consumption of a pulse system with a minimum number of pulses or combustor startup. This type of control scheme is illustrated in Fig. 85. The propellant conditioning system supplies propellant at the controlled pressure and temperature level to two combustor and valve assemblies on the TPU. Each combustor supplies a separate set of nozzles on the partial admission turbine assembly. The low power combustor (sustainer) is always operative and modulates turbine inlet pressure to accommodate power demands from its maximum power capability down to the minimum required power level of the system. The maximum power capability of the sustainer combustor is sized based on the system power profile. A tradeoff exists to size the sustainer combustor large enough to minimize startups of the high power combustor, but small enough to prevent severe throttling penalties. When power demands exceed the sustainer combustor capability, the large combustor is fired and, operating on either a pulse or pressure modulated mode, accommodates the high power demands. To achieve added reliability, the large combustor would be designed to develop maximum power without the aid of the sustaining combustor.

HYBRID POWER CONTROL



PRO

- PERFORMANCE APPROXIMATES IDEAL PULSE POWER CONTROL
- SUBSTANTIALLY REDUCES COMBUSTOR ASSEMBLY CYCLE LIFE REQUIREMENTS

CON

- INCREASED DESIGN COMPLEXITY
- INCREASED CONTROL SYSTEM COMPLEXITY

Figure 85

APU STORABLE PROPELLANT STUDY

OBJECTIVE

The objective of this study was to perform a tradeoff analysis to permit comparison to be made between a hydrogen-oxygen auxiliary propulsion unit (H_2/O_2 APU) and an APU which would operate with storable propellants. In the tradeoff study, the following factors were used in the evaluation:

- Reliability
- Minimum Development Cost
- Minimum Development Risk
- Minimized Weight
- State-of-the-art Materials
- Existing Technology
- Long Life

The major considerations were

- Propellant Selection (monopropellant Vs bipropellant)
- Ignition/Dissociation Control
- Gas Temperature
- Carbon Formation
- Freezing Point
- Performance

Optimization was done considering

- Type of fuel
- Peak and part-load performance

Fuel weight

Dry weight

Tank weight

Safety

Reliability

Life

Development problems

As a result of a comprehensive study, a baseline system was evolved. Figure 86 shows the results, with numbers applied to the booster. The selection of this system will be discussed in the following sections.

PROPELLANT SELECTION

The program plan makes the choice of the propellant to be used one of the program objectives. Both monopropellant and bipropellant storables are to be considered. Figure 87 lists the considerations, limitations, and design factors which must be taken into account.

Because one of the fundamental areas of this program was to determine the trade-off between development cost/risk and performance penalty, the most developed propellants were considered. The evaluation was based on results of a comprehensive study for rocket propulsion previously performed at Rocketdyne. Various propellants were reviewed with respect to performance and applicability.

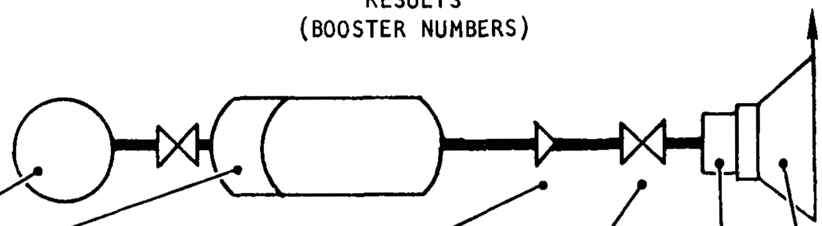
The group of propellants most developed and applicable to the APU are hydrazine and its blends. As shown in Fig. 88 , hydrazine is highly developed and shows a low development risk. However, it freezes at 34.8F and will not be usable to -65F as required unless special provisions are made. The

STORABLE PROPELLANT STUDIES

OBJECTIVES
- RELIABILITY
- MINIMUM DEVELOPMENT COST
- MINIMUM DEVELOPMENT RISK
- MINIMIZED WEIGHT
- STATE-OF-ART MATERIALS
- EXISTING TECHNOLOGY
- LONG LIFE

CONSIDERATION
- PROPELLANT SELECTION
- IGNITION/DISSOCIATION CONTROL
- GAS TEMPERATURE
- CARBON FORMATION
- FREEZING POINT
- PERFORMANCE

RESULTS
(BOOSTER NUMBERS)



TANKAGE
- TITANIUM TANK FOR PRESSURANT
  - 4500 PSI
  - He
  - 6.0 POUNDS
- ALUMINUM TANK FOR HYDRAZINE
  - MINIMUM WALL
  - 11 POUNDS
  - RUBBER BLADDER
  - LOW PRESSURE (<150 PSIA)

PUMP
- GEARBOX MOUNTED
  - MK 36 CENTRIFUGAL
  - 80,000 RPM

VALVE
- PRESSURE MODULATION

TURBINE
- TWO-STAGE, PRESSURE COMPOUND
  - 1700 F GAS
  - 60,000 RPM
  - 1550 FT/SEC TIP SPEED
  - EFFICIENCY = 67%

GAS GENERATOR
- ELECTRICALLY HEATED THERMAL BED
- DEVELOPED SYSTEM
- NO CONTAMINATION PROBLEM
- SLOWER INITIAL RESPONSE
- 1700 F - REQUIRES 60% DISSOCIATION

PROPELLANT
- HYDRAZINE
  - MOST DEVELOPED
  - NO CARBON IN EXHAUST
  - LOWER DISSOCIATION REQUIRED
  - LOWER PERFORMANCE THAN BLEND
    (10-15% ≈ 100 POUNDS)
    SPC = 5.46 LB/HP-HR
  - REQUIRES HEATING (~30W IN ORBIT)

Figure 86

STORABLE PROPELLANT STUDY

MONOPROPELLANTS

BIPROPELLANTS

CONSIDERATIONS

- RELIABILITY
- MINIMUM DEVELOPMENT COST
- MINIMUM DEVELOPMENT RISK
- MINIMIZED WEIGHT
- STATE-OF-ART MATERIALS
- EXISTING TECHNOLOGY
- 1000-HOUR LIFE

LIMITATIONS

- GAS TEMPERATURE
- CARBON FORMATION
- FREEZING POINT
- PRESSURE AT 200F
- IGNITABILITY
- SAFETY

DESIGN FACTORS

• TWO-STAGE TURBINE	
• TEMPERATURE - GAS	1700F
WHEEL	1150F
• OVERTEMPERATURE MARGIN	150F
• UDIMET 700 MATERIAL	

Figure 87

MONOPROPELLANTS

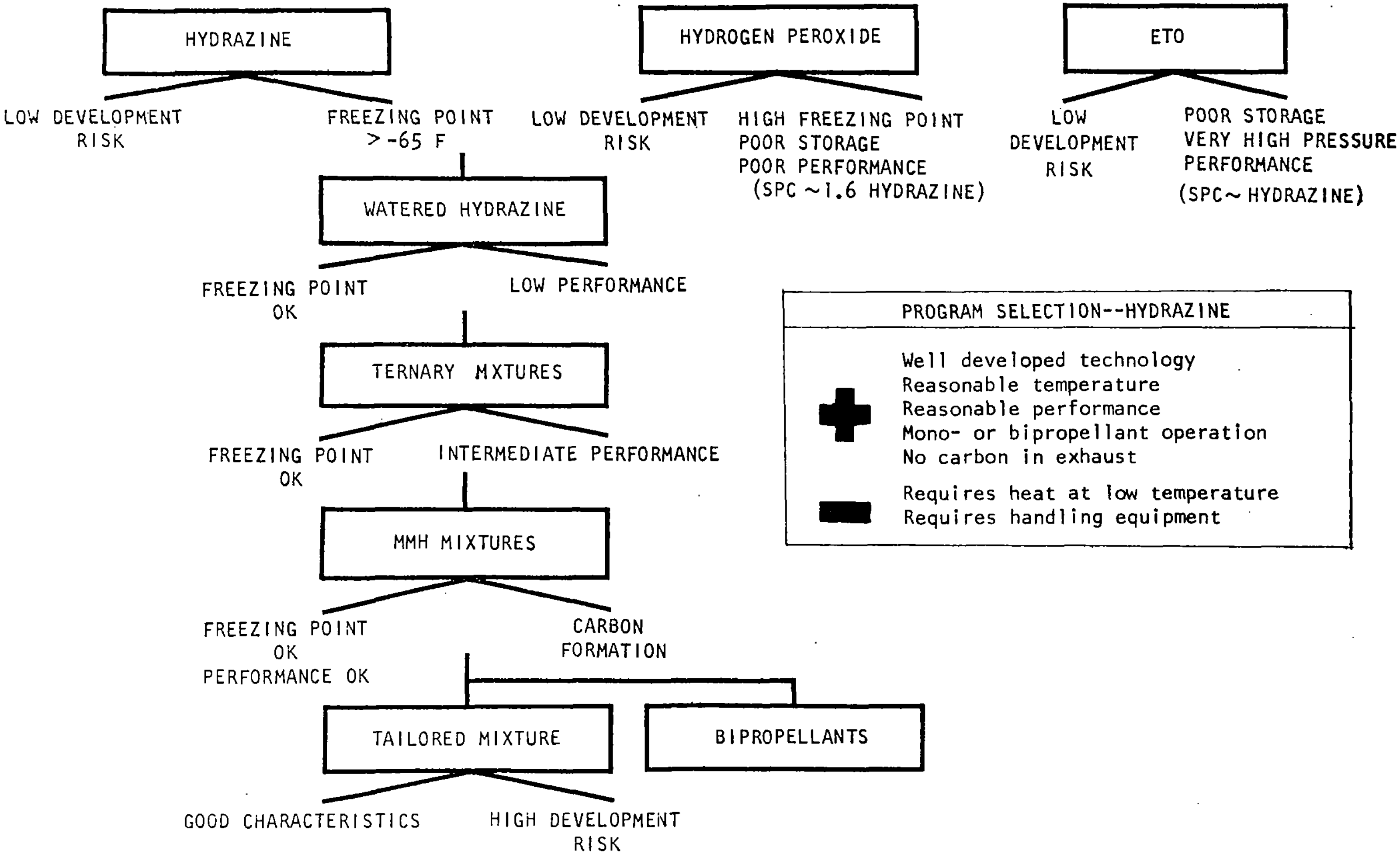


Figure 88

freezing point can be depressed to the desired level by the addition of water. This however lowers performance considerably. Performance can be raised (maintaining low freezing point) by using one of the ternary mixtures (hydrazine, hydrazine nitrate, water) or by using a mixture with MMH. The MMH mixtures (e.g., MHF-3), contain carbon which results in the probability that fouling will occur in the turbine passages due to carbon in the gases (Table 4). It is, of course, possible to consider making a special propellant mixture, tailored to this application. This would provide the required characteristics but would result in an increased development risk.

Hydrazine can also be used as a fuel in a bipropellant system. This does not overcome the freezing point limitation nor does it simplify the system. A survey of other possible bipropellant combinations reveals that none (save hydrogen and oxygen) appear to offer any significant advantages considering the added complexity of two feed systems.

Monopropellants other than the hydrazine blends were also considered. The most likely candidates are hydrogen peroxide and ETO. Both are well developed and present low development risk. However, hydrogen peroxide also requires provision to depress the freezing point to the desired level and has considerably lower performance than hydrazine. ETO has poor performance and its vapor pressure at 180 F is approximately 150 psia which would require heavy tankage.

Propellant selection was, therefore, made from among the hydrazine blends and hydrazine. Figure 89 compares the operation and performance of some

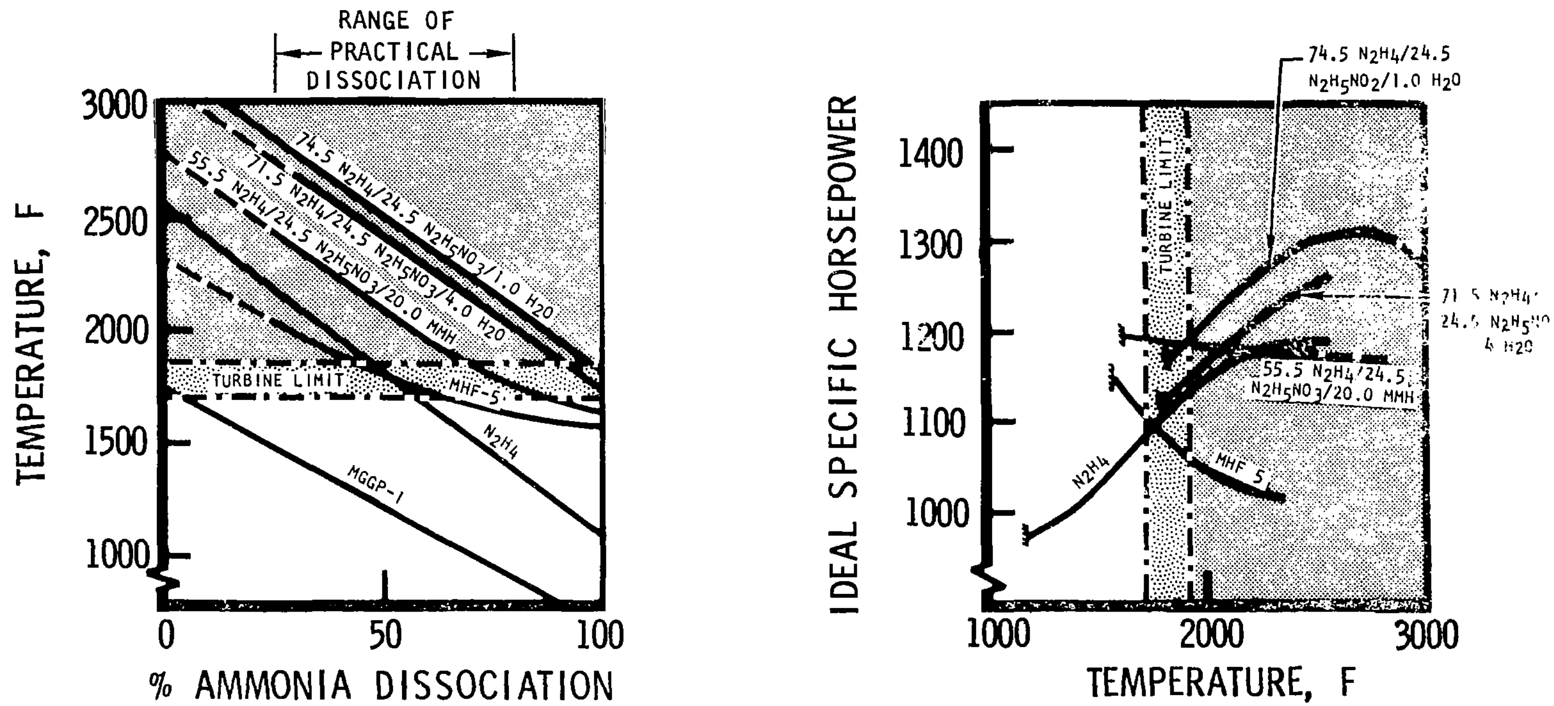
PRODUCTS OF COMBUSTION

	Pc	Tc	PRODUCTS OF COMBUSTION (MOLE/100 GM)*								
	PSIA	^oF	SOLID C	H_2O	NH_3	CH_4	H_2	N_2	CO	CO_2	C_2H_6
MHF-5	1000	1760		0.290	0.014	0.897	3.597	2.298	0.280	0.015	
MHF-3	1000	1700	0.271	-	0.015	1.595	3.263	2.296	-	-	-
98% N_2H_4/2% H_2O	1000	1200	-	0.111	0.242	-	5.768	2.936	-	-	-
50% N_2H_4/50% UDMH	1000	1700	0.103	-	0.015	1.561	3.303	2.384	-	-	-
MMH	150	1450	0.995	-	0.006	1.180	4.150	2.170	-	-	-
	1000	1685	0.517	-	0.015	1.653	3.182	2.163	-	-	-
UDMH	1000	1600	1.379	-	0.014	1.946	2.738	1.657	-	-	0.001
N_2H_4	600	1130	-	-	0.140	-	6.032	3.051	-	-	-
		1475	-	-	1.071	-	4.635	2.585	-	-	-
		1835	-	-	2.088	-	3.109	2.077	-	-	-

* Theoretical Chemical Equilibrium

Table 4

SYSTEM E - STORABLE PROPELLANT SELECTION OF PROPELLANT



- BLENDS GIVE 10-15% BETTER PERFORMANCE
- MMH BLENDS (e.g. MHF-3) CAUSE CARBON FORMATION--POOR LIFE
- BLENDS REQUIRE MORE DISSOCIATION FOR 1700 F
- BLENDS HAVE LOW FREEZING POINT
- ANY CONCLUSION BASED ON HYDRAZINE WILL BE VALID FOR BLENDS WITH MODIFICATION
- HYDRAZINE WELL ESTABLISHED TECHNOLOGY

FIGURE 89

representative propellants. One of the major considerations in a turbine system is the gas temperature. To maintain stress levels at an acceptable level, the gas temperature should be limited to approximately 1700 F. It is readily observed that the hydrazine blends (except MHF-5) result in higher temperature even at 100 percent ammonia dissociation. The low-temperature blend MGGP-1 has relatively poor performance. As shown in the figure, the performance of MHF-5 is only slightly better than that of pure hydrazine at 1700F.

Based on these considerations pure hydrazine was selected to be the propellant used in the study. While it will result in a small (10-15%) performance penalty and must be heated externally to prevent freezing under extreme conditions, it is by far the most developed and best understood of the applicable propellants. It has been used and flight tested and is currently under intensive study for the grand-tour mission (Ref. a). Furthermore, any conclusions drawn using hydrazine will be valid with (at most) small changes for the hydrazine blends. For purposes of this short-term study, the availability of data and experience information appears to outweigh other considerations in the selection.

SYSTEM SELECTION

In selecting the system best suited to the APU application, various types of propellant feed, combustion ignition, and control were considered. Conditions for TPU operation were considered to be the same as for the hydrogen/oxygen APU;

Ref. a. Holcomb, L. B.: Satellite Auxiliary-Propulsion Selection Techniques, Jet Propulsion Laboratory, Technical Report 32-1505, November 1, 1970.

the power profile, flight profile, and power conditioning were taken to be identical. For turbine operation a peak gas temperature of 1700F (which is higher than the 1550F selected for the H_2/O_2 APU) and a tip speed of 1580 ft/sec (which is comparably lower than the 1700 ft/sec for the H_2/O_2 APU) were selected.

The options considered were

Feed System	- Pump Fed
	Pressurized Tank Fed
Ignition System	- Catalytic
	Thermal
	Hypergolic
Control System	- Pressure Modulated
	Pulse-Width Modulated

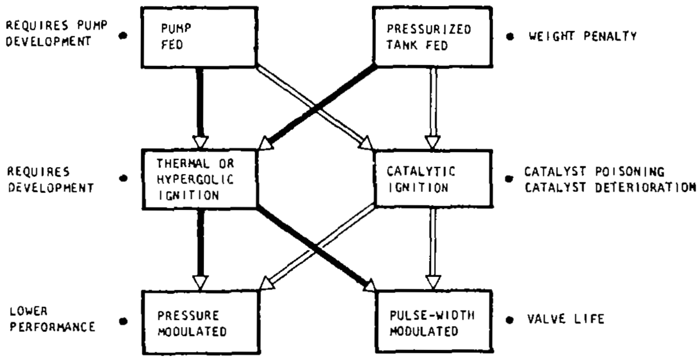
Of the possible combinations, four systems were selected for detailed study These were:

System E1	Pump Fed/Thermal Ignited/Pressure Modulated
E2	Pump Fed/Thermal Ignited/Pulse-Width Modulated
E3	Pressure-Tank Fed/Thermal Ignited/Pressure Modulated
E4	Pressure-Tank Fed/Thermal Ignited/Pulse-Width Modulated

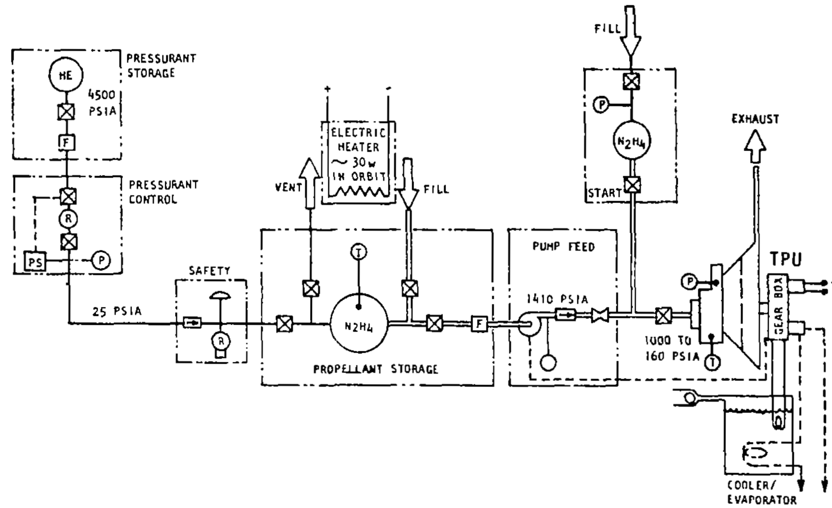
As a result of the tradeoff study discussed below, System E1 was selected as the baseline system. The system selection and the baseline system schematic are shown in Fig. 90.

SYSTEM SELECTION

METHOD



BASELINE SYSTEM (E-1) SCHEMATIC



SYSTEM

E-1 PUMP FED/THERMAL IGNITION/PRESSURE MODULATED

E-2 PUMP FED/THERMAL IGNITION/PULSE-WIDTH MODULATED

E-3 PRESSURE TANK FED/THERMAL IGNITION/PRESSURE MODULATED

E-4 PRESSURE TANK FED/THERMAL IGNITION/PULSE-WIDTH MODULATED

FIGURE 90

Feed System

Propellant feed using either a pump or a pressurized tank was investigated. Pressurized-tank feed is positive and is a proven concept. It has, however, two drawbacks: for multiple start applications the expulsion system must be capable of multiple cycles, and for high feed pressure (desirable for performance) the tank weights become prohibitive. The pump system requires development of a pump to accomplish the particular task of this application and, to maintain the required suction characteristics, a low pressure feed tank continues to be required. Also, the pump can be expected to be relatively low in efficiency, imposing a performance penalty.

The tank weight required is based on data in Ref. b . Figure 91 summarizes these weight criteria. For the booster, approximately 15 cu ft of tank volume is required; it is evident that the pressurized system tank weight is appreciable.

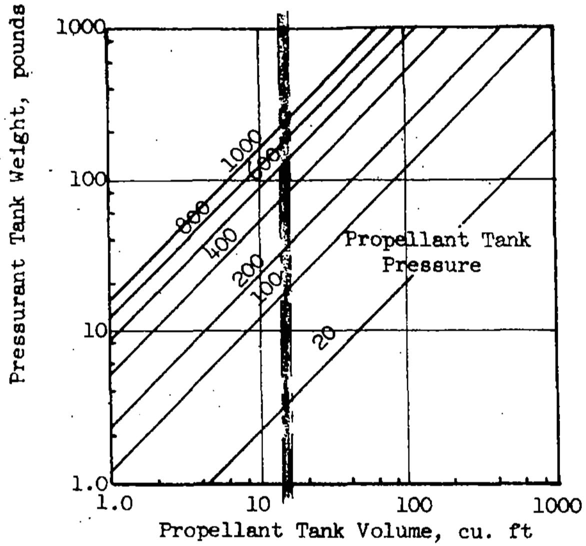
For design purposes the pressure drop between the tank and the combustion chamber has been taken equal to that used in the H_2/O_2 APU:

$$p_{tank} = 1.39\, p_{chamber} + 20$$

The factor includes anticipated pressure losses in the valves, injector, and a cavitating venturi which isolates the supply system from the combustion chamber. A 20 psi line loss is also included.

Ref. b. Space Engine Design Handbook, Rocketdyne, a division of NAR, Report No.R-8000 P-1, 1 January 1969

TANKAGE

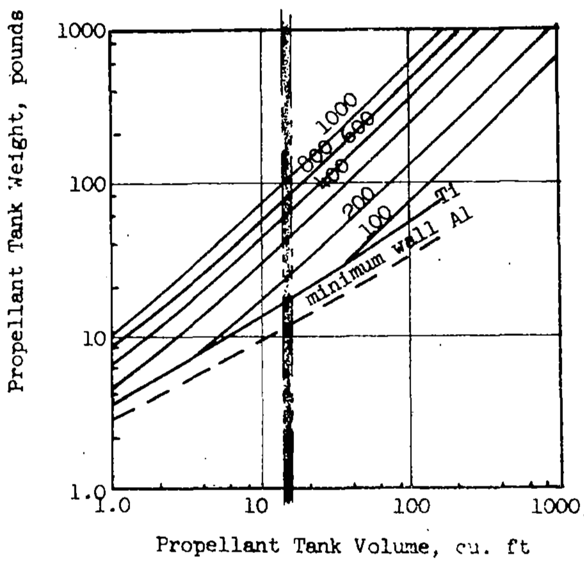


PRESSURANT STORAGE

- Helium Gas @ 4500 psi
- Titanium Tank (6AL 4V)

  s_{ult} = 144,000 psi

  Factor of safety = 1.8
- Spherical Shape



PROPELLANT STORAGE

- Titanium Tank (6AL 4V)

  s_{ult} = 144,000 psi

  Factor of Safety = 1.8
- For Orbiter, minimum wall
- Aluminum is lighter (6066-T6)

  s_{ult} = 47,000 psi

  Factor of Safety = 1.8
- Spherical Shape

FIGURE 91

Tank material can be either titanium or aluminum, both of which are compatible with hydrazine. For pressurized tanks the titanium tanks are lighter at pressures above about 150 psi.

For the pump-fed system a modest 20 psi will be required to maintain the pump in cavitation-free operation. A minimum wall thickness tank will be required, making aluminum lighter. It might be noted that this minimum wall tank will be capable of containing up to 100 psia. At this pressure level the start system shown in Fig. 90 will not be needed, flow can be established directly from the tank during startup.

The propellant tanks (high or low pressure) will require positive expulsion. Three methods have been considered, bladders, bellows, screen tension devices (Fig. 92). The most demonstrated technology is bladder systems which have been flight tested. Cycle life has been demonstrated using rubber (Ethylene-Propylene Terpolymer, 37 cycles to complete expulsion after 30 day's exposure) and extended storage has been performed (463 days at 125F). Because of rubber permeability to helium gas (0.17 $mg/in^2/hr$), some method of gas/liquid separation during long-term exposure will be required. A form of screen tension device appears to be desirable. Bellows can be used if the tank volume is limited. To date, bellows have only been used successfully for tanks up to 3 cu ft volume. It is possible that 5 cu ft could be utilized. This would be sufficient for the orbiter but would require multiple tankage for the booster vehicle. The major advantage of the bellows is positive separation of the gas and liquid. Cycle life has been demonstrated in the limited size.

FEED SYSTEM

POSITIVE EXPULSION

BLADDER

- Demonstrated Technology (Flight Tested)
- Allows Light-weight Tanks (Titanium, Aluminum)
- Cycle Life Demonstrated
  - Rubber ~ 100 cycles
  - Teflon ~ 50 cycles
  - Aluminum ~ 1 cycle (throwaway tank)
- Cylindrical Tank Preferred
  - More weight
  - Less bladder damage
- Requires Gas/Liquid Separation (Screen Tension)

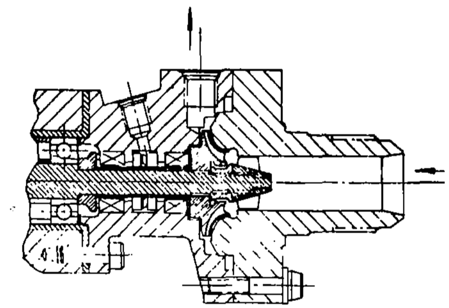
BELLOWS

- Best for Small Tanks (< 3 cu. ft)
- Positive Gas/Liquid Separation
- 100 - 200 cycles Demonstrated
- Temperature Insensitive
- Best for Cylindrical Tanks

SCREEN TENSION

- Fine Mesh Required
  - May accumulate dirt
  - Heavy
- Cylindrical Tanks Preferred
- Slosh Suppression Required
- Infinite Cycle Life
- Relatively Temperature Insensitive

PUMP



HYDRAZINE PUMP FOR 55APU
ROCKETDYNE MK 36 ADAPTATION

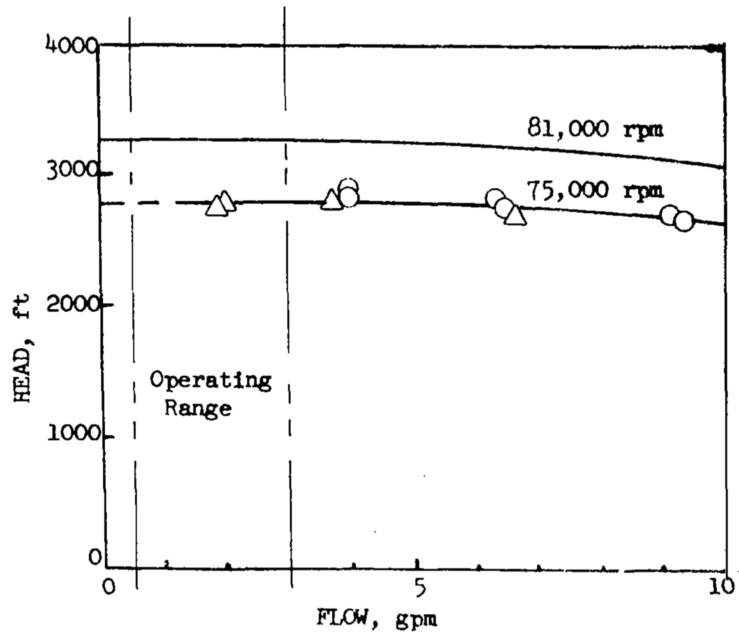


FIGURE 92

Much work has been done in recent years on positive expulsion using screen-tension devices. These are particularly useful in a weightless environment. It is probable that some development effort would be required prior to use of screen-tension expulsion.

<u>For the baseline system a rubber bladder has been selected.</u>

The pump-fed system appears to be more suitable for the multiple start requirement of the APU. Although no developed pump is available, a pump recently tested at Rocketdyne appears to be ideal for this application. Because the large weight penalty associated with the pressurized tankage in the pressure-fed system is not a factor, the pump-fed system can be operated at high combustion pressure. The pump head at a pressure of 1000 psi is 3200 ft which can be achieved by operating the Rocketdyne Mk 36 pump at 81,000 rpm (Fig. 92). This extrapolation is based on well documented test data at 75,000 rpm. The pump curve is extremely flat in the expected operating range indicating good flow control as power changes occur.

<u>For the baseline system, pump feed has been selected.</u>

Ignition System

The generation of gas when monopropellants are used requires the decomposition of the propellant. Unlike bipropellant systems where the injector design is critical, for monopropellants decomposition control is essential.

For hydrazine systems the decomposition of the hydrazine results in the formation of ammonia which then further decomposes into hydrogen and nitrogen at a relatively slow rate. This dissociation is endothermic and results in cooling of the gases with a simultaneous decrease in the molecular weight ($\bar{m}$) of the gas over part of the dissociation range. Between 0 and 75 percent ammonia dissociation the parameter $T/\bar{m}$ varies only $\pm$ 3 percent. For 100 percent dissociation this parameter is reduced by 10 percent. As pointed out previously, for the APU application, a temperature of 1700F, corresponding to 60 percent ammonia dissociation, is desirable. Because this dissociation is relatively slow, provision must be made to enhance it. Adiabatic ammonia dissociation to 29 percent (2100 F) is reported to occur in 35 milliseconds. To ensure reasonable temperature, a long combustion chamber is required. Fortunately, dissociation of ammonia can be speeded catalytically in the presence of many metals. Pulsing rocket engines have been operated at Rocketdyne with 55 to 60 percent dissociation at flowrates comparable to those required by the APU. In such tests, screens (e.g. nickel, stainless steel) were utilized to achieve the dissociation. The screen catalyst is relatively immune to poisoning by exposure to air so that it is useable in a restart application.

To initiate the gas generation process the hydrazine must be decomposed. The most conventional method is to use a catalyst, although the application of heat as in a thermal bed has also been used successfully. Table 5 lists three methods of ignition and decomposition control considered together with the evaluation factors.

