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REPORT MDC E0567

SPACE SHUTTLE
AUXILIARY PROPULSION SYSTEM
DESIGN STUDY

PHASE B REPORT
CANDIDATE RCS CONCEPT COMPARISONS

MCDONNELL DOUGLAS ASTRONAUTICS COMPANY • EAST

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15 FEBRUARY 1972 **REPORT MDC E0567**

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CONTRACT NO. NAS 9-12013

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ABSTRACT

This report describes Phase B of the "Space Shuttle Auxiliary Propulsion System Design Study", Contract NAS 9-12013. The objective of this study was to fully define competing Auxiliary Propulsion concepts and to compare them on the basis of selection criteria such as weight, reliability, and technology requirements. Propulsion systems using both cryogenic oxygen-hydrogen, and earth storable propellants were considered. The main thrust of the cryogenic effort (Phase B) was focused on detailed design and operating analyses for gaseous, oxygen-hydrogen Reaction Control Systems (RCS). This phase of study effort is the subject of this report.

Three high value oxygen-hydrogen Reaction Control System (RCS) concepts were evaluated. In each concept, cryogenic liquid propellants are pumped to high pressure using turbopumps and then thermally conditioned in heat exchangers to desired propellant temperatures. The gaseous propellants are then stored in accumulators until required for RCS thruster operation. All of the concepts employ gas generators to power the pump and heat exchanger, but they differ in the implementation of these assemblies. The three concepts evaluated were: (1) a series-upstream turbine concept which uses the combustion products from a single gas generator to first power the turbopump and then thermally condition the propellants, (2) a series-downstream turbine concept in which the order of gas generator exhaust flow through the heat exchanger and turbine is reversed from that above, and (3) a parallel RCS concept which employs separate gas generators to power the turbopump and heat exchanger independently.

This report provides results of detailed analyses to define preferred controls and optimum system design points for the three RCS concepts. Additionally, the results of analyses to evaluate RCS steady-state and transient operational characteristics and the effect of system malfunctions are presented. Finally, the system concepts are compared based on pertinent selection criteria. The final comparisons demonstrate that all three RCS concepts are viable design approaches. The two series concepts are shown to be virtually identical when all selection criteria are considered and they are shown to have performance superior to a parallel gas generator concept. However, when flexibility and/or growth potential are considered to be more influential criteria, the parallel concept is shown to have distinct advantages.

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1. INTRODUCTION

To provide the technology base necessary for design of the Space Shuttle, the NASA has sponsored a number of technology programs related to Auxiliary Propulsion Systems. Among these has been a series of design studies aimed at providing the system design data necessary for selection of preferred system concepts and for delineation of requirements for complementing component design and test programs. The first of these system study programs considered a broad spectrum of system concepts, but because of high vehicle impulse requirements coupled with safety, reuse,and logistics considerations,only cryogenic oxygen and hydrogen were considered as a propellant combination. Additionally, unknowns in thruster pulse mode ignition and concerns with the distribution of cryogenic liquids served to eliminate liquid - liquid feed systems from the list of candidate concepts. Therefore, only systems which delivered propellants to the thrusters in a gaseous state were considered for the Reaction Control System (RCS). The results of these initial studies, reported in References A through D, indicated that among the many options for design of a gaseous oxygen-hydrogen system, an approach using heat exchangers to thermally condition the propellants and turbopumps to provide system operating pressure would best satisfy requirements for a fully reusable Space Shuttle. These study programs focused attention to this general system type, but did not examine in depth several viable approaches for turbopump system design and control. To fill this need the NASA contracted with McDonnell Douglas Astronautics Company - Eastern Division in July 1971 for additional study of the Shuttle Auxiliary Propulsion Systems. This contract (NAS 9-12013) titled "Space Shuttle Auxiliary Propulsion System Design Study" was under the technical direction of Mr. Darrell Kendrick, Propulsion and Power Division, Manned Spacecraft Center, Houston, Texas.

As originally defined, this design study was a five phase program considering only oxygen and hydrogen propellants. Reference E provides an executive summary of program results and Reference F provides a detailed description of the program plan for each of the five program phases listed below:

1. Phase A-Requirements Definition
2. Phase B-Candidate RCS Concept Comparisons
3. Phase C-RCS/OMS Integration
4. Phase D-Special RCS Studies
5. Phase E-System Dynamic Performance Analysis

Phase A above defined all design and operating requirements for the Auxiliary Propulsion Systems. The results of this phase,which are documented in Reference G, showed that requirements for the booster and orbiter stages were sufficiently similar to allow concentration of all design effort on the orbiter stage, as the results obtained would be applicable to fly-back type booster stages. In Phase B, which is the subject of this report, detailed design and control analyses for the three most attractive gaseous oxygen-hydrogen Reaction Control System concepts were conducted. Phase C was aimed at defining potential for integration of the RCS with the Orbit Maneuvering System (OMS). As defined by the original contract, only oxygen and hydrogen were considered in this phase. However, vehicle studies which were concurrent with this design effort showed that smaller Shuttle orbiters with external, expendable main engine tankage would provide a more cost effective vehicle approach. This change in vehicle design resulted in a significant reduction in APS requirements and this, coupled with a companion Shuttle program decision to allow scheduled system refurbishment, allowed consideration of systems using earth storable propellants for Auxiliary Propulsion. Thus, in November of 1971, the NASA issued a contract change order that extended the scope of Phase C to include earth storable monopropellant and bipropellant systems and redirected Phase E to provide final performance analyses on storable propellant systems. Reference H provides documentation of Phase C effort on oxygen-hydrogen, and Reference I reports the results of both Phase C and E effort on earth storable propellant systems. In addition to the principal contract effort in Phases B and C, the study included an exploratory effort (Phase D) to evaluate two alternatives to gaseous oxygen-hydrogen RCS using turbopumps. Reference J documents the results of the Phase D studies.

This report documents completely the Phase B study effort which compared in detail, three candidate turbopump RCS concepts. There were three primary categories of effort in this phase:

1. Definition of the most sutiable controls for each of the three candidate RCS
2. Evaluation of system performance and operational characteristics with the selected controls
3. Comparison of the candidate RCS on the basis of performance, complexity, flexibility, reliability and required technology.

This report is organized to provide an overview of the above effort in the body of the report with detailed, substantiating data and analyses provided in attached Appendices. The report body provides a summary description of the study approach followed by a brief discussion of the RCS requirements and constraints that are most pertinent to system design and performance. For these requirements, preliminary system design points are defined on the basis of perfect controls, and the rationale for selection of preferred controls is provided. The system design points are updated to reflect controls effects, and the operation of the system with controls is discussed, including system operation under the most significant malfunction conditions. Finally, the systems are compared against the selection criteria. Appendices A and B respectively present the component models and system analyses used to develop RCS design points. Appendix C provides the detailed analyses of system controls, comparing the effectiveness of various control points. Appendix D provides heat exchanger analyses used for final RCS design, and Appendix E provides a detailed failure mode and effects analysis for the RCS.

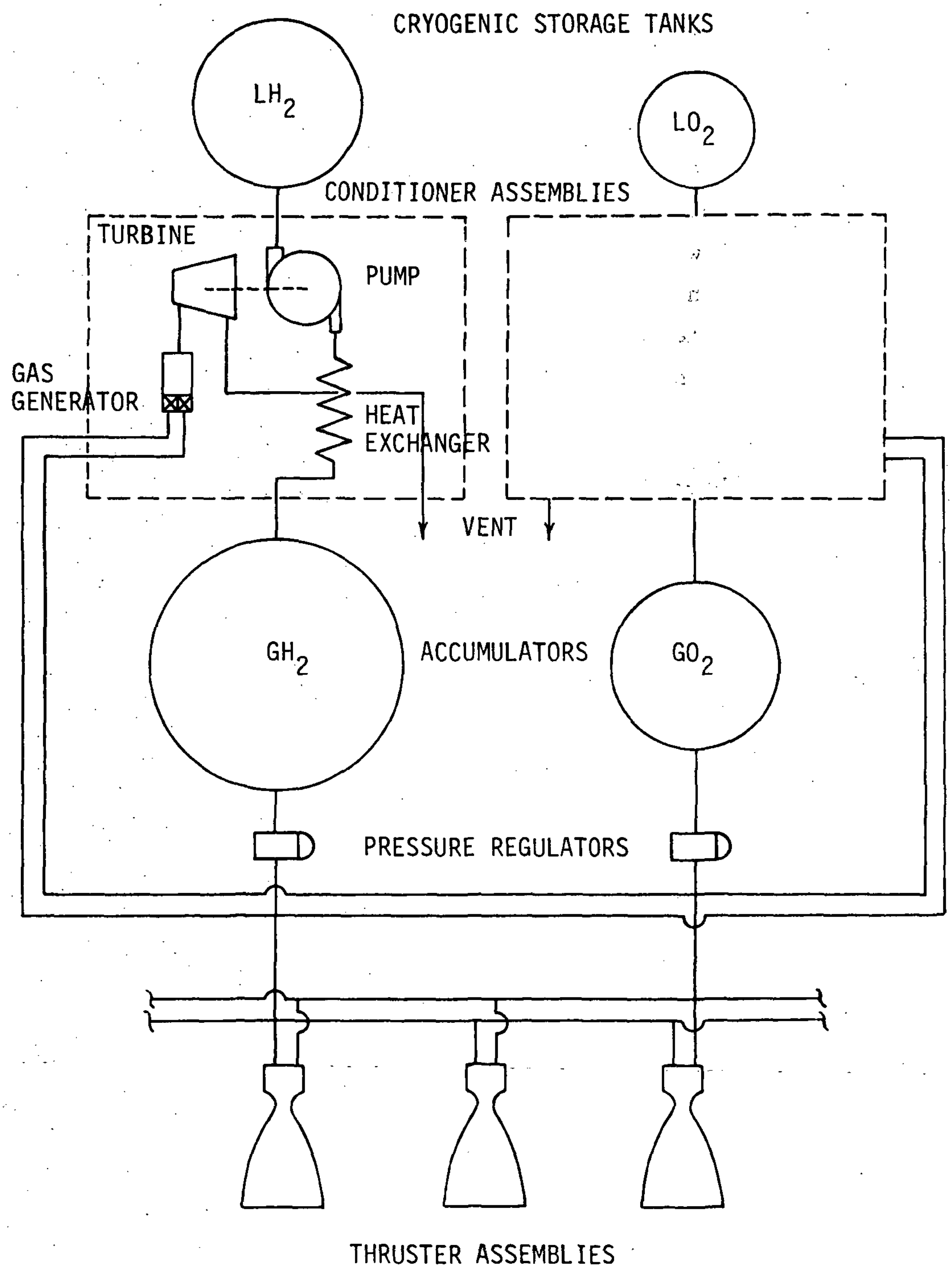
2. CANDIDATE RCS CONCEPTS AND STUDY APPROACH

The basic gaseous oxygen-hydrogen Reaction Control System is shown schematically in Figure 2-1. During system operation, propellants from the cryogenic storage tanks are pumped to high pressure using turbopumps, thermally conditioned to superheated vapors in heat exchangers, and then stored in accumulators. Gaseous propellants are supplied to the thruster assemblies through pressure regulators. The energy for propellant pressure and temperature conditioning is supplied by combustion products from bipropellant gas generators. Oxygen and hydrogen propellants are supplied to the gas generators from the accumulators. The accumulators operate in a blowdown mode with accumulator pressure decaying from a maximum value to a switching value. At the switching value,accumulator resupply is initiated. The gas generators are ignited, providing energy to power the turbopump and heat exchanger. Accumulator pressure continues to decay to a minimum value during the conditioner assembly start transient, and then begins to increase as steady state resupply flowrate is achieved. Resupply flow is maintained by the conditioner assembly until accumulator pressure rebuilds to its maximum value whereupon propellant flow to the gas generators is terminated. The accumulator blowdown/ recharge cycle is repeated as many times as necessary to satisfy mission total impulse requirements. Although the accumulators operate over a wide pressure range, propellant flow is regulated to constant supply pressure downstream of the accumulators, maintaining a constant inlet pressure to the thrusters and gas generators.

Based on previous studies conducted under Contract NAS 8-26248 (References C and D), the NASA identified three high value turbopump concepts differing in the sequence by which energy is extracted from the gas generator combustion products. These three concepts are identified in Figure 2-2 and consist of: (1) a series-upstream turbine concept which uses combustion products from a single gas generator to first power the turbopump and then thermally condition propellant; (2) a series-downstream turbine concept in which the order of gas generator exhaust flow through the heat exchanger and turbine is reversed from the preceeding; and (3) a parallel RCS concept which employs separate gas generators to power the turbopump and heat exchanger independently.

The performance (specific impulse) of these RCS concepts is dictated by the amount of gas generator flow required for propellant conditioning. Gas generator flow requirements for the two series concepts are virtually identical, and are

BASIC RCS CONCEPT



CANDIDATE RCS CONCEPTS

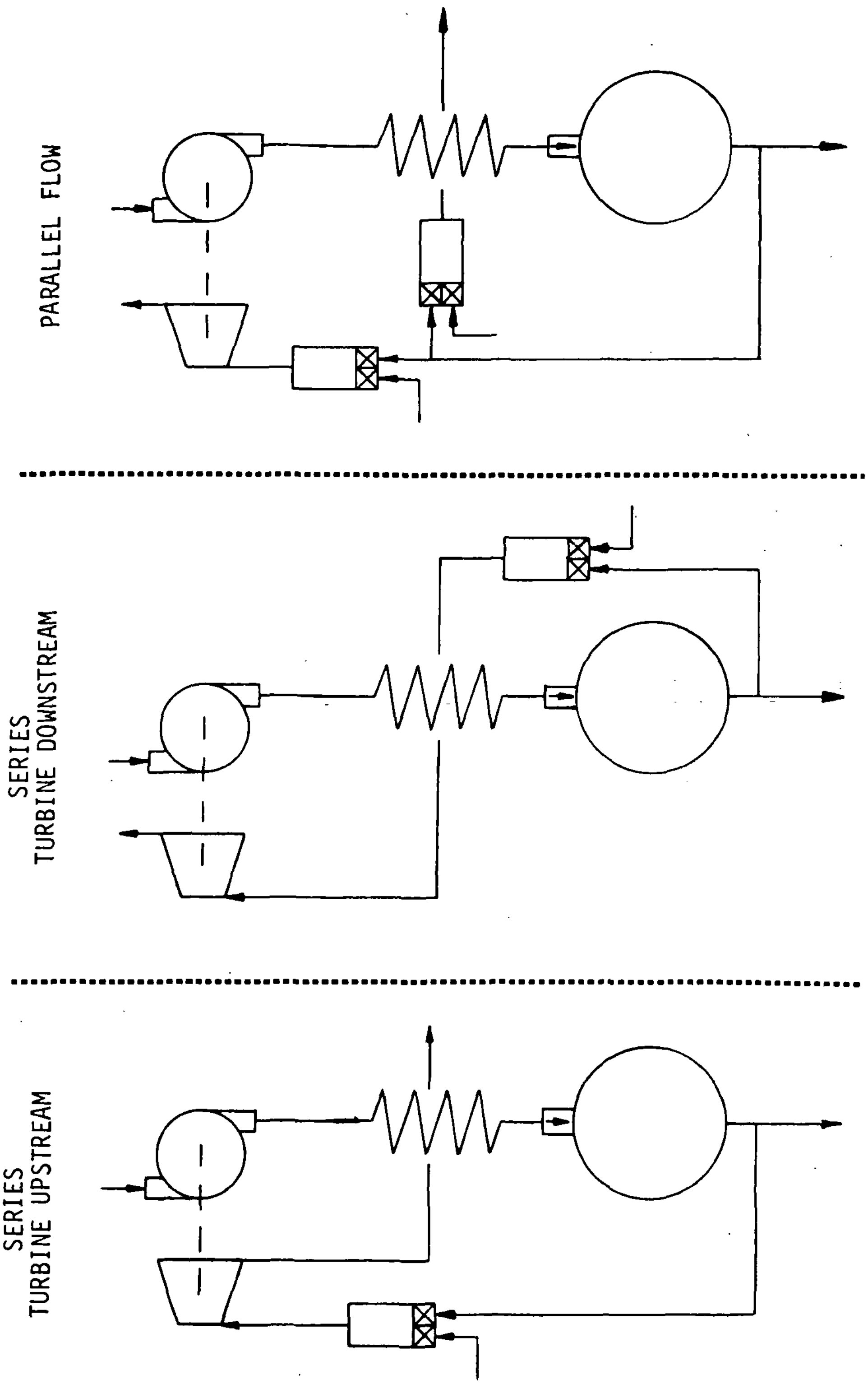


Figure 2-2

slightly less than the parallel concept. The somewhat larger parallel RCS flow requirement results from higher turbine vent exhaust temperatures (higher waste enthalpy) compared with the series concepts. Due to this performance similarity among the three concepts, great attention to detail component/system design differences is required for concept comparison. For example, an important advantage of the series-downstream turbine RCS is its lower turbine rotor blade temperatures, resulting in greater stress margins and higher fatigue life compared with the series-upstream turbine and parallel RCS concepts. A disadvantage is the variation in heat exchanger hot side pressure/temperature drop during off-nominal system operation, causing turbopump delivered power and overall conditioner performance to be less predictable. The primary advantage of the parallel RCS is the use of separate gas generators to power the turbopump and heat exchanger. This eliminates significant interactions between the turbopump and heat exchanger and allows each component to be developed independently.

In previous studies, because of the large numer of system concepts considered, it was not feasible to evaluate each concept on the basis of detailed component design differences. As such, evaluations of the effect of component tolerances on system performance, the effect of component malfunctions on system operational capability, and evaluations of alternate conditioner control approaches were not conducted. The objectives of the current study were to fill these data voids, by comparing the three candidate concepts on a detailed performance basis, and identifing the most attractive concept for the Space Shuttle. The task flow chart for accomplishing these objectives is shown in Figure 2-3. Initially, component analytical models developed under previous study effort (NAS 8-26248) were reviewed and revised as necessary to reflect the RCS requirements defined in Section 3, and also to reflect technology advances occurring after their original formulation. Concurrent with this effort, historical data for component performance tolerances and sensor accuracies were reviewed and compiled for use in subsequent steady-state and transient system performance analyses. Furthermore, system installation drawings showing component locations and line routings were developed, which provided feedline lengths and envelope constraints for primary components. Applying the data generated in these tasks, preliminary system analyses were conducted to establish initial flow balances, operating design points, and system weight sensitivities to design and mission requirements. Reference open-loop (no control) system performance was defined for each concept, applying the component tolerance

PHASE B: CANDIDATE RCS CONCEPT COMPARISONS

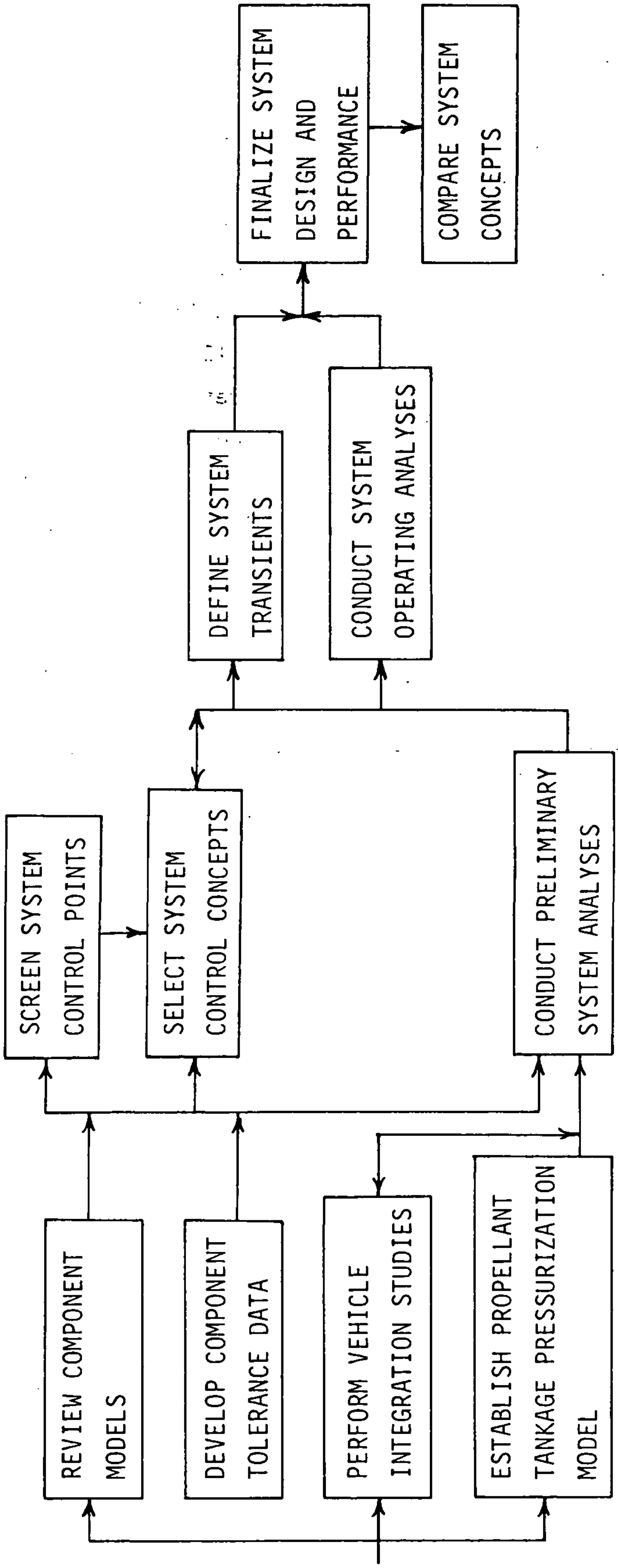


Figure 2-3

data. Numerous control point options were evaluated and compared to the reference open-loop operation in terms of control accuracy and complexity in order to identify the most attractive control approach for each RCS concept. Using the selected control concepts, conditioner transient startup and shutdown performance was defined, and analyses of system transient behavior during critical malfunction modes were conducted. Concurrently, system operating analyses were performed to define variances in system mixture ratio and specific impulse during simulated mission operation. These established required propellant/pressurization tankage weights and volumes. System and component designs were then finalized and the three candidate RCS concepts were compared and ranked on the basis of performance, complexity, flexibility, reliability and required technology.

Pertinent vehicle and system requirements applicable to this Phase B study are defined in the following section, and results from the tasks of Figure 2-3 are summarized in Section 4.

3. VEHICLE AND SYSTEM REQUIREMENTS

The orbiter stage for which the oxygen-hydrogen RCS studies were conducted is illustrated in Figure 3-1. The characteristics of this vehicle were based primarily on the results of MDAC-E studies of fully reusable orbiters and boosters as defined in Reference G. The most distinguishing feature of this orbiter configuration was that the main engine propellant tanks were internal to the vehicle, resulting in a relatively large orbiter stage. Most of the design studies described subsequently in this report used this orbiter as a reference configuration. The exception to this are the growth potential comparisons of Section 5. These show RCS weights at design requirements corresponding to smaller orbiter configurations of the type designed to use external, expendable, main engine tankage.

Reference G provides the detailed analyses, rationale, and vehicle requirements used to develop the RCS requirements tabulated in Figure 3-2. The RCS uses 33 engines at 1150 lbs of thrust each, providing three axis attitude control. The thrust level and thruster arrangement are designed such that, with the failure of any two thrusters the system will provide torque levels sufficient for safe vehicle entry. The total impulse of the system is 2.23 million lbf-sec. This includes total impulse for both attitude control and vernier translation maneuvers less than $\pm$20 ft/sec. The system is capable of sustaining a maximum of 5 thrusters firing (equivalent thrust of 5750 lbf). This corresponds to the use of four control thrusters for a translation maneuver (T/W =0.015) with the equivalent of one additional thruster for vehicle attitude control during the maneuver. The sustained maximum equivalent thrust of the system defines the maximum flow capacity of the turbopumps and heat exchangers. A final requirement that has a major effect on system design is the balance between attitude control impulse and maneuvering impulse. This distribution has a major effect on the life requirements for turbopumps and heat exchangers, as it directly effects the number of operating cycles for these assemblies in each mission. Since the conditioner assemblies must operate each time the accumulators are depleted, a trade-off between increasing accumulator size (and hence system weight) and turbopump-heat exchanger life exists. Reference G defines an attitude control impulse requirement of 965,000 lbf-sec. The remaining RCS impulse (1,265,000 lbf-sec) is used during 10 translation maneuvers. The impulse required for even the smallest translation maneuver is far greater than can reasonably be stored in the gaseous accumulators so the turbopumps must operate during each translation maneuver, independent of accumulator size. The attitude control impulse is considered completely random and this impulse, with the accumulator size, dictates the number of

ORBITER STAGE

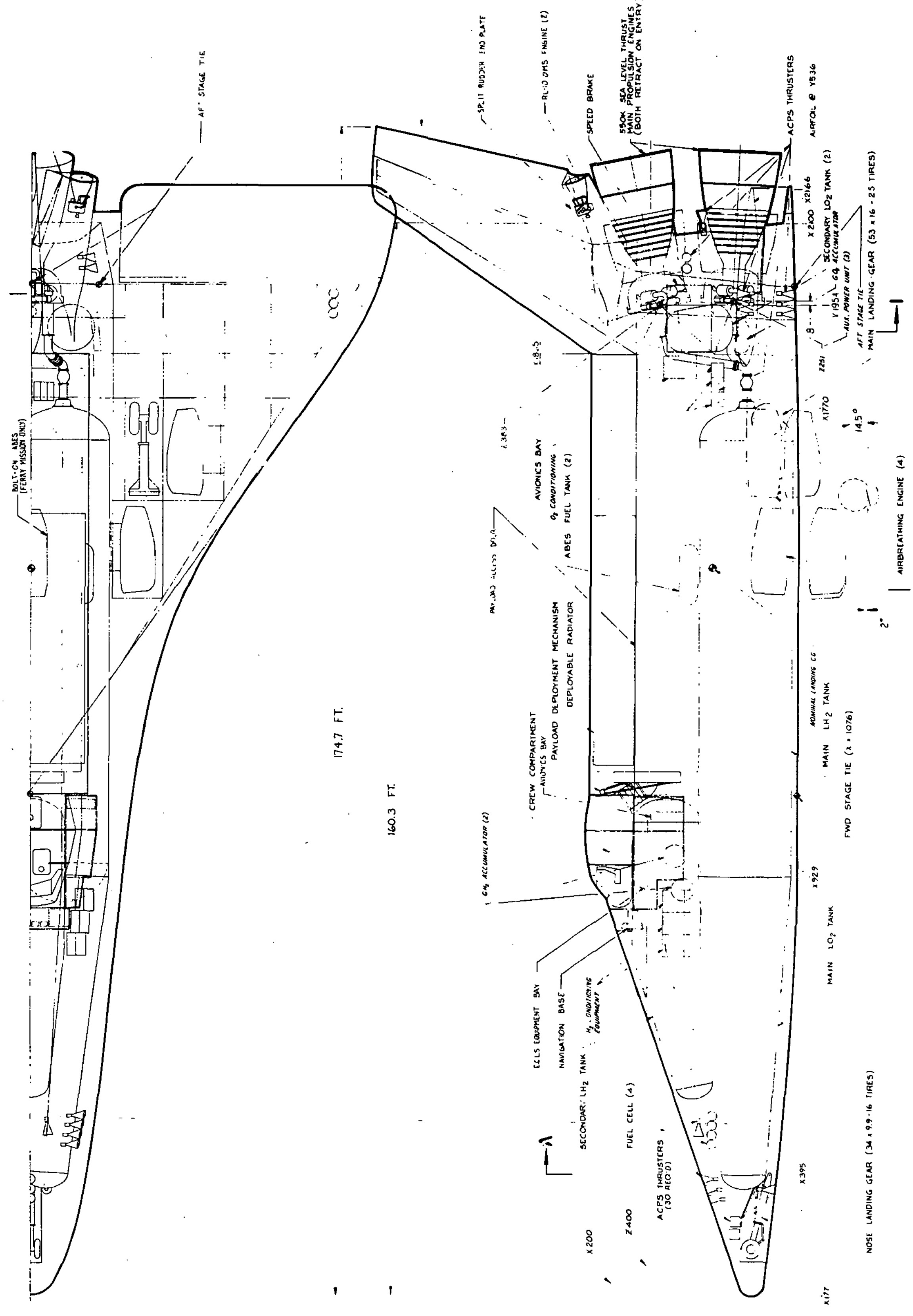


Figure 3-1

additional turbopump-heat exchanger cycles required in each mission. Generally, the systems are sized for a total of 50 cycles of operation in each mission (40 for attitude control, 10 for maneuvers) based on a turbopump life goal of 5,000 cycles in 100 missions.

RCS DESIGN REQUIREMENTS

	ORBITER
NUMBER OF THRUSTERS	33
THRUSTER THRUST (LB)	1150
NUMBER OF CONDITIONERS	3
SYSTEM THRUST (LB)	5750
TOTAL IMPULSE (LB-SEC)	
RESUPPLY	2.23×10^6
EASTERLY LAUNCH	2.23×10^6
SOUTH POLAR	2.15×10^6

Figure 3-2

4. SYSTEM ANALYSES

Presented in this section are the analyses conducted to provide data for concept comparisons. In accordance with the plan of Figure 2-3, initial effort concentrated on development of detailed system schematics, definition of component weight and performance models, and vehicle integration studies. Preliminary system analyses were also conducted to establish optimum design points, weight sensitivities and system pressure/temperature/flow balances. Emphasis then shifted to evaluation of alternate control approaches for maintaining conditioned propellant temperature, pressure and flow-rate within acceptable limits. Concurrent with this evaluation, system steady-state and transient operating analyses were conducted. These determined: (1) the effect of control tolerances on mission propellant utilization; (2) required start-up and shut-down component sequencing; (3) the effect of critical component malfunctions on system operation; and (4) identification of system instrumentation requirements. Contained in the following paragraphs are the results of these analyses.

4.1 Preliminary System Design Points - Preliminary system design points were developed for each of the three RCS concepts based on the assumption of perfect control. The resulting design points are summarized in Figure 4-1 for each of the candidate RCS concepts. Component models employed in the development of these design points are described in Appendix A, and supplementary analyses defining system pressure, temperature and flow balances, and system weight sensitivities to pertinent design parameters are presented in Appendix B. The design points of Figure 4-1, and supplementary analyses of Appendix B provided the basic data necessary for examination of candidate controls.

As shown in Figure 4-1, the two series concepts are nearly identical in bypass flow requirements and system performance. A power balance on these systems requires high hot side heat exchanger flow rate, and at this flow rate, pump power requirements are satisfied with low turbine pressure ratios. As a result, vent pressures are relatively high in the two series concepts. However, in the parallel RCS, low turbine flow rates and corresponding high turbine pressure ratios are required to efficiently utilize the available thermal energy from the gas generator combustion products. Even at these pressure ratios for the parallel RCS, the enthalpy of the exhaust gas is high resulting in a lower system specific impulse. The lower turbine exhaust pressure also increases the turbine vent system weight.

PRELIMINARY RCS DESIGN POINTS

	SERIES RCS UPSTREAM TURBINE		SERIES RCS DOWNSTREAM TURBINE		PARALLEL RCS	
SYSTEM						
Total Impulse, LBF-SEC	2.23M		2.23M		2.23M	
Mixture Ratio	3.11		3.15		2.95	
Specific Impulse, SEC	371		373		355	
THRUSTER						
Thrust Level, LBF	1150		1150		1150	
Mixture Ratio	4.0		4.0		4.0	
Chamber Pressure, LBF/IN^2A	300		300		300	
Expansion Ratio	40:1		40:1		40:1	
Specific Impulse, SEC	433		433		433	
GAS GENERATOR	O_2	H_2	O_2	H_2	O_2	H_2
Combustion Temperature, °R	2061	2060	2061	2058	1997	2000
Flow Rate, LBM/SEC	.811	1.42	.821	1.33	.227 – TPA	.605 – TPA
					.770 – HEX	1.27 – HEX
HEAT EXCHANGER						
Hot Side Inlet Temp, °R	1970	1851	2061	2058	1997	2000
Cold Side Exit Temp, °R	506	263	517	254	473	246
TURBOPUMP ASSEMBLY						
Flowrate, LBM/SEC	11.73	3.80	11.85	3.80	12.11	4.03
Discharge Pressure, LBF/IN^2A	1894	1545	1846	1327	1556	1201
Shaft Horsepower	133	528	129	439	150	560
Turbine Pressure Ratio	2.09	2.59	2.87	7.98	20	20
ACCUMULATOR						
No. Cycles	50	50	50	50	50	50
Volume, FT^3	13.6	41.5	13.6	43.2	14.7	47.7
Pressures, LBF/IN^2A – Max	1550	1300	1551	1265	1448	1195
Switch	666	665	666	660	660	659
Min	571	571	571	571	571	571
WEIGHT	10,168		10,184		10,763	

*BASELINE FOR STUDY

Figure 4-1

4.2 Conditioner Controls Evaluation - At the design points defined in the preceeding paragraph, conditioner performance was evaluated with various control options to determine the best control approaches for each RCS concept. Both passive and active control options were considered and the detailed evaluations are presented in Appendix C. On the basis of these evaluations, it was concluded that active controls are required to provide acceptable conditioner performance. This conclusion was based both on the high weight penalty for open-loop operation (referenced to perfect control) and hardware design limitations associated with passive controls. The preferred controls are identified in Figure 4-2. Gas generator combustion temperature control was considered mandatory to preclude excessive gas temperatures at the turbine/heat exchanger inlet. Based on a comparison of options for this control, modulation of the gas generator O_2 valve was selected. By controlling gas generator combustion temperature, the operating dispersions throughout the system were reduced, resulting in a substantial system weight savings. With gas generator combustion temperature control defined as a baseline, the next most important effects were found to be system weight sensitivity to variations in hydrogen conditioned temperature (all three RCS concepts) and oxygen conditioned temperature (parallel RCS, only). Control of hydrogen conditioned temperature in the two series RCS concepts and both hydrogen and oxygen conditioned temperature in the parallel RCS was best achieved through modulation of heat exchanger cold side bypass flow. (Detailed heat exchanger design analyses with bypass flow are presented in Appendix D.) The final system control maintains pump discharge pressure (flow) by modulation of the gas generator H_2 valve. This control provides a small additional weight reduction, but its primary benefit is maintenance of constant heat exchanger cold side inlet conditions, minimizing the potential for flow instability which has been encountered in previous heat exchanger development programs (Appendix C). Incremental system weight changes associated with these controls are presented in Figures 4-3 through 4-5 for each RCS concept. Typical operating performance maps for the RCS conditioner assemblies are given in Figures 4-6 through 4-8 for the hydrogen conditioners, and conditioner operating bands are tabulated in Figures 4-9 and 4-10 for the selected controls.