While catalytic ignition and decomposition is the most developed, tests have shown that the catalyst will poison in the presence of water vapor or oxygen especially if the catalyst is hot. If the APU will be required to restart

IGNITION

CATALYTIC	WELL DEVELOPED	FLIGHT QUALIFIED AND FLOWN
	SHELL 405 CATALYST	3×10^6 HOT CYCLES DEMONSTRATED UP TO 160 COLD STARTS DEMONSTRATED
	REQUIRES PROTECTION	WATER VAPOR COATS SURFACE OXYGEN CONTAMINATES
	REQUIRES CARE	BRITTLE BREAKUP - PRESSURE PULSES STRATIFIED FLOW ALONG WALLS
THERMAL BED - HEATED	RELIABLE	PRELIMINARY DEVELOPMENT COMPLETE FOR PULSED AND STEADY OPERATION 2 WATTS/LB THRUST
	SLOW INITIAL RESPONSE	TIME REQUIRED TO HEAT BED (POWER SOURCE DEPENDENT)
	LESS DECOMPOSITION CONTROL	REQUIRES GREATER L^* TO ENSURE NH_3 DECOMPOSITION
THERMAL BED - HYPERGOLIC	DEVELOPED, RELIABLE	BI-PROPELLANT TECHNOLOGY
	REQUIRES OXIDIZER SYSTEM	TANKAGE, VALVES, PRESSRANT
	MORE COMPLEX CONTROLS SLOW INITIAL RESPONS	PULSE OXIDIZER FOR SHORT DURATION OXIDIZER ON 30 ms, OFF 570 ms FUEL ON 200 ms, OFF 400 ms REACHES SELF-SUSTAIN IN 6 SEC
	LESS DECOMPOSITION CONTROL	REQUIRES GREATER L^* TO ENSURE NH_3 DECOMPOSITION

TABLE 5

within the atmosphere, then this type of poisoning becomes a major consideration and provision for excluding air from the gas generator upon shutdown would be required. This would normally be a purge system which would have to operate until the bed had cooled to an acceptable level. The weight penalty to be assessed to such a purge system depends on the number of shutdowns required per mission and the ground equipment available if GSE purge is provided. It is evident that more than one or two purge cycles will result in a major weight penalty.

The use of a thermal bed to achieve decomposition is also relatively well developed. Initial response is, of course, delayed until the external heat source has increased bed temperature to the required level. Once decomposition is achieved, the reaction is self-sustaining. There is a small power penalty for the heat source.

Hypergolic initiation of dissociation involves the use of bipropellant operation for short times until the thermal bed reaches self-sustaining temperature. If pulse width modulation is used, this type of operation may require a bipropellant mode for about 5 percent of each pulsing cycle for the first 6 seconds. This system is independent of external power for heat but requires the inclusion of an oxidizer supply system with associated controls.

For the baseline system an externally heated thermal bed was selected.

Control System

Two methods of power control were considered for the storable-propellant APU - pulse width control and pressure-modulated control. The considerations pertaining to these are essentially similar to those for the H_2/O_2 APU except that it is possible to consider higher pressure levels because tankage volume is lower. For purposes of system evaluation, the peak gas generator pressure was limited to 1000 psi. This is somewhat arbitrary and higher pressures could result in lighter-weight systems. The major advantage in using high pressure is that performance degradation due to varying back pressure is minimized, and the pressure change effects due to larger power variation is minimized.
For the baseline system, pressure-modulated control was selected.

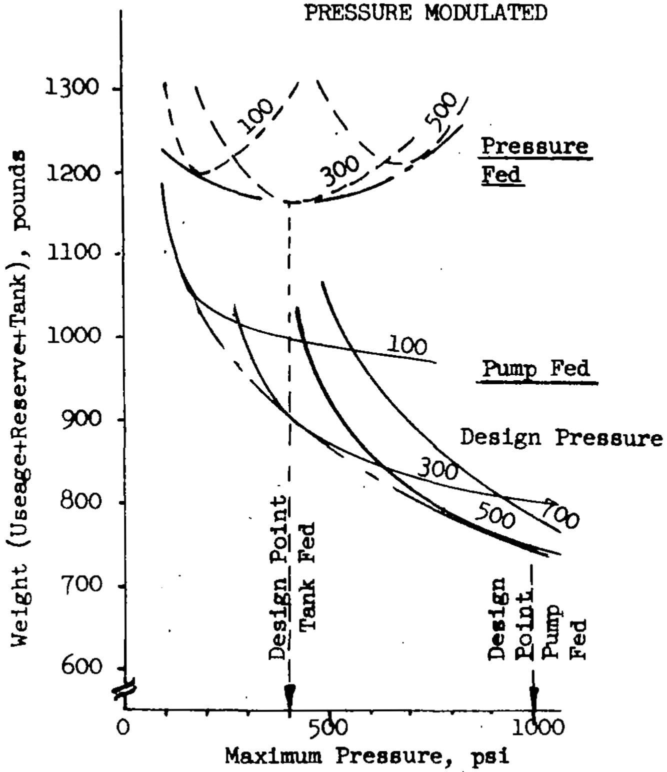
SYSTEM OPTIMIZATION

The selection of the system design conditions was based on a weight optimization for both the booster and orbiter power/flight profile. The results obtained for the booster are shown in Fig. 93 .

Pressure Modulated System

For the pressure modulated system, the required pressure to produce the required power at the exhaust pressure varies over the given flight profile was calculated. Varying design pressures were assumed and off-design penalties as they apply to supersonic turbines were assessed. The propellant usage was calculated and a 5% reserve plus the required tank weight were added to determine weight. The result is a series of curves shown in Fig.93.

SYSTEM OPTIMIZATION



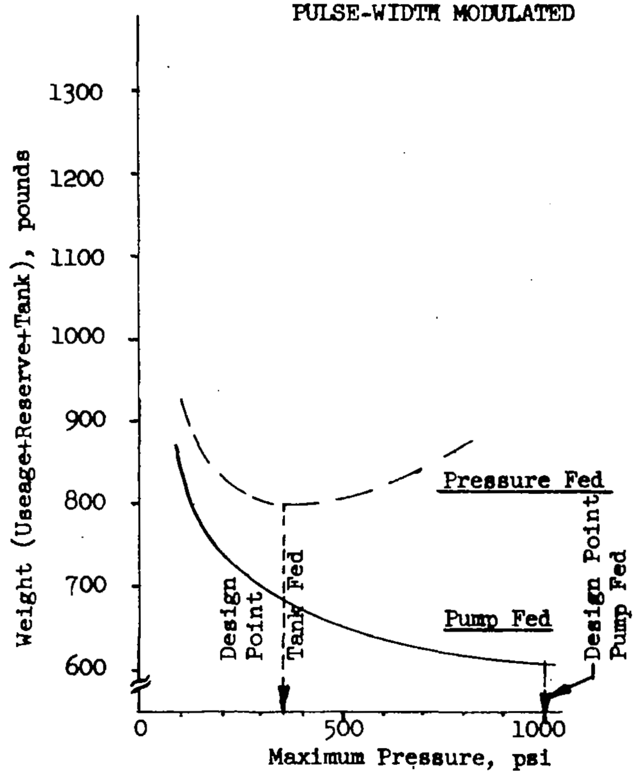


FIGURE 93

which form an envelope as shown. When the system is pump fed, tank weight is low because a minimum wall tank is all that is required. An increase in the maximum allowable pressure is desirable and 1000 psi was selected for system E1 (pump fed/ thermal ignition/pressure modulated). For the pressure-fed system, the tankage weight varies with the supply pressure. The supply pressure was assumed to be

$$p_s = 1.39\ p_r + 20$$

which provides for the pressure drop in the injector, the flow control, and the lines (just as in the H_2/O_2 system). Tank weights were calculated using the data of Fig. 91 as described in the Feed System section. The result (Fig. 93) shows a definite minimum in the weight (usage + reserve + tank) when the design pressure is 300 psi and the maximum pressure is 400 psi (System E3, pressure fed/thermal ignition/pressure modulated).

Pulse-Width Modulated System

When pulse width modulation is used, the design pressure and peak pressure are essentially the same because the pressure is either maximum or zero. (They are not precisely the same because provision is made to obtain peak power when the pulse is on 96 percent of the time, giving a 4 percent reserve. The design pressure has been defined as that required to obtain full power at constant flow.) The two curves related to pulse-width modulated operation are also shown in Fig. 93. Again the best operation for pump feed is the maximum useable (1000 psi) and a minimum exists for pressure feed at 350 psi .

Calculations similar to those plotted in Fig. 95 were made for the orbiter profile. The results are summarized in Table 6 for both vehicles.

TABLE 6
Design and Peak Pressures

	BOOSTER		ORBITER	
System	$(P_T)_{max}$	(P_T)	$(P_T)_{max}$	(P_T)
E1	1000	500	1000	300
E2	1000	960	1000	960
E3	400	300	250	200
E4	350	335	200	190

Figure 94 summarizes the total system weight and specific propellant consumption for the four systems and for the booster and orbiter application. System E1 was selected for the baseline system (even though weight is not minimum) because the pumped-propellant-fed system is considered to be easier to control when pressure modulation is used.

SYSTEM COMPONENTS

Turbopower Unit

Design considerations for the turbopower unit are essentially the same for the storable system as for the H_2/O_2 APU. Because design gas temperature is higher (1700F), the turbine tip speed has been reduced to 1580 ft/sec. Maintaining the same turbine speed results in a tip diameter of 6.040 in.

SYSTEM E STORABLE PROPELLANT, SYSTEM WEIGHT AND SPC

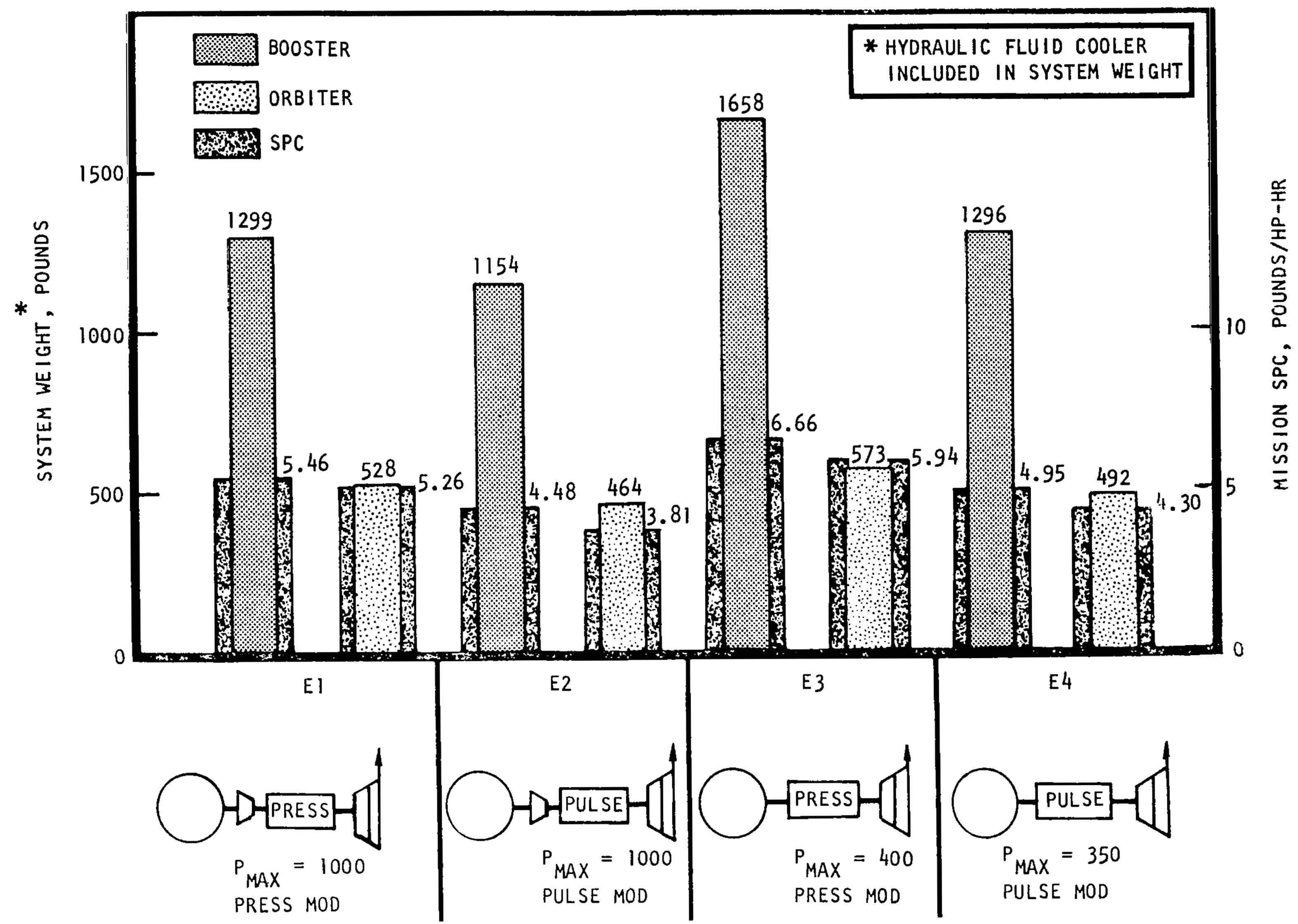


FIGURE 94

The turbine design concept is shown in Fig. 95 . . Two wheels provide pressure staging with the first stage partial admission (72 degrees) and the second stage full admission. As shown in Fig. 95 , burst protection is included.

Hydraulic-Oil Cooler

One disadvantage in the use of the storable system compared to the H_2/O_2 system is the hydraulic-oil cooler. Because it is not possible to utilize the propellant to remove the heat generated in the hydraulic circuit (hydrazine has poor heat capacity and becomes flammable at temperatures above about 400F), an auxiliary system must be provided. It was decided to utilize the latent heat capacity of a liquid to absorb this energy. Of the various candidate materials water has by far the highest latent heat per pound and was, therefore, selected. Figure 96 shows the concept. A valve is set to blow steam off at 20 psi above the ambient. This maintains low Δ P across the tank and minimizes weight. Boiling temperature is set by this pressure independent of power demand. The hydraulic fluid and lube oil flow through a matrix of tubes located at the bottom of the tank. They are sized to maintain hydraulic-oil temperature at 300 F or below. A total of 85 ft of 3/8-inch-diameter tubing is required. The matrix is sized to minimize pressure drop. Provision is made to maintain sufficient water (20 pounds) in the tank at the end of a mission to ensure that tubes are covered. If it becomes necessary to use this technique in a zero-g environment, some form of screen-tension device will be required to maintain the liquid mass on the tubes.

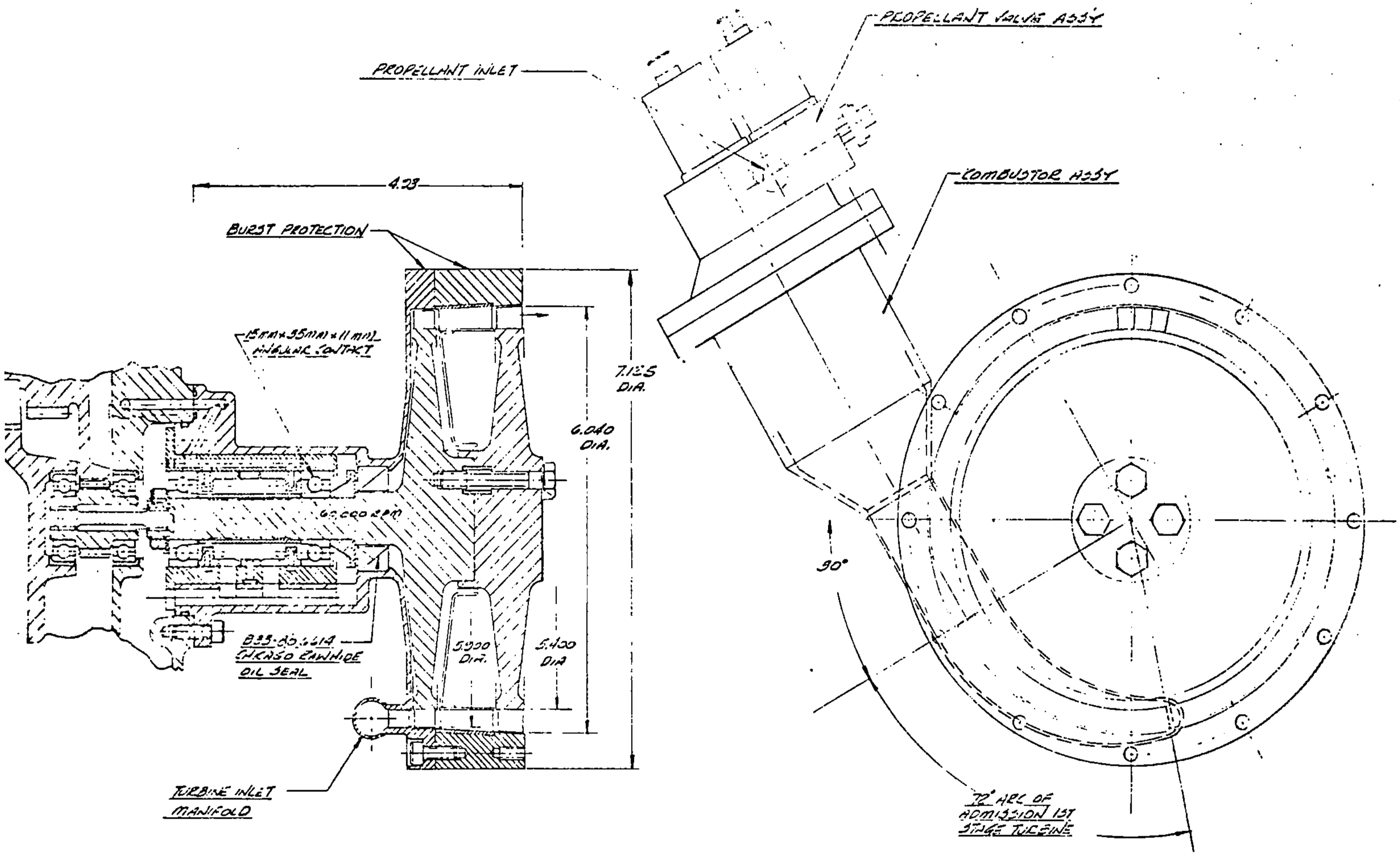


FIGURE 95
SSAPU TURBINE ASSY
STORABLE PROPELLANT SYSTEM

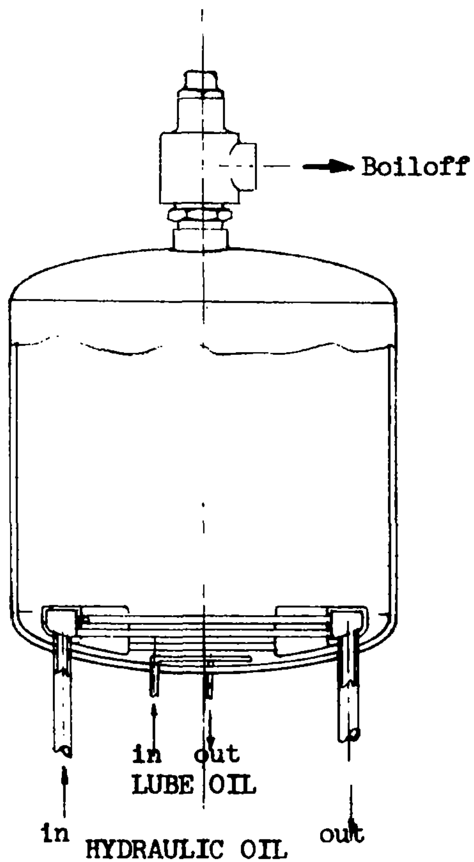
HYDRAULIC OIL COOLER

POOL BOILER

Water
High Latent Heat/Pound
Easy Handling

Valve Set @ 20 psid
Booster Mission Requirement

166	pounds water for boiloff
20	pounds water tare
64	pounds tank, valve, tubes
250	pounds



HYDRAULIC OIL

Maintain Temperature 300 F
Minimize pressure drop

FIGURE 96

Materials

Hydrazine has been shown to be compatible with a wide variety of materials for this type of application. Table 7 summarizes compatibility with the more prominent candidate materials for an APU.

MATERIALS

COMPATIBILITY FOR HYDRAZINE OR BLEND

MATERIALS	GENERAL SERVICE	LIMITED SERVICE	UNACCEPTABLE
ALUMINUM	■		
INCONEL		■	
IRON			■
MONEL		■	
NICKEL		■	
CRES - 304, 347,	■		
- 302, 321, 410, 430		■	
- 303, 315, 316, 317, 329, 416, 440			■
STELLITE		■	
TITANIUM	■		
PLASTICS - Hycar, Kel-F, Nylon, Polyvinyl		■	
- Teflon, Polyethylene	■		
RUBBER		■	
GRAPHITE		■	
SILICONE		■	

TABLE 7

Controls

The controls for the storable propellant APU are essentially similar to those used on the H_2/O_2 APU except that no propellant conditioning is required and, with a monopropellant, gas generator control is different. Table 8 lists the control tradeoffs considered together with the selections for the baseline system.

TABLE 8

APU SYSTEM CONTROLS TRADEOFF

CONTROL OPTIONS AND SELECTION	ADVANTAGES	DISADVANTAGES
● Power Control, Monitoring		
● Torque Load (Variable Speed)		
● Clutch or Dissipative Load (Constant Speed)		
✱ RPM (Constant Speed)	● Simple System ● High Reliability ● Low Weight ● Optimum System Operation at Constant Speed ● Reliable Control (RPM Monitor) ● Allows Operation at Off Design with Minimum Penalties ● Minimum Propellant Consumption	● Large Number of System Cycles Required. Improved Component Performance, Life and Durability Required. ● Heat Soadback and Thermal Conditioning Critical
● Gas Generator Operation - Propellant Flow		
● Pulse (Bang-Bang Valve)		
✱ Modulating	● Constant Turbine Speed ● Minimizes Thermal Design	● More Complex ● Possible Sensitivity to Acceleration Loads ● Instrumentation Calibration Accuracy and Valve Positioning Calibration Critical. Subject to Changes with g-Load, Wear, Friction, Contamination, Temperature. ● Higher Propellant Consumption ● Components Designed to Operate Over Wider Range of Pressures, Flows. Results in Lower Efficiency. ● Instability Potential ● Servo Fluid Overboard Dump

✱ Denotes Selection for Baseline System

TABLE 8 (Continued)

APU SYSTEM CONTROLS TRADEOFF

CONTROL OPTIONS AND SELECTION	ADVANTAGES	DISADVANTAGES
● Gas Generator Operation - Combustion Temperature		
● Closed Loop		
✱ Open Loop	● Simplicity. Temperature Controlled by Gas Generator Bed Design	● No System Shutdown for Turbine Protection at High Temperature.
● Gas Generator Ignition		
● Catalytic		
● Thermal-Hypergolic		
✱ Thermal-Externally Heated	● Inert to Environment ● Simple	● Requires External Heat Source ● Slow Startup
● Gas Generation		
● Catalytic		
✱ Thermal	● Inert to Environment and Materials ● Simplicity	● Design Over Operating Range Critical to Prevent Quenching ● Requires Hypergolic Start or Heater

TABLE 8 (Concluded)

APU SYSTEM CONTROLS TRADEOFF

CONTROL OPTIONS AND SELECTION	ADVANTAGES	DISADVANTAGES
● Propellant Fee System		
● Cold Gas, Pressure Fed		
✱ Pump Fed	● Lighter Weight ● Compact ● High Efficiency-High Pressure System	● Million Cycle Pump Development. ● Pump NPSH Requires Pressurization System ● Requires Startup System (Pump Connected to Turbine Gear Box Drive).
● Feed System Propellant Expulsion Technique		
● Surface Tension		
● Bellows		
✱ Bladder		
- Teflon	● Conventional. Demonstrated and Flown ● Easy to Clean ● High Volumetric and Expulsion Efficiency ● Smooth Expulsion ● Gas Free Expulsion with Gas Trap	● Limited Cycle Life ● Sensitive to Handling Abuses ● Sensitive to Dynamic Loads, Especially Sloshing ● Sophisticated Tank Gauging and Filling Techniques Required ● Pressurant Gas Permeation of Bladder
- Rubber	● Good Cycle Life ● Insensitive to Handling Abuses and Dynamic Loads ● Easy to Clean ● High Volumetric and Expulsion Efficiency ● Smooth Expulsion ● State-of-the-Art ● Gas Free Expulsion with Gas Trap	● Sophisticated Tank Gauging and Filling Techniques Required. ● Pressurant Gas Permeation of Bladder (Greater Than Teflon). ● Development Required

All required technology for the turbopower unit has been developed. Demonstration in the selected size range is required.

Controls similar to those which will be required have been demonstrated and fully developed in rocket technology. Unique requirements of decomposition maintenance and multiple ignition has been demonstrated.

TECHNOLOGY STATUS

In order to determine the useability of storable propellants for the APU the status of the various applicable portions of the technology were reviewed. Figure 18 shows the resultant assessment for various options considered.

The propellant selected (hydrazine) has been used in a number of flight systems and has been tested as a propellant in the size and power range required here. It is considered to be well developed.

The pumping system will require adaptation of an existing (Mark 36) design with subsequent confirmation testing. The technology has been demonstrated on other, similar systems.

For ignition, thermal bed heating is selected. This type of system has been developed for thrustors and has proven to be reliable. It requires adaptation and final development in the proper size.

Combustion systems similar to those required for this application have been developed. Sufficient L* for decomposition control needs to be demonstrated.

Development of an appropriate hydraulic oil cooler will be required. The design is established and all necessary technology has been identified as being available.

Availability of materials compatible with hydrazine service is established.

RELIABILITY

Each of the systems evaluated in this study was subjected to a preliminary analysis to determine the relative reliability. A separate set of calculations was made for the H_2/O_2 and storable propellant systems. Relative failure rates were estimated and nominal and maximum expected system reliabilities were determined using established NASA failure data.

CRYOGENIC SYSTEM

System Analysis

The candidate systems were studied in detail to determine the type of component best suited for the intended application. Ground rules and assumptions essential to (and made prior to) the analysis are listed in Table 9.

As an example of the analysis, the type of servo system required for the regulation of propellants through the hot-gas heat exchanger was determined. It proved to be essential to ascertain whether a mechanical or electronic type of system was required for the application. The unreliabilities of the two systems are significantly different.

A system matrix identifying the type and quantity of components required for each of the candidate systems was constructed as shown in Table10. Component relative unreliabilities were for incorporation into the system reliability models and system reliabilities and unreliabilities were

TABLE 9

GROUND RULES FOR RELIABILITY ANALYSIS

- o Hydraulic pressure feedback to motor (hydraulic) servo valve will not require electronic gear for detection, amplification, etc., but is a direct hydraulic pressure control mechanism.

- o Heat exchanger bypass valves are considered for this analysis to be electronic servos (torque motor, solenoid, etc.) with electronic feedback from a thermocouple temp pickup requiring amplification, etc.

- o Oil and hydraulic cooler controls are considered for this analysis to be simple thermostat devices requiring no electronic equipment.

- o The bipropellant valve (whether modulating or pulsing) will require electronic equipment to translate the speed reading to an electrical output to the valve. The combustor throttling assembly will also utilize a similar feedback electronic control assembly.

- o All system valving (purge control, venting, etc.) are assumed to be remotely operated solenoid valves even though the system schematic does not indicate electrical actuation. The only exceptions to this assumption are the fill and vent valves utilized on the propellant tanks when the tankage is provided with the APU. In flight purge is a requirement.

- o Oxidizer regulator performance requirement for the high pressure system is assumed to be $\pm$ 1% whereas the low pressure system required $\pm$ 0 2%.

- o Bipropellant valve actuation for the high pressure systems will be pneumatic whereas the low pressure system will utilize hydraulic pressure.

- o Because they are not shown on the schematic, the following were not considered in the reliability analysis:

    a) Hydraulic system components such as reservoirs, accumulators and system valving.

    b) 28 volt power supply with the proper support electronics (apart from the electronic control loops) for heater consumption, valve power, etc.

    c) Lubricating oil supply equipment.

- o The pressure controlling device in the propellant tank is a mechanical device rather than a resistance bridge type transducer which sends a signal to the controller, etc.

- o All temperature instrumentation will be thermocouples with individual failure rates but the failure rates will not be included in the controller failure rate.

Component Name	A-1	A-2	A-3	A-4	A-5	A-6	B-1	B-2	B-3	B-4	B-5	B-6	C-1	C-2	C-3	C-4	C-5	C-6	C-7	C-8	Relative Unreliability Q	NASA Report SE-2A-63-1C-6	AD Report -36 441-ML -19	Beach & Chesapeake	PTAF 1000 Hrs. 3 Hr./Mission =4.1x10⁻⁶ Q Failure/Hr	Reliability	1-4.1x10⁻⁶ Q
	System Designation																				Unreliabilities (Failure Rates)						
1. Hydrogen Tank						X						X									2	ND	ND	0.07	360	.999 92	
2. Hydrogen Start Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	1.7	11.0	30	.999902	
3. Hydrogen Purge Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	1.7	11.0	30	.999902	
4. H_2 Purge Check Valve	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	4	112	2.0	2.3	180	.999984	
5. Hydraulic Motor/H_2 Pump	X	X	X	X	X	X															180	[illegible]	ND	4.5	4	.999262	
6. Hydraulic Servo Valve	X	X	X	X	X	X															8	[illegible]	[illegible]	[illegible]	.90	.999967	
7. H_2 H/E Bypass Valve (Servo)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	72	666	1.7	30.0	10	.999706	
8. H_2/Hot Gas Heat Exchanger	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	8	633	<0.1	5.0	90	.999967	
9. O_2 & H_2 Fill Valve (Manual)	X	X	X	X		2					X	2									4	250	ND	0.112	180	.999984	
10. O_2 & H_2 Vent Valve (Manual)	X	X	X	X		2					X	2									4	250	ND	0.112	180	.999984	
11. O_2 & H_2 Relief Valve	X	X	X	X		2					X	2									12	330	1.7	5.7	61	.999951	
12. Prop Heater & Control Unit	X	X	X	X		X					X	2									54	260	ND	7.67	13	.999778	
13. Oxygen Tank	X	X	X	X		X					X	X									2	ND	ND	0.07	360	.999992	
14. Oxygen Start Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	1.7	11.0	30	.999902	
15. Oxygen Check Valve	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	4	112	2.0	2.3	180	.999984	
16. Oxygen Purge Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	1.0	11.0	30	.999902	
17. O_2 H/E Bypass Valve (Servo)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	72	666	1.7	30.0	10	.999706	
18. O_2/Hot Gas Heat Exchanger	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	8	633	<0.1	5.0	90	.999967	
19. Hydraulic Oil Cooler	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	8	633	<0.1	1.67	90	.999967	
20. Lubricating Oil Cooler	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	8	633	<0.1	1.67	90	.999967	
21. H_2 Start Accum Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X									24	690	1.7	11.0	30	.999902	
22. H_2 Accum Charge & Vent Valve	X	X	X	X	X	X	X	X	X	X	X	X									24	690	1.7	11.0	30	.999902	
23. H_2 Start Accumulator	X	X	X	X	X	X	X	X	X	X	X	X									4	155	N.D.	7.2	180	.999984	
24. Main H_2 Check Valve	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	4	112	2.0	2.3	180	.999984	
25. H_2/O_2 Temp Equalizer	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	8	635	<0.1	5.0	90	.999967	
26. H_2 Purge Vent Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	2.0	11.0	30	.999902	
27. O_2 Purge Vent Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	2.0	11.0	30	.999902	
28. O_2/H_2 Press Regulator (1%)	X	X	X	X	X	X	2	2	2	2	2	2									81	1075	N.D.	3.0	9	.999668	
29. O_2 Press Regulator (0.2%)													X	X	X	X	X	X	X	X	126	<1075	N.D	<3.0	6	.999484	
30. Pulsing Biprop Valve	X		X		X		X		X		X		X		X		X		X		72	428	N.D	372	10	.999706	
31. Combustor	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	16	40	1.4	N.D	45	.999934	
32. Combustor Purge Valve (Solenoid)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	24	690	1.7	11.0	30	.999902	
33. Turbopower Unit (4 stage)			X	X	X	X			X	X											32	2554	5.8	10.0	23	.999869	
34. Turbopower Unit (2 stage)	X	X					X	X			X	X	X	X	X	X					24	ND	ND	ND	30	.999902	
35. Turbopower Unit (1 stage)																	X	X	X	X	16	ND	ND	ND	45	.999934	
36. Gear Box	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	20	40	N.D.	N.D.	36	.999918	
37. Electronic Control Assy	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	30	641	4.4	N.D	24	.999877	
38. Modulating Biprop Valve		X		X		X		X		X		X		X		X				X	84	666	0.4	30.0	9	.999755	
39. Variable Geometry Comb Inlet																		X			96	N.D	N.D	ND	8	.999606	
40. Bypass Valve Elect Control Assy	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	30	641	4.4	N.D	24	.999877	
41. Hyd/Oil Cooler Thermostat	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	1.6	ND	2.24	360	.999992	
42. Speed Transducer	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	2	ND	ND	ND	360	.999992	
43. Temp Transducer	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	4	ND	ND	ND	180	.999984	
44. O_2 Shut-off Valve (Solenoid)		X		X		X		X		X		X		X	2	X		X		X	24	690	1.7	11.0	30	.999902	
45. H_2 Shut-off Valve (Solenoid)		X		X		X		X		X		X		X	2	X		X		X	24	690	1.7	11.0	30	.999902	
Total Score	1015	1075	1023	1083	947	1105	832	892	840	900	908	1044	712	772	776	772	704	776	704	764							

TABLE 10

calculated and compared. Results of the system comparisons were analyzed, conclusions drawn and recommendations for improvement were compiled.

Determination of Relative Unreliabilities

Two basic methods were utilized to jointly arrive at relative component unreliabilities. One method was a search of available failure data on similar components. Three sources of failure data were utilized as listed in the reference section as Reference a), b) and c). The second method was a systematic scheme of deriving a relative unreliability ranking which considered four attributes: a) performance, b) stress, c) state-of-the-art, and d) complexity.

Generic Data Search - Three sources of data were found to contain unreliability or failure rate data which could be utilized in this trade study:

Reference a) is a compilation of failure rates which NASA has compiled in its experience with rocket engines up to 1965. The data are applicable due to the cryogenic component application of the reported data.

Reference b) is a summary of failure data compiled from Rocketdyne records on the J-2 and F-1 programs. Failure rates from only stage static testing (after engine delivery and installation into the vehicle stage) were considered in the report.

Reference c) is data from an independent organization which compiled all available failure rate data from aerospace contractors. Direct application of the data is limited due to a lack of usage description and necessity for application of various influence factors.

Reliability Estimate - In obtaining the estimate of component relative unreliability a system was devised to incorporate the influence of characteristics such as a) relative tightness of the performance requirement, b) relative degree of severity of stresses, c) relative degree of previous usage or advancement of state-of-the-art, and d) relative complexity. Each component was analyzed relative to the ground rules delineated in Table 9 and given a rating from 1 to 5 as described. The product of the four factors was obtained to arrive at the predicted relative unreliabilities which are listed for each component in Table 10.

Tests for Rationality

In order to validate the predicted relative unreliabilities (failure rates), generic data were compared with the estimates, and components were compared with components. With these two cross checks the unreliability estimates were reviewed and revised as dictated by the analysis.

The results of this analysis are summarized below:

1. The relative system failure rates were determined and are summarized in Table 11.
2. The most significant influences on relative system failure rates are provided by three areas listed below in order of importance.

   Pressure System Elements. The low pressure subsystem is best because it has the fewest valves.

TABLE 11
SSAPU RELATIVE UNRELIABILITIES

SYSTEM DESIG.	VEHICLE H_2	APU H_2	VEHICLE O_2	APU O_2	1 STAGE TURB.	2 STAGE TURB.	4 STAGE TURB.	PULSE MODULATED	PRESS MODULATED	AREA MODULATED	Q (UNRELIABILITY)	RANK	Δ RATING	% ABOVE MIN.	EXPECTED (NOM) RELIABILITY	MAXIMUM RELIABILITY
A1	X			X		X		X			1015	15	311	144.2	.9705	.9958
A2	X			X		X			X		1075	18	371	152.7	.9687	.9956
A3	X			X			X	X			1073	16	319	145.3	.9702	.9958
A4	X			X			X		X		1083	19	379	153.8	.9685	.9956
A5	X		X				X	X			947	14	243	134.5	.9724	.9961
A6		X		X			X		X		1105	20	401	157.0	.9678	.9955
B1	X		X			X		X			832	9	128	118.2	.9758	.9966
B2	X		X			X		X			892	11	188	126.7	.9746	.9963
B3	X		X				X	X			840	10	136	119.3	.9756	.9966
B4	X		X				X		X		900	12	196	127.8	.9738	.9963
B5	X			X		X		X			908	13	204	129.0	.9736	.9963
B6		X		X		X			X		1044	17	340	148.3	.9696	.9957
C1	X		X			X		X			712	3	8	101.1	.9793	.9971
C2	X		X			X			X		772	5	68	109.7	.9775	.9968
C3	X		X			X		X			776	7	72	110.2	.9774	.9968
C4	X		X			X			X		772	5	68	109.7	.9775	.9968
C5	X		X		X			X			704	1	0	100.0	.9795	.9971
C6	X		X		X					X	776	7	72	110.2	.9774	.9968
C7	X		X		X				X		704	1	0	100.0	.9795	.9971
C8	X		X		X				X		764	4	60	108.5	.9778	.9969

The pressurized subsystem (supercritical) has more valves and is significantly less desirable than the low pressure system.

The pumped system is least desirable primarily because of the necessity for a hydraulic motor-driven hydrogen pump.

A high-pressure gas system, while not included in the analysis, would be rated similar to the low-pressure gas system.

Turbine Combustor Propellant Control. The pulse-modulated system utilizes a single bipropellant valve which introduces propellants to the combustor at controlled average flow rates and provides positive shutoff for system nonoperating periods.

Modulating valve and variable geometry injection may require additional positive shutoff valves, which can increase the system unreliability.

The variable geometry injection concept appears the least reliable because of system complexity and development risk.

Tankage - Vehicle propellant tanks (e.g., those to be used for the ACS) are better than separate APU tanks because they can be used with essentially no reliability penalty for the vehicle and eliminate the need for APU tanks and associated fill, drain, and relief controls.

3. The major reliability problem areas identified by this study are:

Complex Control Systems - Controls systems, especially those requiring transducer-electronic control-electrohydraulic control loops contribute substantially to the system unreliability. Both hardware complexity and critical performance requirements are influential factors.

Components - Individual components which have the highest relative failure rates, thus incluencing the reliability of the system in which they are used are:

Hydrogen pump and hydraulic motor

Modulating bypass valves and high precision pressure regulators

4. Reliability improvement can be made in the following areas:

Alternate control systems with less complexity or wider performance tolerance ranges.

Improved valve and regulator or pump technology.

In-system redundancy.

STORABLE SYSTEMS

This section presents the results of a reliability study of eight conceptual storable propellant systems. The objectives of the study were to compare the storable propellant systems to previously evaluated oxygen/hydrogen systems as well as to compare the eight concepts. In addition, a brief maintainability study was conducted to identify any unique storable propellant system problems or costs which should be considered in the comparison of the storable and cryogenic systems.

Reliability Trade Study Method

Figure 97 is a matrix of the major system differences and designations. The eight storable systems considered were various combinations of:

1. Pump fed or blowdown system concepts
2. Pulsing or modulating main propellant valve
3. Catalytic or thermal combustion devices

In order to compare the eight candidate systems a complete components list was created for each system (many components common), and relative unreliabilities were listed for each component. By simply summing the component unreliabilities, the system unreliabilities were obtained. Table 12 is the list of components, the system usage, the component unreliability and the resultant system unreliability. Ranking of the systems by the relative unreliabilities can be accomplished for comparison only, but cannot be used on an absolute basis.

Relative unreliabilities were estimated and tested for rationality in the same manner as previously described for cryogenic systems. Ground rules used were:

1. Fill valve and purge valve unreliability was included only when the manual valve was required to provide a positive seal in the closed position. If the manual valve is open during the whole mission and

STORABLE PROPELLANT SYSTEM - RANKING CHART

SYSTEM DESIGNATION	Blowdown System	Pump-fed System	Catalytic Combustion	Thermal Bed Combustion	Modulating Valve	Pulsing Valve	RELATIVE UNRELIABILITY FACTOR	RANK	% ABOVE MINIMUM
E-1		X		X	X		564	6	27.2
E-2		X		X		X	535	5	20.8
E-3	X			X	X		472	2	6.5
E-4	X			X		X	443	1	0
E-5		X	X		X		602	8	35.8
E-6		X	X			X	600	7	35.4
E-7	X		X		X		510	4	15.0
E-8	X		X			X	508	3	14.7

FIGURE 97

COMPONENT NAME	E-1	E-2	E-3	E-4	E-5	E-6	E-7	E-8	Relative Unreliability factor	SRI DATA x 10^-6 Cycles	SRI DATA x 10^-6 Seconds	Cryogenic APU Trade Study Factor
	SYSTEM DESIGNATION											
High Pressure Helium Tank - 4500 psi			X	X			X	X	3	50	.0001	-
Low Pressure Helium Tank - 1700 psi	X	X			X	X			2	50	.0001	-
Helium Fill Manual Valve (Unregulated)	X	X	X	X	X	X	X	X	8	30	-	4
Helium Supply Solenoid Valve (Unreg.)	X	X	X	X	X	X	X	X	22			24
Helium Regulator - 4500 to 500 psi			X	X			X	X	68	600	.3	<81
Helium Regulator - 1000 to 500 psi	X	X			X	X			36	600	.3	<81
Helium Supply Check Valve	X	X	X	X	X	X	X	X	2	30	.1	4
Helium Relief Valve	X	X	X	X	X	X	X	X	9			12
Helium Vent Valve (Low Press) Manual	X	X	X	X	X	X	X	X	4			4
Hydrazine Tank (Screen)	X	X	X	X	X	X	X	X	48	100	.001	2
Hydrazine Manual Valve (Resupply)	X	X	X	X	X	X	X	X	4			4
Hydrazine Solenoid Valve	2	2	X	X	2	2	X	X	18	100	.01	24
Heating Blanket	X	X	X	X	X	X	X	X	16			
Temp Sensing and Control Device	X	X	X	X	X	X	X	X	2			4
Main Propellant Valve - Modulating	X		X		X		X		63			84
Main Propellant Valve - Pulsing		X		X		X		X	34			72
Speed Pickup	X	X	X	X	X	X	X	X	2			2
Prop Control Electronic Assembly	X	X	X	X	X	X	X	X	30			30
Catalytic Combustion Device					X	X	X	X	81 54	- -	100	-
Thermal Bed Combustion Device & Temp Control	X	X	X	X								
Latching Solenoid Valve (Hydrazine)	2	2	X	X	2	2	X	X	25	200	.1	16
Turbine Assembly	X	X	X	X	X	X	X	X	24			16-32
Turbine Exhaust Gate Valve					X	X	X	X	72			-
Gear Box	X	X	X	X	X	X	X	X	20			20
Combustor and Injector	X	X	X	X	X	X	X	X	16	500	50	16-32
Hydrazine Pump (Low Pressure Tank)	X	X			X	X			90			<180
Upstream Purge Solenoid Valve (Latching)	X	X			X	X			25			24
Pump Outlet Supply Solenoid (Latching)	X	X			X	X			25			24
SYSTEM UNRELIABILITY TOTALS	564	535	472	443	602	600	510	508				

TABLE 12
COMPONENT/SYSTEM DESIGNATIONS

only closed during ground operations, its unreliability was not included in that the only failure mode would be external leakage such as lines and fittings.

2. Lines and fittings unreliability was not included.
3. Redundant regulators or redundant components were exclused because no redundancy was included in the cryogenic system study.
4. The purge gas system component unreliabilities were not included, consistent with the cryogenic study ground rule. Only the purge control valve unreliability is included.
5. Filter unreliabilities were excluded.
6. No hydraulic pump or alternator unreliability is included in the tabulations.
7. No pressure or temperature transducer unreliability is considered in the summation unless it is an integral part of the control servo loop.

Subsystem Comparison

Table13 is an analysis of the basic subsystems within overall systems. In other words, eliminating common components, the blowdown system advantage over the pump-fed system was quantified. Also the table shows the quantified advantage of the pulsing propellant valve over the modulating propellant valve.

Comparison With Cryogenic Systems

Comparison of the catalytic and thermal combustion systems shows very small differences. Some recent publications have indicated, however, that failure rates of catalyst beds are considerably higher than implied by the numbers used in the present study. Under such conditions the thermal bed would be strongly favored.

TABLE 13

SUBSYSTEM COMPARISON

1. BLOWDOWN SYSTEM AND PUMPED SYSTEM

a)	One Pump	+ 90
b)	One Latching Solenoid	+ 25
c)	One Solenoid Valve	+ 18
d)	One Hydrazine Start Tank	+ 48
e)	One Helium Fill Valve (Manual)	+ 4
f)	One Hydrazine Fill Valve	+ 4
g)	One Start System Supply Solenoid	+ 18
		+207
a)	Low Pressure Helium Tank (Not High)	- 1
b)	Low Pressure Single Stage Reg.	- 32
c)	Lower Press. Manual Valve	- 4
d)	Lower Press. Solenoid Valve	- 4
		- 41

Net Diff. + 166

Blowdown Advantage over Pump Fed System + 166

2. MODULATING PROP. VALVE AND PULSING PROP. VALVE

a) For Catalyst System (Advantage of Pulse Valve)

1)	Valve Difference	+ 29
2)	Detrimental to Catalyst	- 27
	Net Diff.	+ 2

Pulse Advantage over Modulating + 2

b)	For Thermal Bed Combustor	+ 29
1)	Valve Difference	+ 29
2)	Detrimental to Thermal Bed	0

Pulse Advantage over Modulating + 29

Quantitative comparisons of the storable propellant system unreliabilities with cryogenic system unreliabilities are shown in Table 14. The two systems (cryogenic and storable) can be compared in this fashion because consistent ground rules and rating procedures were used.

TABLE 14

STORABLE VS CRYOGENIC RELATIVE UNRELIABILITY COMPARISON	
Cryogenic Range	704 to 1105
Storable Range	508 to 602

The apparent advantage of the storable systems is due to several factors:

1. Pump power is taken off the gear box compared to separate drive for cryogenic system

		Amount
a)	No hydraulic motor	+ 90
b)	No hydraulic servo	+ 8
		+ 98

2. Blowdown system is for single propellant for storable and dual propellant for cryogenic system.
3. Cryogenic regulators and servo valves contributed significant unreliability.

A significant factor in the consideration of a storable system is the requirement for maintenance to achieve a high "in flight" reliability. As the number of flights is increased, reliability is significantly degraded on all hydrazine components requiring close tolerances, tight clearances, and sliding fits due to detrimental effects of propellant residuals. Hydrazine in extended use leaves a residue which can reduce valve response times or in the extreme prevent valve movement. Achieving high reliability levels therefore is dependent on frequent maintenance with its attendant costs, and this aspect is discussed next.

Maintainability

The maintainability characteristics of the monopropellant systems were compared on a preliminary basis. Both scheduled maintenance (checkout, servicing, etc.) and unscheduled maintenance (corrective: replace, repair, etc.) were considered.

The monopropellant systems are expected to require significantly fewer unscheduled maintenance actions than cryogenic systems because the failure rate is less. This advantage is partially offset by increased average time for corrective action on the propellant subsystem caused by the potentially hazardous nature of residual propellants. The monopropellant advantage is amplified by a reduced spares inventory, especially of costly complex devices (regulators, etc.).

Scheduled maintenance is more expensive for the monopropellant systems. Because no other space shuttle subsystem would require hydrazine, certain unique non-recurring expenses must be attributed to a monopropellant APU. These include:

1. Facilities. APU-unique hydrazine loading and safing facilities and GSE would be required. It is assumed that for the most part, these are not now in the NASA inventory.
2. Support requirements would be mor extensive and costly in the areas of training, handbooks, and personnel skill levels because of the APU-unique propellants.
3. Safety requirements for hydrazine, while within state-of-the-art capabilities, are complex and inconvenient.

4. As a minimum, inspection (boroscope or direct visual) of the catalyst would be required each mission duty cycle. Replacement prior to completion of 100 starts (the NASA goal for the Space Shuttle) is expected, with replacement every flight possible.

Another possible contributor to high scheduled maintenance cost may be the necessity for extensive post-flight decontamination of propellant system components. Residual hydrazine can cause operational problems with controls components which incorporate close-tolerance fits between moving parts.

The hydrazine systems employed in current commercial aircraft APUs are for emergency operation only; they are not normally used. Aircraft APUs are generally gas turbine devices utilizing aircraft propellants.

Conclusions

1. Thermal-bed systems appear to have a reliability advantage over the cryogenic systems if they are not degraded by residual propellant.
2. Catalytic decomposition appears to present the greatest reliability risk because of the uncertainty of the catalyst bed reliability, especially for restart.
3. Blowdown systems are more reliable than pump-fed systems.
4. The pulsing valve offers no clear reliability advantage over the modulating valve. The pulsing valve would degrade the catalyst; effect on the thermal bed would be less detrimental. The modulating valve must incorporate close tolerance dynamic fits; this is potentially susceptible to degradation from residual propellant.

5. Maintenance costs for the storable propellant systems will be significantly higher than for the cryogenic systems. The primary cost penalties are

Special facilities and equipment required for the storable propellants

More extensive post flight decontamination that may be required for components subject to malfunction caused by residual propellants

Higher cost of corrective maintenance per action on the propellant subsystem, necessitated by safety precautions.

CONTROL SYSTEM RELIABILITY COMPARISON

This section presents the results of a reliability analysis of two candidate propellant control systems:

1. Pulsing bipropellant valve to introduce oxygen and hydrogen to the turbine combustor along with an associated turbine speed feedback control loop

2. Modulating bipropellant valve and control loop.

The earlier reliability study for the purpose of system comparison considered only relative failure rate. It concluded that there was little difference between a modulating and pulsing system, and that difference was with the valves. This study explored the severity of effect as well as probability of failure and reevaluated the failure rates based on better definition of component conceptual design and APU mission duty cycle.

A failure mode and effect analysis was conducted to:

1. identify the predominant component failure modes

2. determine the effect of each failure on APU operation

3. categorize the criticality of the effect

4. apportion the estimated failure rates among the criticality categories to evaluate the relative probabilities of more severe failure modes.

Functional Analysis

In order to fully define the component functions in the entire control loop, functional block diagrams were created. Figure 98 is the functional diagram for the pulsing system and Figure 99 is the diagram for the modulating control system.

Although the valve designs are in the concept stage, some basic assumptions were made as to their configurations:

PULSING SYSTEM - BLOCK DIAGRAM

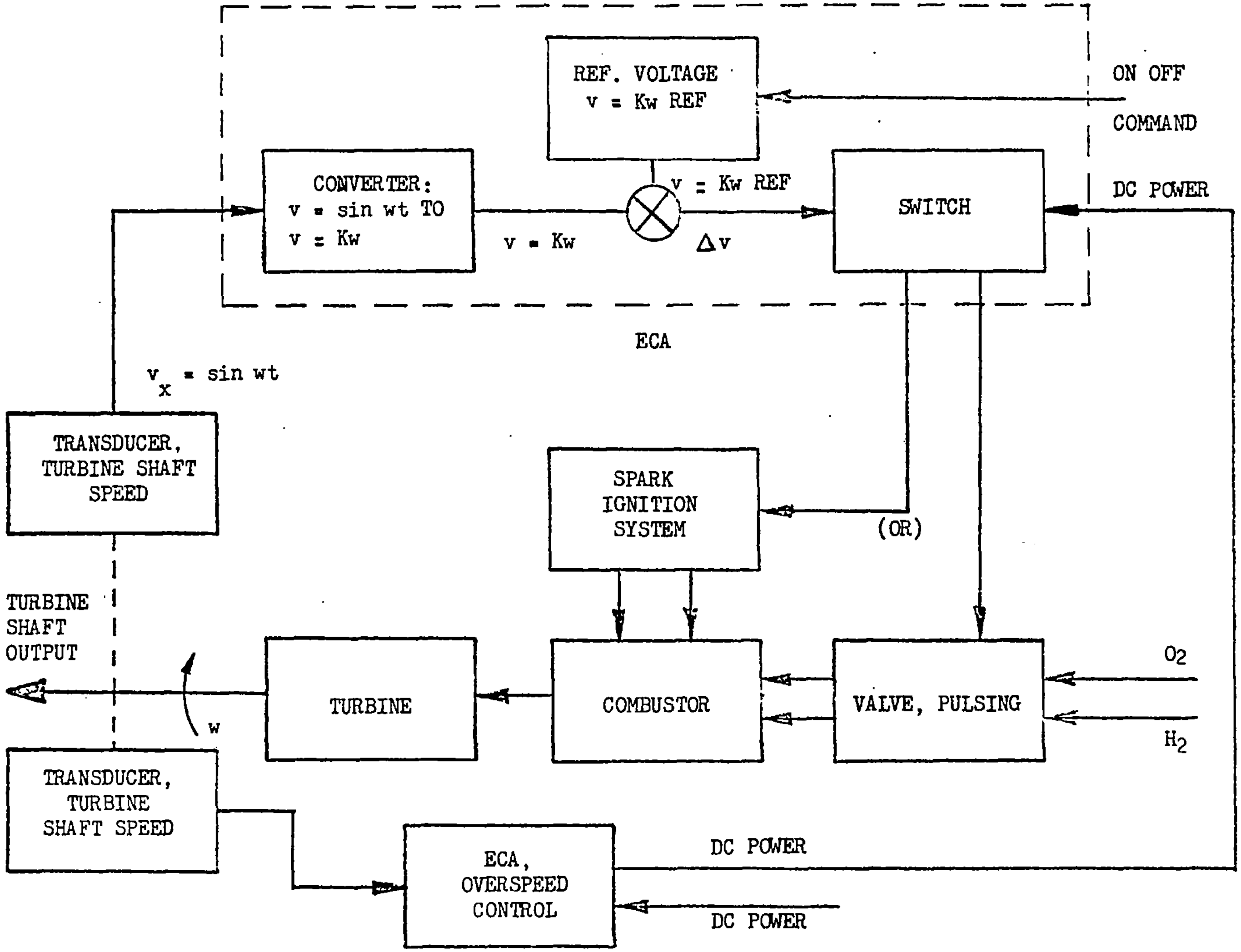


FIGURE 98

MODULATING SYSTEM - BLOCK DIAGRAM

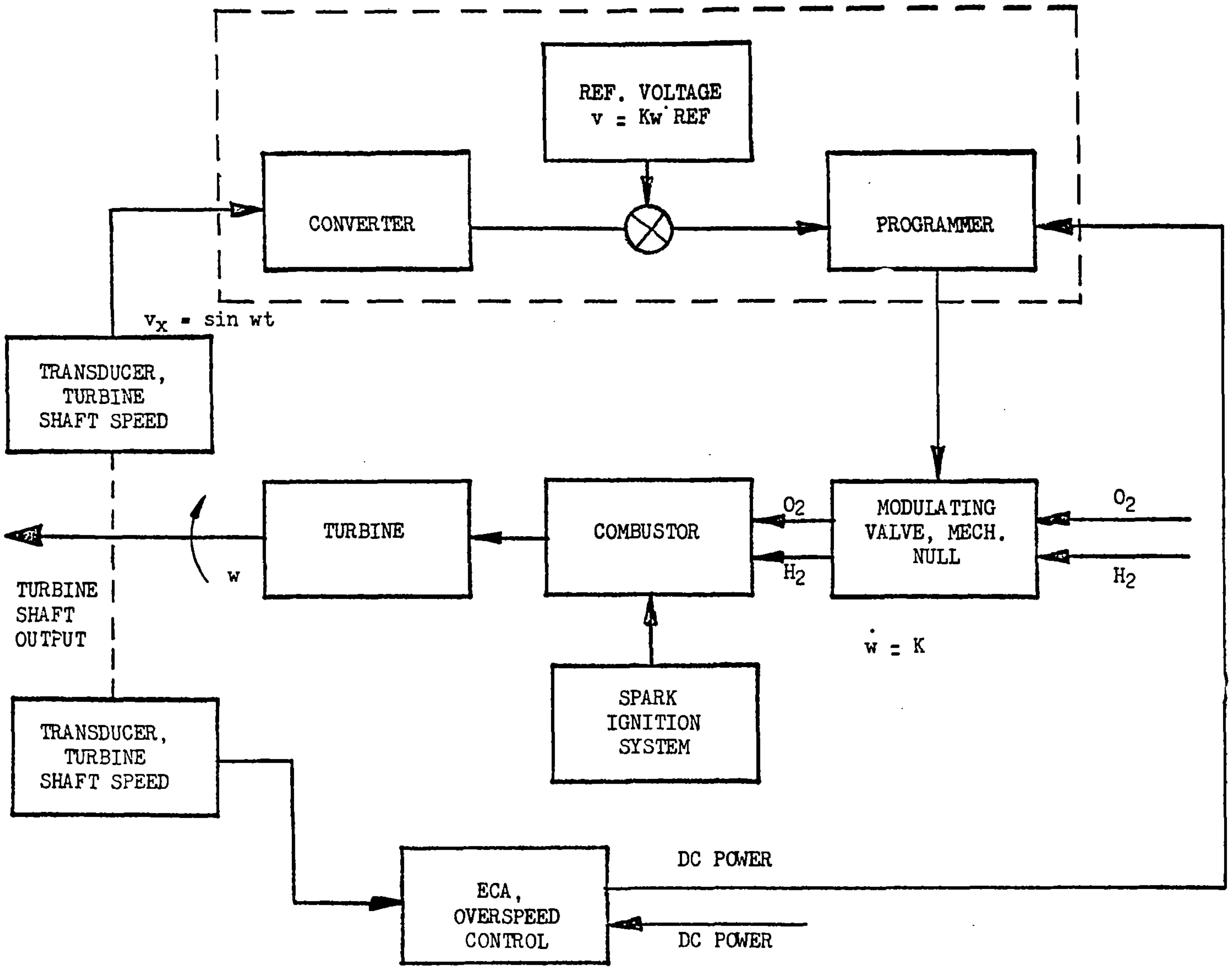


FIGURE 99

1. The valves would be a singly actuated bipropellant valve rather than two separate valves electrically linked.

2. The main flow control devices would be mechanically linked to provide positive mixture ratio control.

3. The pulsing valve would be a torque motor actuated flapper type valve similar to the RS-14 bipropellant valve.

4. The modulating valve would be a torque motor mechanically linked valve, flow resistance being inversely proportional to applied voltage. The valve would be spring-loaded closed. Flow control could be by pintles, spools, balls or cylinders, or gates.

Consideration was given to a modulating system which employed a two-stage valve that remained in a fixed position until commanded to translate. This system concept was rejected without detailed study because:

1. The valve and electronic control assembly (ECA) failure rates would be higher than for the other systems.

2. A predominance of failure modes would result in more critical failure effects.

A safety cutoff subsystem is necessary for both the modulating and pulsing subsystems in order to prevent turbine overspeed failure from continuing to turbine destruction. The safety cutoff subsystem must have the capability of shutting down the APU in the event of a permanent or transient condition allowing turbine overspeed or of a gearbox failure. The gearbox output transducer (perhaps the alternator) and associated ECA must be independent of the primary APU controls and should have the capability for manual reset to allow reuse of the APU in the event a transient condition was responsible for the overspeed cutoff. The overspeed cutoff circuit must also close the propellant supply solenoid valves in the event a valve open failure is responsible for the overspeed. Thus, the overspeed cutoff was considered as being part of the system rather than just a recommendation and the result was a lessening of the severity of the criticality of some of the failure modes.

This study also pointed out the need for determining the redundancy and failure detection/compensation equipment necessary to prevent catastrophic failures and/or significantly reduce system probability of failure.

Failure Mode and Effects Analysis

Each operating component in the control loop shown in Figs. 98 and 99was analyzed and the Failure Mode and Effects Analysis included the following:

1. Component functional description
2. Failure modes including typical causes for the failure mode and conditions in the system giving the appearance of the failure mode
3. Failure rate apportionment - (described below)
4. Criticality rating (Table 15)
5. Effect of the component failure mode upon the operation of the APU for each operational phase
6. Provisions for detecting the failure mode and methods for compensating or overriding the effects of the failure mode, as applicable.

The principal output of the Failure Mode and Effects Analysis is summarized in Tables 16 and 17 . Table 15 shows a summary reliability comparison based on the breakdown presented in Tables 16 and 17. Four levels of criticality are defined:

I. Loss with Hazard. Permanent disablement of APU with hazard to vehicle or crew.

II. Loss. Permanent disablement of APU, no hazard, including shutdown and failure to start or restart.

III. Shutdown. APU shutdown safety, but can be restarted (shutdown by overspeed overspeed protection subsystem).

IV. Performance. APU steady state or dynamic performance is outside specified limits, with total mission requirements not met in the extreme case.

TABLE 15

RELIABILITY COMPARISON

CRITICALITY OF FAILURE	RELATIVE PROBABILITY OF FAILURE (FAILURE RATE)	
	Pulsing	Modulating
I. Loss with Hazard . Permanent disablement of APU with hazard to vehicle or crew.	3.8	2.8
II. Loss. Permanent disablement of APU, no hazard, including shutdown and failure to start or restart.	16.6	10.2
III. Shutdown. APU shutdown safety, but can be restarted (shutdown by overspeed overspeed protection subsystem).	5.1	7.0
IV. Performance. APU steady state or dynamic performance is outside specified limits, with total mission requirements not met in the extreme case.	178.5	141.0
TOTAL	204	161

TABLE 16

RELIABILITY RATING

MODULATING SYSTEM

COMPONENT **Failure Mode**	FAILURE MODE RATING	CRITICALITY OF FAILURE: LOSS WITH HAZARD	LOSS	SHUTDOWN	PERFORMANCE LOSS
Transducer, Turbine Speed					
1. Sum of All Modes	2.0			2.0	
Transducer - TOTAL	2.0			2.0	
Electronic Control Assembly (ECA)					
1. Fail to Provide Power	1.0		1.0		
2. Power Level Low	3.0		0.5		2.5
3. Throttle Transition Shift-Low	2.0				2.0
4. Throttle Transition Shift-High	2.0		0.5		1.5
5. Non-Linear Response	2.0				2.0
6. Hysteresis	2.0				2.0
7. Control Voltage Shift-High	9.0			1.0	8.0
8. Control Voltage Shift-Low	9.0				9.0
ECA - TOTAL	30.0		2.0	1.0	27.0
Valve, Modulating					
1. Hard Over-Full Open	1.0			1.0	
2. Hard Over-Full Closed	3.0		3.0		
3. Inaccurate Position Control	40.0			2.0	38.0
4. Slow Opening	14.0				14.0
5. Slow Closing	8.0			1.0	7.0
6. External Leakage	16.0	0.1	0.9		15.0
7. Flow Restriction	2.0		0.1		1.9
Valve - TOTAL	84.0	0.1	4.0	4.0	75.9
Ignition System					
1. Fail to Spark	0.3		0.3		
2. Fail to Terminate Spark	0.1				0.1
3. Low Power	4.6		0.4		4.2
Ignition - TOTAL	5.0		0.7		4.3
Combustor					
1. Improper Mixing	12.0	0.5			11.5
2. External Leakage or Rupture	2.0	0.2			1.8
3. Extinguishment of Combustion			2.0		
Combustor - TOTAL	16.0	0.7	2.0		13.3
Turbine					
1. Low Efficiency	17.5	0.5	1.5		17.0
2. Fails to Start	1.5				
3. Structural Failure	1.0	1.0			
4. External Leakage	4.0	0.5			3.5
Turbine - TOTAL	24.0	2.0	1.5		20.5
MODULATING SYSTEM - TOTAL	161.0	2.8	10.2	7.0	141.0

TABLE 17

RELIABILITY RATING

PULSING SYSTEM

COMPONENT Failure Mode	FAILURE MODE RATING	CRITICALITY OF FAILURE			
		LOSS WITH HAZARD	LOSS	SHUTDOWN	PERFORMANCE LOSS
Transducer, Turbine Speed					
1. Sum of all Modes	2.0			2.0	
Transducer - TOTAL	2.0			2.0	
Electronic Control Assembly (ECA)					
1. Fail to Provide Power	1.0		1.0		
2. Power Level Low	3.0		0.5		2.5
3. Switch Locked Off	1.0		1.0		
4. Switch Locked On	1.0			1.0	
5. Drift in Output-Low	15.0				15.0
6. Drift in Output-High	15.0			1.0	14.0
ECA - TOTAL	36.0		2.5	2.0	31.5
Valve, Pulsing					
1. Fail to Open	3.0		3.0		
2. Opens Slowly	2.0				2.0
3. Fail to Close	1.0			1.0	
4. Closes Slowly	3.0			0.1	2.9
5. Internal Leakage	45.0	0.2	1.8		43.0
6. External Leakage	16.0	0.1	0.9		15.0
7. Flow Restriction	2.0		0.1		1.9
Valve - TOTAL	72.0	0.3	5.8	1.1	64.8
Ignition System					
1. No Ignition	1.8		1.8		
2. Fail to Terminate Spark	0.2				0.2
3. Low Power	52.0	1.0	5.0		46.0
Ignition - TOTAL	54.0	1.0	6.8		46.2
Combustor					
1. Improper Mixing	12.0	0.5			11.5
2. External Leakage or Rupture	4.0	0.5			3.5
Combustor - TOTAL	16.0	1.0			15.0
Turbine					
1. Low Efficiency	1.8	0.5			17.5
2. Fails to Start	1.5		1.5		
3. Structural Failure	0.5	0.5			
4. External Leakage	4.0	0.5			3.5
Turbine - TOTAL	24.0	1.5	1.5		21.0
PULSING SYSTEM - TOTAL	204.0	3.8	16.6	5.1	178.5

The failure modes for each of the operating components in the control loop were identified and a value rating was assigned, as shown in the first column in Table . This rating was then apportioned among the four levels of criticality. From this it is possible to assign a relative probability of system failure for each of the levels, (total) as shown in Table . This was done for both the pulsing and the modulating systems to permit an overall comparison to be made. As shown in Table 15 the modulating system was adjudged superior to the pulsing system based on a comparison of the relative failure rates for the four criticalities.