4.3 Simulated Mission Operation - The system weights shown in Figures 4-3 through 4-5 were developed by simulating RCS mission duty cycles to determine the effect of conditioner operating bands on system mixture ratio and specific impulse. This was accomplished in two steps: (1) conditioner operating bands for each control

RECOMMENDED CONTROL CONCEPTS

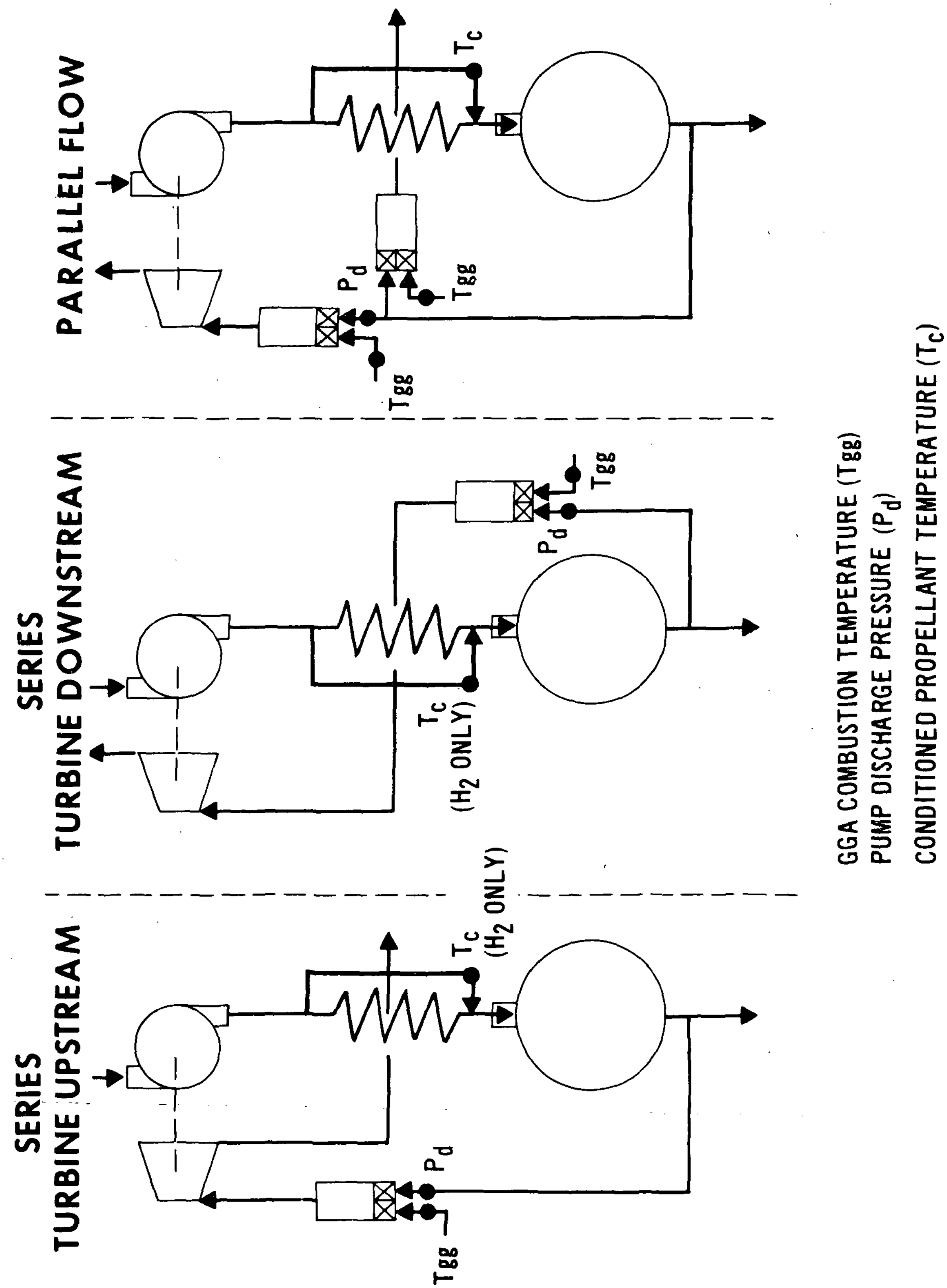


Figure 4-2

SERIES RCS (TURBINE UPSTREAM) CONTROLS EVALUATION

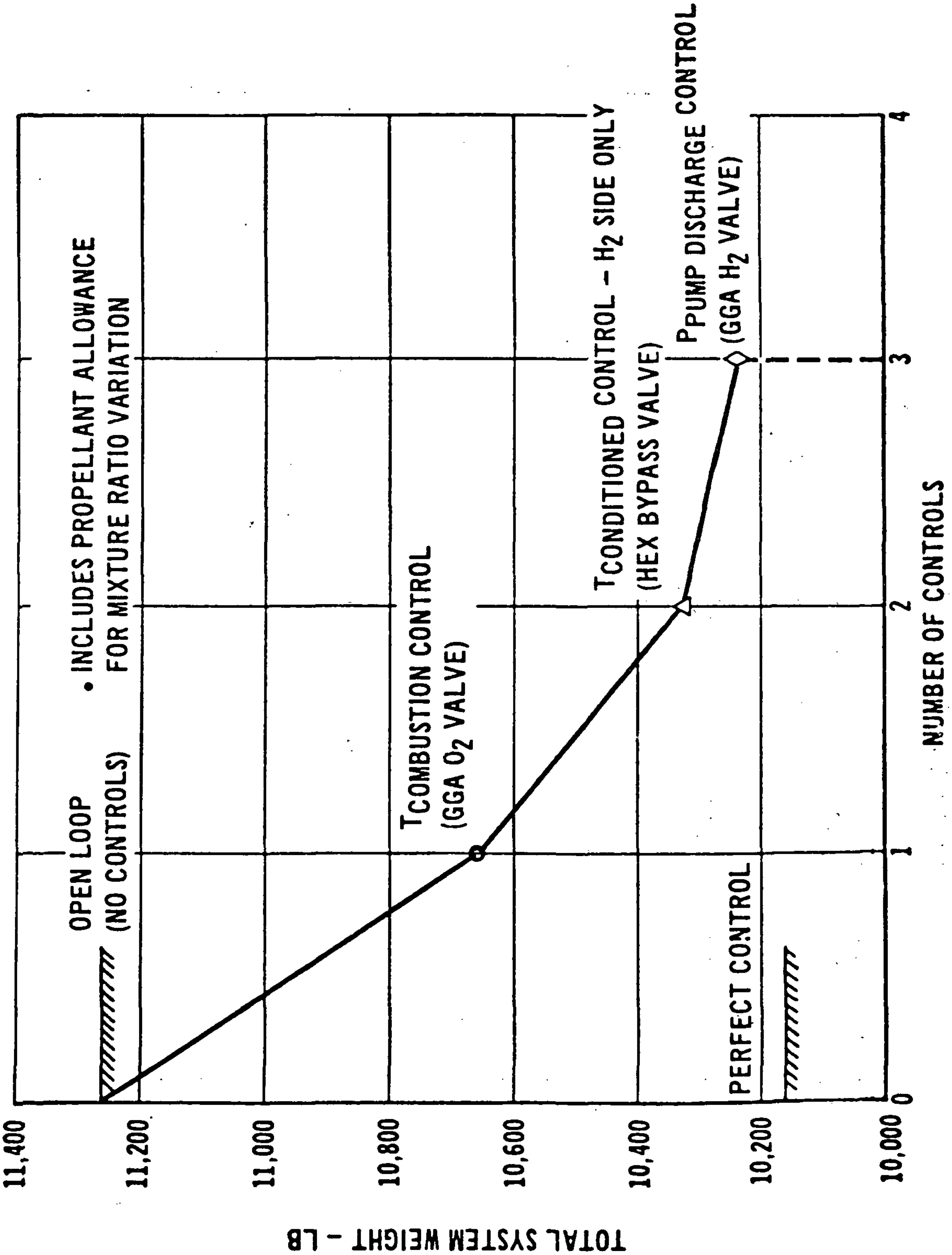
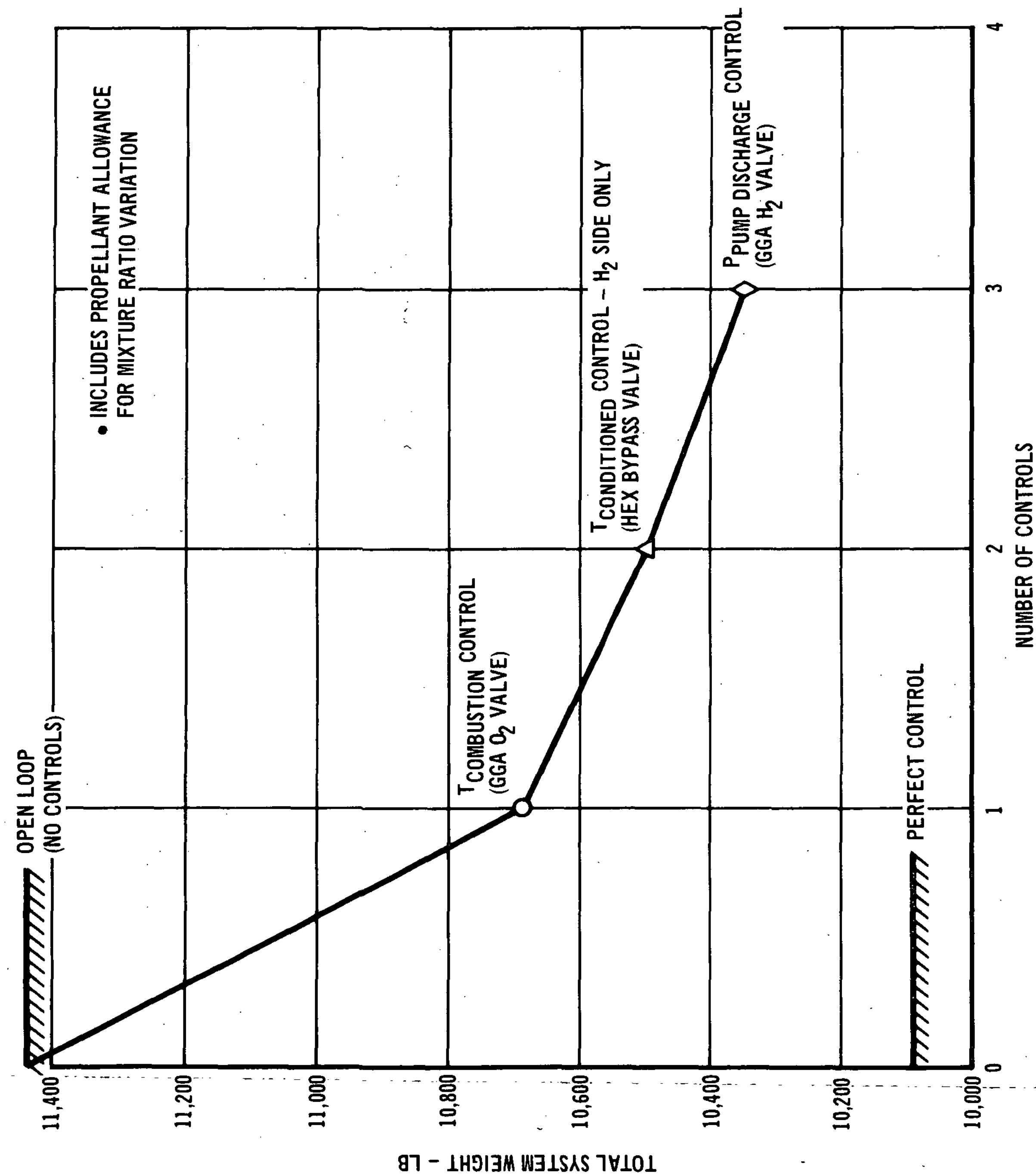