The relative failure rates used for each component (valve, ECA, transducer, etc) are compatible with those of the original trade study. As a result of this analysis, however, some of the rates were adjusted prior to making the final summation. Failure ratios which were changed from the previous study are listed below accompanied with the reason for change.

1. PULSING SYSTEM ECA - The pulsing system ECA was increased to 36 while holding the modulating system to the previous 30 because of difficulty in maintaining constant DC reference voltages in transient conditions associated with the constant pulsing system.
2. IGNITION SYSTEM - The modulating system ignition device was reduced to a relative rate of 5 while holding the pulsing ignition at 54. The difference was believed necessary due to the 1000:1 ratio of operating pulses (pulse system to modulating system).

Conclusions

It was concluded that the pressure modulating system is somewhat more reliable than the pulse modulating system. The difference is not sufficient to warrant choosing the pressure modulating system. Other factors such as system weight, cost, etc. must be considered simultaneously and they may easily outweigh the reliability difference. Also influencing the comparison made here is the status of the design. Neither control system has yet been defined in detail at the component level. Significant variations in reliability are possible due to component choices and redundancy. No redundancy was assumed in the present analysis. An example of application of redundancy would be use of two ignition systems. This adds little weight and cost but would eliminate about one fourth of the relative unreliability of the pulsed system while hardly affecting the pressure modulated systems. The net result of this redundancy would be to make the two systems nearly equal in relative unreliability.

In summary, the final choice between the pulse and pressure modulated systems must be based on more refined estimates of reliability as well as on system factors such as system weight, operational characteristics, etc.

SYSTEM RELIABILITY CONSIDERATIONS

The reliability of alternative subsystems and systems was discussed above. This section considers system aspects.

If the relative unreliabilities for the systems are converted to estimated reliability, the results are somewhat below the desired level. This should not be surprising because no efforts have het been made to optimize the components or systems from a reliability standpoint. Further reliability analysis is planned for Phase II.

One of the first steps to be taken will be to extend the Failure Mode and Effects Analysis to cover the rest of the APU system. (Only the control subsystem has been done so far.) Results of this will highlight component areas where design and development effort might fruitfully be applied to improve specific components.

Application of redundancy in some areas will prove to be desirable. Detained study of means of sensing failures will aid in avoiding potentially catastrophic results. Sensing can also be used in some instances to detect incipient failures during ground checkout, thereby allowing repairs to be made.

In summary, it appears that an adequate reliability level over the design life of the APU can be attained. Effort in Phase II will be directed toward finding and assessing means of improving reliability.

RELIABILITY

REFERENCES

(a) SR-QUAL-65-63, "MFSC Missile and Space Booster Failure Rates", dated December 30, 1965.

(b) SSE 4.4.1 MC-09, "Maintainability Analysis - Projected SSME Problem Rates for Establishment of Unscheduled Maintenance Actions", dated August 26, 1970.

(c) "Reliability Engineering Data Series - Failure Rates", by D. R. Earles and M. F. Eddins, dated April 1962 (Avco Corporation Research and Advanced Development Division, Wilmington, Massachusetts).

APPENDIX A
PRELIMINARY ANALYSIS-PHASE IA

In the Phase IA study, the primary candidate systems were synthesized on the basis of preliminary system and component evaluations. Various component oriented combinations were evaluated for each of the three major subsystems, i.e., Propellant Feed, Turbo Power Unit and Power Controls. The Propellant Feed Subsystem includes the necessary tankage and pressurizing or pumping equipment as well as the propellant conditioning subsystem. The subsystems were chosen and combined into various system combinations as shown in Fig. A1 to allow comparative evaluation for selection of the "best" combination of components and subsystems. The propellant systems chosen are representative of the wide range of vehicle influenced propellant supplies which may be available.

The evaluation is intended to include a full assessment of the penalty associated with selection of a particular propellant supply. For example, if a vehicle subcritical hydrogen supply is evaluated (as in Systems A-1, 2, 3, 4, 5), the portion of the large propellant tank used (SSE tankage) is charged to the system.

SYSTEMS

Three primary system types were evaluated:

System A - Pumped LH_2*

System B - Pressurized Supercritical H_2*

System C - Low Pressure Gaseous H_2/O_2

* In all cases, the O_2 tankage was taken as pressurized-supercritical.

SYSTEMS EVALUATED

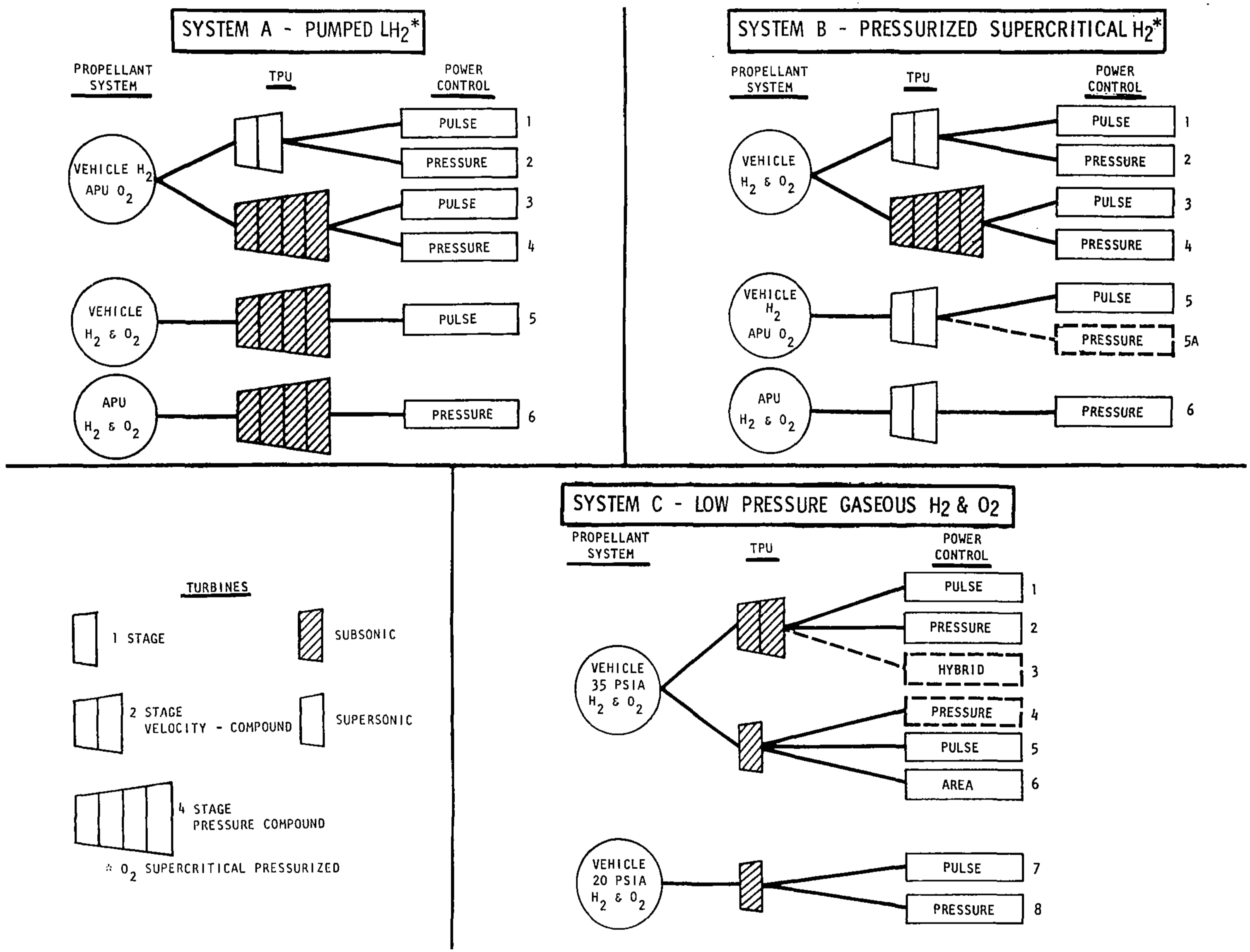


FIGURE AI

The mission profile used for the preliminary study is shown in Fig. A2. The power flow for the corresponding peak, mode and idle conditions are shown in Fig. A3. A simplified road map identifying the various aspects of the most promising candidate systems is shown in Fig. A4. Following selection of these systems from the 21 systems evaluated (Fig. A1) the remainder of Phase I effort is devoted to selecting the best subsystems comprising a single system.

System Results

Figures A5, A6 and A7 present some key system optimization results obtained. Optimum supply pressure (P_T) and pressure rations (P_T/P_e) can be selected for System A and B for various combinations including pressure modulation and pulse power control for two-stage supersonic velocity compound and four-stage subsonic pressure compounded TPU's. The results are representative of data generated through use of a direct access digital computer system optimization program. This program utilizes the various component performance and weight characteristics generated under the component screening portion of Phase I, along with the Statement-of-work (SOW) mission profile as per Fig. A2. The program was constructed in a flexible manner so that input changes can easily be implemented, i.e. mission profile modifications, vehicle application changes, orbiter tankage conditions as compared to booster, or component characteristics such as off design turbine performance. Figure A7 illustrates the low pressure gas system optimization.

SOW MISSION PROFILE

PHASE 1A STUDY

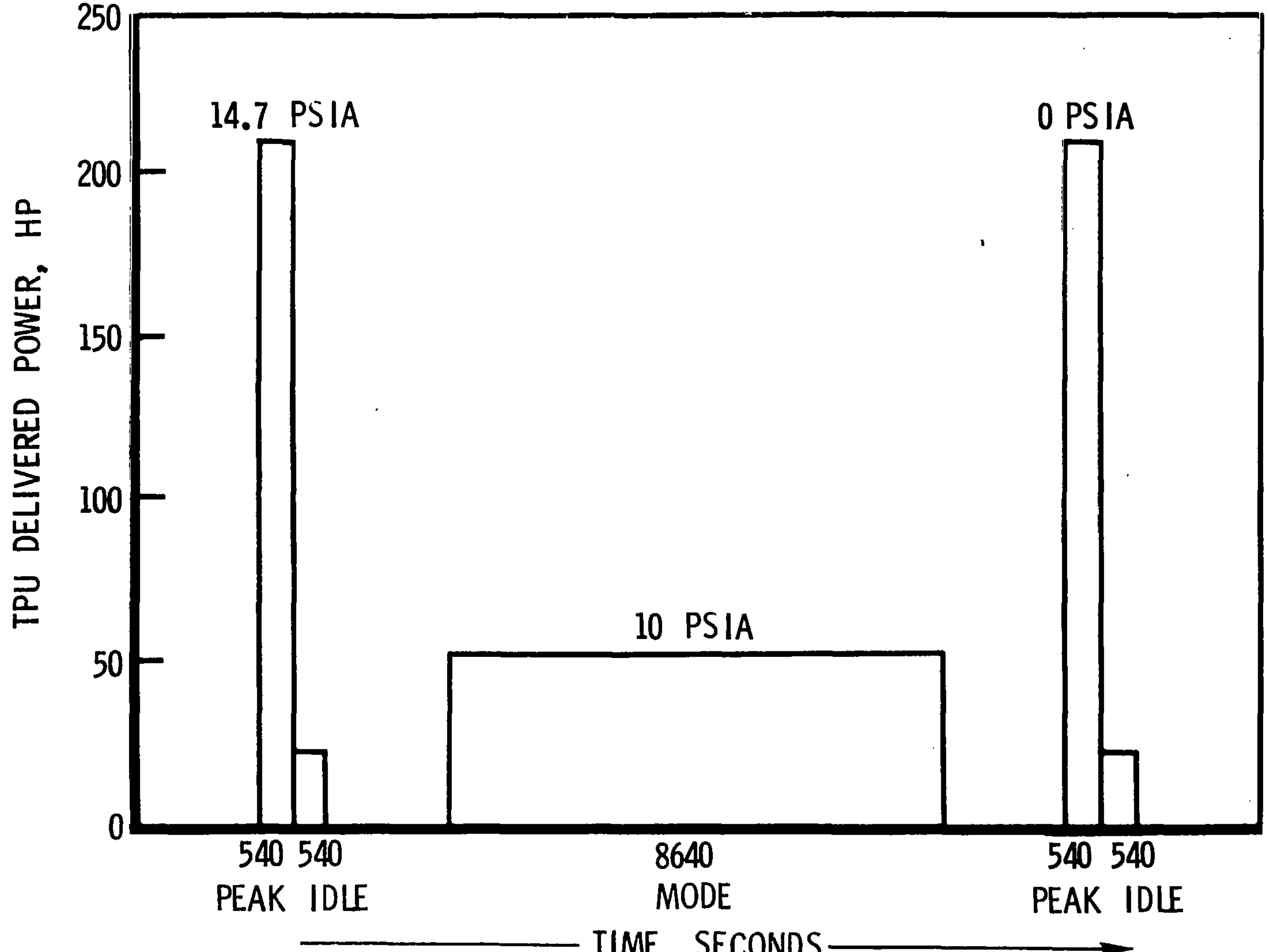


FIGURE A2

TYPICAL APU POWER FLOW

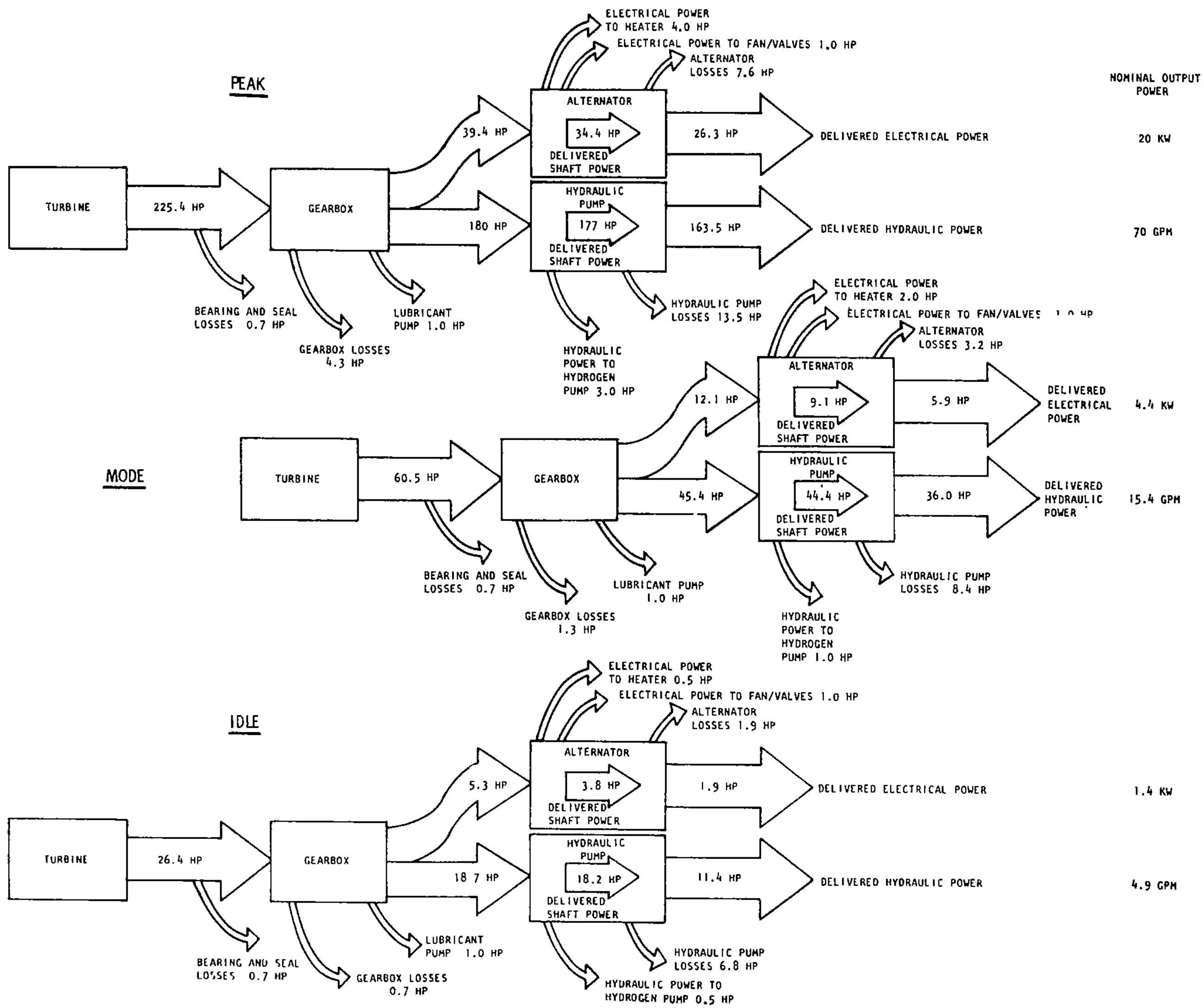


FIGURE A3

EVALUATION TECHNIQUE

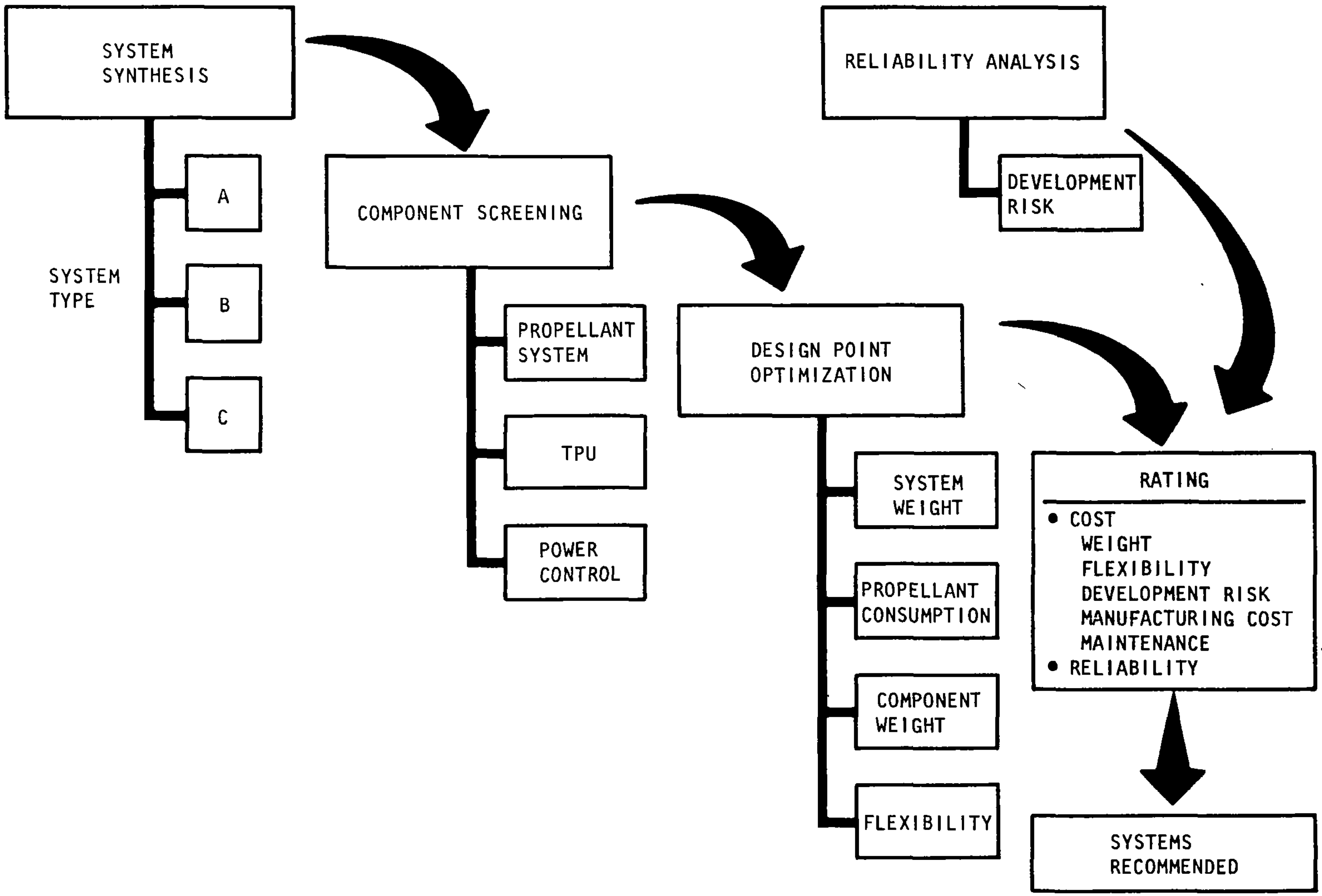


FIGURE A4

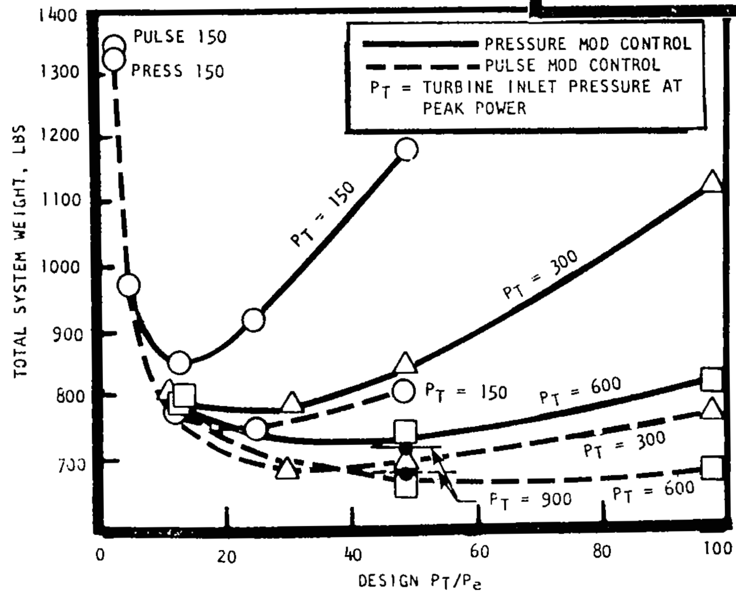
DESIGN OPTIMIZATION
SYSTEM A

SOW MISSION PROFILE
(BOOSTER)

WEIGHT INCLUDES
- 5% PROPELLANT RESERVE
- 5% ULLAGE
- 95% EXPULSION EFFICIENCY

ALL COMPONENTS LESS
HYDRAULIC PUMP AND ALTERNATOR

SYSTEM A-1, A-2
2-STAGE VEL. COMP. TURBINE
PUMPED HYDROGEN-VEHICLE (SSE)
SUPERCRITICAL OXYGEN-APU



SYSTEM A-3, A-4, A-6
4-STAGE PRESS. COMP. TURBINE
PUMPED H_2 - SSE INTEGRATED
SUPERCRITICAL O_2 - APU

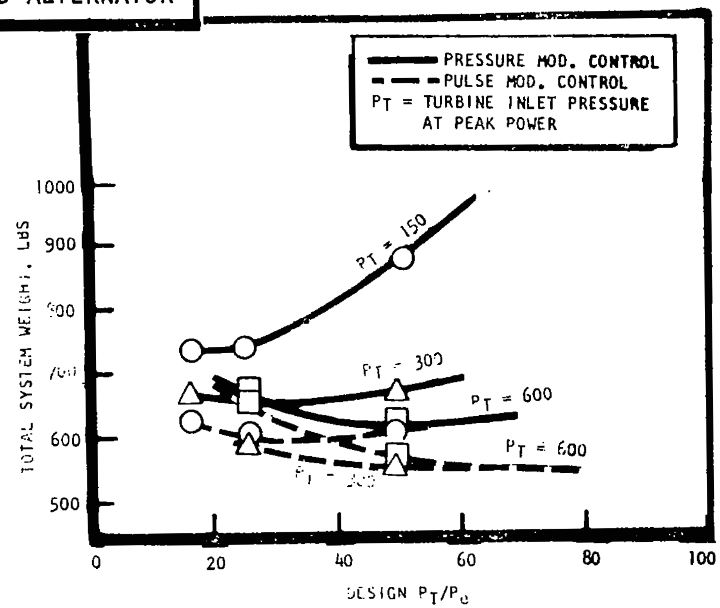


FIGURE A5

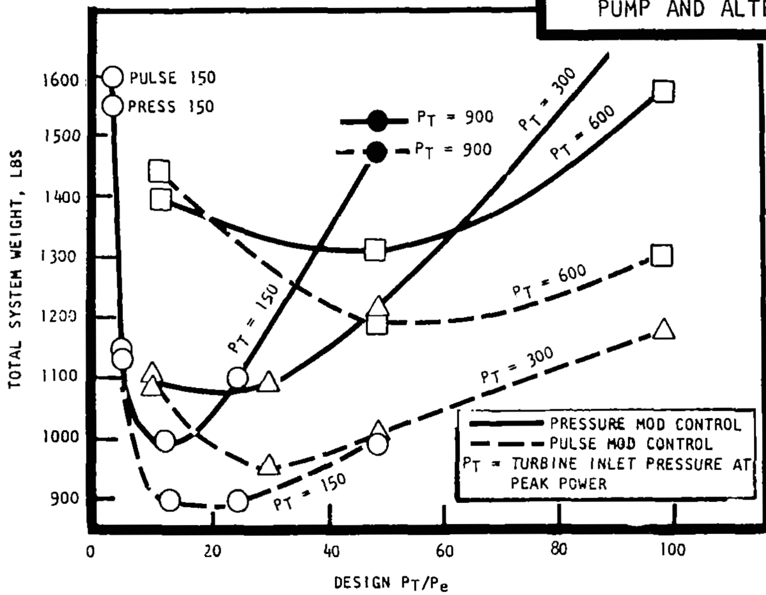
DESIGN OPTIMIZATION
SYSTEM B

SOW MISSION PROFILE
(BOOSTER)

WEIGHT INCLUDES
5% PROPELLANT RESERVE
5% ULLAGE
95% EXPULSION EFFICIENCY

ALL COMPONENTS LESS HYDRAULIC PUMP AND ALTERNATOR

SYSTEM B-1, B-2
2-STAGE VEL. COMP. TURBINE
SUPERCRITICAL HYDROGEN-RCS INT.
SUPERCRITICAL OXYGEN-RCS INT.



SYSTEM B-3, B-4
4-STAGE PRESS. COMP. TURBINE
SUPERCRITICAL H_2-RCS INTEGRATED
SUPERCRITICAL O_2-RCS

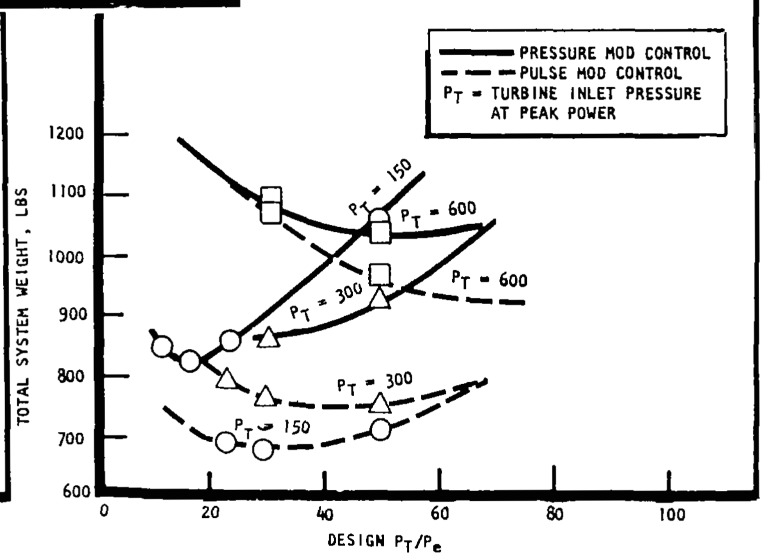


FIGURE A6

DESIGN OPTIMIZATION
SYSTEM C

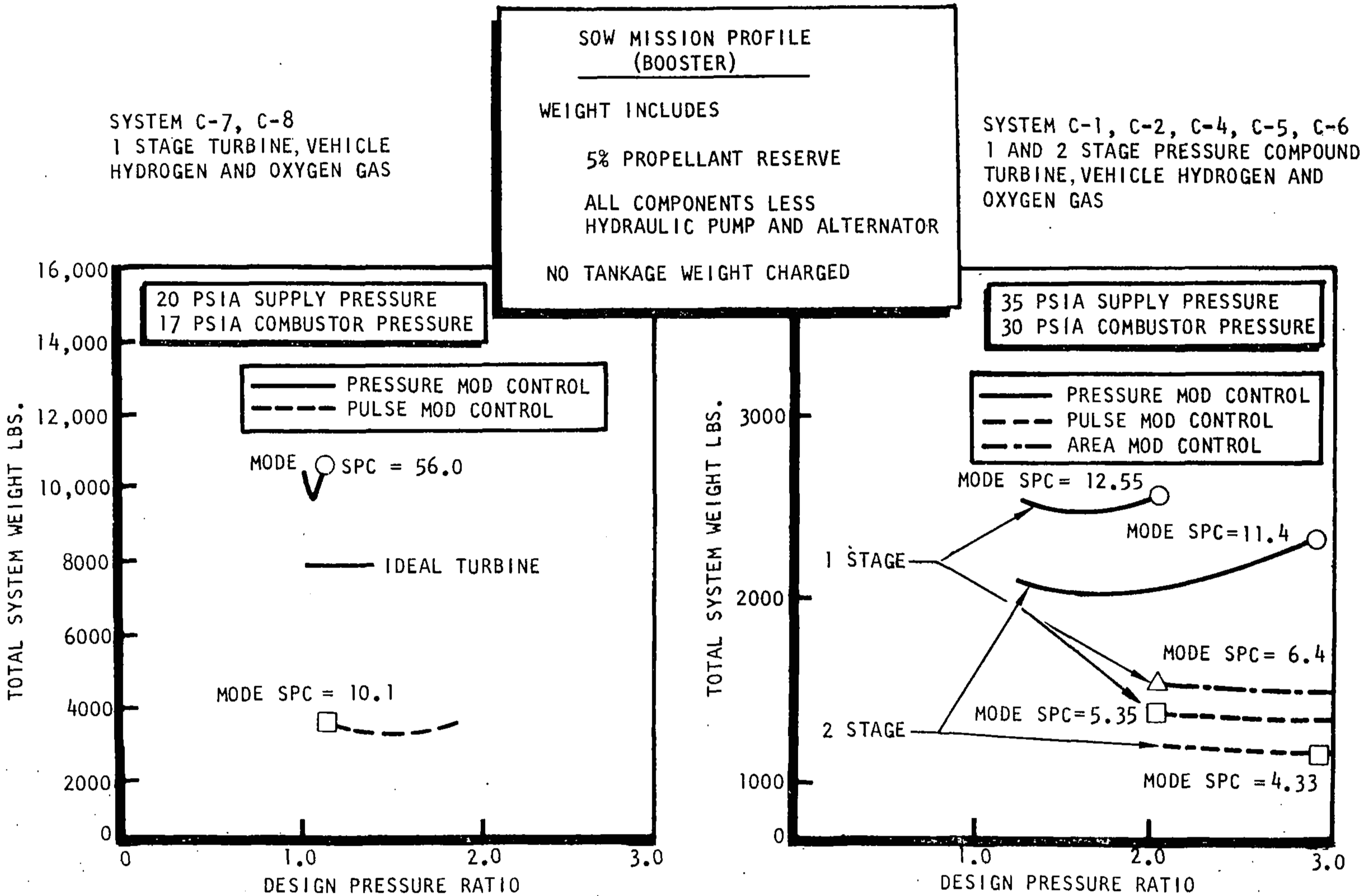


FIGURE A7

For this analysis, all the gaseous propellant used plus 5% reserve was assumed chargeable to the APU system.

Figures A8 and A9 illustrate the propellant portion of the system weights associated with the various systems.