Figure 4-3

SERIES RCS (TURBINE DOWNSTREAM) CONTROLS EVALUATION



PARALLEL RCS CONTROLS EVALUATION

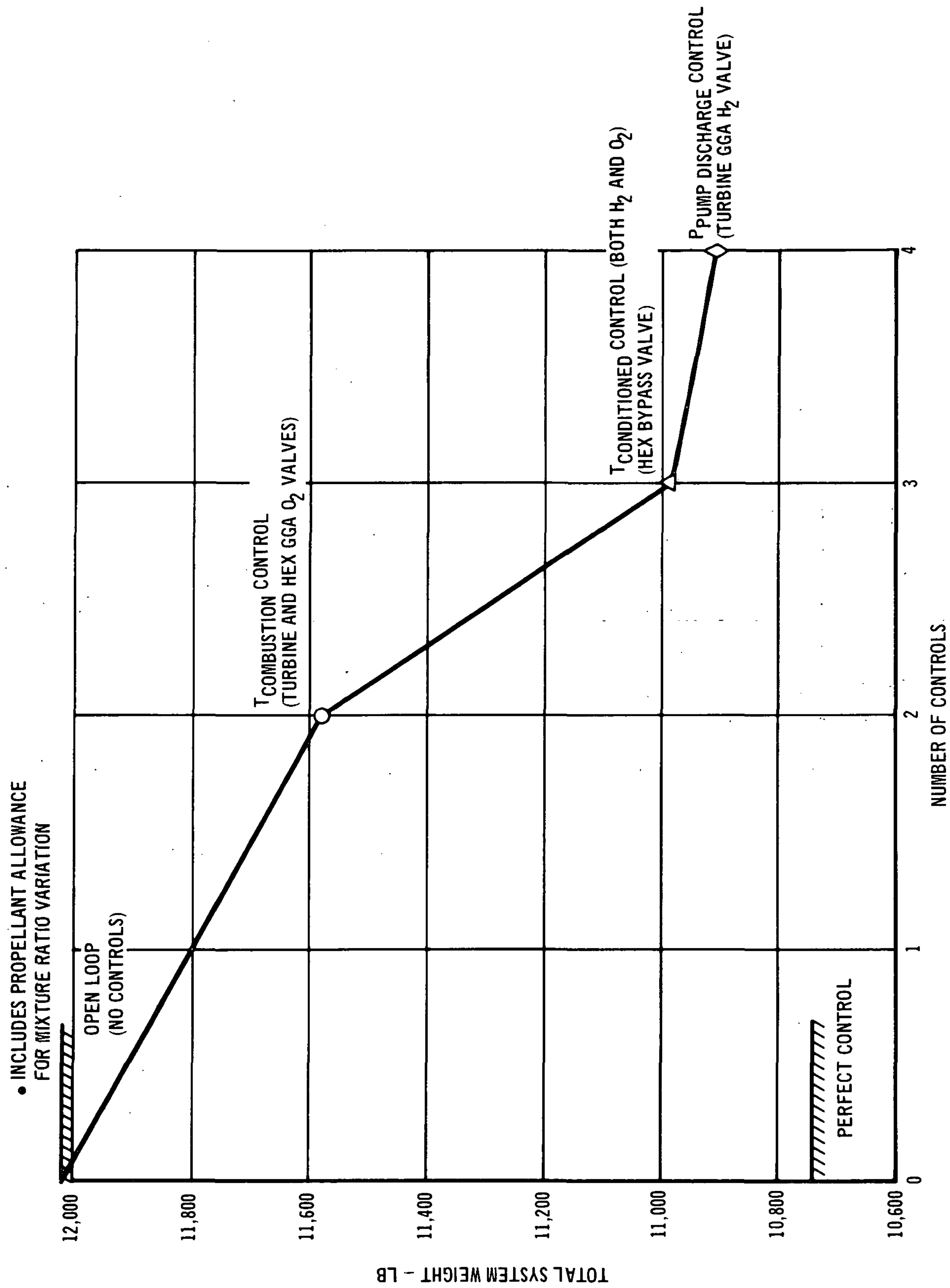


Figure 4-5

SERIES RCS (TURBINE UPSTREAM) CONDITIONER PERFORMANCE MAP

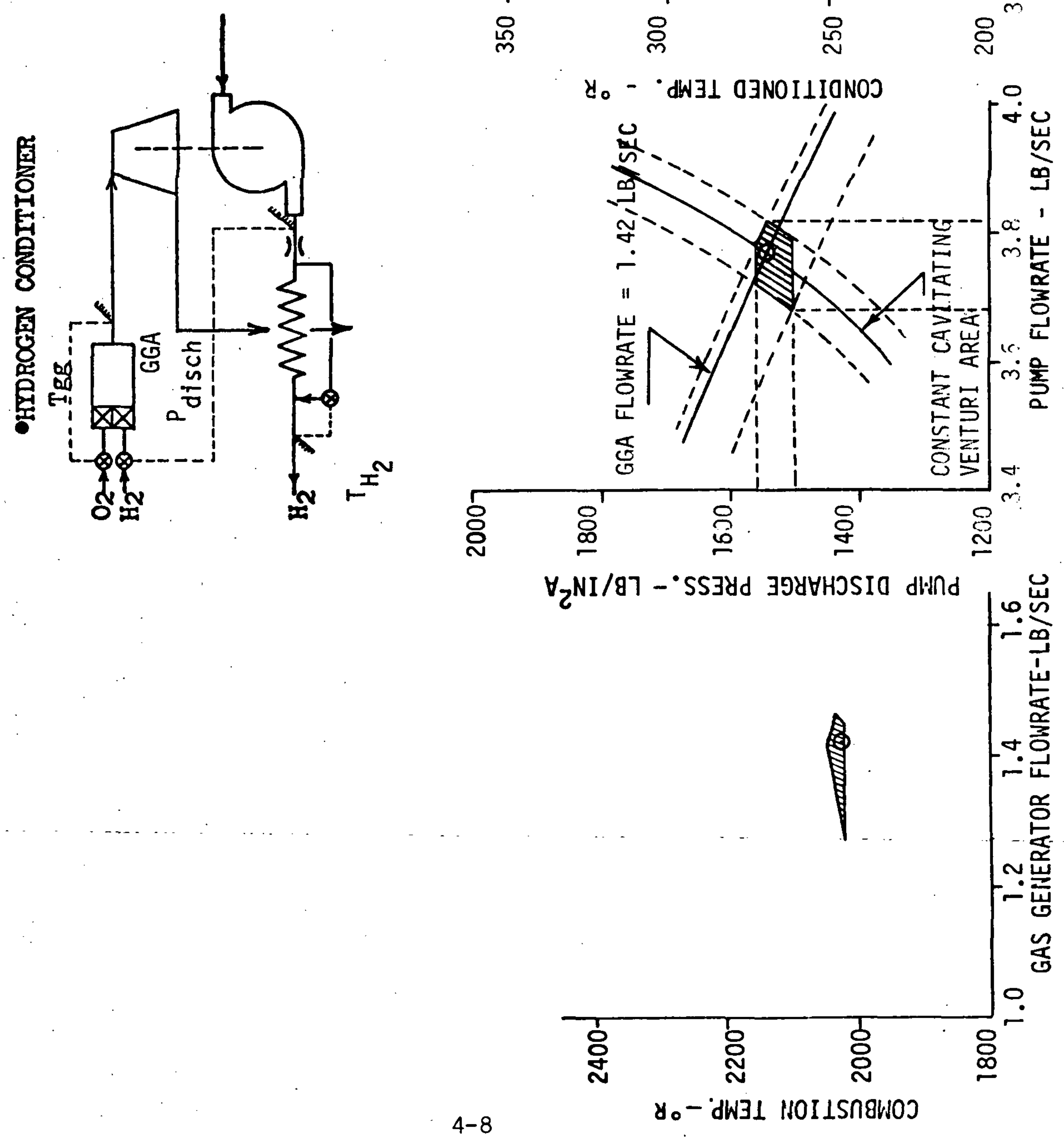


Figure 4-6

SERIES RCS (TURBINE DOWNSTREAM) CONDITIONER PERFORMANCE MAP

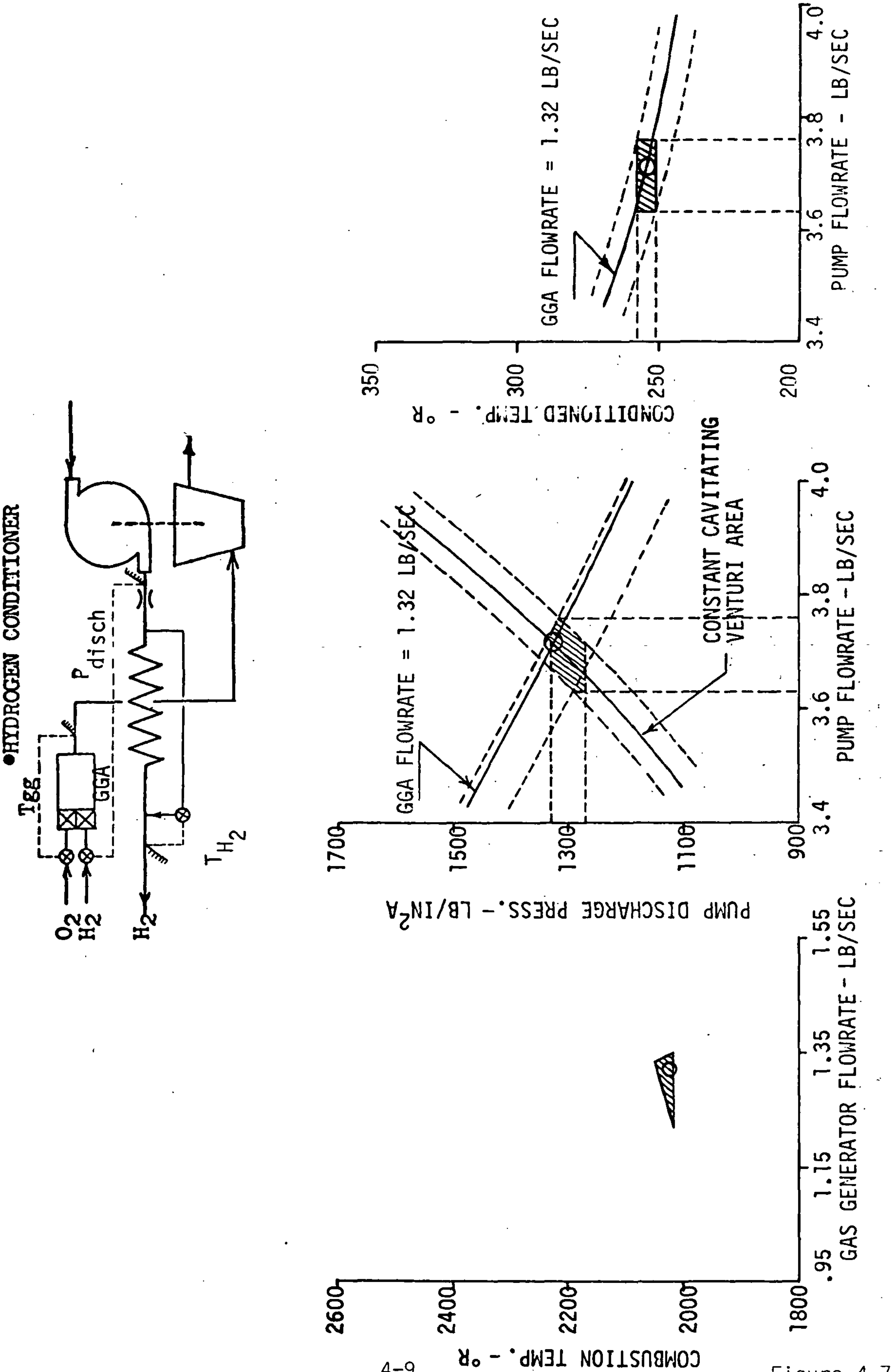


Figure 4-7

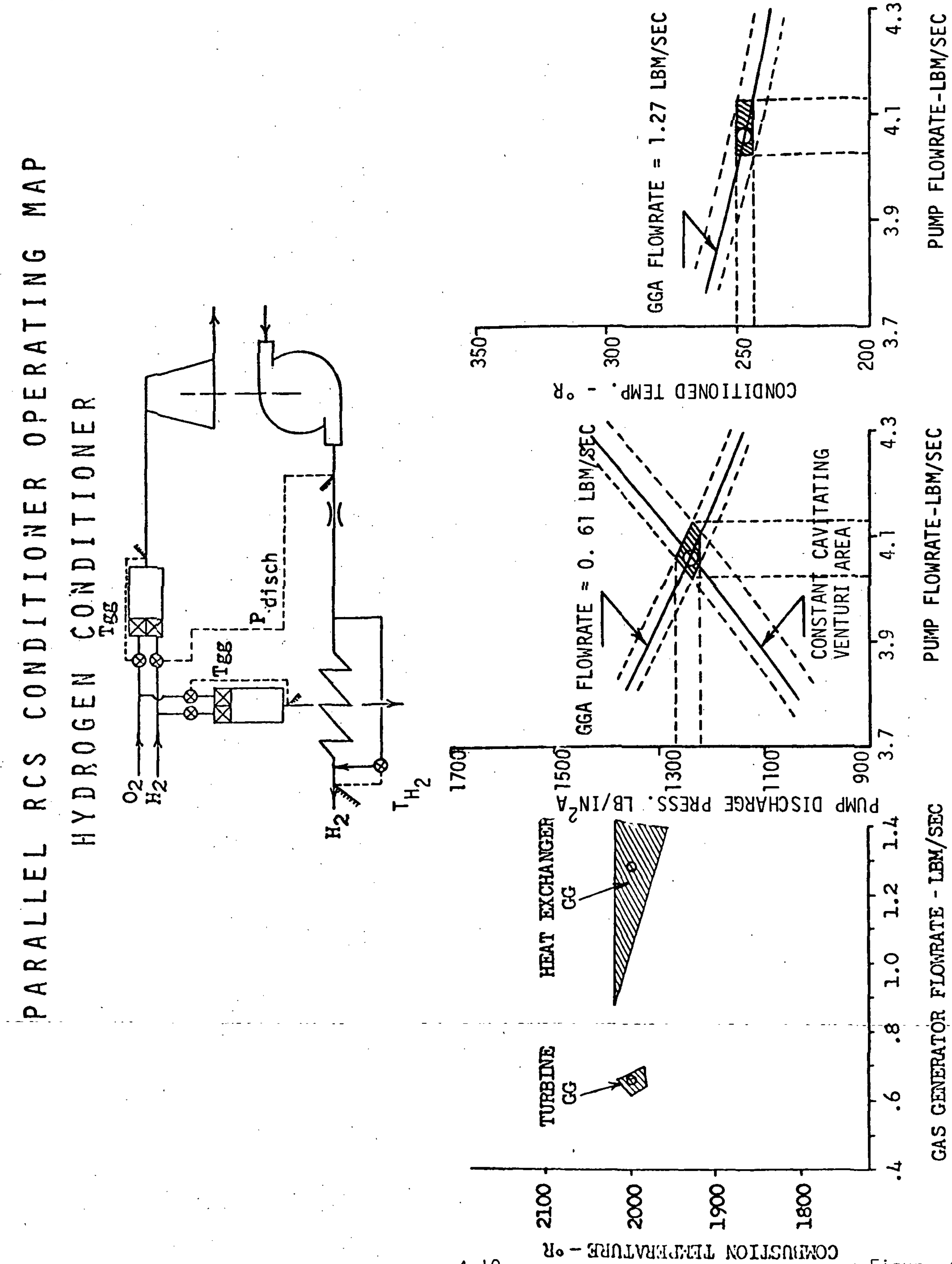
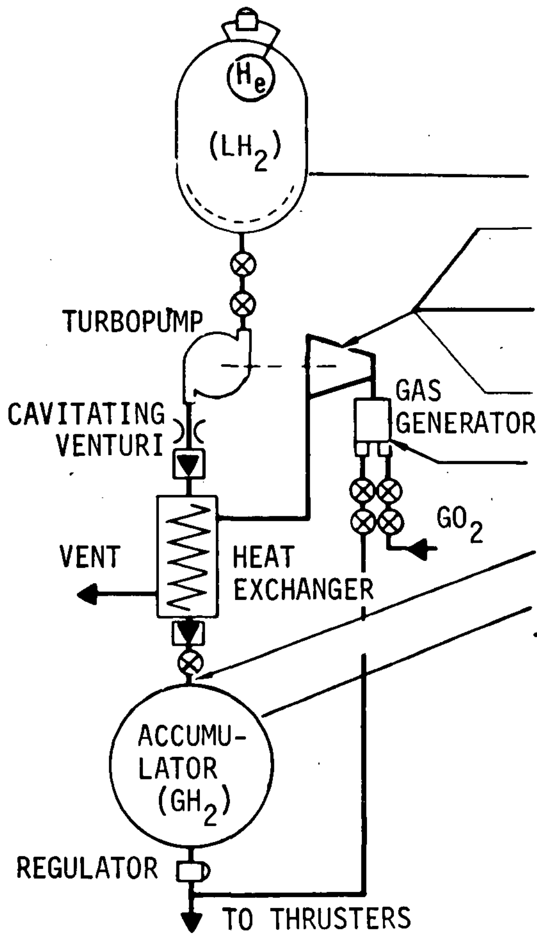

PARALLEL RCS CONDITIONER OPERATING MAP
HYDROGEN CONDITIONER
O_2
H_2
T_{gg}
T_{gg}
P_{disch}
H_2
T_{H_2}
GGA FLOWRATE = 1.27 LBM/SEC
CONDITIONED TEMP. - °R
350
300
250
200
3.7
3.9
4.1
4.3
PUMP FLOWRATE-LBM/SEC
GGA FLOWRATE = 0. 61 LBM/SEC
CONSTANT CAVITATING VENTURI AREA
PUMP DISCHARGE PRESS. LB/IN^{2}A
1700
1500
1300
1100
900
3.7
3.9
4.1
4.3
PUMP FLOWRATE-LBM/SEC
TURBINE GG
HEAT EXCHANGER GG
COMBUSTION TEMPERATURE - °R
2100
2000
1900
1800
.4
.6
.8
1.0
1.2
1.4
GAS GENERATOR FLOWRATE - LBM/SEC

Figure 4-8

DESIGN AND OPERATING CHARACTERISTICS (HYDROGEN SIDE)

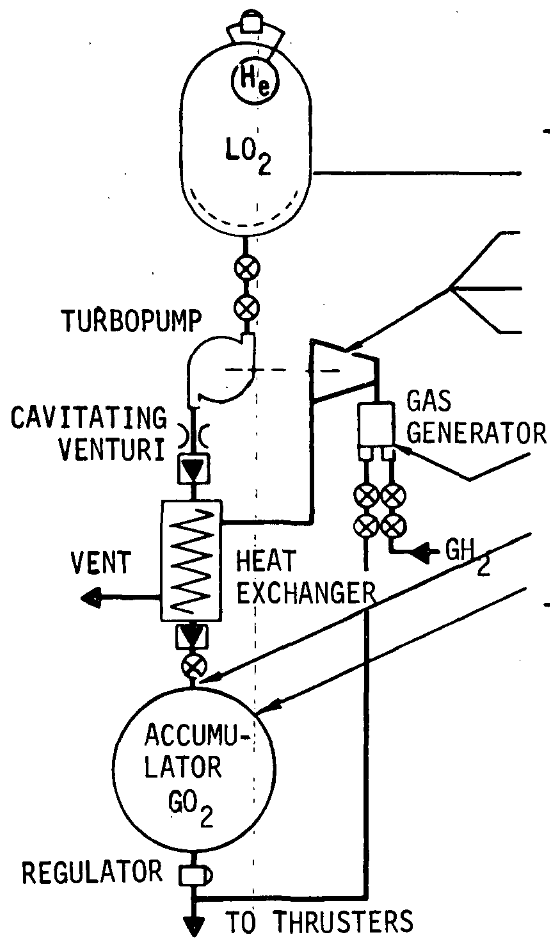


	SERIES RCS (TURBINE UPSTREAM)	SERIES RCS (TURBINE DOWNSTREAM)	PARALLEL RCS
LIQUID HYDROGEN WEIGHT, LB (INCLUDING Δ MR EFFECTS)	1713	1729	1926
TURBINE HORSEPOWER	715 - 762	596 - 617	571 - 598
PUMP FLOW RATE, LBM/SEC	3.68 - 3.81	3.64 - 3.77	4.02 - 4.13
PUMP DISCHARGE PRESSURE, LB/IN^{2}A	1505 - 1533	1290 - 1332	1223 - 1261
GAS GENERATOR COMBUSTION TEMPERATURE, °R	2020 - 2037	2016 - 2022	{1961 - 2020 / 1966 - 2016}*
CONDITIONED PROPELLANT TEMPERATURE, °R	260 - 266	250 - 256	241 - 249
ACCUMULATOR VOLUME, FT^3	42.5	44.8	49.5

* {HEAT EXCHANGER GGA / TURBINE GGA}

Figure 4-9

DESIGN AND OPERATING CHARACTERISTICS (OXYGEN SIDE)



	SERIES RCS (TURBINE UPSTREAM)	SERIES RCS (TURBINE DOWNSTREAM)	PARALLEL RCS
LIQUID OXYGEN WEIGHT, LB (INCLUDING Δ MR EFFECTS)	4653	4708	4928
TURBINE HORSEPOWER	174 - 195	164 - 177	153 - 161
PUMP FLOW RATE, LBM/SEC	11.40-12.49	11.23-12.14	12.16-12.50
PUMP DISCHARGE PRESSURE, LB/IN^2A	1795-2079	1725-1919	1599-1650
GAS GENERATOR COMBUSTION TEMPERATURE, °R	2019-2071	2009-2052	{1968-2020* / 1968-2016}
CONDITIONED PROPELLANT TEMPERATURE, °R	495-531	477-523	462-470
ACCUMULATOR VOLUME, FT^3	14.1	14.9	15.1

* {HEAT EXCHANGER GGA / TURBINE GGA}

Figure 4-10

option were applied to determine the accumulator volumes required to provide 50 recharge cycles (i.e., 40 random cycles and 10 maneuver cycles); and (2) at these accumulator volumes and operating bands, the average system mixture ratio and average specific impulse were determined for a typical mission duty cycle. Accumulator sizing was accomplished as described in Appendix A, Paragraph A.5 and the mission duty cycle was simulated using the operational performance program described in Reference K. A typical mission duty cycle simulation is presented in Figure 4-11 for the series-upstream turbine RCS with the selected controls. System mixture ratio and specific impulse variances for each control option are tabulated in Figures 4-12 through 4-14 for the three RCS concepts. Considering the results of Figure 4-12 for the series-upstream turbine RCS, it is seen that impulsive propellant requirements can be reduced from 6780 lbm for the open loop (no control) case, to 6055 lbm with preferred controls. A summary of mission operational parameter variances for each RCS with selected controls is presented in Figure 4-15. These analyses provided assurance that conditioner control concepts were compared on the basis of valid system weight determinations.

4.4 Transient/Malfunction Analyses - Transient analyses were conducted using the conditioner assembly transient computer program (Reference L) to define conditioner startup and shutdown times, and to establish component sequencing for each RCS concept. These are described in the following paragraphs along with startup and shutdown component sequencing, and selected malfunction analyses. Conditioner instrumentation requirements derived from these analyses and a complete conditioner failure mode and effects analysis (FMEA) applicable to all three RCS concepts are contained in Appendix E.

4.4.1 Startup/Shutdown Transients - Typical start transients for the series-upstream turbine RCS, hydrogen and oxygen loops, are presented in Figures 4-16 and 4-17, respectively. As described in Appendix A, a recirculation loop is employed for pump cooling. Therefore, a simple valve-orifice assembly was modeled to bypass liquid propellant back into the tank prior to and during the start transient. The bypass loop was sized for 50% design pump flow at minimum accumulator pressure. When pump discharge pressure exceeded 200 $lbf/in.^2a$, a 200 ms latching valve was signalled closed, stopping the bypass flow. As shown in the example of Figure 4-16, the startup time to 75% of design pump flow is approximately 0.5 seconds. Similar results are shown in Figure 4-17 for the series-upstream turbine oxygen loop, and Figure 4-18 for the series-downstream turbine hydrogen loop. However, as shown in Figure 4-19, the parallel RCS yields somewhat slower response due to the contribution

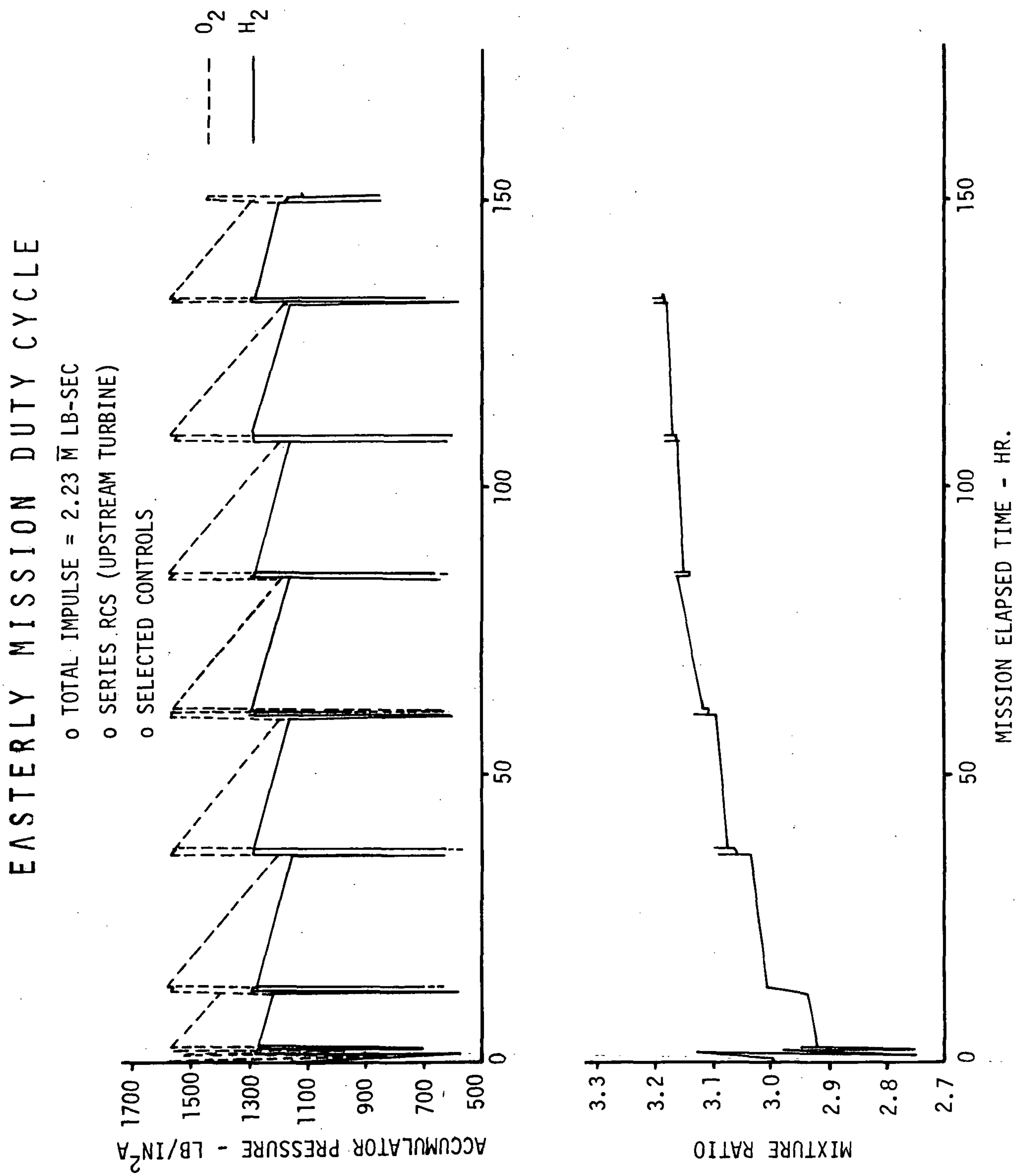
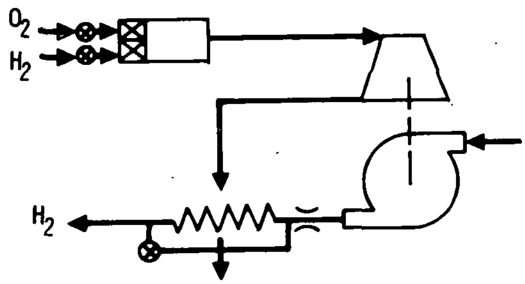

EASTERLY MISSION DUTY CYCLE
o TOTAL IMPULSE = 2.23 $\overline{M}$ LB-SEC
o SERIES RCS (UPSTREAM TURBINE)
o SELECTED CONTROLS
O_2
H_2
ACCUMULATOR PRESSURE - LB/IN^{2}A
1700
1500
1300
1100
900
700
500
0
50
100
150
MIXTURE RATIO
3.3
3.2
3.1
3.0
2.9
2.8
2.7
MISSION ELAPSED TIME - HR.