Figure A10 shows how System A and B are weight optimized for various power control and turbine types. System A, the hydrogen pump fed system, tends to optimize at medium to high turbine inlet pressures while System B, the hydrogen supercritical storage system, optimized at the lowest pressure consistent with keeping the hydrogen stored supercritically. It should be noted that the entire weight penalty associated with supercritical storage for the booster vehicle is included in this optimization*. Optimization with various possible booster tanks was conducted, but the integration with possible RCS tanks was used for illustration in Fig. A10. Figure A11 compares system A, B and C for typical tankage configuration, and Fig. A12 shows specific propellant consumption for some representative systems at the most important mission conditions.

The sensitivity of the systems to changes in total energy requirements associated with possible mission profile changes was evaluated and is illustrated in Fig. A13 and A14. Booster conditions were improved on the optimization. For orbiter profiles, the tankage would tend to be heavier on a specific weight basis due to the longer storage requirements.

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* Tankage Support Structures (mounting brackets, etc.) were not included due to their dependence on concurrent vehicle studies in progress.

PROPELLANT WEIGHT OPTIMIZATION

SYSTEMS A AND B

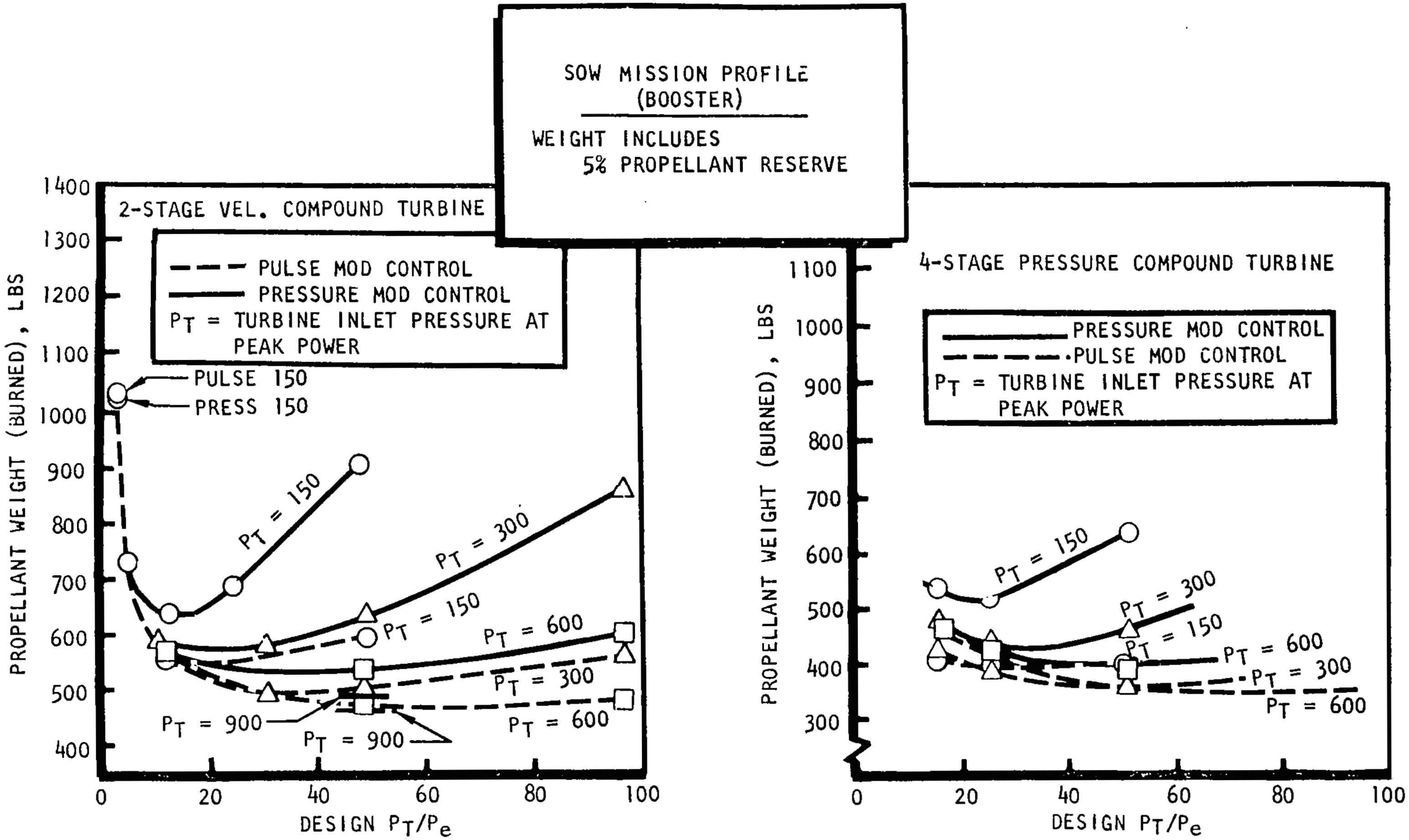


FIGURE A8

PROPELLANT WEIGHT SUMMARY

SYSTEM C-LOW PRESSURE-35 PSIA SUPPLY

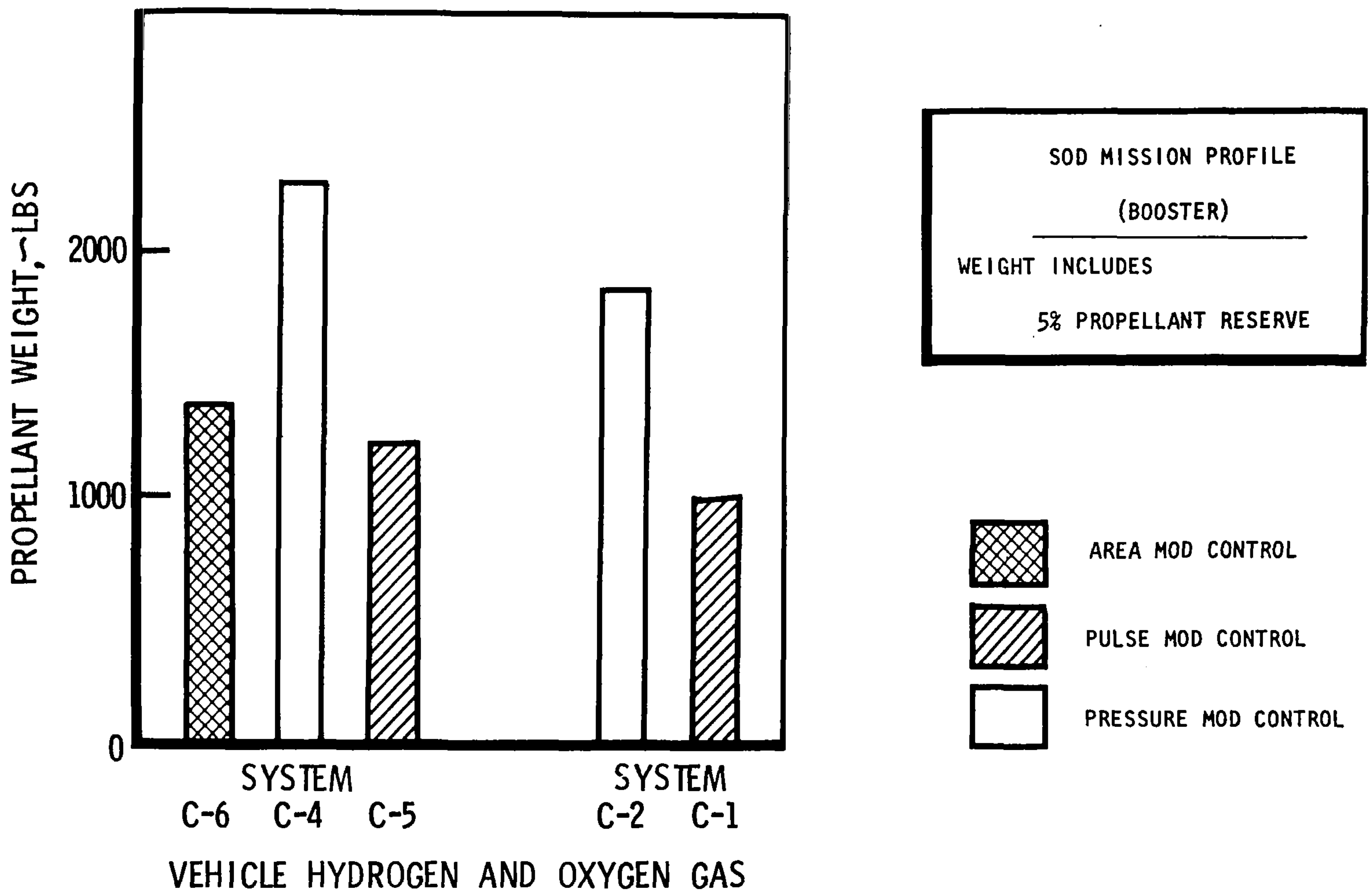


FIGURE A9

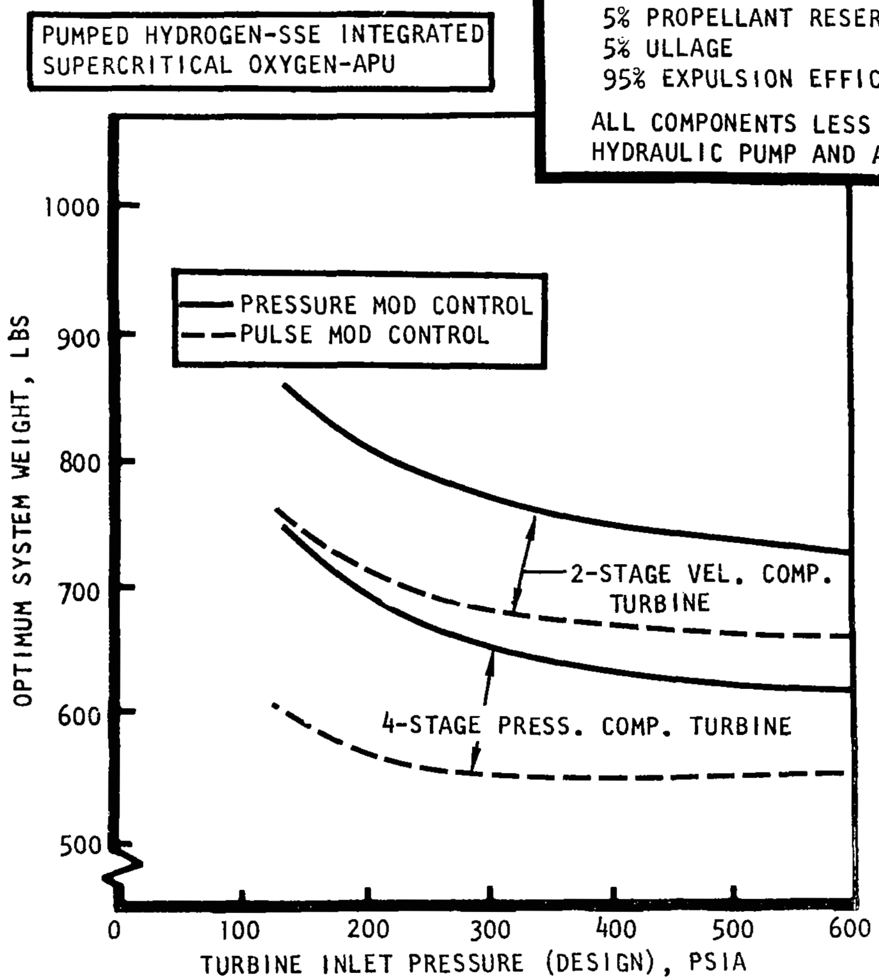
OPTIMUM SYSTEM WEIGHT

SOW MISSION PROFILE
(BOOSTER)

WEIGHT INCLUDES
5% PROPELLANT RESERVE
5% ULLAGE
95% EXPULSION EFFICIENCY

ALL COMPONENTS LESS
HYDRAULIC PUMP AND ALTERNATOR

SYSTEM A



SYSTEM B

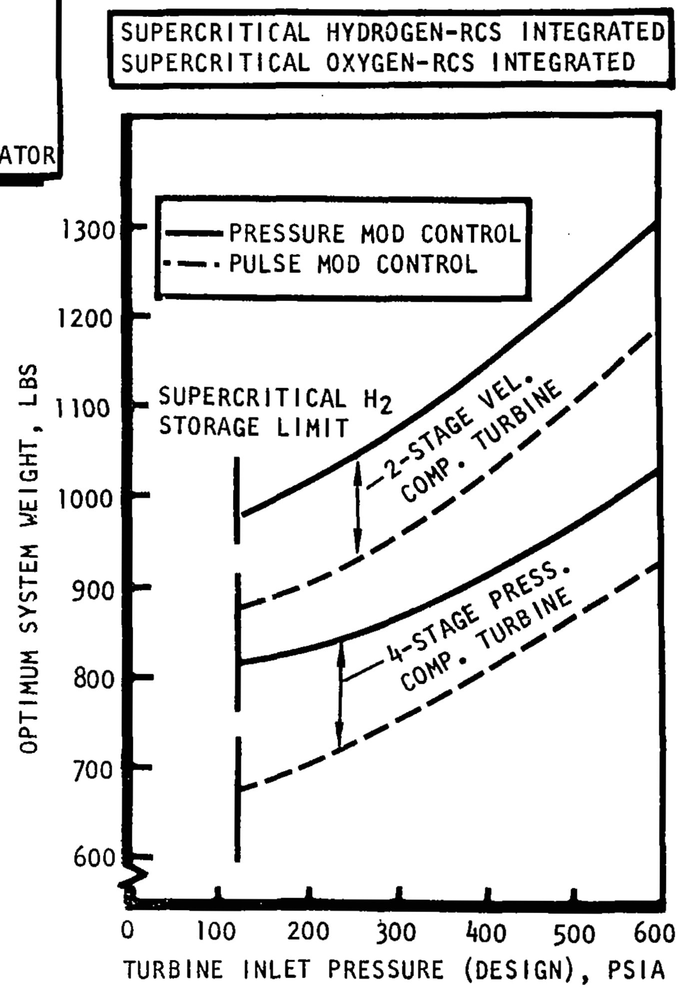


FIGURE A10

TYPICAL OPTIMUM SYSTEM WEIGHT COMPARISON
SYSTEMS A, B AND C

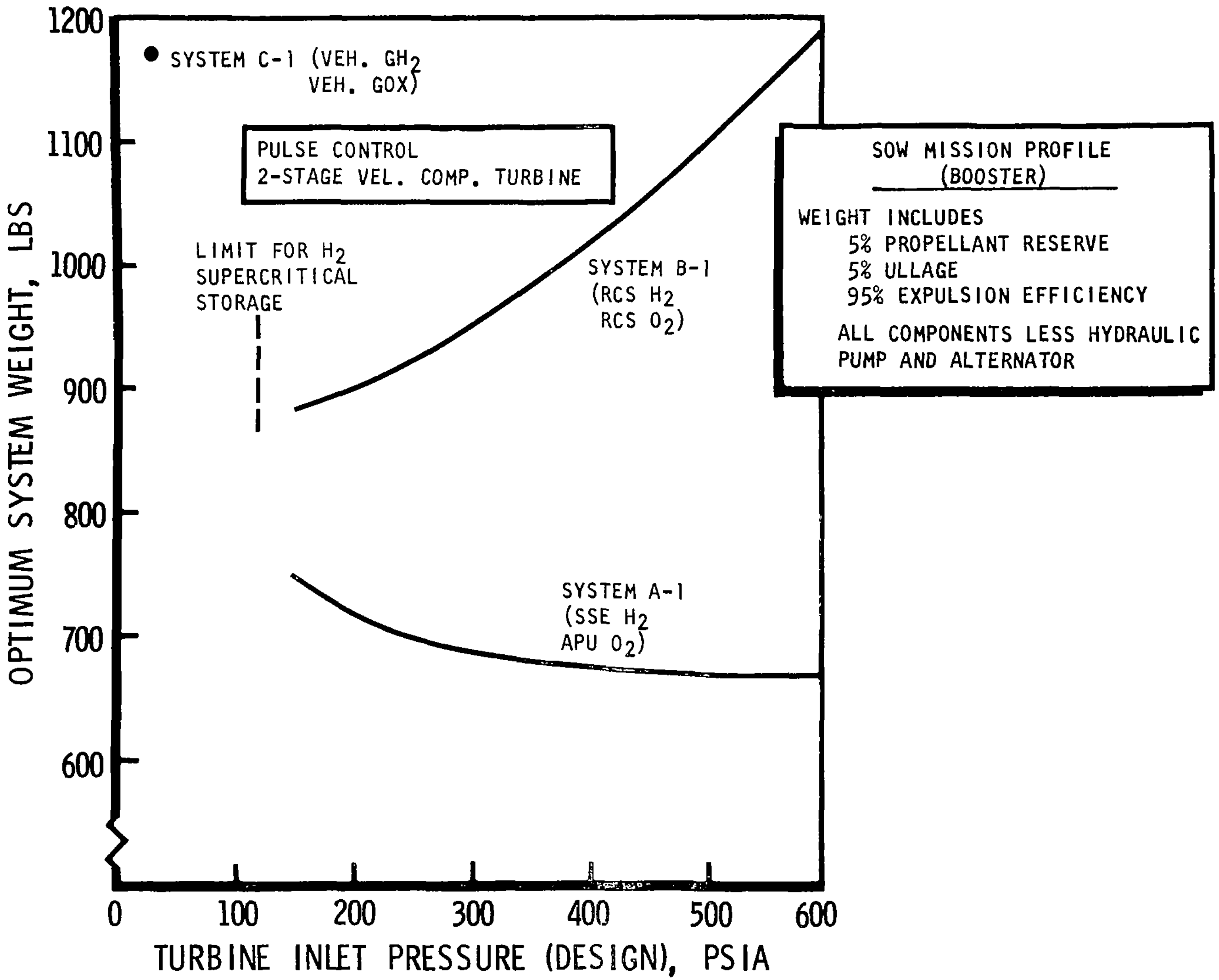
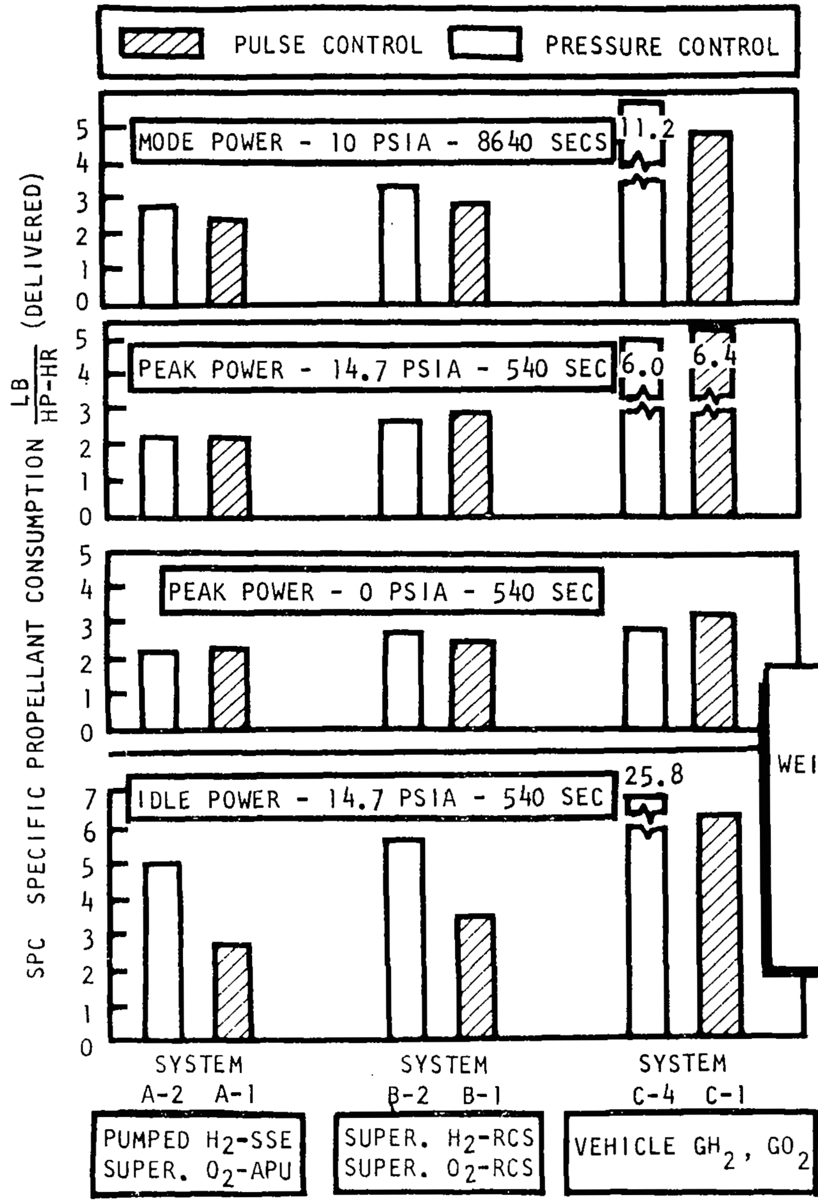


FIGURE A11

TYPICAL OPTIMUM SYSTEM SPECIFIC PROPELLANT CONSUMPTION COMPARISON



SOW MISSION PROFILE
(BOOSTER)

WEIGHT INCLUDES

5% PROPELLANT
5% ULLAGE
95% EXPULSION EFFICIENCY

ALL COMPONENTS LESS
HYDRAULIC PUMP AND ALTERNATOR

FIGURE A12

DESIGN OPTIMIZATION

SOW MISSION PROFILE (BOOSTER), 100 % AND 50 % DURATION

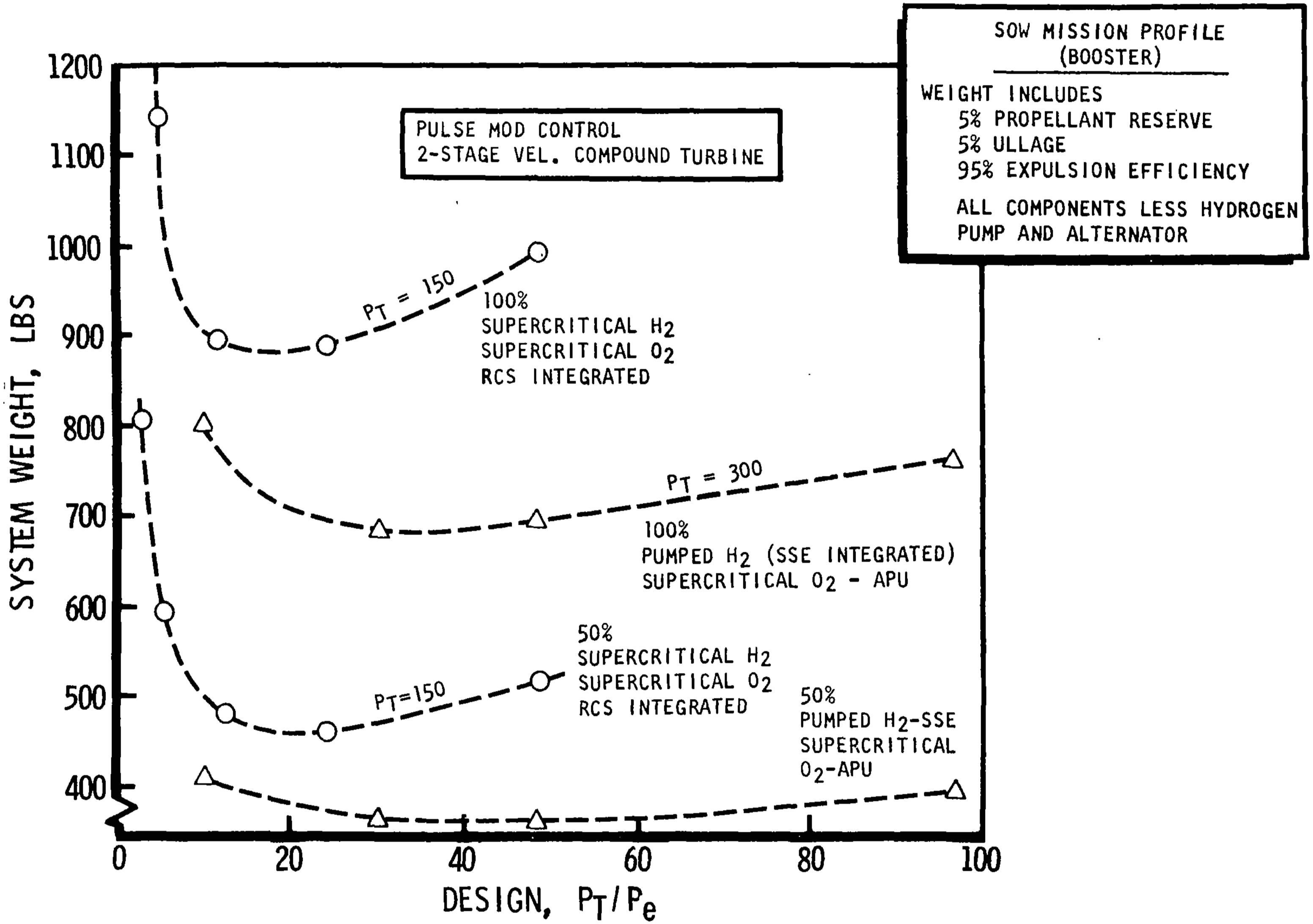
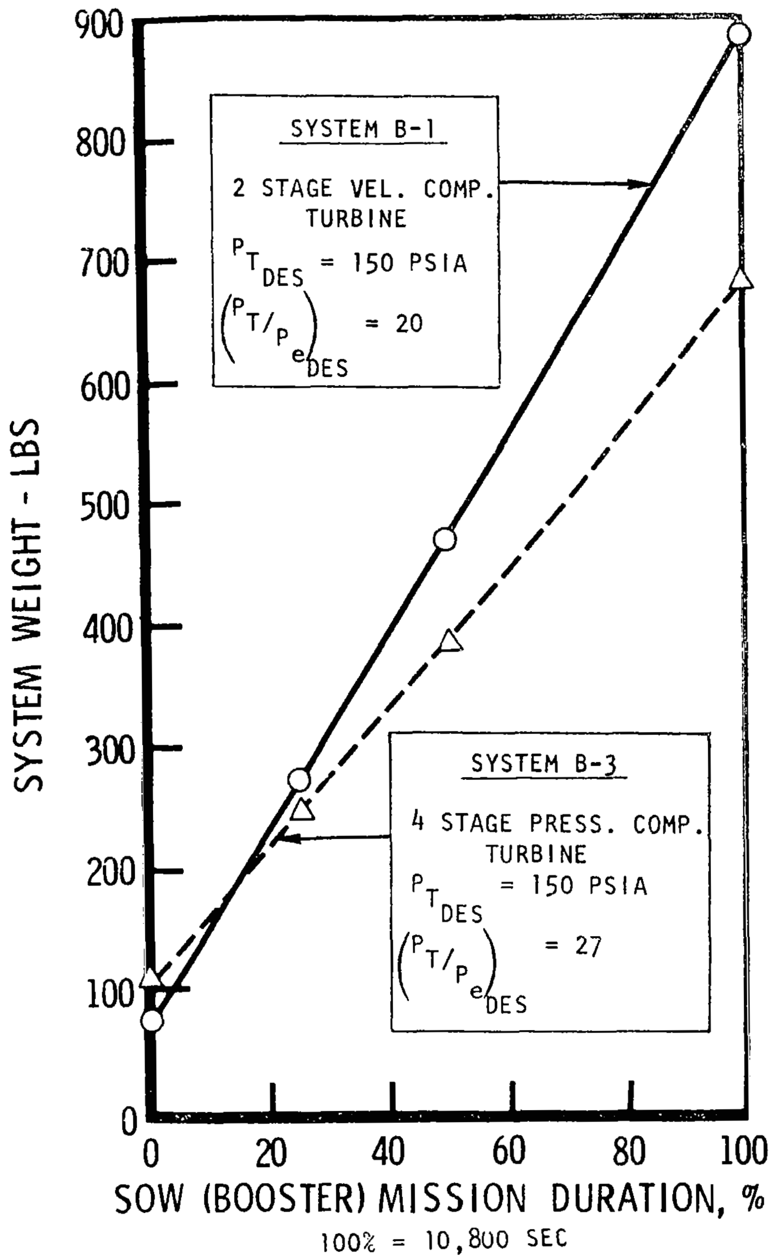


FIGURE A13



EFFECT OF MISSION DURATION ON SYSTEM WEIGHT

- SUPERCRITICAL H_2 - RCS INTEGRATED
- SUPERCRITICAL O_2 - RCS INTEGRATED
- PULSE MOD CONTROL

SOD MISSION PROFILE

(BOOSTER)

WEIGHT INCLUDES

5% PROPELLANT RESERVE

5% ULLAGE

95% EXPULSION EFFICIENCY

ALL COMPONENTS LESS HYDRAULIC PUMP AND ALTERNATOR

FIGURE A14

System Evaluation

The purpose of preliminary system evaluation was to provide a quantitative basis for selection of the most promising components, subsystems, and systems. A survey was conducted of the various criteria and techniques currently in use on the various Space Shuttle vehicle, engine and auxiliary system studies. On the basis of the survey and discussion held with the NASA program monitor, a rating technique was devised, as illustrated in Fig.A15, A16 and A17. The weighting as shown in Fig.A15 relates all items to cost and reliability. There are indications that in later work, all items including reliability can be put on a cost basis.

A strong emphasis (20%) was placed on flexibility. In referring to Fig. A17 it should be noted that items such as mission profile, power levels, turn down ratios, requirements for Orbiter/Booster commonality and propellant sources are all quite elastic at this stage of the overall Space Shuttle Vehicle Studies. For this reason, it is highly desirable to select APU designs which are "flexible" in their ability to accommodate to these vehicle/mission imposed changes. Figure A16 illustrates the ranges and technique used to quantity ratings for weight and reliability based on the spread in quantitative results described previously in the system synthesis work. Manufacturing cost was done in a similar manner. Figure A17 illustrates the multiple ballot technique used to quantify the three qualitative rating items, flexibility, development risk, and maintainability. Three systems oriented engineers, closely associated with the APU project, were given

RATING TECHNIQUE

ITEM			RATING
COST			65
	• WEIGHT	25	
	• FLEXIBILITY	20	
	• DEVELOPMENT RISK	10	
	• MAINTAINABILITY	5	
	• MANUFACTURING COST	5	
RELIABILITY			35
TOTAL POINTS			100

FIGURE A15

QUANTITATIVE RATING ITEMS

ITEM	FACTOR	RATING
WEIGHT 25 POINTS	500 TO 1500 POUNDS MINIMUM WEIGHT SYSTEM 550 POUNDS 3 LOW PRESSURE SYSTEMS ABOVE 1500 POUNDS	25 TO 0
RELIABILITY 35 POINTS	0 TO 1500 UNRELIABILITY SYSTEM RANGE 764 TO 1105	35 TO 0

FIGURE A16

QUALITATIVE RATING ITEMS

ITEM	CONSIDERATIONS		RATING
FLEXIBILITY 20 POINTS	• POWER CHANGE	LEVEL	3
		PEAK/MODE RATIO	3
	• MISSION CHANGE		4
	• ORBITER/BOOSTER COMMONALITY		6
	• FEED SYSTEM CHANGE		4
DEVELOPMENT RISK 10 POINTS	• PROPELLANT SYSTEM	TANKAGE	1
		FEED	3
		CONDITIONING	1
	• TPU	COMBUSTOR	1
		TURBINE	2
	• CONTROL SYSTEM		2
MAINTAINABILITY 5 POINTS	• MODULAR DESIGN ADAPTABILITY		1
	• MINIMUM COMPLEXITY		3
	• READILY MONITORED		1

MULTIPLE BALLOT TECHNIQUE TYPICAL FORM

MAINTAINABILITY 5 POINTS				
SYSTEM	MODULAR DESIGN ADAPTABILITY	MINIMUM COMPLEXITY	READILY MONITORED	TOTAL
	1	3	1	5
A-1	0.8	1.3	0.8	2.9
A-2	0.8	0.3	0.7	1.8
A-3	0.7	1.1	0.8	2.6
A-4	0.7	0.1	0.7	1.5
A-5	0.7	1.1	0.8	2.5
A-6	0.7	0.0	0.7	1.4
B-1	1.0	2.3	1.0	4.3
B-2	1.0	1.3	0.9	3.2
B-3	0.9	2.1	1.0	4.0
B-4	0.9	1.1	0.9	2.9
B-5	1.0	2.0	1.0	4.0
B-6	1.0	0.8	0.9	2.7
C-1	1.0	2.8	1.0	4.8
C-2	1.0	1.8	0.9	3.7
C-3	1.0	2.3	1.0	4.3
C-4	1.0	1.8	0.9	3.7
C-5	1.0	3.0	1.0	5.0
C-6	1.0	2.0	0.9	3.9

3 EVALUATIONS PERFORMED - RESULTANT RATINGS AVERAGED

FIGURE A17

ballots similar to the "typical" ballot of Fig.A17 and were asked to rate each system on each item. The results of these ballots were then tabulated, inspected for significant deviations and averaged to obtain the necessary quantitative ratings.