Figure 4-11

MISSION OPERATING PERFORMANCE VARIATIONS



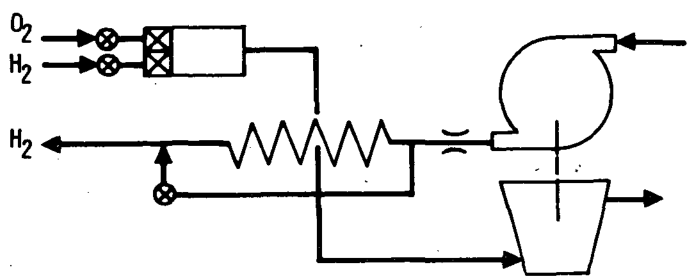
- SERIES – UPSTREAM TURBINE RCS
- EASTERLY MISSION
- TOTAL IMPULSE = 2.23 $\bar{M}$ LB-SEC

	SYSTEM AVERAGE I_{s_p} – SEC	SYSTEM AVERAGE MR	MAXIMUM IMPULSIVE H_2 – LBM	MAXIMUM IMPULSIVE O_2 – LBM	MAXIMUM IMPULSIVE PROPELLANT – LBM
• OPEN LOOP (NO CONTROLS)	346–370*	2.30–4.03*	1950	4830	6780
• Tgg CONTROL	358–369	2.75–3.59	1660	4730	6390
• Tgg + P_d CONTROL	361–369	2.86–3.46	1600	4694	6294
• Tgg + T_{H_2} CONTROL	367–368	3.07–3.20	1495	4610	6105
• PERFECT CONTROL	368	3.13	1467	4588	6055

*SPECIFIED RANGE IS THE RESULT OF CONDITIONER OPERATING TOLERANCES

Figure 4-12

MISSION OPERATING PERFORMANCE VARIATIONS



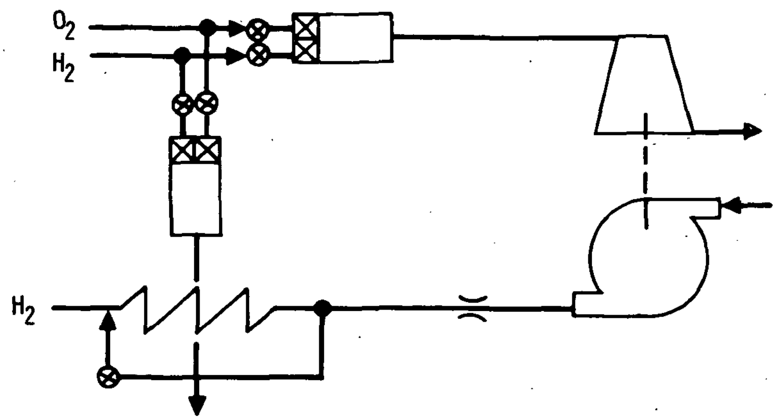
- SERIES – DOWNSTREAM TURBINE RCS
- EASTERLY MISSION
- TOTAL IMPULSE = 2.23 $\bar{M}$ LB-SEC

	SYSTEM AVERAGE I_{sp} – SEC	SYSTEM AVERAGE MR	MAXIMUM IMPULSIVE H_2 – LBM	MAXIMUM IMPULSIVE O_2 – LBM	MAXIMUM IMPULSIVE PROPELLANT – LBM
• OPEN LOOP (NO CONTROLS)	344-370*	2.20-4.09*	2030	4840	6870
• Tgg CONTROL	359-369	2.80-3.55	1635	4720	6355
• Tgg + T_{H_2} CONTROL	362-368	2.93-3.37	1570	4667	6237
• Tgg + T_{H_2} + P_d CONTROL	363-368	2.95-3.37	1555	4666	6221
• PERFECT CONTROL	368	3.13	1467	4588	6055

*SPECIFIED RANGE IS THE RESULT OF CONDITIONER OPERATING TOLERANCES

Figure 4-13

MISSION OPERATING PERFORMANCE VARIATIONS



- PARALLEL RCS
- EASTERLY MISSION
- TOTAL IMPULSE = 2.23 $\bar{M}$ LB–SEC

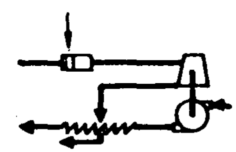
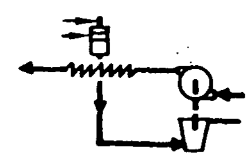
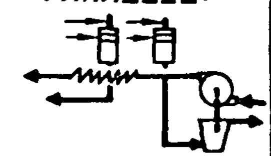
	SYSTEM AVERAGE I_{sp} – SEC	SYSTEM AVERAGE MR	MAXIMUM IMPULSIVE H_2 – LBM	MAXIMUM IMPULSIVE O_2 – LBM	MAXIMUM IMPULSIVE PROPELLANT – LBM
• OPEN LOOP (NO CONTROLS)	328–352*	2.14–3.88*	2165	5040	7205
• Tgg CONTROL (BOTH TURBINE & HEX GGA'S)	337–351	2.49–3.47	1897	4930	6827
• Tgg + T_{H_2} CONTROL	346–350	2.76–3.13	1718	4830	6548
• Tgg + T_{H_2} + T_{O_2} + P_d CONTROL	347–349	2.86–3.02	1666	4800	6466
• PERFECT CONTROL	350	2.93	1620	4750	6370

*SPECIFIED RANGE IS THE RESULT OF CONDITIONER OPERATING TOLERANCES

Figure 4-14

MISSION OPERATIONAL VARIANCES

- o $I_t = 2.23\ \overline{M}$ LB - SEC
- o EASTERLY MISSION
- o SELECTED CONTROLS

	SERIES: 	SERIES: 	PARALLEL: 
SYSTEM MIXTURE RATIO *	3.07 - 3.20	2.95 - 3.37	2.69 - 3.25
SYSTEM SPECIFIC IMPULSE, SEC. *	367 368	363 - 368	347 - 349
ACCUMULATOR TEMP., °R			
HYDROGEN	236 - 500	227 - 500	220 - 500
OXYGEN	380 - 555	382 - 555	352 - 555
IMPULSIVE PROPELLANT USAGE, LBM			
HYDROGEN	1495	1555	1666
OXYGEN	4610	4666	4800

* SPECIFIED RANGE IS THE RESULT OF CONDITIONER OPERATING TOLERANCES

Figure 4-15

SERIES (TURBINE UPSTREAM)
HYDROGEN CONDITIONER STARTUP TRANSIENTS

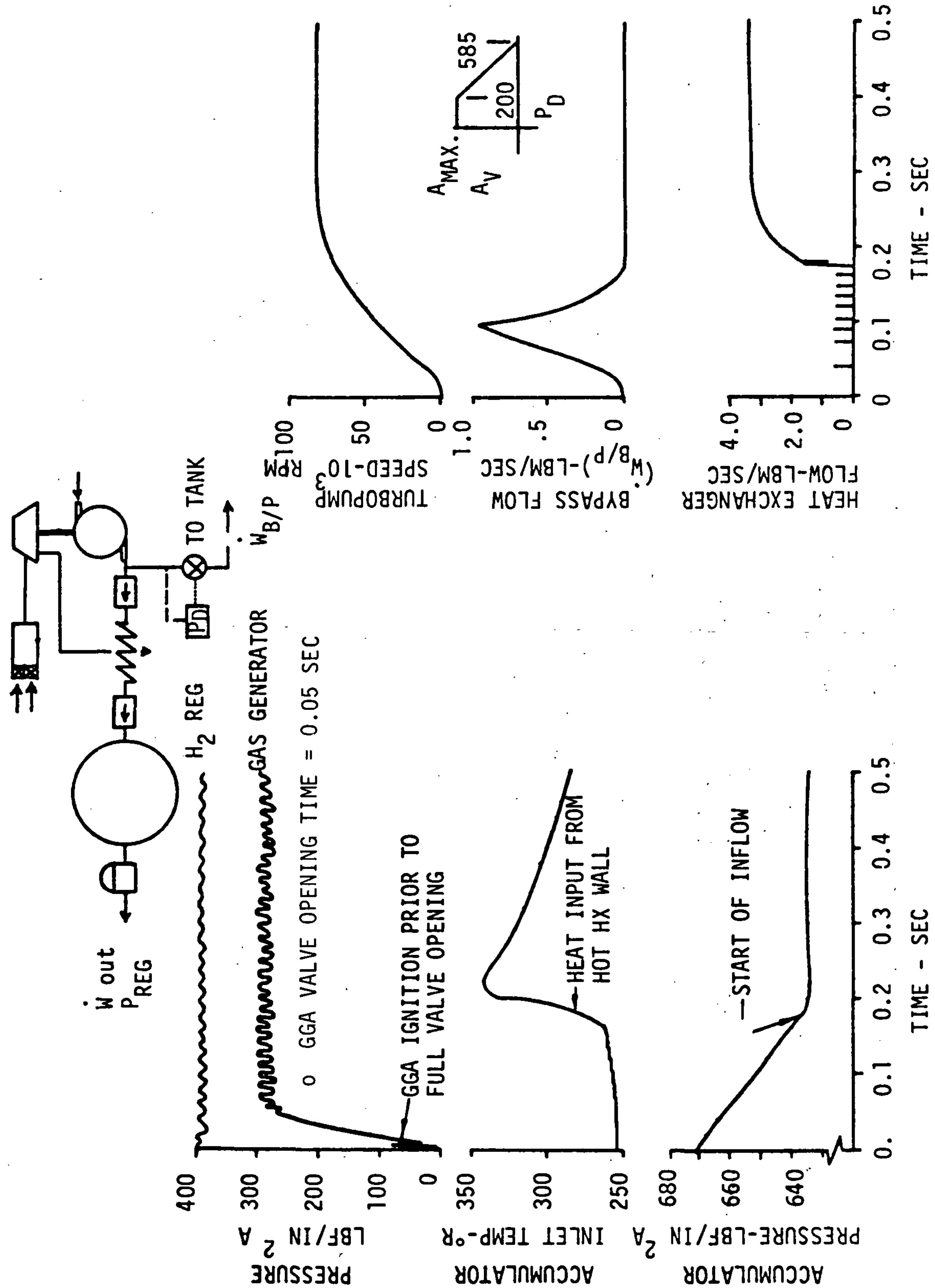
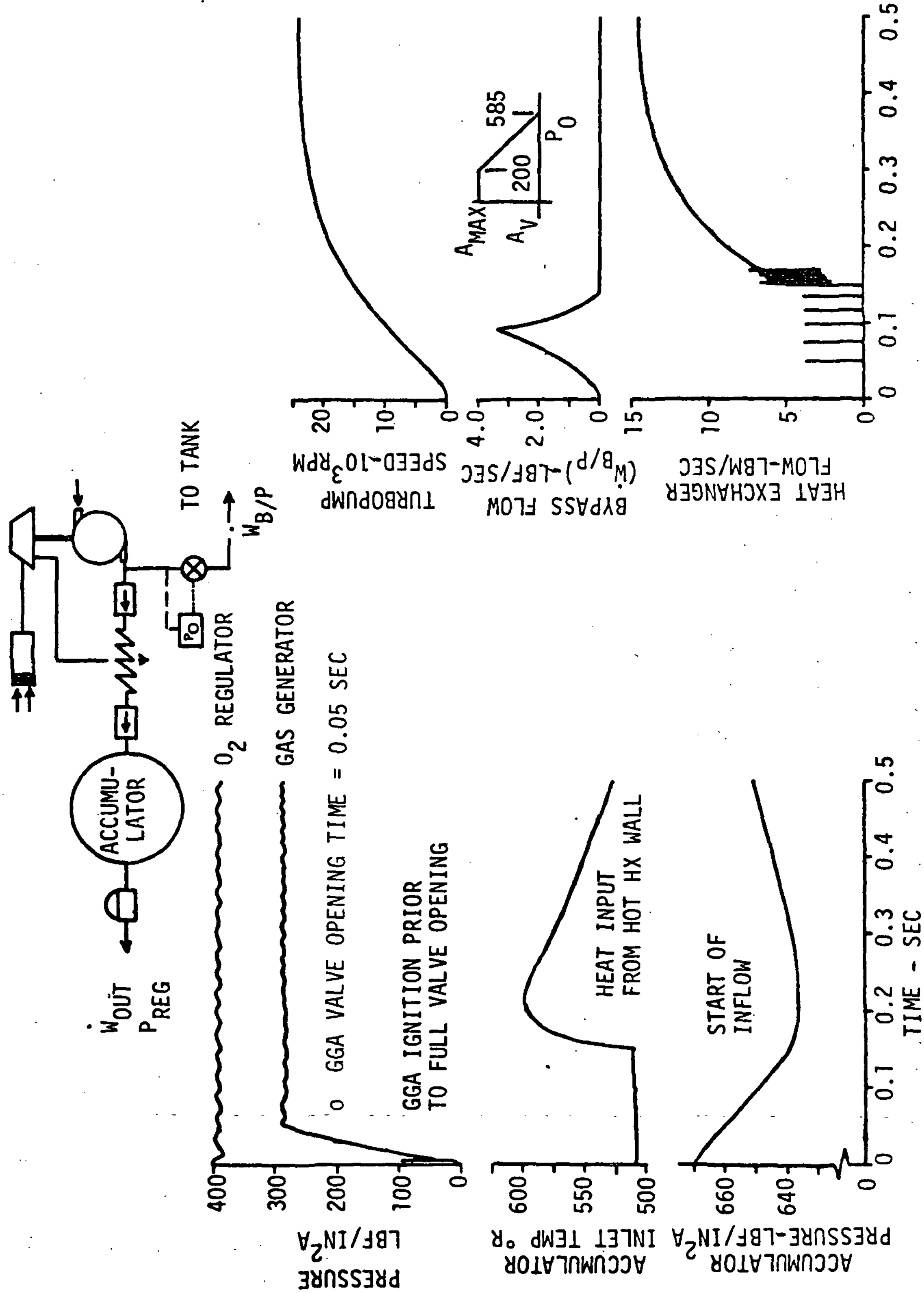