The ratings for each system are summarized in Fig. A18, A19 and A20. Data is also presented for the actual system optimized weight and for the design conditions, i.e. turbine inlet pressure (P_T) and pressure ratio (P_R).

Fig. A21 summarizes the ratings for each of the systems for convenient comparisons and presents the optimum weight as well.

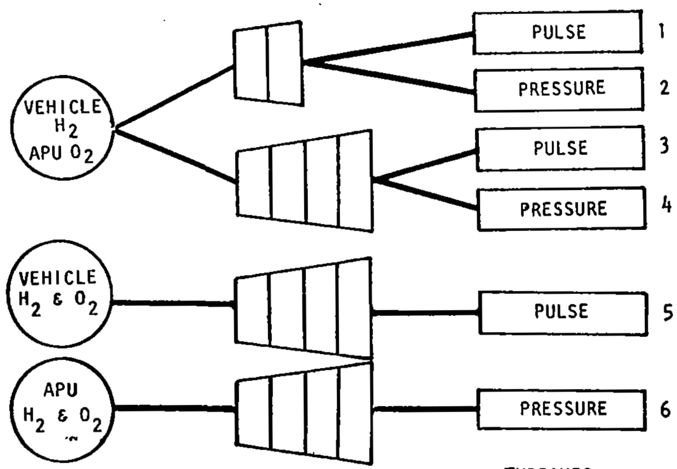
The system evaluation was conducted in a manner to allow certain key comparisons to be made on a consistent basis. Fig. A22 shows which of these comparisons have been carried out to date. The results of these comparisons and some key conclusions are shown in Figs. A23 through A27.

PROPELLANT FEED SYSTEM

The propellant and feed system design considerations included in the study have been:

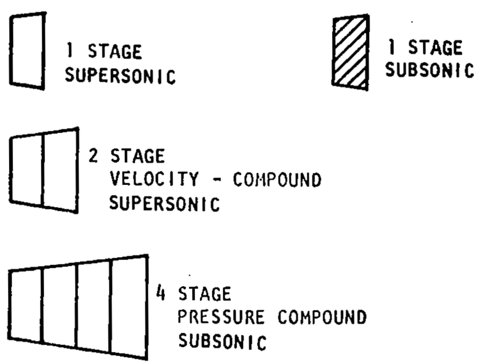
TANKAGE SUBSYSTEM	SEPARATE APU TANKAGE (1) SINGLE APU (2) MULTIPLE APU	VS	INTEGRATED TANKAGE (1) SSE (2) RCS

SYSTEM A—(PUMPED LH²)* EVALUATION



	Power control	Weight	Conditions P_t	Conditions PR	Rating: Weight (25)	Flexibility (20)	Dev. Risk (10)	Mfg. Cost (5)	Maint. (5)	Reliability (35)	Total (100)
1	PULSE	685 (665)	300 (600)	30 (50)	20.2	16.9	5.9	4.6	2.8	11.3	61.7
2	PRESSURE	730	600	40	19.3	11.9	5.7	4.5	2.0	9.9	53.3
3	PULSE	550	300	50	24.8	16.9	4.5	4.3	2.5	11.1	64.1
4	PRESSURE	620 (660)	600 (300)	50 (25)	22.1	12.0	4.9	3.7	1.8	9.7	54.2
5	PULSE	550	300	50	24.8	16.4	4.9	4.7	2.7	12.9	66.4
6	PRESSURE	650	600	50	21.2	13.0	4.8	2.5	1.3	9.2	52.0

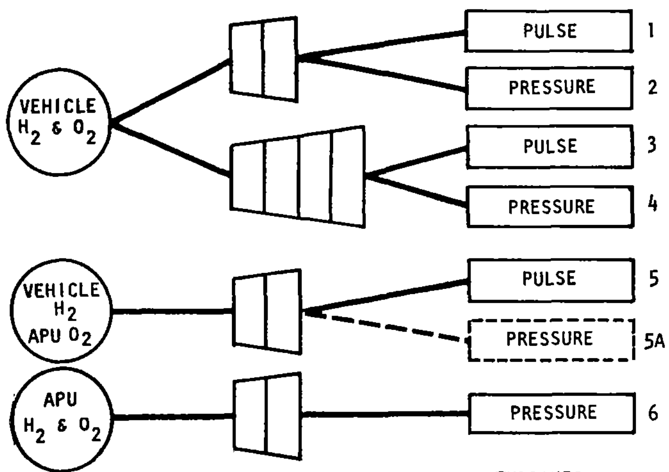
TURBINES



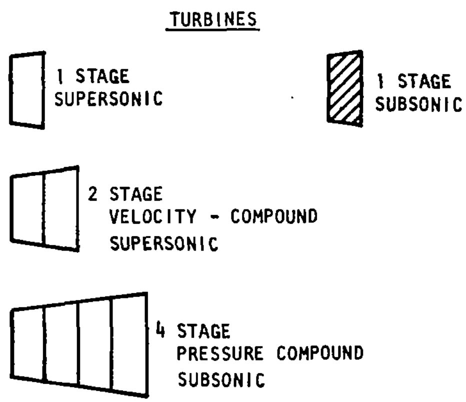
* O_2 SUPERCRITICAL PRESSURIZED

FIGURE A18

SYSTEM B-(PRESSURIZED SUPERCRITICAL H_2)* EVALUATION



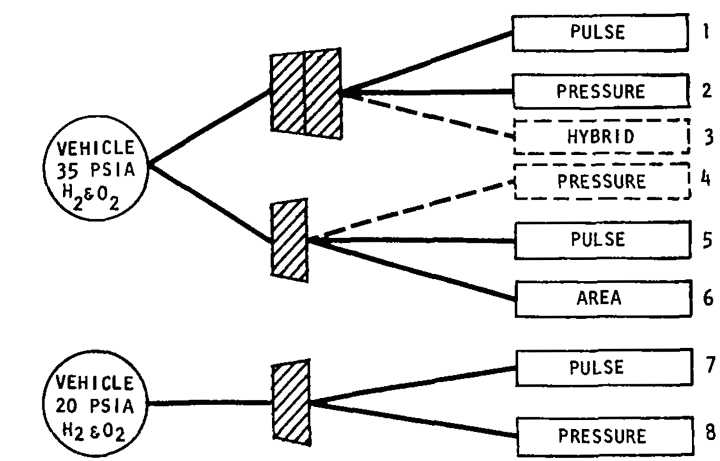
No.	WEIGHT	CONDITIONS P_t	CONDITIONS PR	RATING: WEIGHT 25	FLEXI-BILITY 20	DEV. RISK 10	MFG. COST 5	MAINT. 5	RELIA-BILITY 35	TOTAL 100
1	870	150	20	15.7	15.0	8.6	4.6	4.4	15.6	63.9
2	990	150	12	12.7	11.2	8.3	4.5	3.7	14.2	54.6
3	680	150	30	20.3	14.9	7.1	4.5	4.1	15.4	66.3
4	820	150	20	17.1	11.2	7.3	3.9	3.4	14.0	56.9
5	870	150	20	15.7	15.5	8.3	4.4	4.0	13.8	61.7
5A	990	150	12	--	--	--	--	--	--	NR
6	1000	150	12	12.7	12.7	8.0	0	3.0	10.7	46.9



* O_2 SUPERCRITICAL PRESSURIZED

FIGURE A19

SYSTEM C - LOW PESSURE GASEOUS H_2 & O_2



	PROPELLANT SYSTEM	POWER CONTROL	WEIGHT	CONDITIONS P_t	CONDITIONS PR	RATING WEIGHT 25	RATING FLEXIBILITY 20	RATING DEV. RISK 10	RATING MFG. COST 5	RATING MAINT. 5	RATING RELIABILITY 35	RATING TOTAL 100
1	VEHICLE 35 PSIA H_2&O_2	PULSE	1170	30	2.9	8.4	4.1	5.9	3.7	4.5	18.4	45.0
2	VEHICLE 35 PSIA H_2&O_2	PRESSURE	2330	30	2.9	NR	1.1	5.6	3.7	3.4	17.0	NR
3	VEHICLE 35 PSIA H_2&O_2	HYBRID	1170	30	2.9	8.4	4.5	4.9	3.4	3.8	16.9	40.1
4	VEHICLE 35 PSIA H_2&O_2	PRESSURE	2460	30	1.2	NR	---	---	---	---	----	NR
5	VEHICLE 35 PSIA H_2&O_2	PULSE	1350	30	2.9	3.2	3.8	6.3	5.0	4.6	18.5	42.0
6	VEHICLE 35 PSIA H_2&O_2	AREA	1490	30	3.3	0.2	2.7	3.9	5.0	3.3	16.9	32.0
7	VEHICLE 20 PSIA H_2&O_2	PULSE	3400	17	1.4	NR	---	---	---	---	18.5	NR
8	VEHICLE 20 PSIA H_2&O_2	PRESSURE	10000	17	1.1	NR	---	---	---	---	17.1	NR

TURBINES

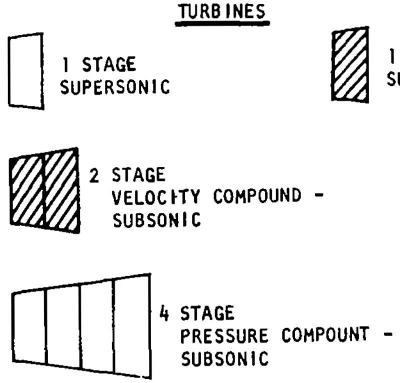
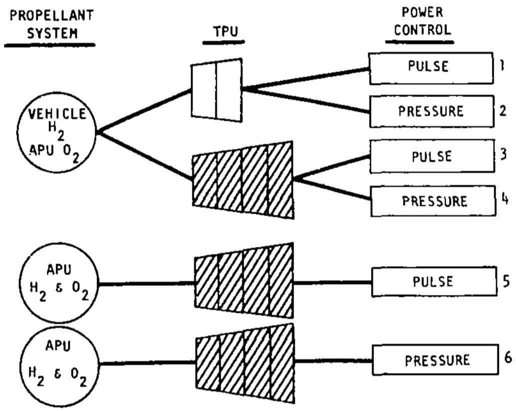


FIGURE A20

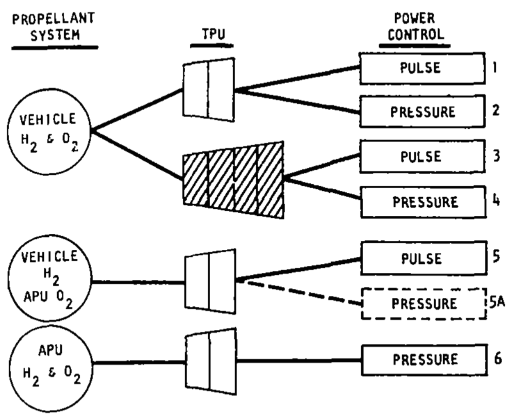
SYSTEM EVALUATION SUMMARY

SYSTEM A - PUMPED LH_2 *

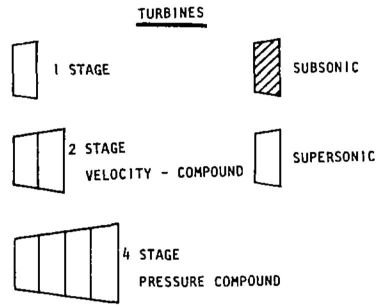


	POWER CONTROL	SYSTEM WEIGHT	RATING
1	PULSE	685 (665)	61.7
2	PRESSURE	730	53.3
3	PULSE	550	64.1
4	PRESSURE	620 (660)	54.2
5	PULSE	550	66.4
6	PRESSURE	650	52.0

SYSTEM B - PRESSURIZED SUPERCRITICAL H_2 *

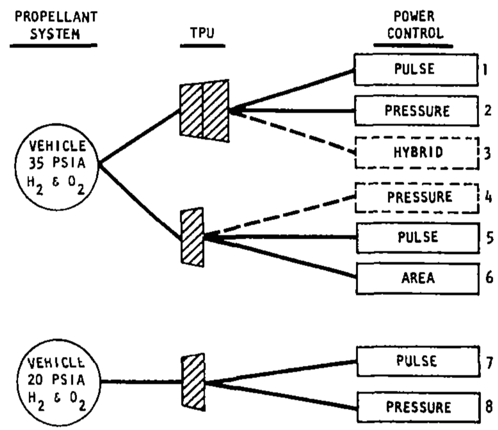


	POWER CONTROL	SYSTEM WEIGHT	RATING
1	PULSE	870	63.9
2	PRESSURE	990	54.6
3	PULSE	680	66.3
4	PRESSURE	820	56.9
5	PULSE	870	61.7
5A	PRESSURE	990	52.4
6	PRESSURE	1000	46.9



* O_2 SUPERCRITICAL PRESSURIZED

SYSTEM C - LOW PRESSURE GASEOUS H_2 & O_2



	POWER CONTROL	SYSTEM WEIGHT	RATING
1	PULSE	1170	45.0
2	PRESSURE	2330	NR
3	HYBRID	1170	40.1
4	PRESSURE	2460	NR
5	PULSE	1350	42.0
6	AREA	1490	32.0
7	PULSE	3400	NR
8	PRESSURE	10000	NR

FIGURE A21

PRIMARY EVALUATIONS

- BASIC SYSTEMS - A (PUMPED) VS, B (SUPERCRITICAL) VS, C (LOW PRESSURE GAS)
- TURBINE STAGE NUMBER - 4 STAGE SUBSONIC VS 2 STAGE SUPERSONIC
- TANKAGE - VEHICLE INTEGRATED VS APU TANKAGE
- POWER CONTROL - PULSE VS PRESSURE MODULATION
- SUPERCRITICAL VS PUMPED - SYSTEM A VS SYSTEM B

FIGURE A22

BASIC SYSTEM COMPARISON

SYSTEM	WEIGHT RANGE	RATING RANGE						
		TOTAL 100	WEIGHT 25	FLEXIBILITY 20	DEVELOPMENT RISK 10	MANUFACTURING COST 5	MAINTENANCE 5	FLEXIBILITY 35
A-PUMPED	550-730	51.8-66.4	19.3/24.8	11.9/16.9	4.5/5.9	2.5/4.6	1.3/2.7	9.2/11.3
B-SUPERCRITICAL	680-1000	46.9-66.3	12.5/20.3	11.2/15.5	7.1/8.6	3.9/4.6	3.0/4.4	10.7/15.6
C-LOW PRESSURE GAS	1170-2460	32.0-45.0	0.2/8.4	1.1/4.5	3.9/6.3	3.7/5.0	3.3/4.6	16.9/18.5

CONCLUSIONS

- SYSTEMS A AND B MOST PROMISING
- SYSTEM A MOST FLEXIBLE/MOST RISK/MOST EXPENSIVE DEVELOPMENT
- SYSTEM B LEAST RISK
- SYSTEM C MOST RELIABLE/SEVERE WEIGHT PENALTY

BEST

WORST

FIGURE A23

TURBINE TYPE COMPARISON

FOUR STAGE SUBSONIC vs TWO STAGE SUPERSONIC

TURBINE TYPE		RATING	WEIGHT	SYSTEM			
				TYPE	DESCRIPTION		CONDITIONS
4 STAGE		64.1	550	A-3	PUMPED PULSE CONTROL	VEHICLE H_2	P_T = 300 PR = 50
	2 STAGE	61.7	685	A-1		APU O_2	P_T = 300 PR = 30
4 STAGE		54.2	620	A-4	PUMPED PRESSURE MOD. CONTROL	VEHICLE H_2	P_T = 600 PR = 50
	2 STAGE	53.3	730	A-2		APU O_2	P_T = 600 PR = 40
4 STAGE		66.3	680	B-3	SUPERCRITICAL STORAGE PULSE CONTROL	VEHICLE H_2	P_T = 150 PR = 30
	2 STAGE	63.9	870	B-1		VEHICLE O_2	P_T = 150 PR = 20

CONCLUSIONS

- 4 STAGE
  - STRONG POTENTIAL WEIGHT PAYOFF
  - HIGHER DEVELOPMENT COST AND RISK
  - LOWER RELIABILITY POTENTIAL
  - WEIGHT ADVANTAGE - STRONGLY DEPENDENT ON MISSION TOTAL ENERGY REQUIREMENT
  - MAY RESULT IN PENALTY FOR ORBITER APPLICATION

FIGURE A24

VEHICLE vs APU TANKAGE - BOOSTER

TANKAGE TYPE		RATING	WEIGHT	SYSTEM			
H_2				TYPE	DESCRIPTION		CONDITIONS
VEHICLE		54.2	620	A-4	PUMPED 4 STAGE PRESSURE MOD.	APU O_2	P_T = 600 PSIA PR = 50
	APU	52.0	650	A-6			
VEHICLE		52.4	990 *	B-5A	SUPERCRITICAL STORAGE 2 STAGE PRESSURE MOD.	APU O_2	P_T = 150 PSIA PR = 12
	APU	46.9	1000 *	B-6			

* LARGER WEIGHT DIFFERENCE WOULD OCCUR IN ORBITER

TANKAGE TYPE		RATING	WEIGHT	SYSTEM			
				TYPE	DESCRIPTION		CONDITIONS
VEHICLE SUPERCRITICAL O_2		63.9	870	B-1	SUPERCRITICAL PROPELLANTS 2 STAGE PULSE	VEHICLE H_2	P_T = 150 PSIA PR = 20
	APU SUPERCRITICAL O_2	61.7	870	B-5			

CONCLUSIONS

- VEHICLE TANKAGE FAVORABLE FOR H_2 AND O_2
- VEHICLE TANKAGE MORE IMPORTANT FOR H_2

FIGURE A25

PUMPED vs SUPERCRITICAL STORAGE H_2 BOOSTER

FEED SYSTEM TYPE		RATING	WEIGHT	SYSTEM			
				TYPE	DESCRIPTION		CONDITIONS
PUMPED		61.7	685	A-1	2 STAGE PULSE	APU O_2	P_t = 300 PSIA PR = 30
	SUPERCRITICAL STORAGE	61.7	870	B-5			P_t = 150 PSIA PR = 20
PUMPED		66.4	550	A-5	4 STAGE PULSE	VEHICLE O_2	P_t = 300 PSIA PR = 50
	SUPERCRITICAL STORAGE	66.3	680	B-3			P_t = 150 PSIA PR = 30
PUMPED		53.3	730	A-2	2 STAGE PRESSURE MOD.	APU O_2	P_t = 600 PSIA PR = 40
	SUPERCRITICAL STORAGE	52.4	990	B-5A			P_t = 150 PSIA PR = 20

CONCLUSIONS

- COMPARIBLE RATINGS
- SUPERCRITICAL WEIGHT PENALTY 130-260 POUNDS
- PUMP OPERATION AND DEVELOPMENT RISK REQUIRE FURTHER EVALUATION
- ORBITER SENSITIVITY REQUIRES DEFINITION

FIGURE A26

PULSE vs PRESSURE MODULATION CONTROL

CONTROL TYPE		RATING	WEIGHT	TYPE	SYSTEM DESCRIPTION		SYSTEM CONDITIONS
PULSE		61.7	685 (665)	A-1	2 STAGE	VEHICLE H_2 / APU O_2	P_T = 300 PSIA PR = 30 (P_T = 600 PSIA PR = 50)
	PRESSURE MOD.	53.3	730	A-2	2 STAGE	VEHICLE H_2 / APU O_2	P_T = 600 PSIA PR = 40
PULSE		64.1	550	A-3	4 STAGE	VEHICLE H_2 / APU O_2	P_T = 300 PSIA PR = 50
	PRESSURE MOD.	54.2	620	A-4	4 STAGE	VEHICLE H_2 / APU O_2	P_T = 600 PSIA PR = 50
PULSE		63.9	870	B-1	2 STAGE	VEHICLE H_2 / VEHICLE O_2	P_T = 150 PSIA PR = 20
	PRESSURE MOD.	54.6	990	B-2	2 STAGE	VEHICLE H_2 / VEHICLE O_2	P_T = 150 PSIA PR = 12

CONCLUSIONS

- PULSE ALWAYS BEST
  - 8 TO 10 POINTS
  - 45 TO 120 LBS
- PRESSURE MODULATION WEIGHT PENALTY EXCESSIVE WITH SUPERCRITICAL STORAGE
- PULSE MORE FLEXIBLE
  - COMPATIBLE WITH HIGH PEAK/MODE POWER RATIOS
  - OPTIMIZIZES WITH REDUCED INLET PRESSURES
  - ACCOMODATES BETTER TO ORBITER REQUIREMENTS
- PULSE OPERATION REQUIRES FURTHER EVALUATION

FIGURE A27

	LOW PRESSURE VS SUPERCRITICAL STORAGE
	ORBITER VS BOOSTER INSULATION
PUMP/COMPRESSOR SUBSYSTEM	PUMP TYPE: POSITIVE DISPL. VS DYNAMIC CONTROL TYPE: N = COST. W/BYPASS N = VARIABLE, INTERMITTENT FLOW
PROPELLANT CONDITIONING	HIGH PRESSURE VS LOW PRESSURE HEAT EXCHANGER
	THERMAL DYNAMICS
	CONDITIONING CONTROL

Certain of the key factors entering into the component design are summarized in Fig.A28.

Tankage

Fig. A29 and A30 illustrate some of the results of the tankage study. These results represent the weight of tankage associated with a particular quantity of propellant typical of a booster mission (Fig.A29) and orbiter mission (Fig.A30). Data has been generated for tankage weight as a function of propellant weight and some of this data is presented in Fig.A31 for both booster and orbiter. In Fig.A31 the effect of utilizing different propellant supply tankage is illustrated. The results are expressed in pounds of chargeable tankage weight per APU. Tankage for various conditions such as one tank for each APU $(\frac{\text{one}}{\text{APU}})$, one tank for four APU $(\frac{\text{one}}{4\text{APU}})$ and main vehicle tankage for each APU (SSE-H_2, RCS-O_2) is illustrated.

COMPONENT EVALUATION

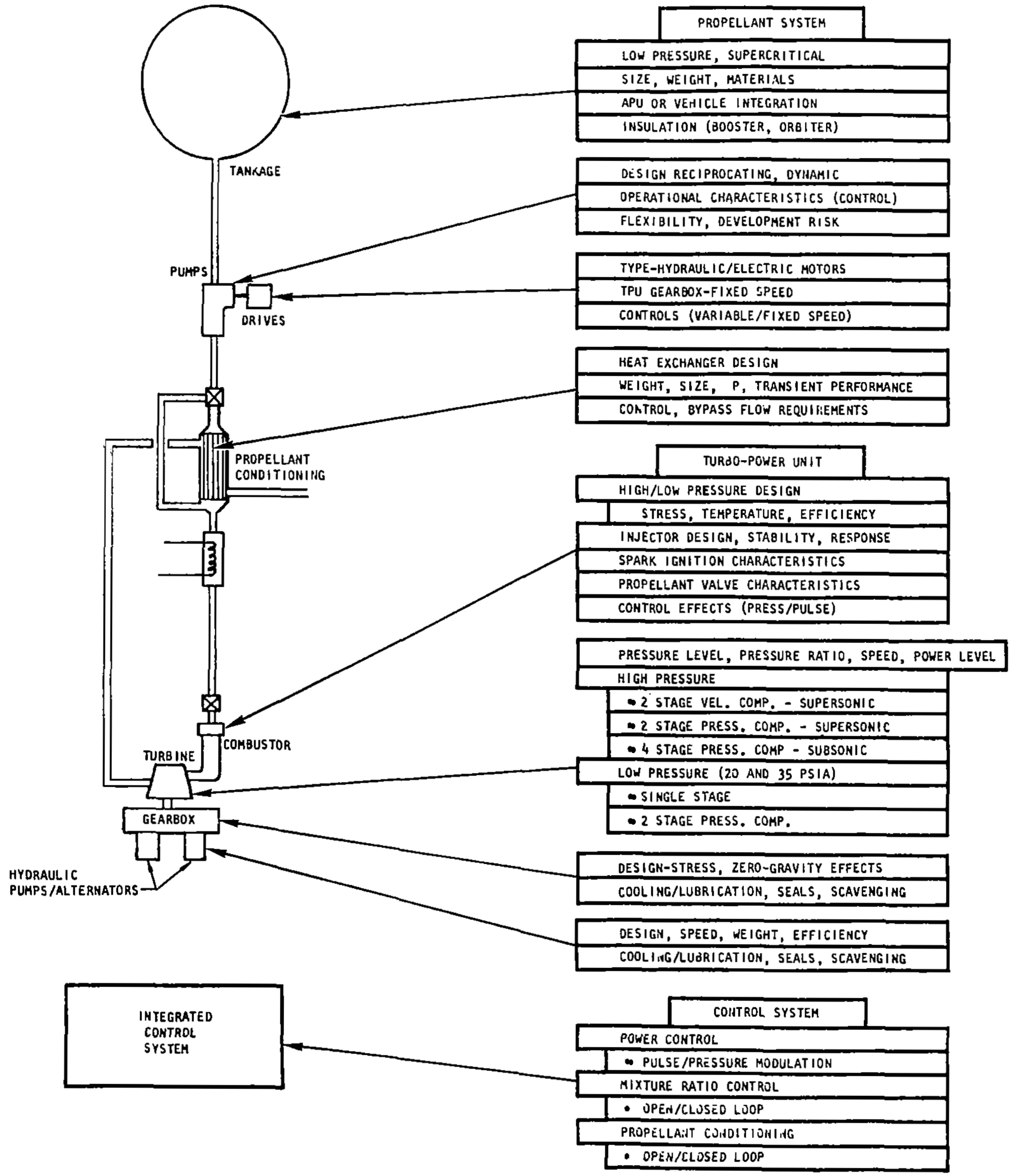
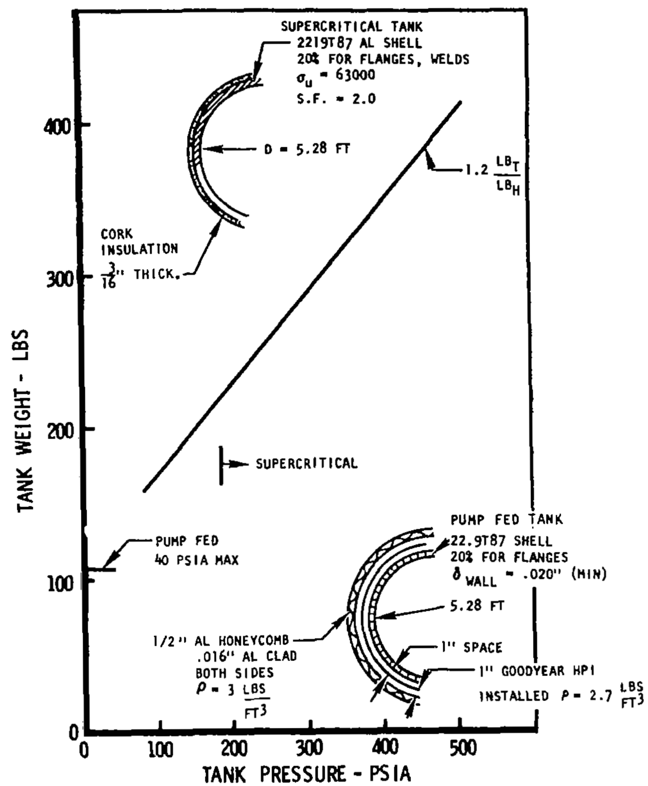


FIGURE A28

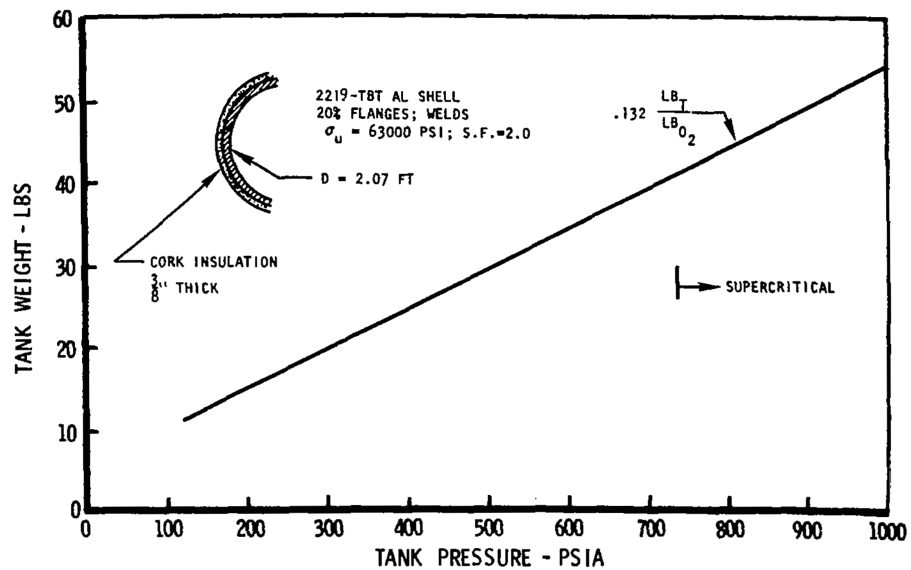
HYDROGEN AND OXYGEN APU TANK WEIGHT
TYPICAL BOOSTER

W_H = 300 LBS (BURNED)
W_H = 332 LBS (LOADED)



HYDROGEN

W_O = 300 LBS (BURNED)
W_O = 332 LBS (LOADED)



OXYGEN

FIGURE A29

HYDROGEN AND OXYGEN APU TANK WEIGHT

TYPICAL ORBITER

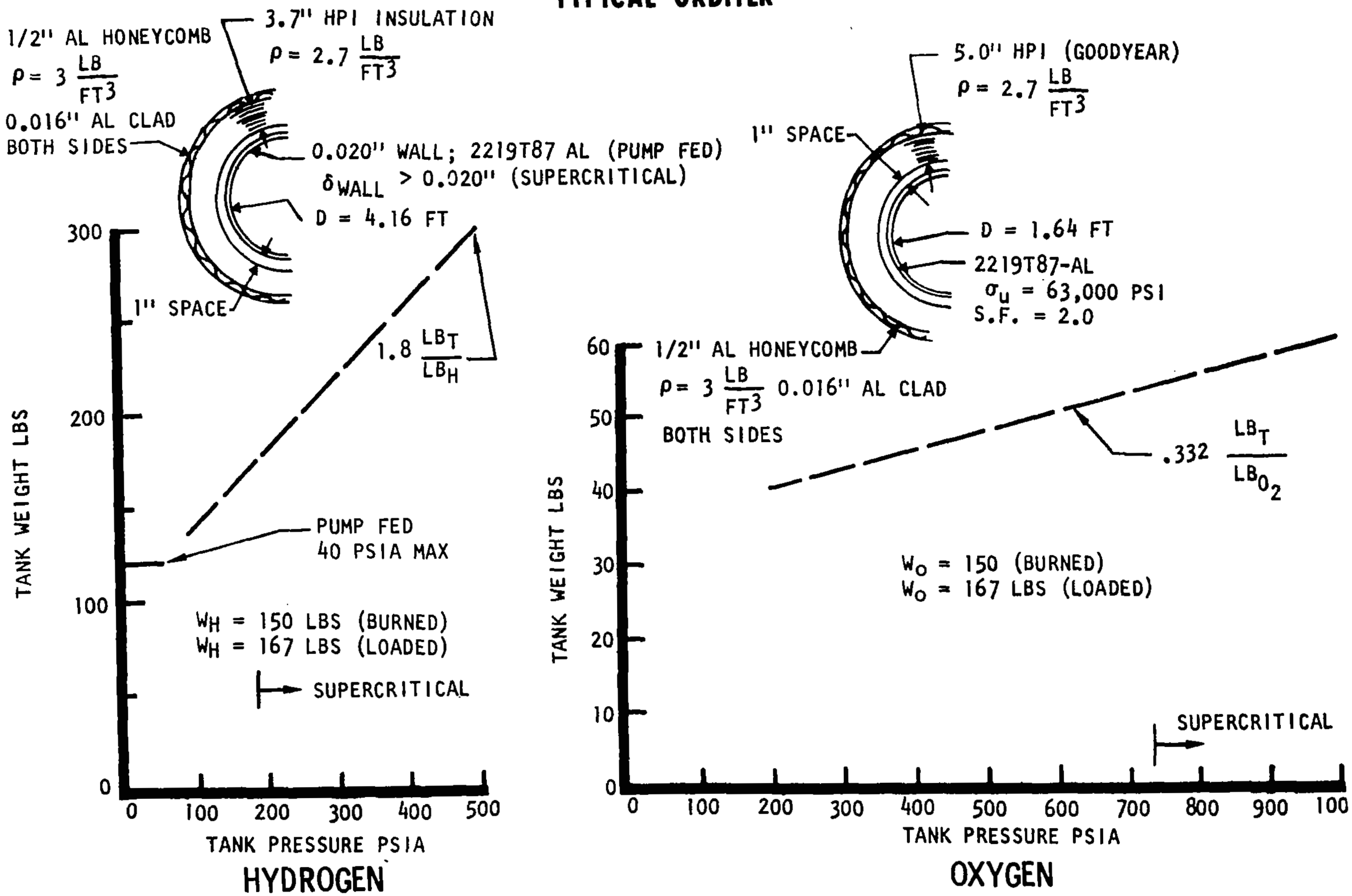
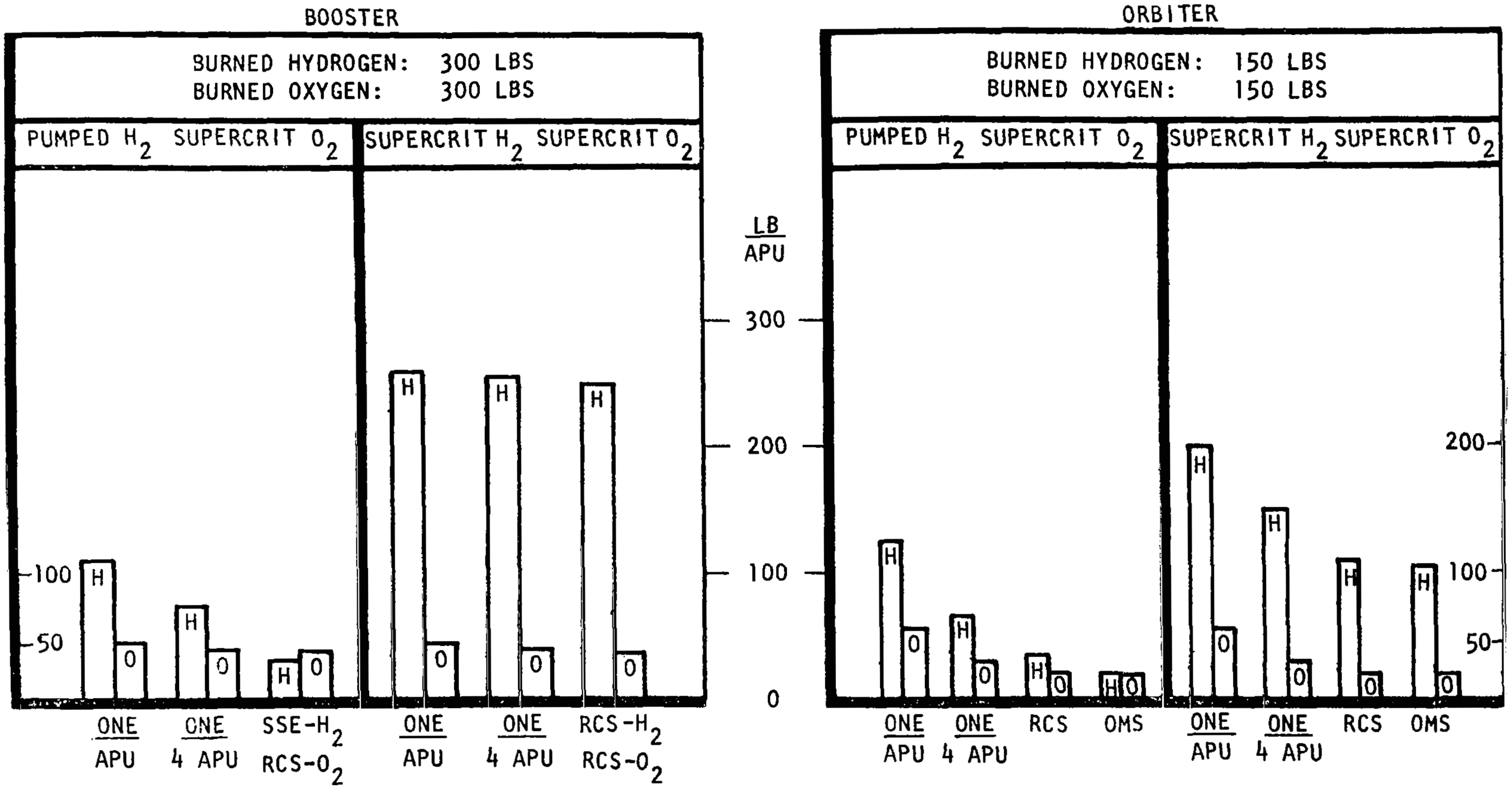


FIGURE A30

TANK WEIGHT CHARGEABLE TO APU FOR SHARED PROPELLANT TANKS



- SHELL MATERIAL: 2219T87-AL; σ_u = 63000 PSI; S.F. =2.0 (20% FOR FLANGES)
- BOOSTER INSULATION: CORK (PREVENTS CRYOPUMPING)
- ORBITER INSULATION: GOODYEAR HPI, $\rho = 2.7 \frac{LB}{FT^3}$ (INSTALLED)
  AL HONEYCOMB OUTER SHELL, $\rho = 3$ LB/FT^3
  CLAD BOTH SIDES .016" AL
- EXPULSION 95%; 5% ULLAGE END OF LOADING; 5% PROPELLANT RESERVE
- SUPERCRITICAL HYDROGEN PRESSURE = 250 PSIA
- SUPERCRITICAL OXYGEN PRESSURE = 800 PSIA
- PUMPED HYDROGEN TANK PRESSURE = 40 PSIA

FIGURE A31

Fig. A32 shows the Rocketdyne generated specific tankage data compared with certain predictions presently being used in the Space Shuttle vehicle studies.

Compressor Augmentation

The application of a compressor was investigated to boost the pressure of propellant gas from the low pressure tankage source, where ullage gas may be available. Fig. A33 indicates that for low ΔP the shaft power requirement is acceptable if saturated gas in the range of $40^{\circ}R$ is available. For superheated gas at higher temperatures, however, the power required becomes excessive, particularly at higher ΔP. At this time, accurate estimates of the expected inlet gas temperature from the low pressure tankage are not available.

Fig. A34 illustrates the difference in requirements for compressors operating at $43^{\circ}R$ and $232^{\circ}R$ and compares these compressors with a pump sized to perform a similar job. It appears then that, while inlet gas temperatures may be expected to be somewhere between $40^{\circ}R$ and $230^{\circ}R$ if low ΔP is acceptable, compressors may offer a viable solution to the use of low pressure ullage gas from main propellant tankage.

Pumping Systems

A study is in progress to select the most promising hydrogen pump system for use in high combustor pressure systems. Fig. A35 illustrates four of the most promising systems under evaluation. Fig. A36 summarizes the

PROPELLANT TANKAGE SPECIFIC WEIGHT

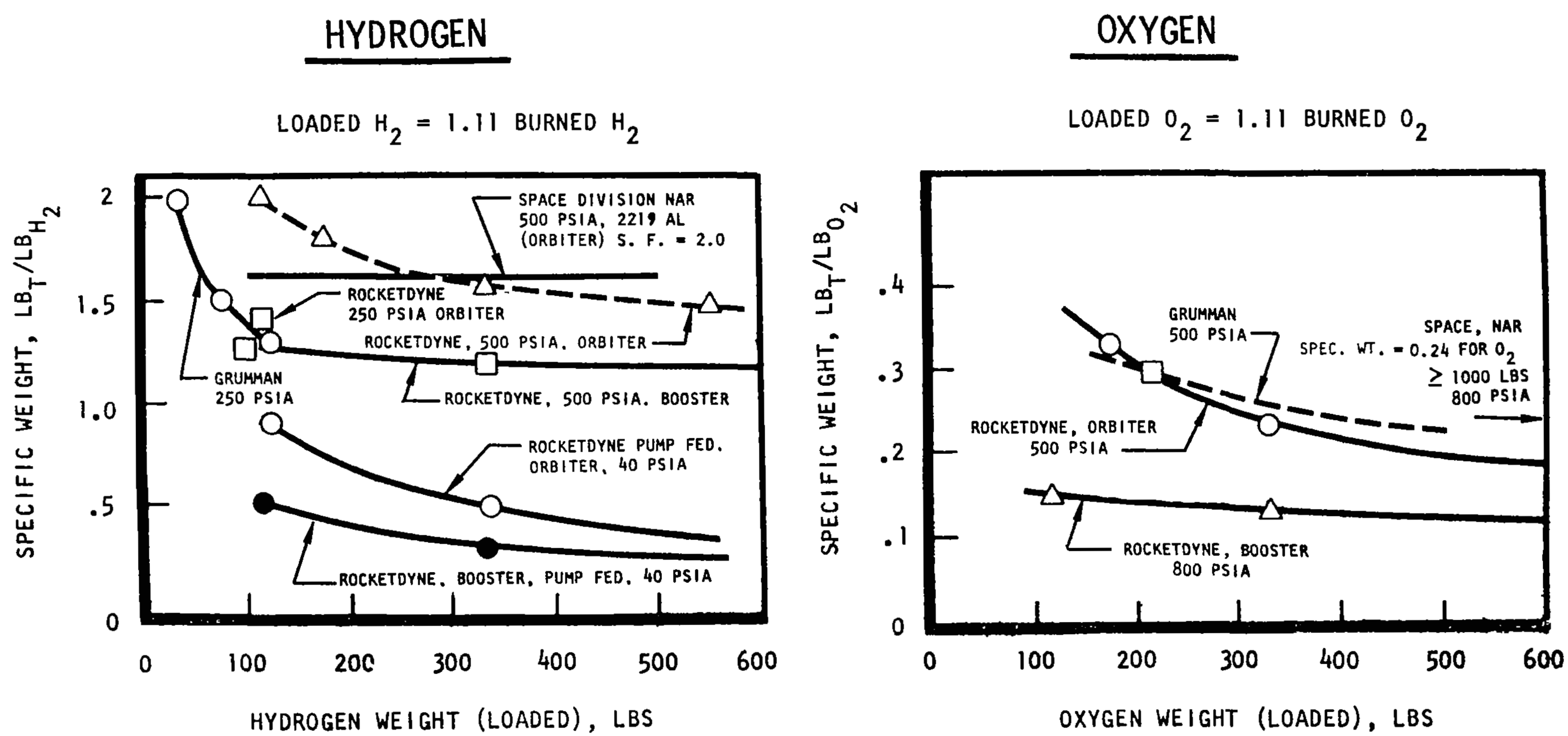


FIGURE A32

COMPRESSOR POWER REQUIREMENTS

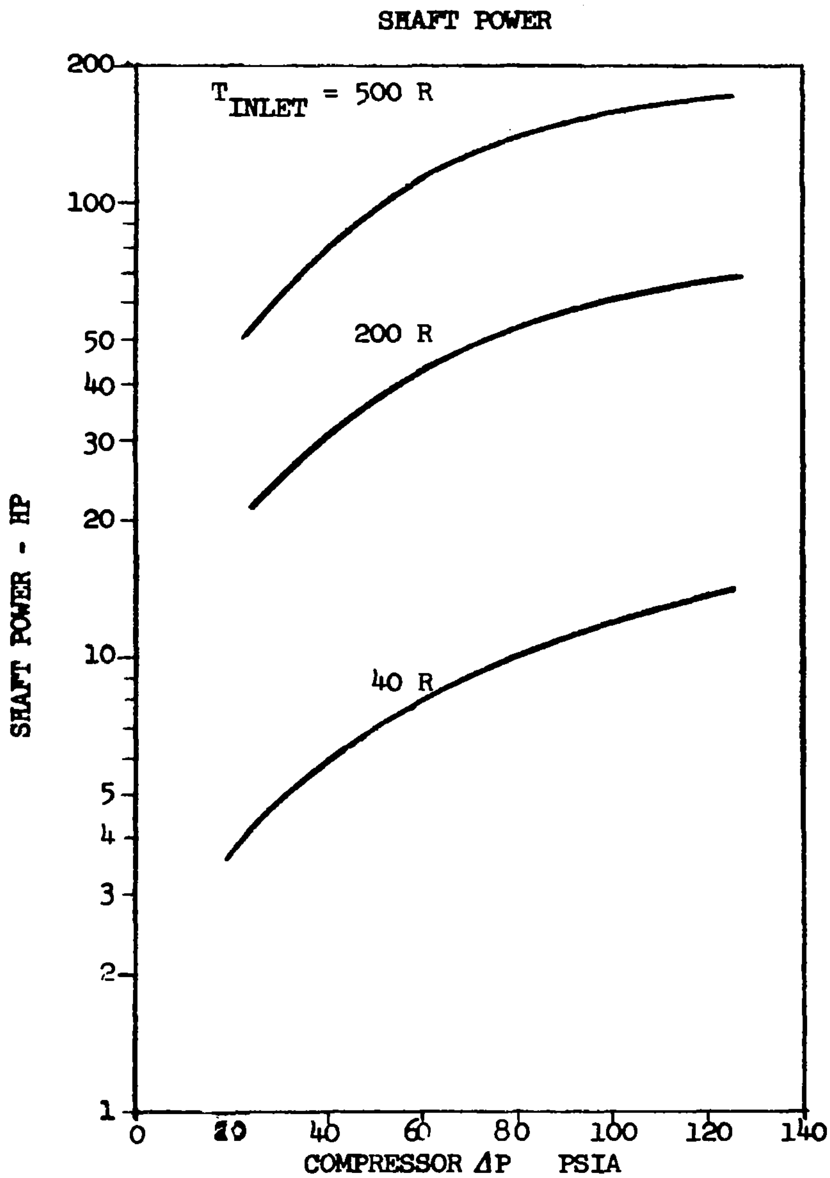
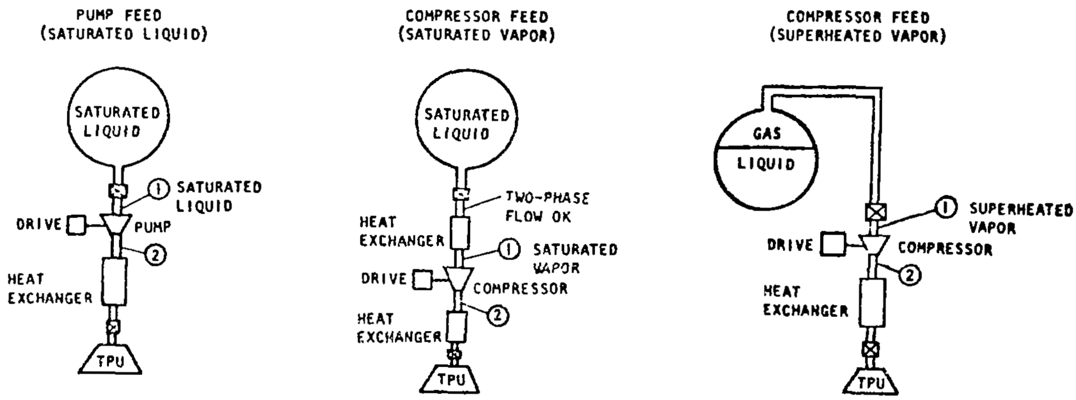


FIGURE A33

PUMP COMPRESSOR DELIVERY COMPARISON



$\dot{w}_{H_2} = \dot{w}_{O_2} \approx 0.07$ LB/SEC (PEAK POWER)

	PARAMETER	SATURATED LIQUID SUPPLY		SUPERHEATED GAS SUPPLY
		PUMP	COMPRESSOR	COMPRESSOR
1	INLET TEMPERATURE, R	40	43	232
2	DISCHARGE TEMPERATURE, R	45	130	500
1	INLET PRESSURE, PSIA	25	40	40
2	DISCHARGE PRESSURE, PSIA	400	400	400
	SPECIFIC SPEED	0.4	0.4	0.4
	SPEED, RPM	3340	4060	5720
	OVERALL η, %	70	70	70
	ADIABATIC HEAD, FEET	10,900	109,000	565,000
	SHAFT POWER, HP	2.0	20	104
	PUMP/COMPRESSOR WEIGHT, POUNDS	10	90	200
	DRIVE WEIGHT*, POUNDS	1	10	52
	TOTAL, POUNDS	11	100	252

* HYDRAULIC MOTOR

PUMP

- POSITIVE DISPLACEMENT
- LOW WEIGHT/POWER
- PROPELLANT ACQUISITION
  - CRITICAL CONSIDERATION
- MEDIUM DEVELOPMENT RISK

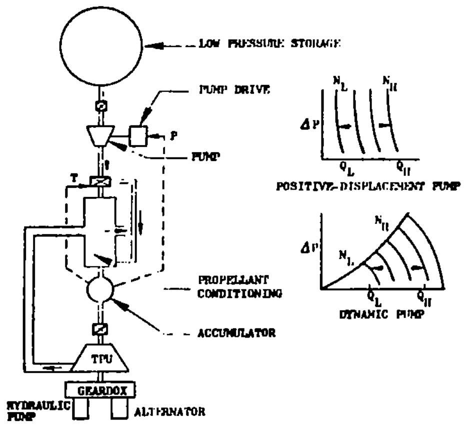
COMPRESSOR

- POSITIVE DISPLACEMENT
- HIGH WEIGHT/POWER
- REMOTE TANKAGE/APU MOUNTED
  - COMPRESSOR FEASIBLE
- EXCELLANT ZERO-G PROPELLANT
- ACQUISITION
- MEDIUM DEVELOPMENT RISK

FIGURE A34

APU PUMP FED SYSTEMS

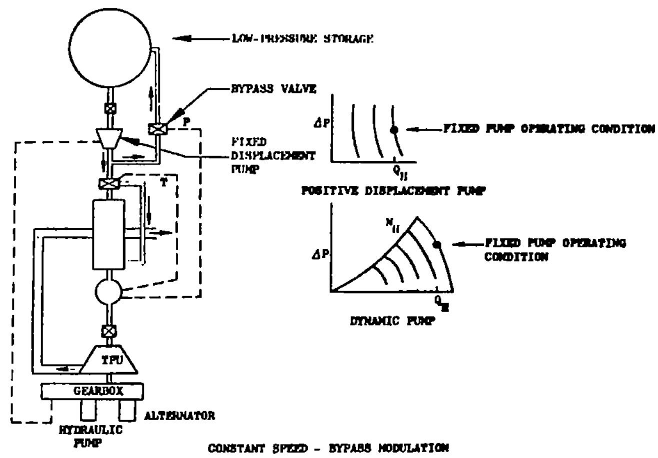
SYSTEM A



VARIABLE SPEED - FLOW MODULATION

- LOW WEIGHT
- MEDIUM DEVELOPMENT RISK
- HIGH FLEXIBILITY
- FAST RESPONSE

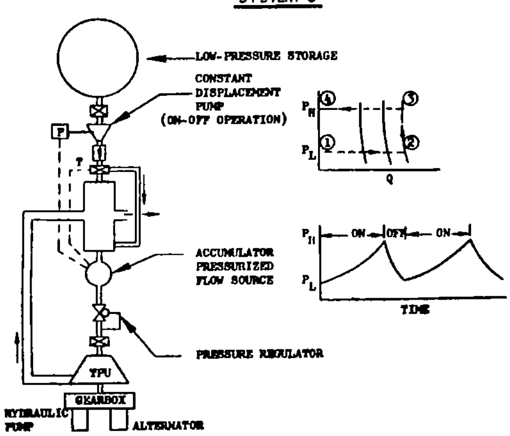
SYSTEM B



CONSTANT SPEED - BYPASS MODULATION

- LOW WEIGHT
- MEDIUM DEVELOPMENT RISK
- MEDIUM FLEXIBILITY
- HIGH VENT LOSS PENALTY
- FAST RESPONSE

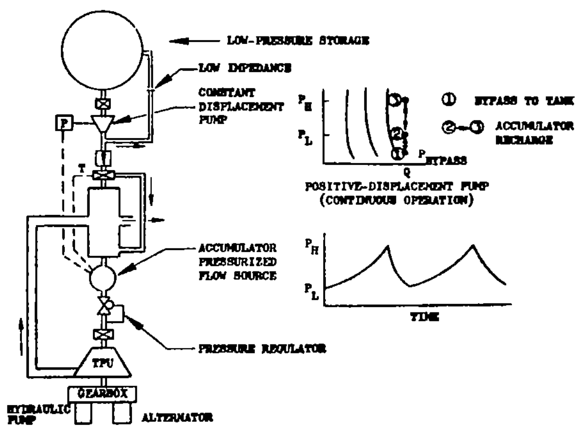
SYSTEM C



CONSTANT SPEED - INTERMITTENT FLOW

- MEDIUM WEIGHT
- MEDIUM DEVELOPMENT RISK
- PUMP SPEED TRANSIENTS
- LOW FLEXIBILITY (CYCLE LIFE LIMITED)
  - CYCLES VS ACCUMULATOR SIZE
- FAST RESPONSE
- CONDITIONED PROPELLANT TRANSIENTS
  - ACCUMULATOR DISCHARGE

SYSTEM D

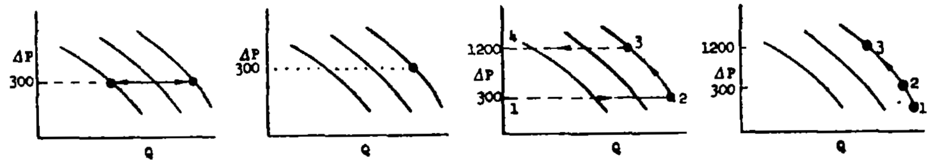


CONSTANT SPEED - INTERMITTENT FLOW WITH BYPASS

- MEDIUM WEIGHT
- MEDIUM DEVELOPMENT RISK
- NO PUMP SPEED TRANSIENTS
- LOW FLEXIBILITY (CYCLE LIFE LIMITED)
  - CYCLES VS ACCUMULATOR SIZE
- LOW VENT LOSS
- FAST RESPONSE
- CONDITIONED PROPELLANT TRANSIENTS
  - ACCUMULATOR DISCHARGE

FIGURE A35

PUMP CHARACTERISTICS AND APPLICABILITY



	GENERAL CHARACTERISTICS					APPLICABILITY			
	Dev. Risk (Relative)	Speed 1000		NPSH	Size	VARIABLE SPEED FLOW MODULATION	CONSTANT SPEED BYPASS MODULATION	START-STOP INTERMITTENT FLOW	CONSTANT SPEED INTERMITTENT FLOW
PISTON PUMP	High	2-6	0.8	Good	Med.	Yes ● (N/N) = .15 ● η/η = .53	Yes Med. Vent Loss	Yes ● η/η = .88 ● Q/Q = .89	Yes ● Q(bypass) = 1.1 Des. ● η(bypass) = 0.9 ● Low vent loss.
PILOT PUMP	Med.	40-50	0.35	Good	Med.	Yes ● N/N = .82 ● η/η = .17	Yes High vent loss	No H-Q too flat	No H-Q too flat
DRAG PUMP	Med.	10-30	.52	Poor	Med.	Yes ● N/N = 0.4 ● η/η = 0.33	No High vent loss W/ high NPSH req'd	Yes ● η/η = .65 ● Q/Q = .55 ● By Pass Controls	Yes ● Q(by pass) = 1.25 Des. ● η(by pass) = .75 ● Low vent loss
CENTRIFUGAL PUMP. Full emission Partial emission (Barske)	Low	60-100	.60	Med.	Low	No Limited flow range	Yes High vent loss W/ Med. NPSH req'd	Yes ● η/η = .55 ● Q/Q = .6 ● by pass stall control	Yes ● Cavitation limit prevents head reduction
ROOTS	Med.	20-60	.50	Poor	Med.	Yes	No High vent loss W/ High NPSH req'd	Yes	Yes
VALVE PUMP	Low	10-20	.8	Poor	Med.	Yes	Yes High NPSH Req'd	Yes	Yes

FIGURE A36

investigation presently in progress, to select a pump for the "best" of the various pumping systems of Fig.A35. At present, no selection has been made since characterization of the various most promising pumps is in progress. Application of the Mark 36 centrifugal pump in single or a two-stage configuration is being investigated. Figure A37 shows some experimental results of the Mark 36 compared with the APU requirements.

Propellant Conditioning Heat Exchanger

A design study was undertaken to determine the requirements for conditioning the hydrogen using the TPU exhaust gas. The study indicated that steady state heat exchanger requirements are not expected to be severe and the next aspect of the study will be concerned with transient conditions including startup and shutdown.

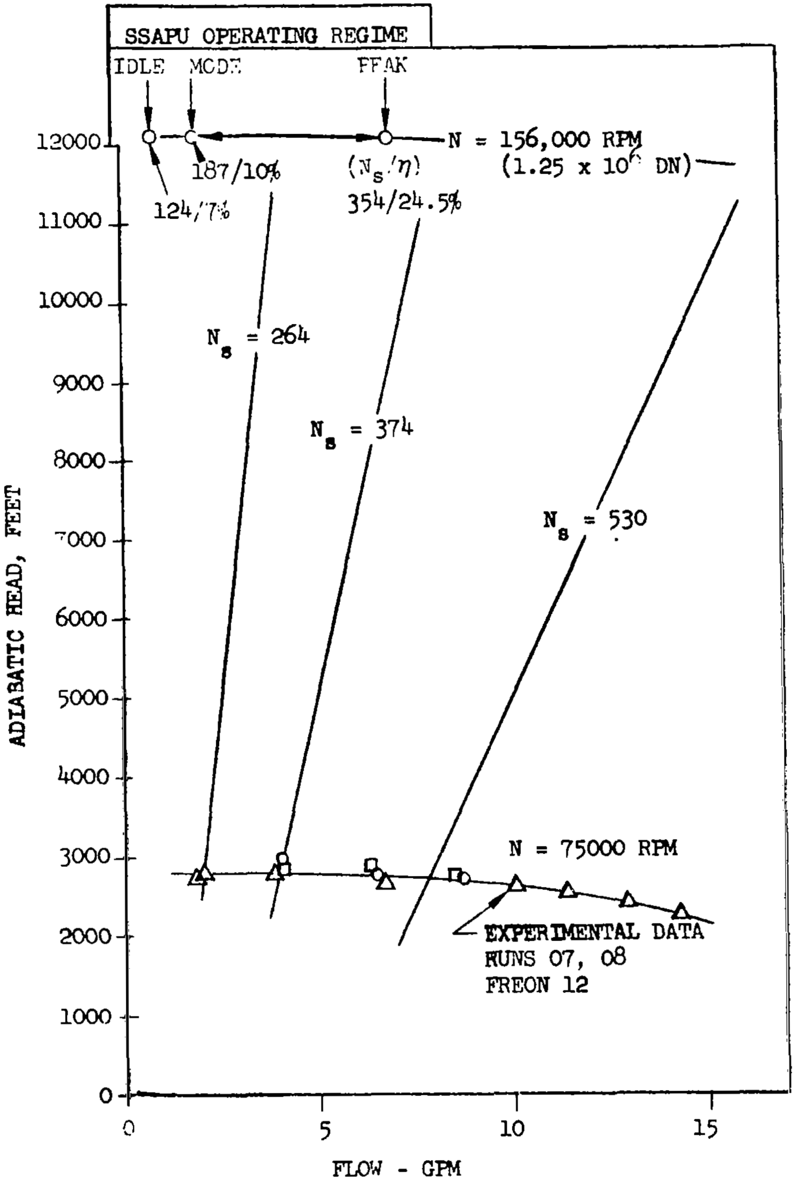
Propellant Feed System Line and Valve Losses

Representative line and valve loss conditions are shown in Fig. A38 for the two most sensitive systems B and C. Weights quoted in the Systems section were based on components and lines sized for these pressure drops.

TURBINE POWER UNIT

Turbine Designs

Information was generated for families of turbines in order to provide component performance and weight information for the system optimization program. Design point and off design data were generated for each of system design points shown in Figures A5, A6 and A7. Figures A39 through A42



MARK 36 PUMP

PERFORMANCE CHARACTERISTICS

(NAS 3-12022)

FIGURE A37

PROPELLANT FEED SYSTEM LINE AND VALVE LOSSES

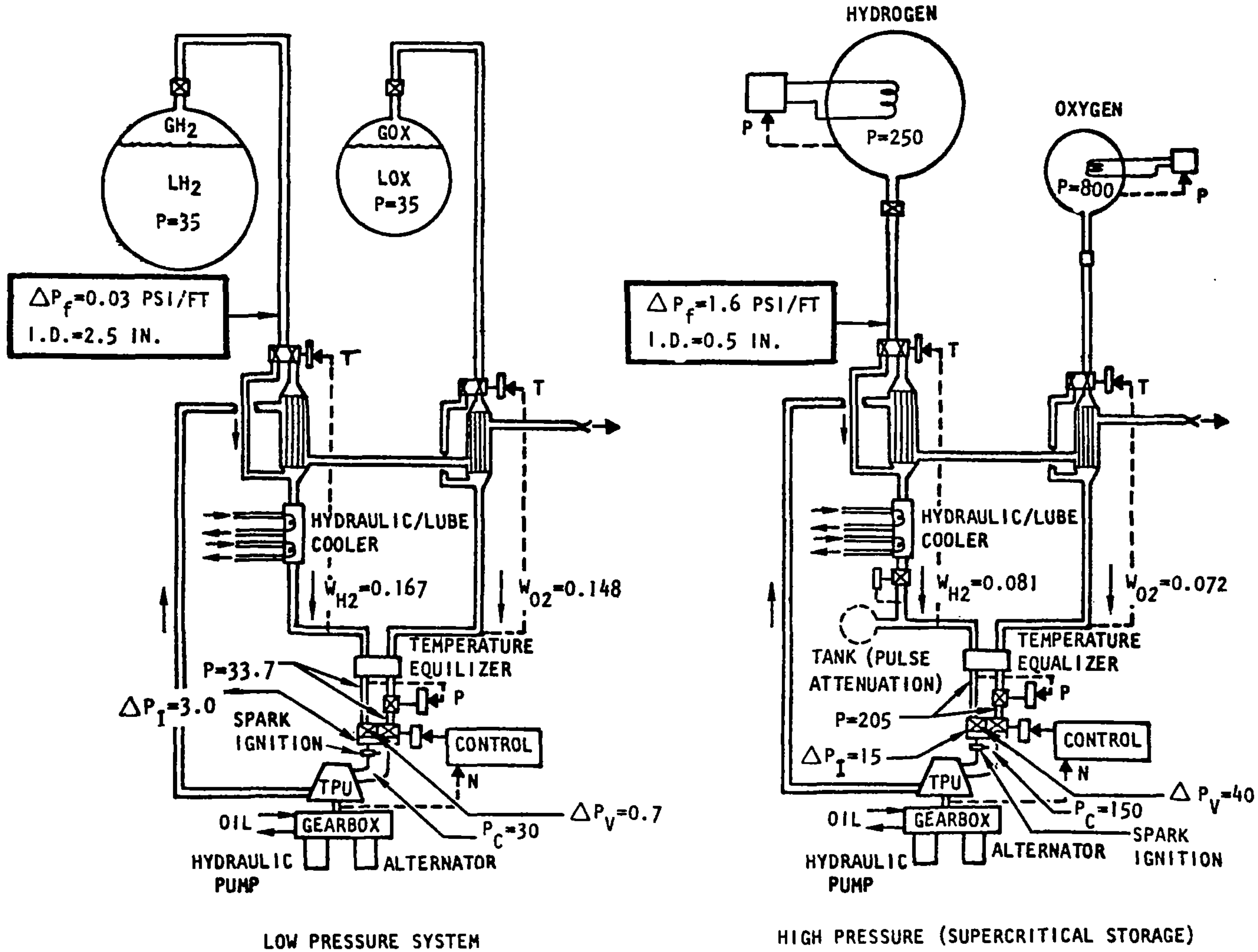


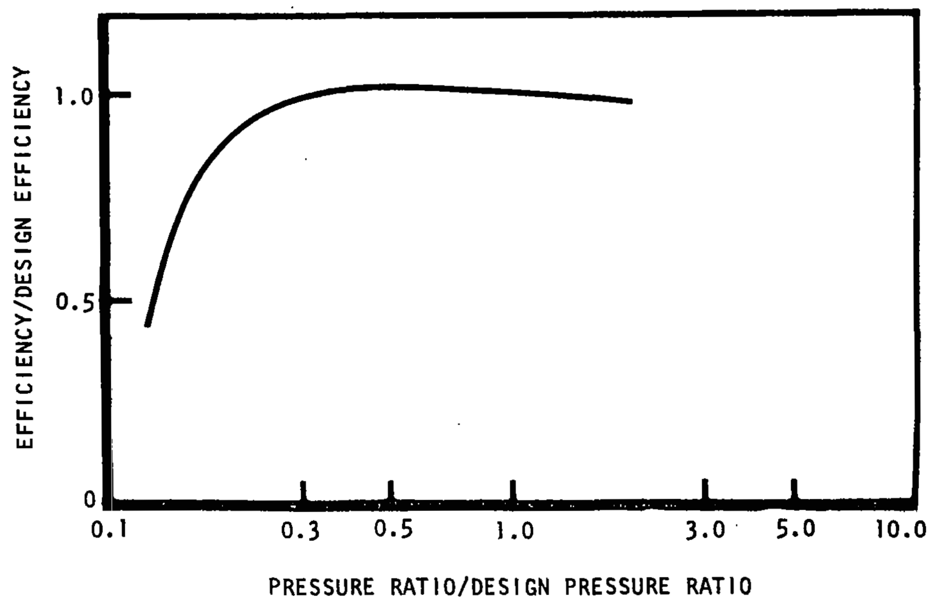
FIGURE A38

HIGH-PRESSURE, FOUR-STAGE TURBINE
TYPICAL FOR SYSTEM A3

DESIGN POINT

• COMBUSTION PRESSURE	300 PSIA
• EXHAUST PRESSURE	6.2 PSIA
• COMBUSTION TEMPERATURE	1470 F
• TIP SPEED	1800 FT/SEC
• ROTATIVE SPEED	80,000 RPM
• DIAMETER	5.15 IN.
• DESIGN POINT - EFFICIENCY	.719
• DESIGN POINT - SPC	1.55 $\frac{\text{LB}}{\text{HP - HR}}$
• TURBINE ASSEMBLY - WEIGHT	51.9 LB
• TURBINE ASSEMBLY - INERTIA	9.4×10^{-3} LB-FT-SEC^2