Figure 4-16

SERIES (TURBINE UPSTREAM) OXYGEN CONDITIONER STARTUP TRANSIENTS



SERIES (TURBINE DOWNSTREAM) HYDROGEN CONDITIONER STARTUP TRANSIENTS

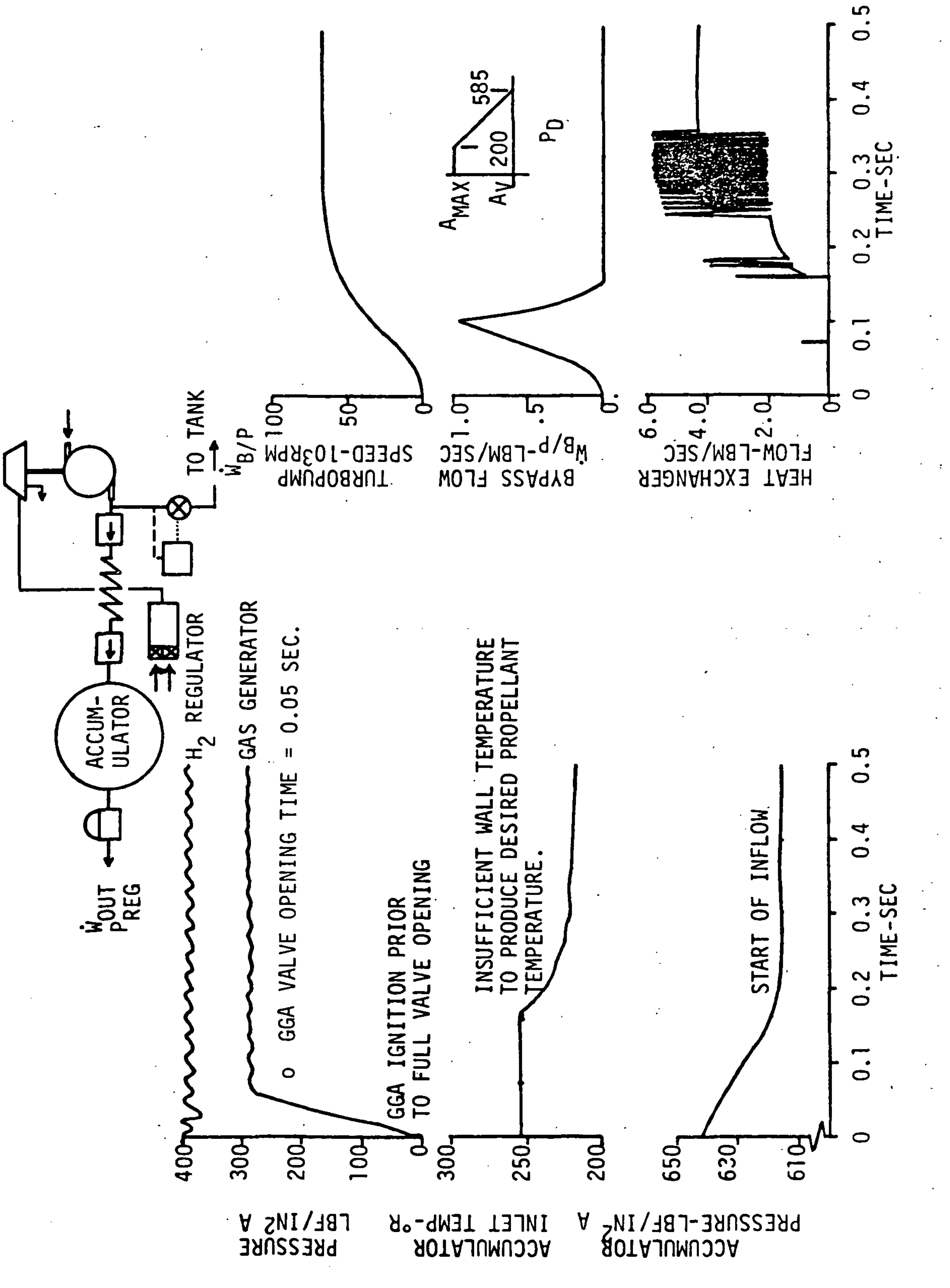


Figure 4-18

PARALLEL HYDROGEN CONDITIONER STARTUP TRANSIENTS

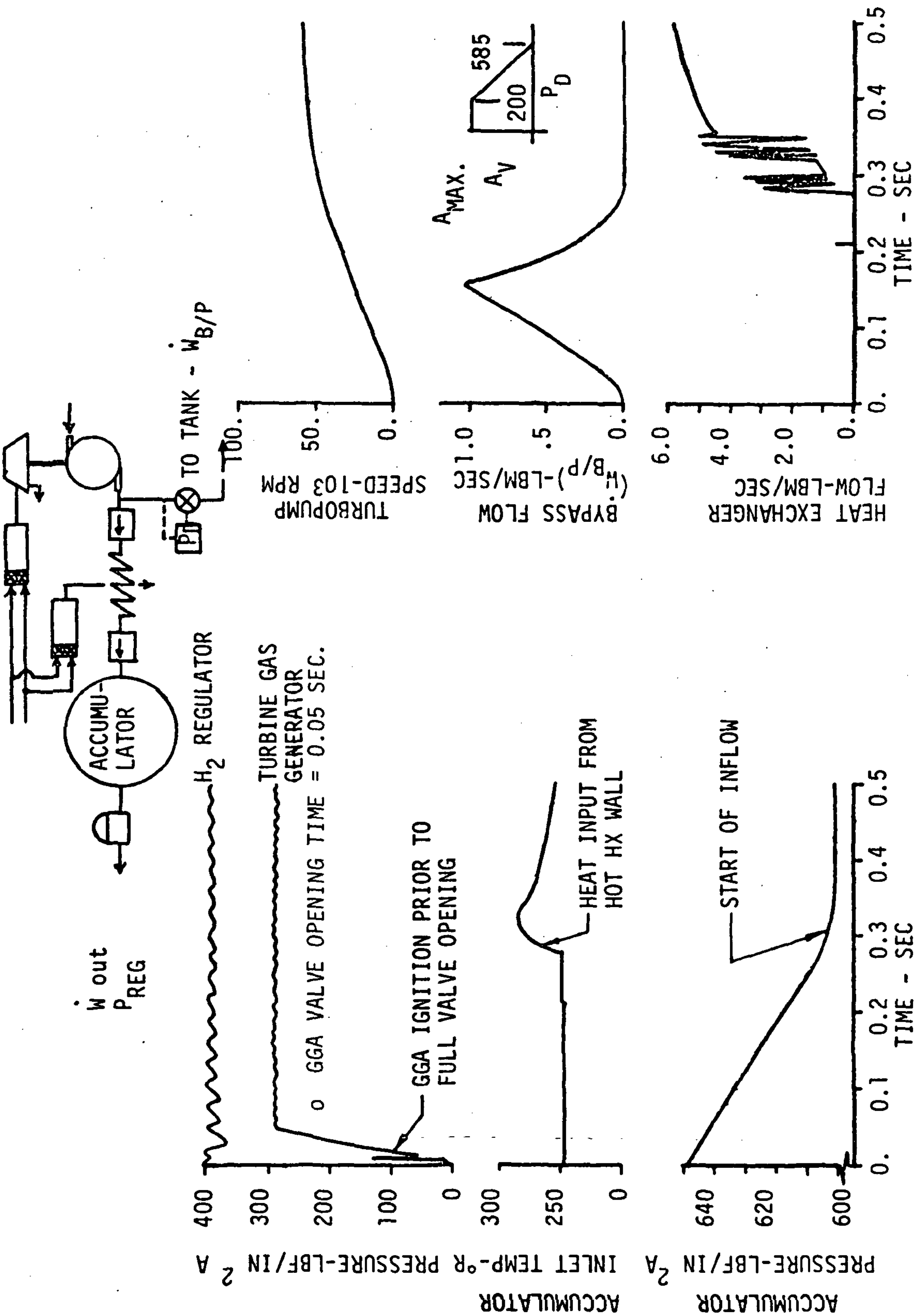


Figure 4-13

of an additional turbine stage and the corresponding increase in turbine inertia. The above results were based on gas generator valve response times of 50 ms. The most significant result from the startup response analyses of Figures 4-16 through 4-19 is the high hydrogen turbopump shaft accelerations (approximately 260,000 RPM/sec for the series concepts and 165,000 RPM/sec for the parallel RCS). In an effort to reduce these accelerations, a controlled conditioner startup was considered in which the gas generator throttle valves were ramped open from an area setting of 50% to 100% in one-half second. As shown in Figure 4-20 for the series-upstream turbine RCS, this results in a more gradual build-up in gas generator flow rate and turbine torque, reducing the probability of bearing "sledding" which can result from excessive shaft acceleration. Shaft acceleration experienced during this 0.5 second ramp interval was approximately 200,000 RPM/sec compared with 260,000 RPM/sec for the uncontrolled startup (Figure 4-16). Current experience with propellant-cooled bearings is approximately 40,000 RPM/sec, and thus, even with ramped turbine power over a reasonable start interval, pump bearing design must be regarded as critical technology area.

The principle concern during conditioner shutdown is the potential for pump backsurge and/or "water hammer" effects which can damage or impair performance of the propellant tank surface tension device. To minimize these effects, fast acting valves in the pump suction lines were avoided and a propellant bypass (tank return) circuit was incorporated. The bypass circuit is opened at pump shutdown to alleviate propellant temperature rise as the kinetic energy of pump rotating parts is dissipated via joule heating to the propellant. The effectiveness of the bypass circuit is illustrated in Figure 4-21 for the series-upstream turbine RCS. As shown, a pump backsurge of approximately 1.0 lbm/sec is encountered with no bypass (Case A) whereas backsurge is completely eliminated with a bypass designed to handle 80% pump flow rate (Case B). Figure 4-22 provides a more complete history of Case B shutdown characteristics.

4.4.2 <u>Component Sequencing</u> - The conditioner startup and shutdown transients of Figures 4-16 through 4-22 were used to formulate the sequence charts of Figures 4-23 and 4-25 for the series and parallel RCS, respectively. The sequence is basically the same in both instances. Referring to the appropriate schematics, Figures 4-24 and 4-26, it is seen that the pumps are maintained in a wetted condition with sump-mounted recirculation pumps providing the necessary cooling flow between conditioner cycles. The start sequence for the series RCS is initiated when accumulator sensors signal that gas pressure has decayed to approximately

TYPICAL START TRANSIENT

- o SERIES RCS/TURBINE UPSTREAM (HYDROGEN SIDE)
- o GGA H_2 & O_2 VALVES RAMPED FROM 50% TO FULL OPEN IN 0.5 SEC.

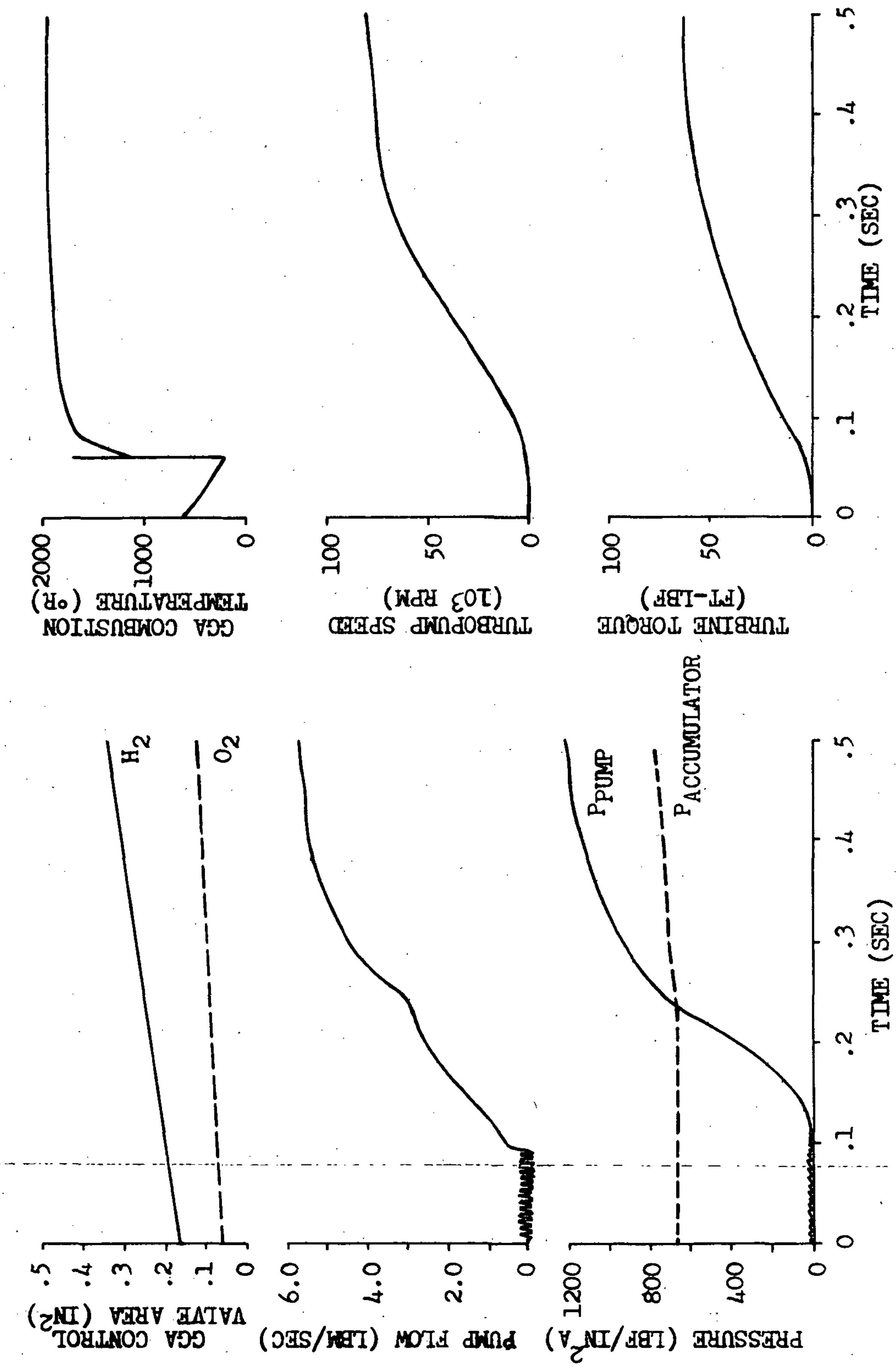


Figure 4-20

RCS SHUTDOWN TRANSIENT: WITH AND WITHOUT BYPASS

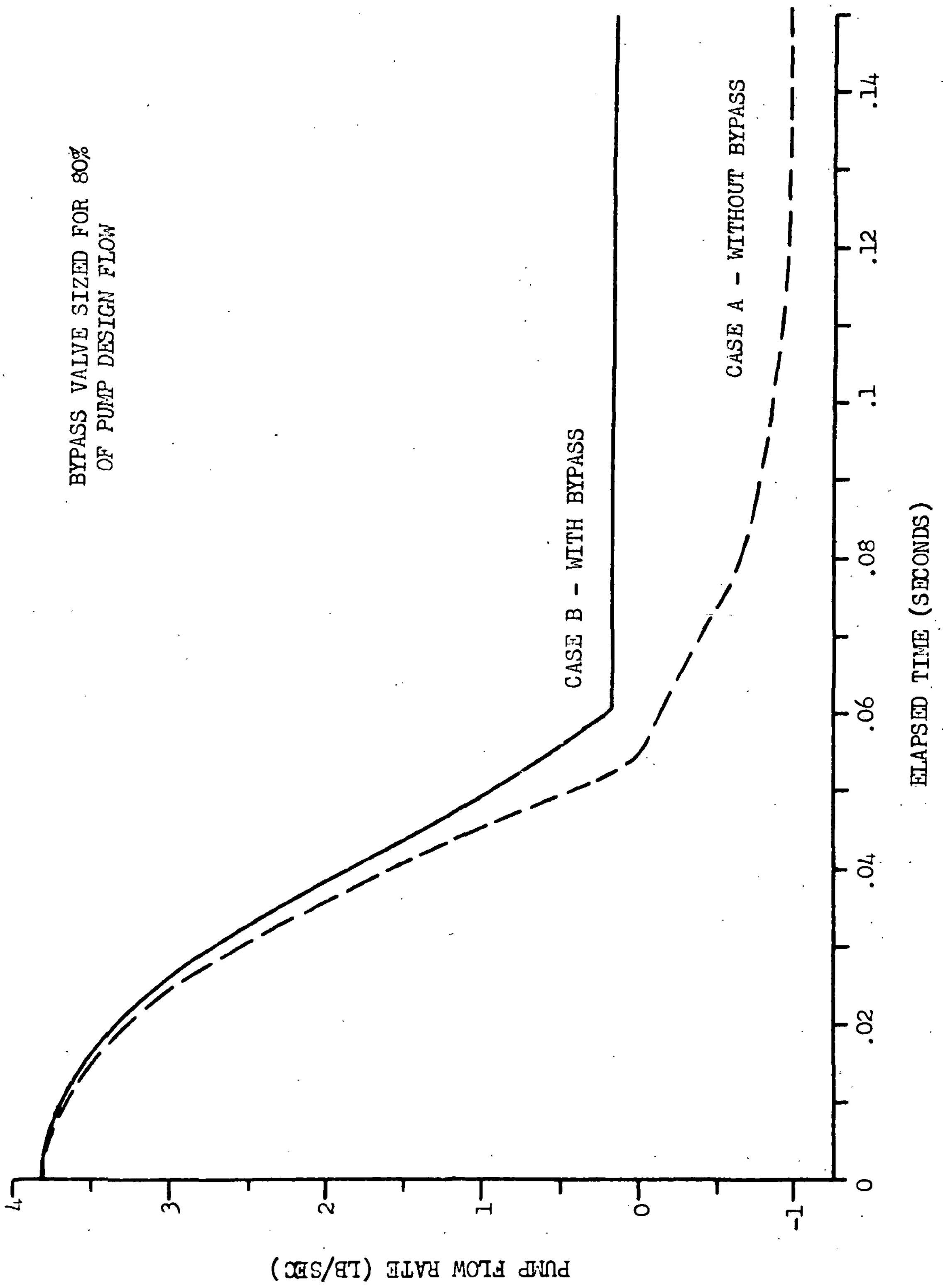


Figure 4-21

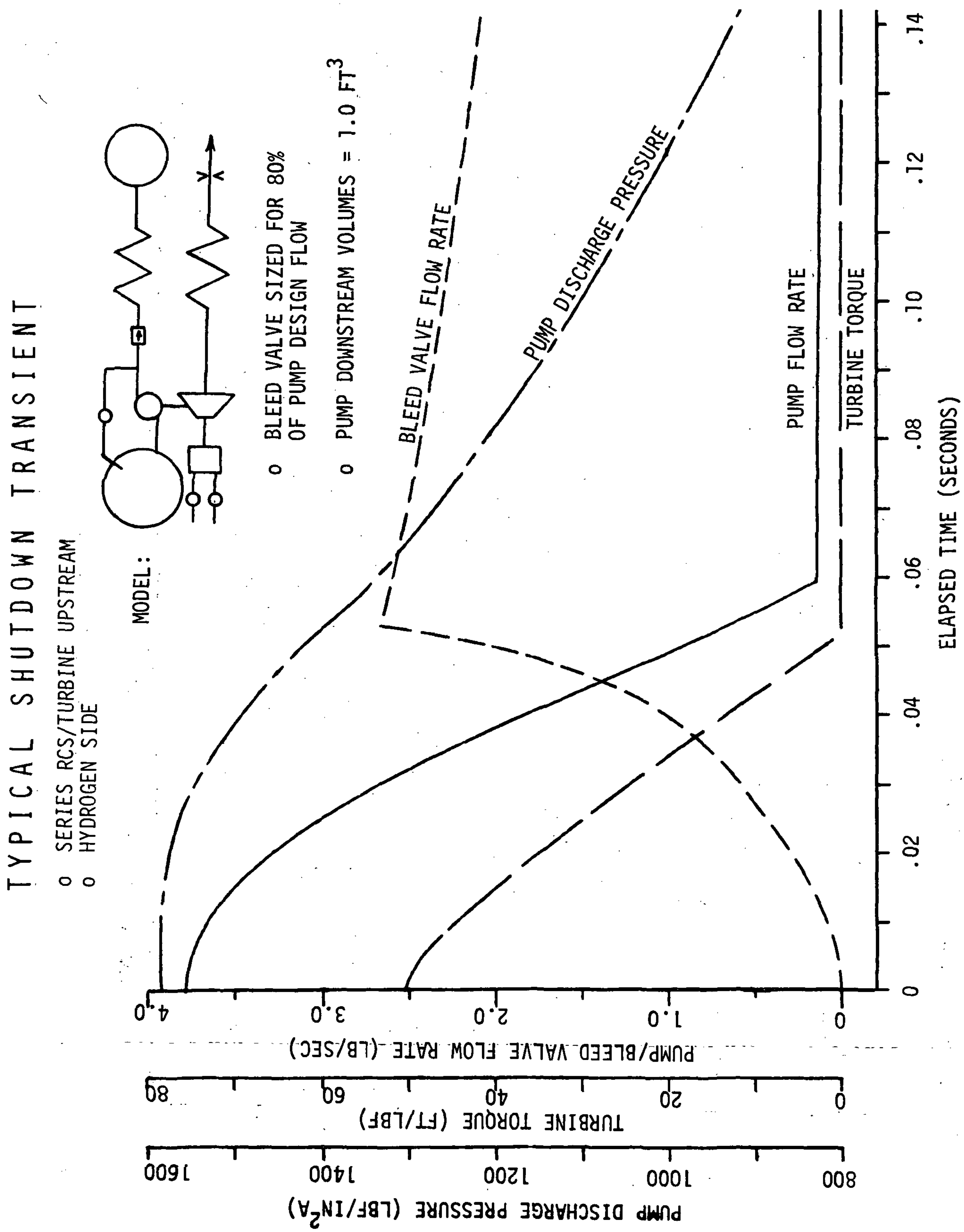
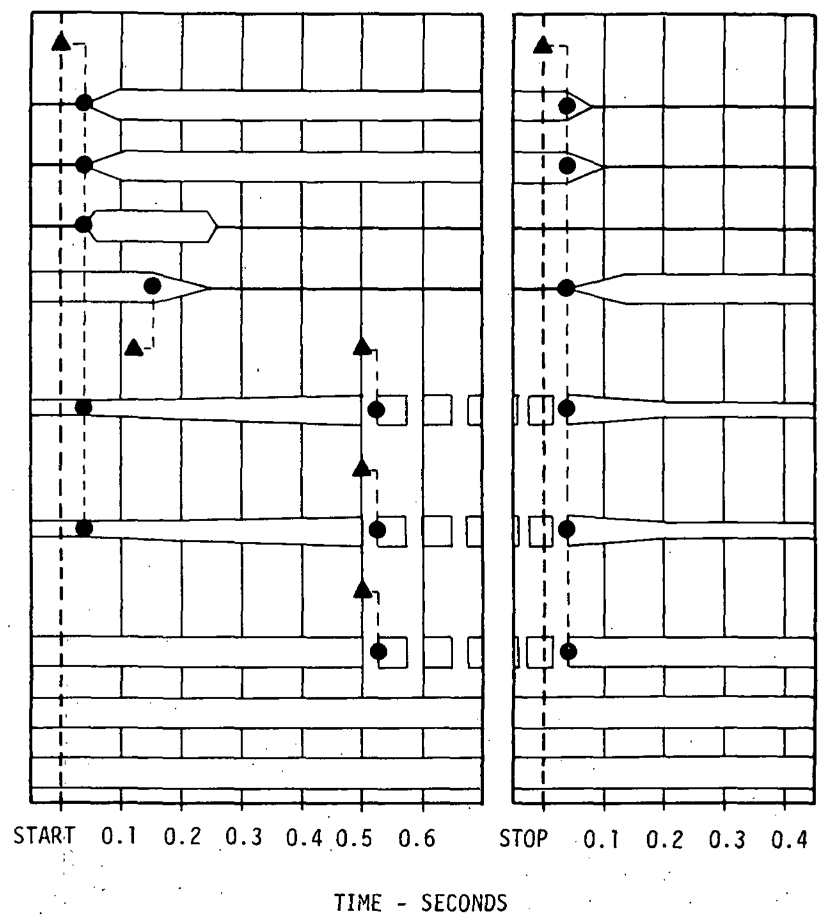