OFF - DESIGN



STAGE DATA

STAGE NUMBER	1	2	3	4
• VELOCITY RATIO, U/Co	0.283	0.292	0.313	0.335
• BLADE HEIGHT, IN	0.144	0.268	0.350	0.371
• TIP CLEARANCE, IN.	0.012	0.012	0.012	0.012
• DEGREE OF ADMISSION	0.22	0.29	0.54	0.7
• RELATIVE MACH NUMBER	0.877	0.905	0.869	0.860

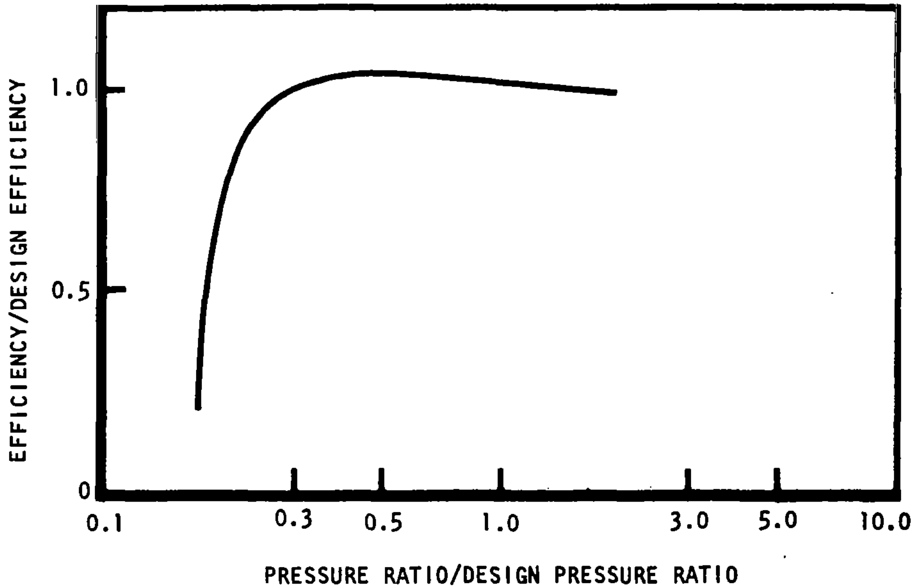
FIGURE A39

HIGH-PRESSURE, VELOCITY-STAGED TURBINE
TYPICAL OF SYSTEMS B1, B3, B4, AND B5

DESIGN POINT

• COMBUSTION PRESSURE	150 PSIA
• EXHAUST PRESSURE	6.2 PSIA
• COMBUSTION TEMPERATURE	1470 F
• TIP SPEED	1800 FT/SEC
• ROTATIVE SPEED	80,000 RPM
• DIAMETER	5.15 IN.
• DESIGN POINT - EFFICIENCY	.548
• DESIGN POINT - SPC	2.28 $\frac{LB}{HP - HR}$
• TURBINE ASSEMBLY - WEIGHT	20.2 LB
• TURBINE ASSEMBLY - INERTIA	3.46×10^{-3} LB-FT-SEC^2

OFF - DESIGN



STAGE DATA

STAGE NUMBER	1	2
• VELOCITY RATIO, U/Co	0.178	0.371
• BLADE HEIGHT, IN.	0.366	0.535
• TIP CLEARANCE, IN.	0.012	0.012
• DEGREE OF ADMISSION	0.60	1.0
• RELATIVE MACH NUMBER	2.03	0.659

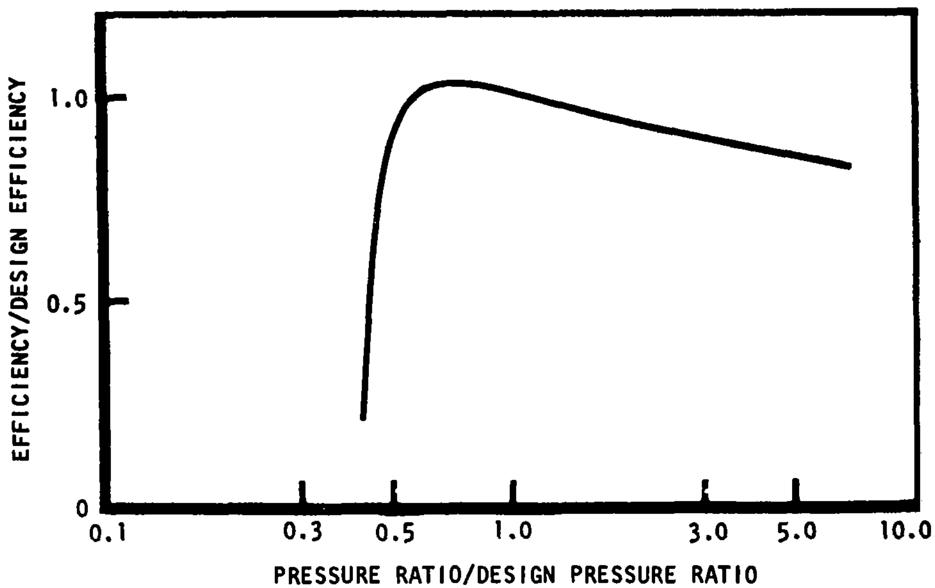
FIGURE A40

LOW-PRESSURE (35 PSIA), TWO-STAGE TURBINE
TYPICAL OF SYSTEMS C1, C2, AND C3

DESIGN POINT

• COMBUSTION PRESSURE	30. PSIA
• EXHAUST PRESSURE	10.45 PSIA
• COMBUSTION TEMPERATURE	1470 F
• TIP SPEED	1800 FT/SEC
• ROTATIVE SPEED	60,000 RPM
• DIAMETER	6.87 IN.
• DESIGN POINT - EFFICIENCY	.695
• DESIGN POINT - SPC	4.16 $\frac{\text{LB}}{\text{HP - HR}}$
• TURBINE ASSEMBLY - WEIGHT	43.7 LB
• TURBINE ASSEMBLY - INERTIA	1.16×10^{-2} LB-FT-SEC^2

OFF - DESIGN



STAGE DATA

STAGE NUMBER	1	2
• VELOCITY RATIO, U/Co	0.374	0.371
• BLADE HEIGHT, IN.	0.62	0.62
• TIP CLEARANCE, IN.	0.016	0.016
• DEGREE OF ADMISSION	1.0	1.0
• RELATIVE MACH NUMBER	0.555	0.586

FIGURE A41

LOW-PRESSURE (20 PSIA), SINGLE STAGE TURBINE
TYPICAL OF SYSTEMS C7 AND C8

DESIGN POINT

• COMBUSTION PRESSURE	17. PSIA
• EXHAUST PRESSURE	15. PSIA
• COMBUSTION TEMPERATURE	1470 F
• TIP SPEED	1800 FT/SEC
• ROTATIVE SPEED	20,000 RPM
• DIAMETER	20.6 IN.
• DESIGN POINT - EFFICIENCY	.868
• DESIGN POINT - SPC	25.1 $\frac{\text{LB}}{\text{HP - HR}}$
• TURBINE ASSEMBLY - WEIGHT	309. LB
• TURBINE ASSEMBLY - INERTIA	0.26 LB-FT-SEC^2

STAGE DATA

STAGE NUMBER	1
• VELOCITY RATIO	0.744
• BLADE HEIGHT, IN.	1.94
• TIP CLEARANCE, IN.	0.050
• DEGREE OF ADMISSION	1.00
• RELATIVE MACH NUMBER	0.164

FIGURE A42

illustrate some of the data available from the study. Each of the figures describe a turbine where an evaluation has been conducted to establish that a "real" turbine is being represented. For example, stress margins have been evaluated and blade heights, admission arc and flow angles revised from optimum with appropriate performance penalties where required. In a like manner, realistic tip clearances were utilized and performance penalties included.

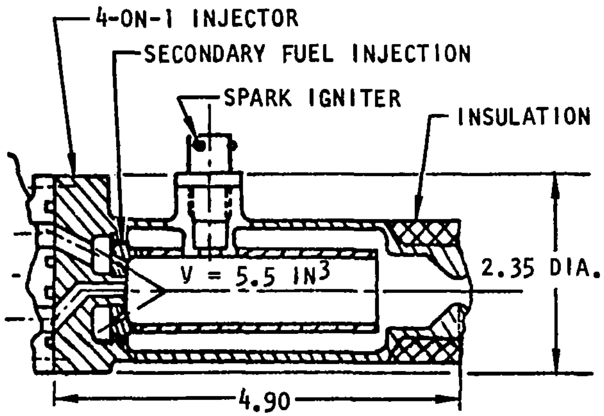
Combustor Assembly

The primary effort involved a comparison between high pressure (System A and B) and low pressure (System C) combustor assemblies. The result of these studies is summarized in Fig. A43. A digital dynamic model was constructed for both combustors and as seen in Fig.A44 andA45 both combustors appear to lend themselves to acceptable pulse power control operation. The problem of combustion stability for pressure modulation power control was investigated and is summarized in Fig.A46. While the high pressure combustor should be acceptable in both control modes, the low pressure combustor is seen to be acceptable for pulse mode control though marginal with pressure modulation control. Turbine nozzle area control appears unacceptable.

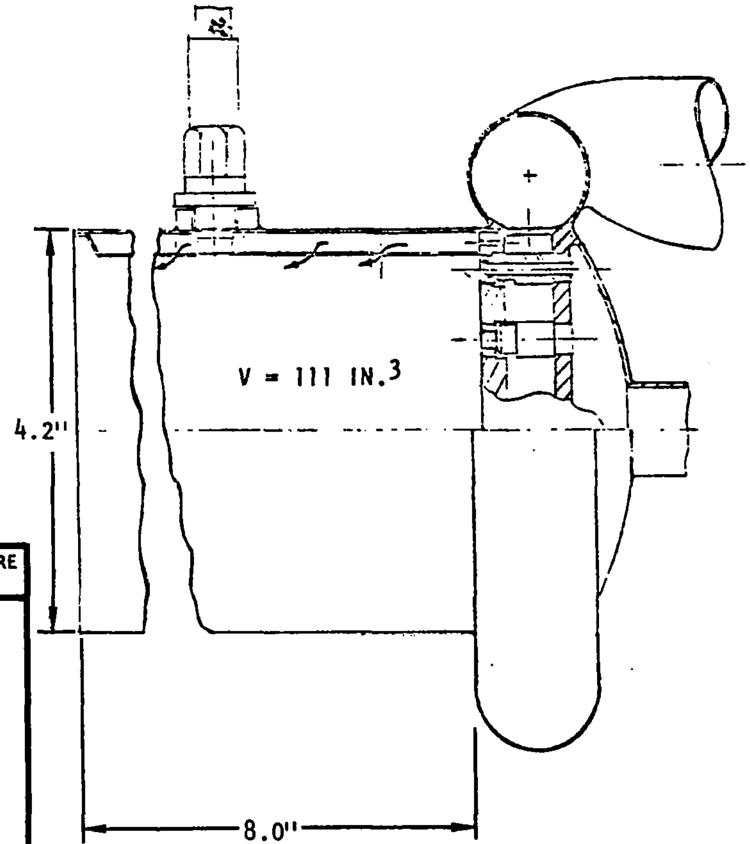
High and Low Pressure TPU Comparison

Fig. A47 illustrates the high and low pressure TPU assemblies for comparison purposes and Fig.A48 shows two typical linked bipropellant valves for comparative purposes. The turbine assembly is seen to be comparable for high and low pressure while the low pressure combustor assembly represents the major size and weight difference between assemblies.

TYPICAL GAS GENERATOR ASSEMBLY DESIGN



HIGH PRESSURE COMBUSTOR



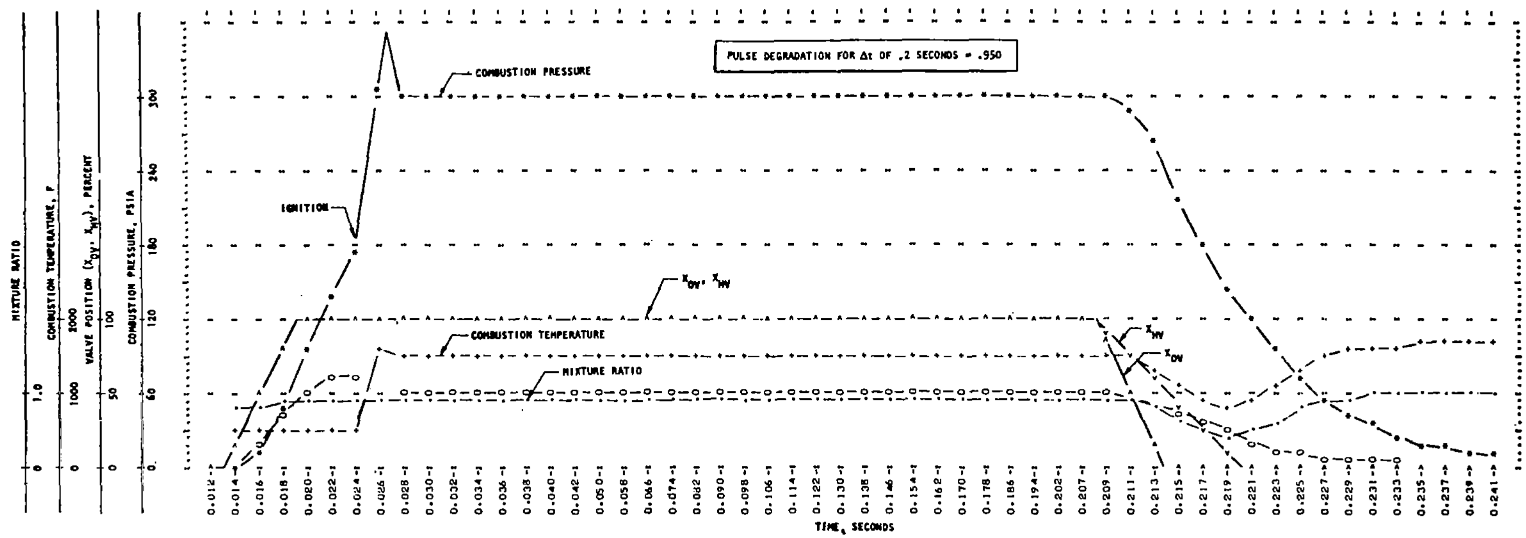
LOW PRESSURE COMBUSTOR

TYPICAL CHARACTERISTICS
(PEAK POWER)

PARAMETER	LOW PRESSURE SYSTEM	HIGH PRESSURE SYSTEM
CHAMBER PRESSURE, PSIA	30	300
FLOWRATE (TOTAL, INSTANTANEOUS), LB/SEC	0.315	0.152
GO_2 (INSTANTANEOUS), LB/SEC	0.148	0.0715
GH_2 (INSTANTANEOUS), LB/SEC	0.167	0.0805
MIXTURE RATIO, O/F	0.89	0.89
COMBUSTION TEMPERATURE, F	1550	1550
CHARACTERISTIC VELOCITY, FT/SEC	7600	7600
COMBUSTION EFFICIENCY (η_{c^*}), PERCENT	95	98
COMBUSTION PRESSURE TRANSIENT START PLUS CUTOFF, MILLISECONDS	38	26

FIGURE A43

COMBUSTOR PULSE TRANSIENT CHARACTERISTICS-HIGH PRESSURE SYSTEM



NOTES:

(1) LINKED BIPROPELLANT VALVE:

(a) SIMULTANEOUS OPENING IN 6 MILLISECONDS

(b) 6 MILLISECONDS H_2 VALVE LAG ON CLOSURE TO INSURE SAFE (< 1.0) MIXTURE RATIO TRANSIENT

(2) IDEAL REGULATION OF PROPELLANT GAS CONDITIONS AT VALVE INLET, i.e., ZERO DIFFERENTIAL PRESSURE AND TEMPERATURE

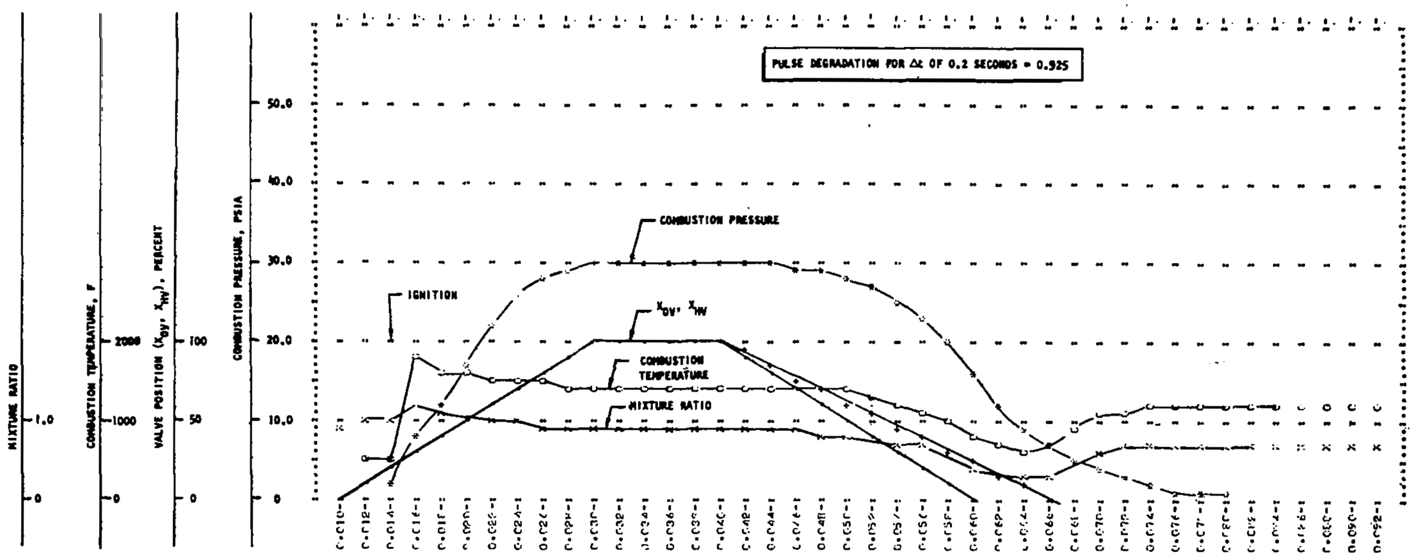
(3) IGNITION DELAYED FOLLOWING VALVE OPENING DUE TO ASSUMED SPARK RATE OF 200 SPARKS/SEC AND CRITERIA OF P_c > 1.0 AND MR > 0.9

(4) (a) 6 MILLISECONDS COMBUSTION PRESSURE BUILDUP TIME INCLUDES VALVE ACTUATION TIME, SPARK DELAY, AND COMBUSTOR CAPACITANCE LAG

(b) 20 MILLISECONDS COMBUSTION PRESSURE DECAY TIME INCLUDES VALVE ACTUATION (WITH H_2 VALVE LAG), AND COMBUSTOR CAPACITANCE LAG

FIGURE A44

COMBUSTOR PULSE TRANSIENT CHARACTERISTICS—LOW PRESSURE SYSTEM



NOTES:

(1) LINKED BIPROPELLANT VALVE:

(a) SIMULTANEOUS OPENING IN 20 MILLISECONDS

(b) 6 MILLISECONDS H_2 VALVE LAG ON CLOSURE TO INSURE SAFE (< 1.0) MIXTURE RATIO TRANSIENT

(2) IDEAL REGULATION OF PROPELLANT GAS CONDITIONS AT VALVE INLET, i.e., ZERO DIFFERENTIAL PRESSURE AND TEMPERATURE

(3) IGNITION DELAYED FOLLOWING VALVE OPENING DUE TO ASSUMED SPARK RATE OF 200 SPARKS/SEC AND CRITERIA OF $P_c > 1.0$ AND $MR > 0.9$

(4) (a) 15 MILLISECONDS COMBUSTION PRESSURE BUILDUP TIME INCLUDES VALVE ACTUATION TIME, SPARK DELAY, AND COMBUSTOR CAPACTANCE LAG

(b) 23 MILLISECONDS COMBUSTION PRESSURE DECAY TIME INCLUDES VALVE ACTUATION (WITH H_2 VALVE LAG), AND COMBUSTOR CAPACITANCE LAG

FIGURE A45

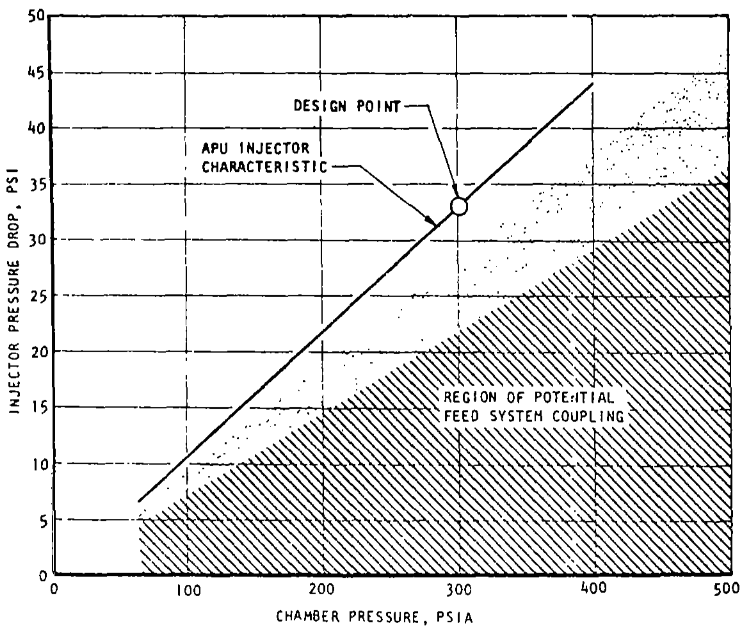
COMBUSTION STABILITY

HIGH PRESSURE

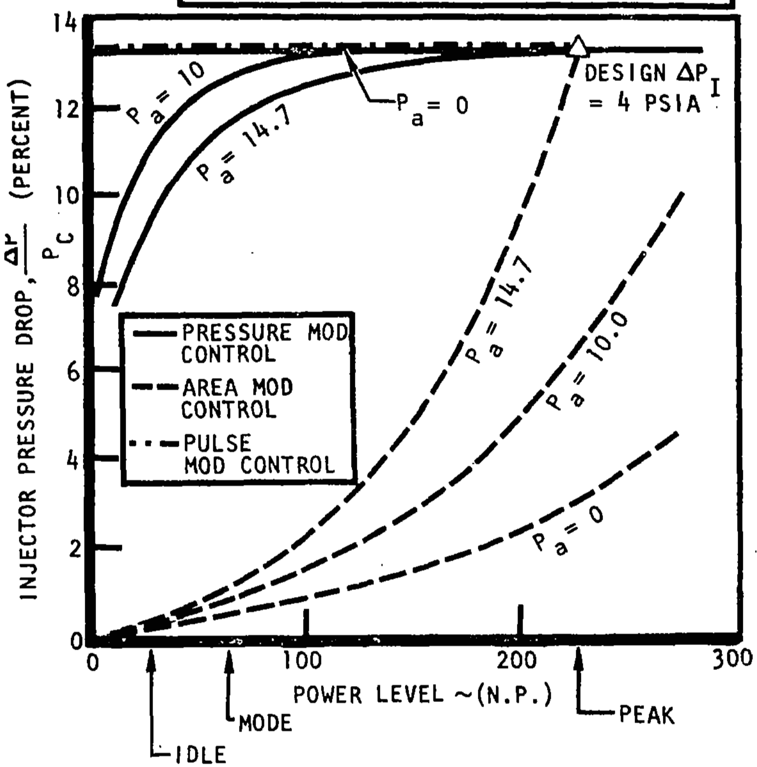
- PULSE MOD CONTROL $\Delta P = \text{CONSTANT}$
- PRESSURE MOD CONTROL $\Delta P/P_c \approx \text{CONSTANT}$

LOW PRESSURE

- PULSE MOD CONTROL $\Delta P = \text{CONSTANT}$
- PRESSURE MOD CONTROL $\Delta P/P_C \approx \text{CONSTANT}$ (HIGH ALTITUDE)
  $\Delta P/P_C < \text{DESIGN}$ (LOW ALTITUDE)
- AREA MOD CONTROL $\Delta P/P_C \ll \text{DESIGN}$ (LOW ALTITUDE)



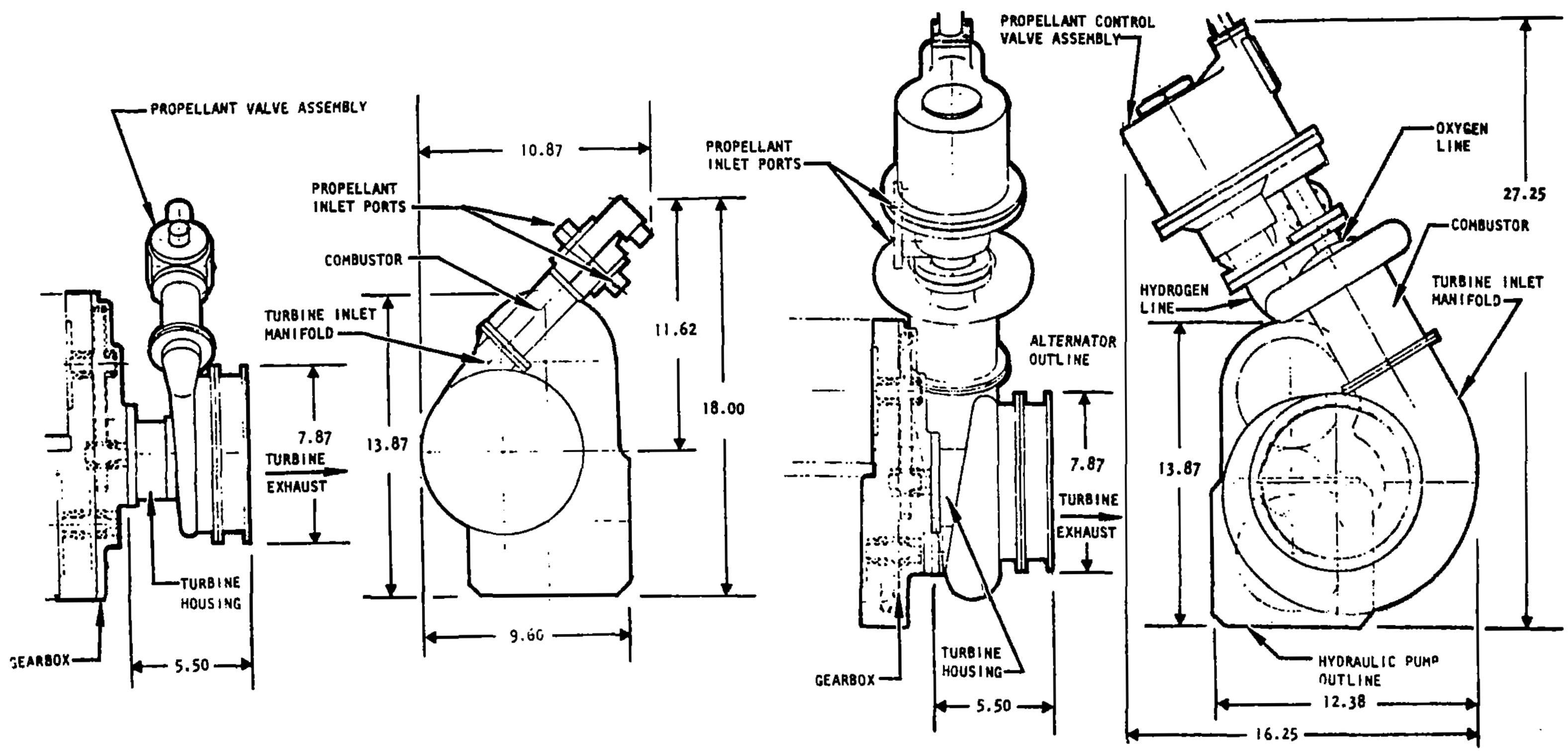
INJECTOR STABILITY CHARACTERISTIC



INJECTOR FLOW CHARACTERISTICS

FIGURE A46

HIGH AND LOW PRESSURE TPU COMPARISON

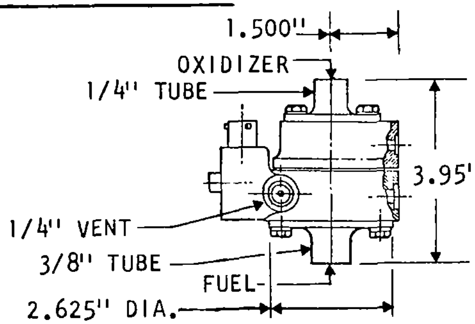


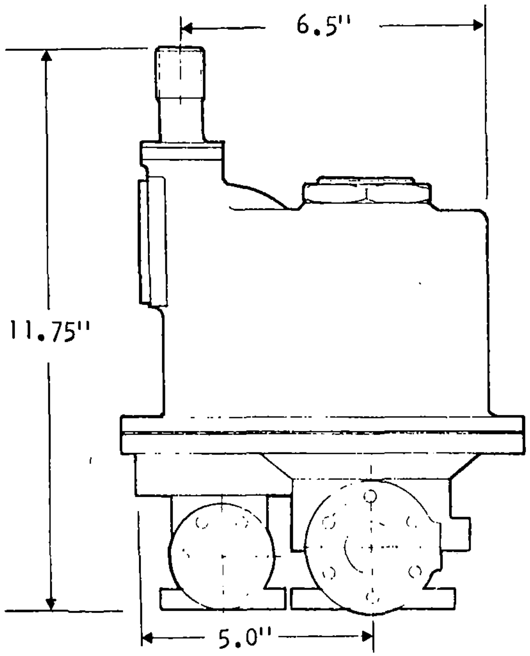
- ROTATIVE SPEED 60,000 RPM
- DESIGN PRESSURE RATIO 48.4
- DESIGN POINT EFFICIENCY .53

- ROTATIVE SPEED 60,000 RPM
- DESIGN PRESSURE RATIO 2.87
- DESIGN POINT EFFICIENCY .70

FIGURE A47

HIGH PRESSURE SYSTEM





LOW PRESSURE SYSTEM

BI-PROPELLANT VALVES
COMPARISON OF HIGH PRESSURE AND LOW PRESSURE SYSTEMS

TYPICAL VALVE CHARACTERISTIC

VALVE TYPE	HIGH PRESSURE	LOW PRESSURE
	LINKED ON-OFF BIPROPELLANT POPPET	LINKED ON-OFF BIPROPELLANT POPPET
WEIGHT (LBS)	5	21
GO_2 INLET PORT (IN)	0.142	1.25
GH_2 INLET PORT (IN)	0.286	2.5
H_2 POPPET SIZE DIA, (IN)	0.5	2.5
VALVE PRESSURE DROP (PSI)	20	0.8
VALVE INLET PRESSURE	350	34
FLOWRATE		
GO_2 (LB/SEC)	0.072	0.148
GH_2 (LB/SEC)	0.081	0.167
ACTUATION TYPE	PNEUMATIC	HYDRAULIC
RESPONSE, MILLISEC	6	20
DESIGN LIFE CYCLES	1×10^6	1×10^6

FIGURE A48

NASA-Langley, 1971 — 3 E-6410