TYPICAL SHUTDOWN TRANSIENT
o SERIES RCS/TURBINE UPSTREAM
o HYDROGEN SIDE
MODEL:
o BLEED VALVE SIZED FOR 80% OF PUMP DESIGN FLOW
o PUMP DOWNSTREAM VOLUMES = 1.0 FT³
BLEED VALVE FLOW RATE
PUMP DISCHARGE PRESSURE
PUMP FLOW RATE
TURBINE TORQUE
0
.02
.04
.06
.08
.10
.12
.14
ELAPSED TIME (SECONDS)
4.0
3.0
2.0
1.0
0
PUMP/BLEED VALVE FLOW RATE (LB/SEC)
80
60
40
20
0
TURBINE TORQUE (FT/LBF)
1600
1400
1200
1000
800
PUMP DISCHARGE PRESSURE (LBF/IN²A)

Figure 4-22

SERIES RCS CONDITIONER SEQUENCE

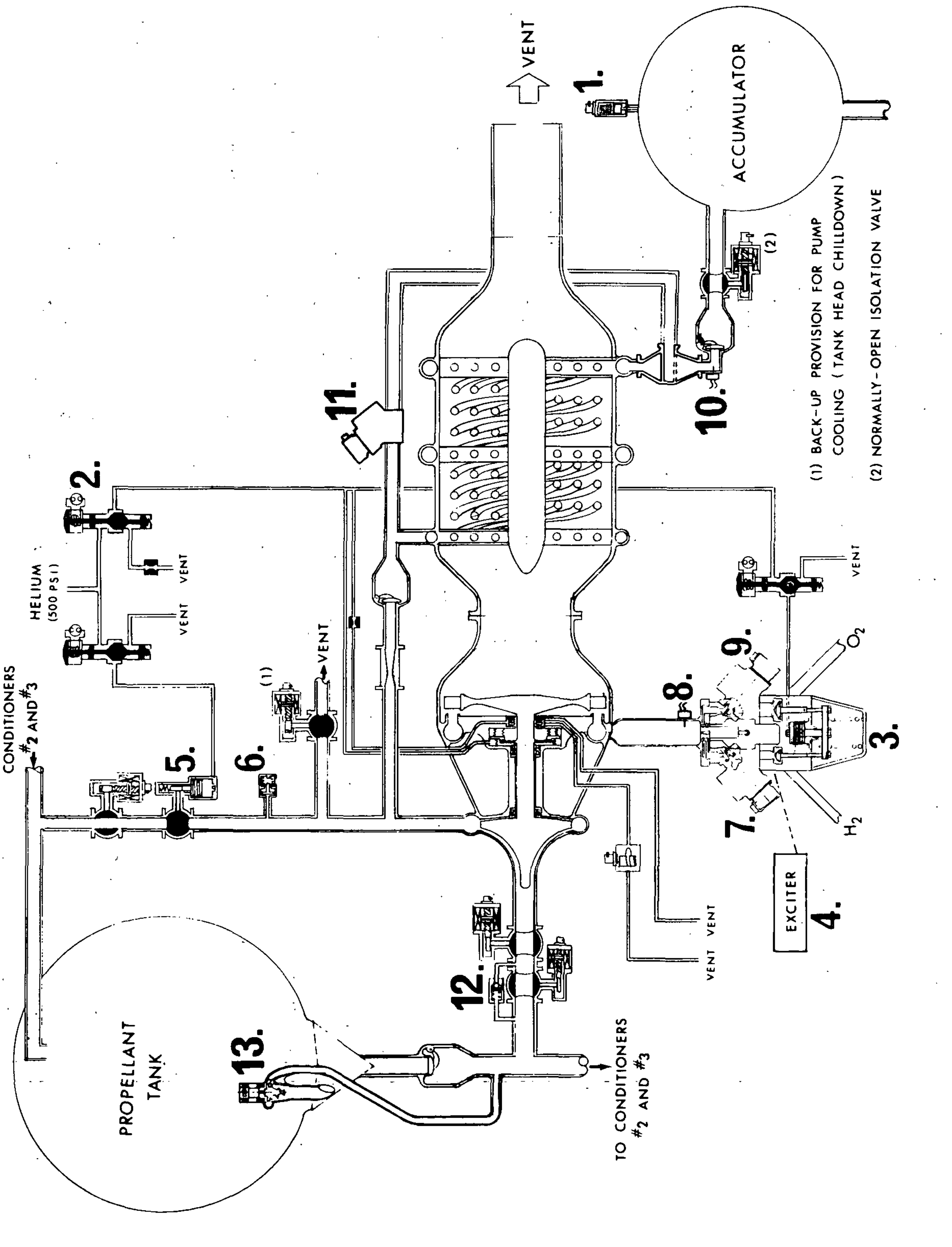
1. ACCUMULATOR PRESSURE SENSOR
2. PUMP LIFT-OFF SEAL AND SEAL PURGE VALVE
3. GGA PROPELLANT SHUT-OFF VALVE
4. GGA IGNITION EXCITER
5. PUMP BYPASS VALVE
6. PUMP DISCHARGE PRESSURE SENSOR
7. GGA FUEL THROTTLE VALVE
8. GGA COMBUSTION TEMPERATURE SENSOR
9. GGA OXIDIZER THROTTLE VALVE
10. HEX COLD SIDE OUTLET TEMPERATURE SENSOR
11. HEX COLD SIDE BYPASS THROTTLE VALVE
12. PUMP SUCTION VALVE
13. PROPELLANT RECIRCULATION PUMP



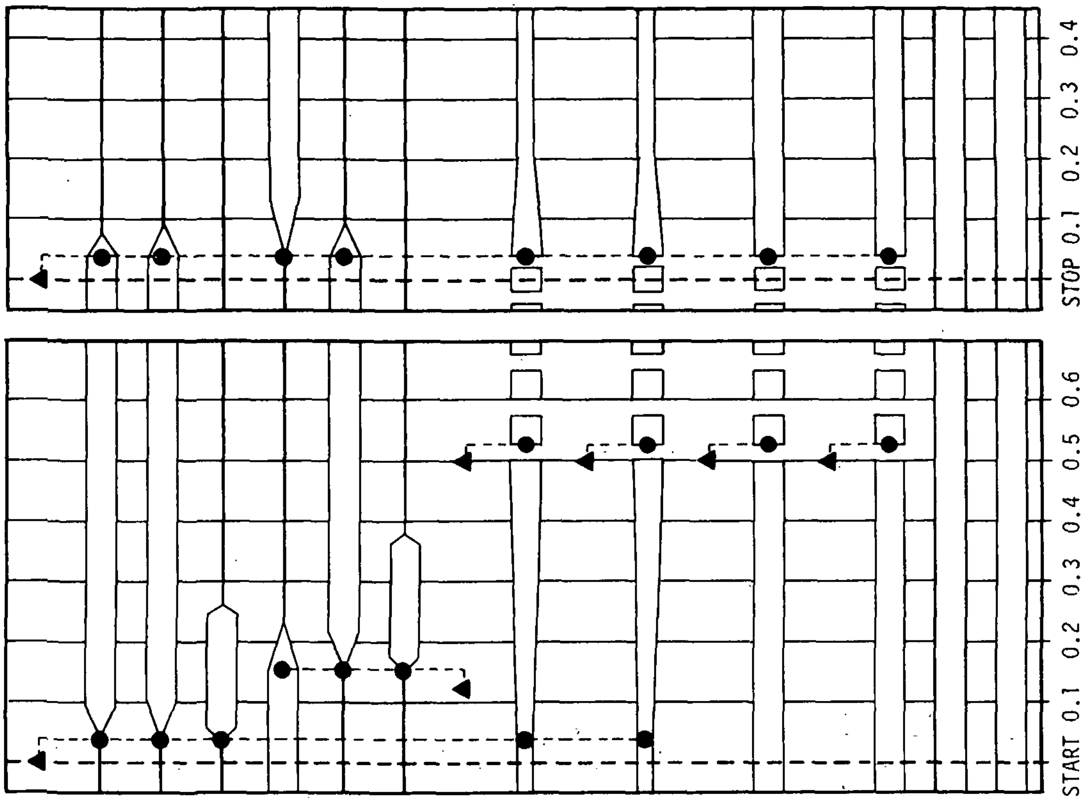
FULL OPEN
MODULATED
PARTIALLY OPEN
FULL CLOSED

ITEMS MARKED ▲ OCCUR AUTOMATICALLY AND CAUSE SUCCEEDING EVENTS ●

SERIES/TURBINE UPSTREAM
RCS CONDITIONER SCHEMATIC



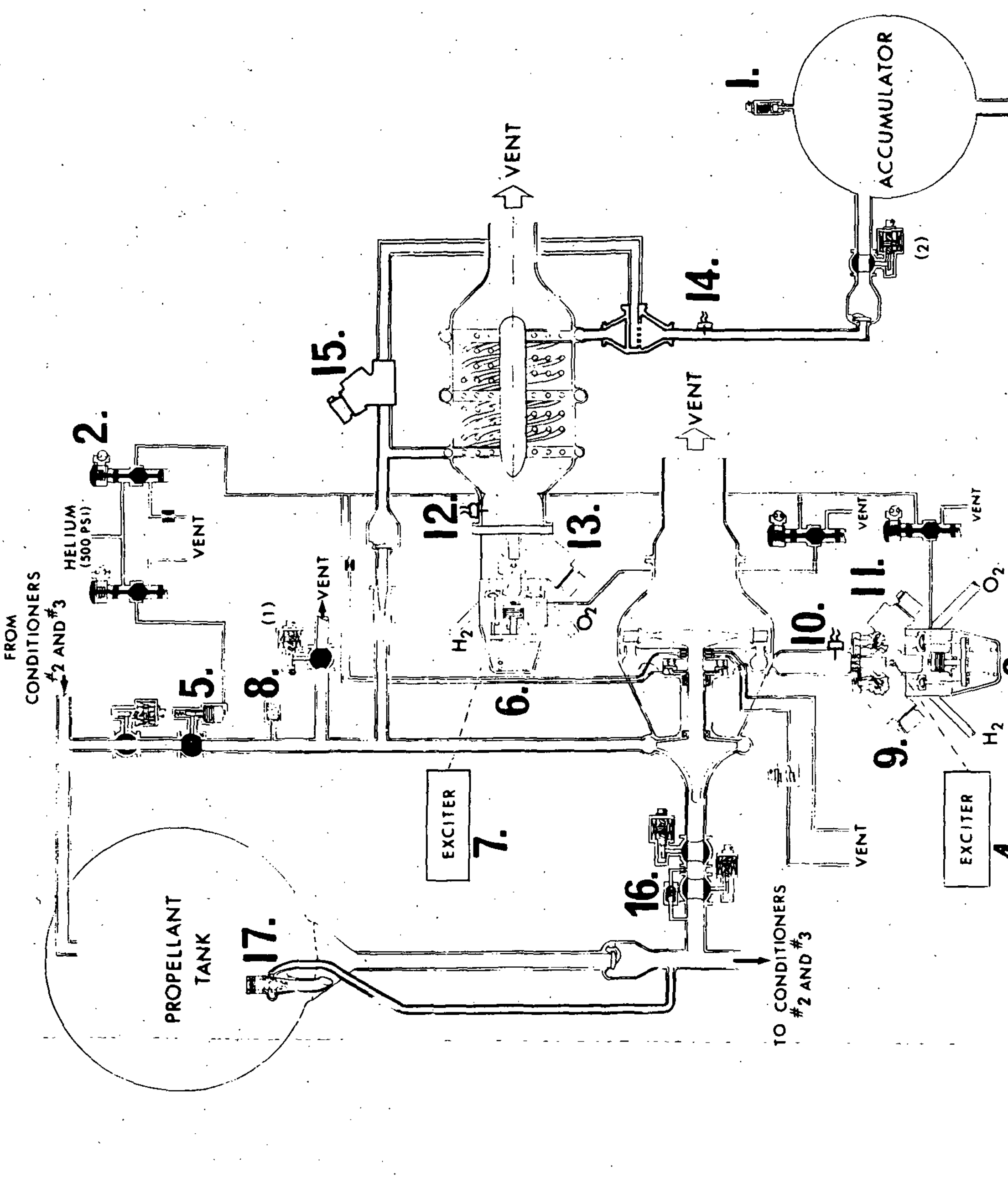
PARALLEL RCS CONDITIONER SEQUENCE



1. ACCUMULATOR PRESSURE SENSOR
2. PUMP LIFT-OFF SEAL AND SEAL PURGE VALVE
3. TURBINE GGA PROPELLANT SHUT-OFF VALVE
4. TURBINE GGA IGNITION EXCITER
5. PUMP BYPASS VALVE
6. HEX GGA PROPELLANT SHUT-OFF VALVE
7. HEX GGA IGNITION EXCITER
8. PUMP DISCHARGE PRESSURE SENSOR
9. TURBINE GGA FUEL THROTTLE VALVE
10. TURBINE GGA COMBUSTION TEMPERATURE SENSOR
11. TURBINE GGA OXIDIZER THROTTLE VALVE
12. HEX GGA COMBUSTION TEMPERATURE SENSOR
13. HEX GGA OXIDIZER THROTTLE VALVE
14. HEX COLD SIDE OUTLET TEMPERATURE SENSOR
15. HEX COLD SIDE BYPASS THROTTLE VALVE
15. PUMP SUCTION VALVE
17. PROPELLANT RECIRCULATION PUMP

Figure 4-25

PARALLEL FLOW
RCS CONDITIONER SCHEMATIC



(1) BACK-UP PROVISION FOR PUMP COOLING (TANK HEAD CHILLDOWN)
(2) NORMALLY-OPEN ISOLATION VALVE

Figure 4-26

665 lbf/in^2a. This pressure signal triggers the simultaneous actuation of: (1) the oxygen pump lift-off seal and intermediate seal purge valves, (2) the gas generator shut-off valves, and (3) a spark train to the gas generator igniter. Also, the gas generator H_2 and O_2 throttle valves are ramped open from an area setting of 50% to 100% in one-half second. When pump discharge pressure exceeds 200 lbf/in^2a, the propellant bypass valve in the tank return circuit is signaled closed, thus promoting a rapid rise in heat exchanger propellant flow rate. Feed-back control signals to the gas generator and heat exchanger bypass valves are locked out for one-half second to provide time for the conditioner flow parameters and sensor output signals to stabilize. Accumulator recharge is accomplished in approximately 6 seconds (no thruster flow) as shown in the example of Figure 4-27. Upon achieving the accumulator maximum (cutoff) pressure; the gas generator shutoff valves, oxygen pump lift-off seal valve, and purge valves are closed; the pump bypass valve is opened; and the gas generator throttle valves are returned to a 50% area setting. The sequence for both series RCS concepts is the same, and the sequence for the parallel RCS (Figure 4-25) differs only in the delayed start of the heat exchanger gas generator. Here, heat exchanger hot side flow is delayed until propellant flow is sensed at the pump cavitating venturi to assure a cold side flow lead into the heat exchanger. Complete instrumentation requirements of the series and parallel RCS for operational control and malfunction detection are presented in Appendix E.

4.4.3 Malfunction Analyses - To provide a more complete RCS operational description, three potentially critical malfunctions were simulated to determine system response during failure detection and isolation. These included: (1) conditioner failure during startup (series RCS), (2) loss of gas generator combustion temperature control (series RCS), and (3) parallel RCS heat exchanger gas generator ignition failure.

Two separate examples of conditioner startup failure are shown in Figure 4-28 for the series-upstream turbine RCS. The first assumes the failure of the H_2 conditioner with nominal operation of the O_2 conditioner (solid curves), while the second assumes an O_2 conditioner failure with nominal operation of the H_2 conditioner (dashed curves). A 0.5 second delay for malfunction detection, shutdown and startup command signal to the backup conditioner was assumed for these two cases. Typical conditioner malfunction detection and isolation logic is illustrated in Figure 4-29. As shown, the controller initiates the accumulator recharge command when accumulator pressure decays to its switching level. However, if after a 500 ms

ACCUMULATOR RECHARGE CHARACTERISTICS

o SERIES RCS/TURBINE UPSTREAM

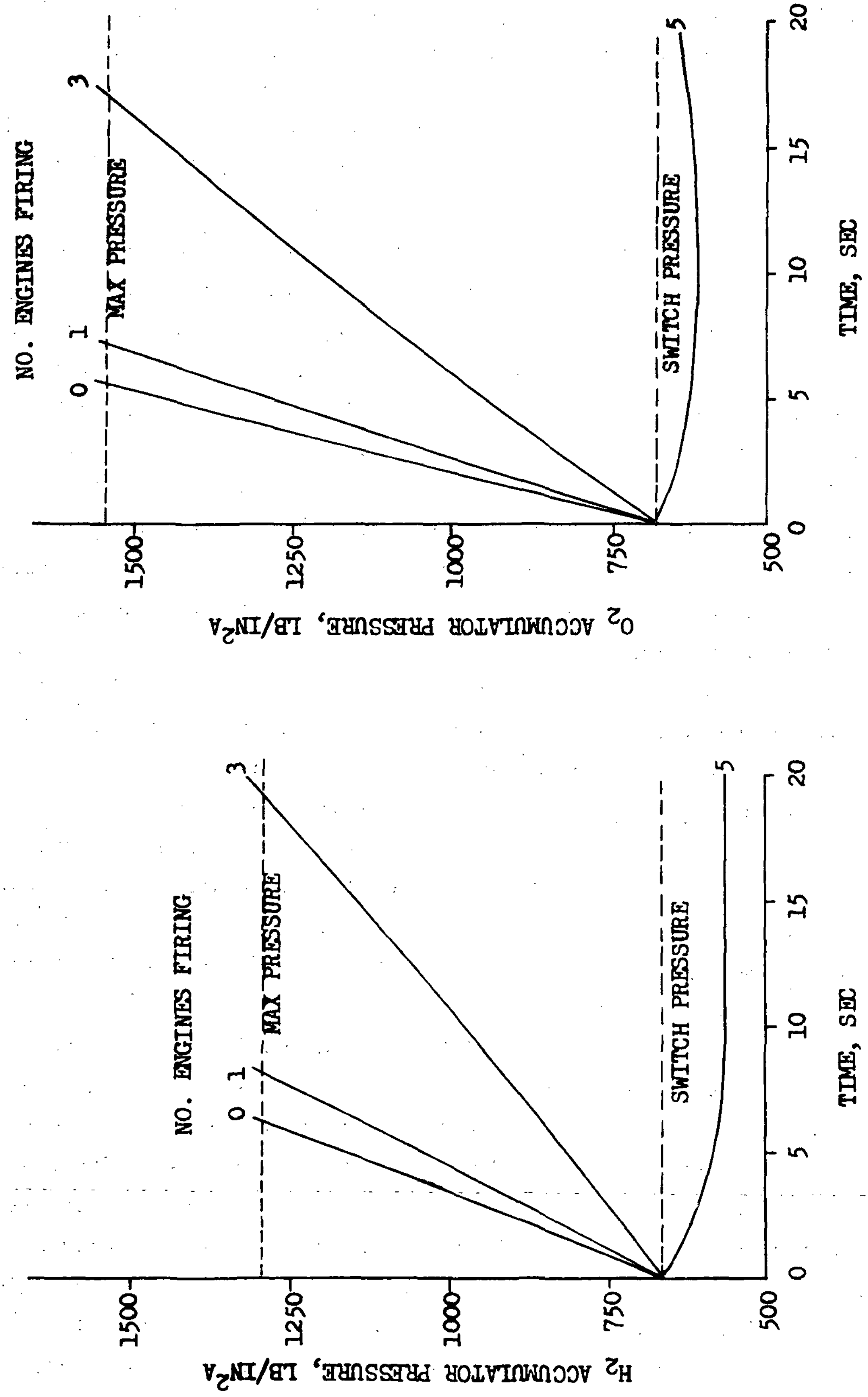


Figure 4-27

SIMULATED CONDITIONER FAILURE DURING START-UP

- SERIES RCS TURBINE UPSTREAM (5 ENGINES FIRING)
- 0.5 SEC DELAY TO DETECT CONDITIONER FAILURE

——— H_2 CONDITIONER FAILURE
- - - - O_2 CONDITIONER FAILURE

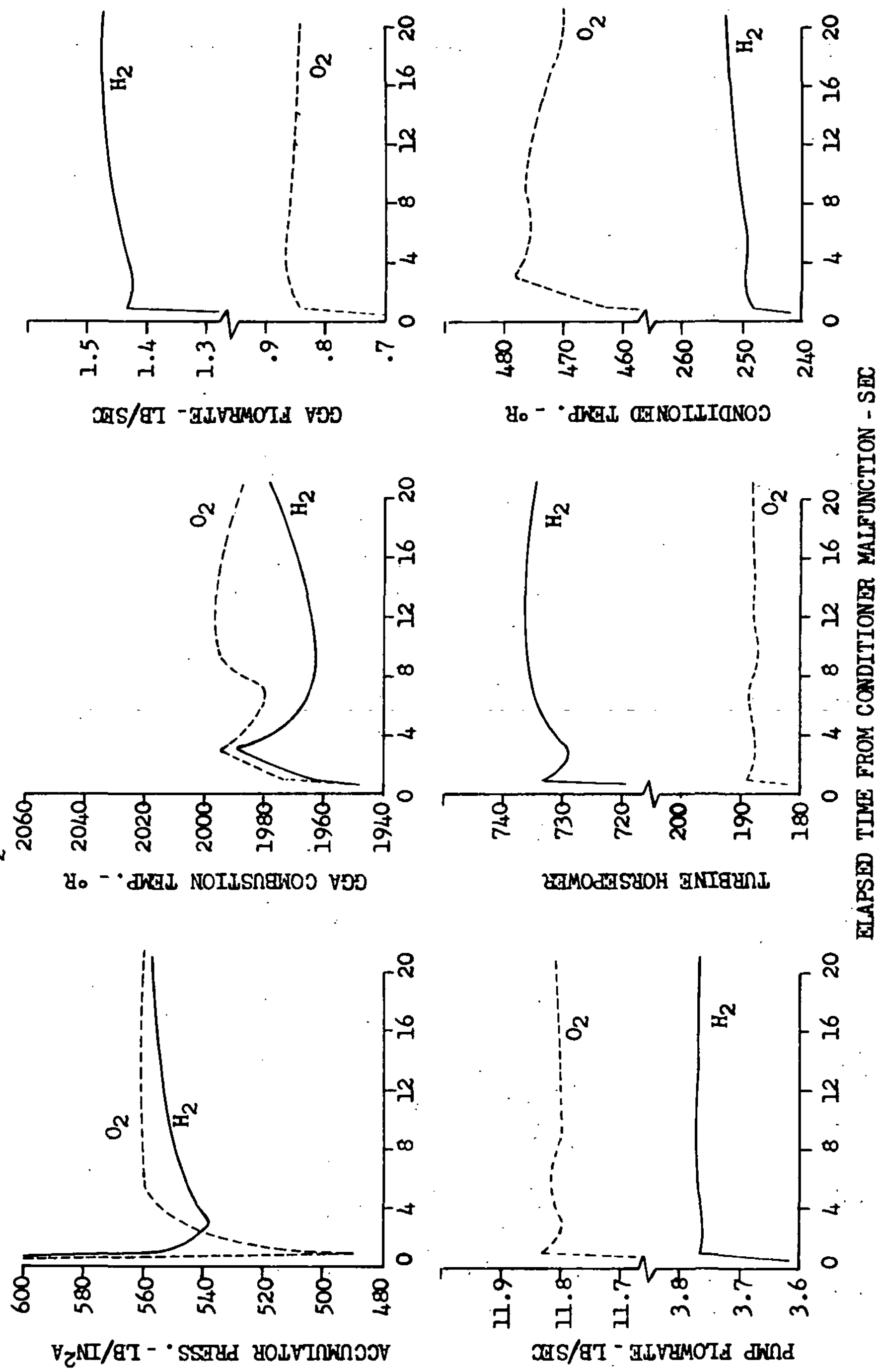


Figure 4-28

CONDITIONER MALFUNCTION DETECTION AND ISOLATION

o PRIMARY CONDITIONER FAILS TO START

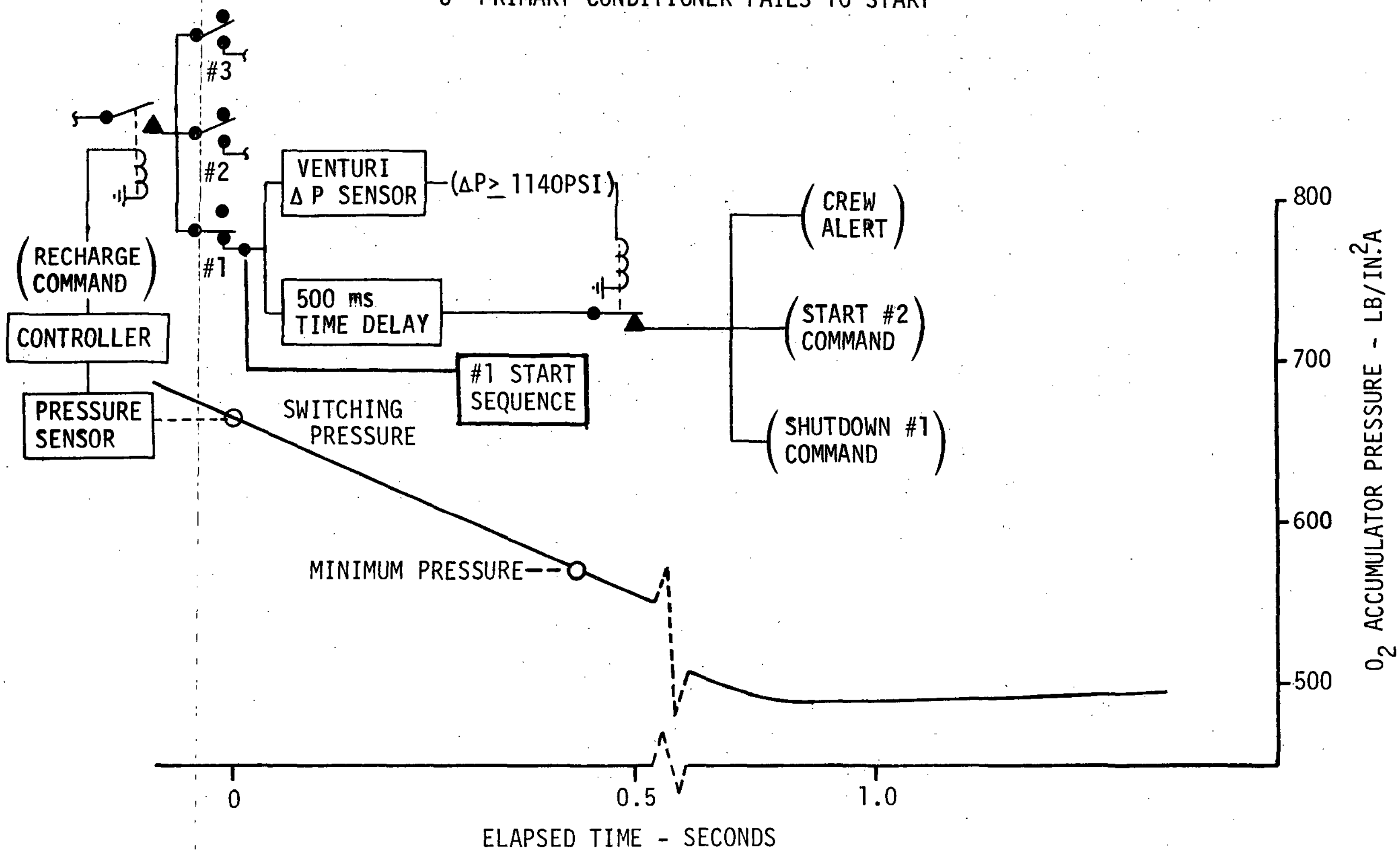


Figure 4-29

time delay, pump cavitating venturi ΔP (inlet total pressure minus throat static pressure) does not achieve 80% of its design value (1140 lbf/in^{2}d) the nominal start sequence is interrupted. When this occurs, the primary conditioner is commanded to shutdown, the backup conditioner (No. 2) is commanded to start up, and the crew is alerted. During this 0.5 second interval, (Figure 4-28) accumulator pressure decays from its switching level (665 lbf/in^{2}a) to 540 (H_2) and 490 (O_2) lbf/in^{2}a as a result of propellant flow to five thrusters. However, following startup of the back-up conditioner, accumulator pressure recovers and acceptable conditioner performance is maintained throughout the recharge cycle.

A second malfunction example, depicting conditioner malfunction under nominal operating conditions is presented in Figure 4-30 for the series-upstream turbine RCS. In this example, an open circuit in the thermocouple leads to the gas generator temperature sensor is simulated. This sensor is used to control gas generator O_2 throttle valve area in response to combustion temperature. The loss of this sensor signal is interpreted by the controller as a low combustion temperature, and therefore, the controller drives the O_2 throttle valve to its full open position in approximately 80 ms. As shown in Figure 4-30, gas generator combustion temperature increases sharply from 2020°R to 3900°R in the 80 ms interval, necessitating conditioner shutdown (separate temperature sensors are used for gas generator malfunction detection - Appendix E). The impact of these high combustion temperatures on the time to failure of critical downstream components (turbine and heat exchanger) is presented in Figure 4-31. As shown, design limitations of the turbine rotor blades are more critical than those of the heat exchanger tubing or outer shell. Critical turbine blade stresses are achieved near the blade root (Station 17) approximately 130 ms after malfunction occurrence, whereas critical heat exchanger outer shell hoop stresses are reached approximately 160 ms following malfunction. Heat exchanger tube wall temperatures, on the other hand, are far less critical, reaching thermal equilibrium below the maximum allowable temperature. Based on these results, the response requirements for the malfunction detection and isolation system must be on the order of 120 ms, indicating that shielded thermocouples may not be suitable for the gas generator malfunction sensors because of their large time constant. Fluidic or resistance-type temperature sensors, however, will provide the necessary response.

The third failure mode considered was a simulated malfunction of the igniter shutoff valve in the heat exchanger gas generator (parallel RCS) during startup. In this failure mode, cold liquid propellant is passed through the heat exchanger

SIMULATED LOSS OF GGA COMBUSTION TEMPERATURE CONTROL

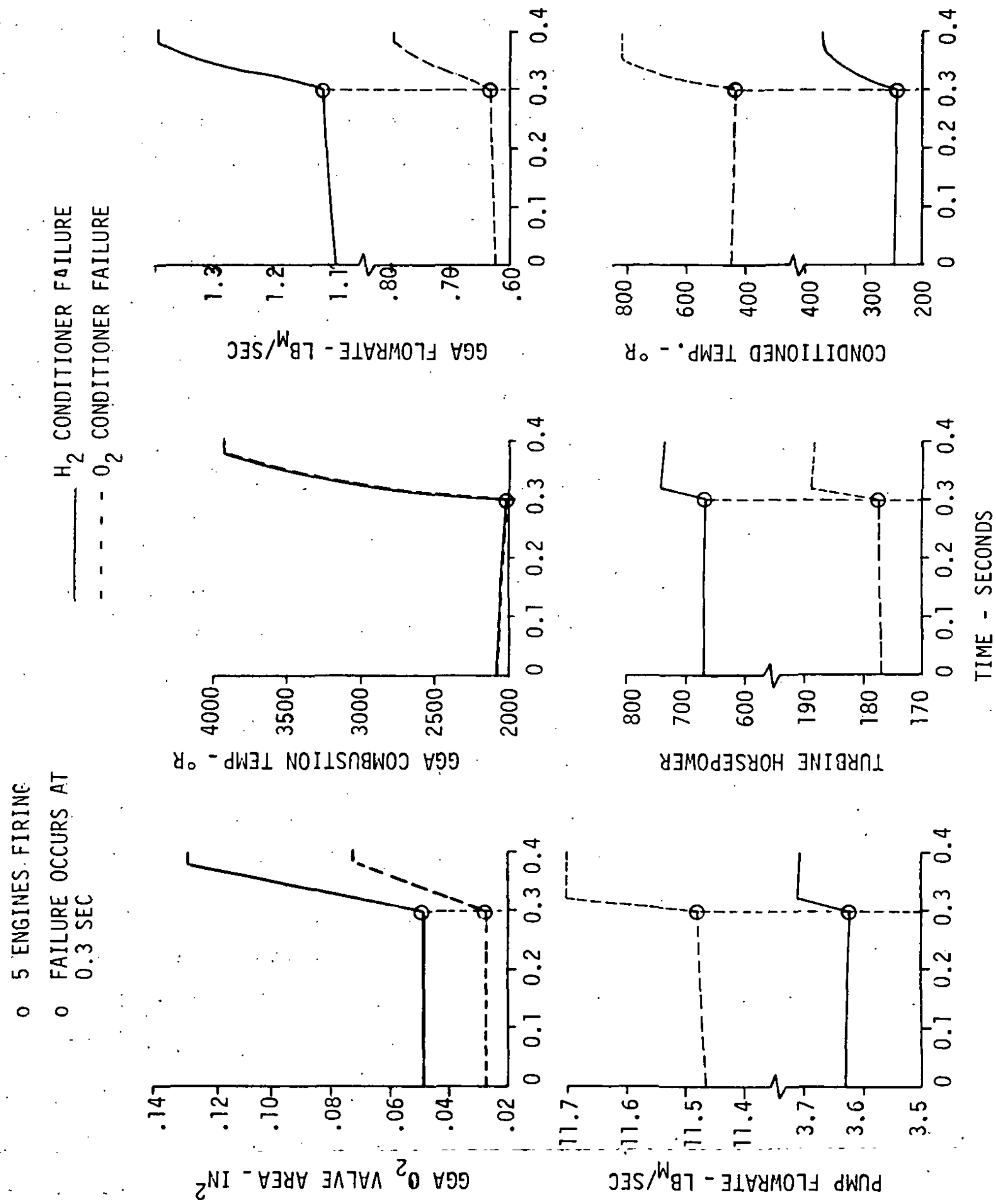


Figure 4-30

COMPONENT THERMAL RESPONSE FOLLOWING OPEN CIRCUIT IN COMBUSTION TEMPERATURE FEEDBACK LOOP

TURBINE BLADE THERMAL RESPONSE

o BLADE MATERIAL - UDIMET 700

HEAT EXCHANGER THERMAL RESPONSE

o TUBING/SHELL MATERIAL - INCONEL X 750

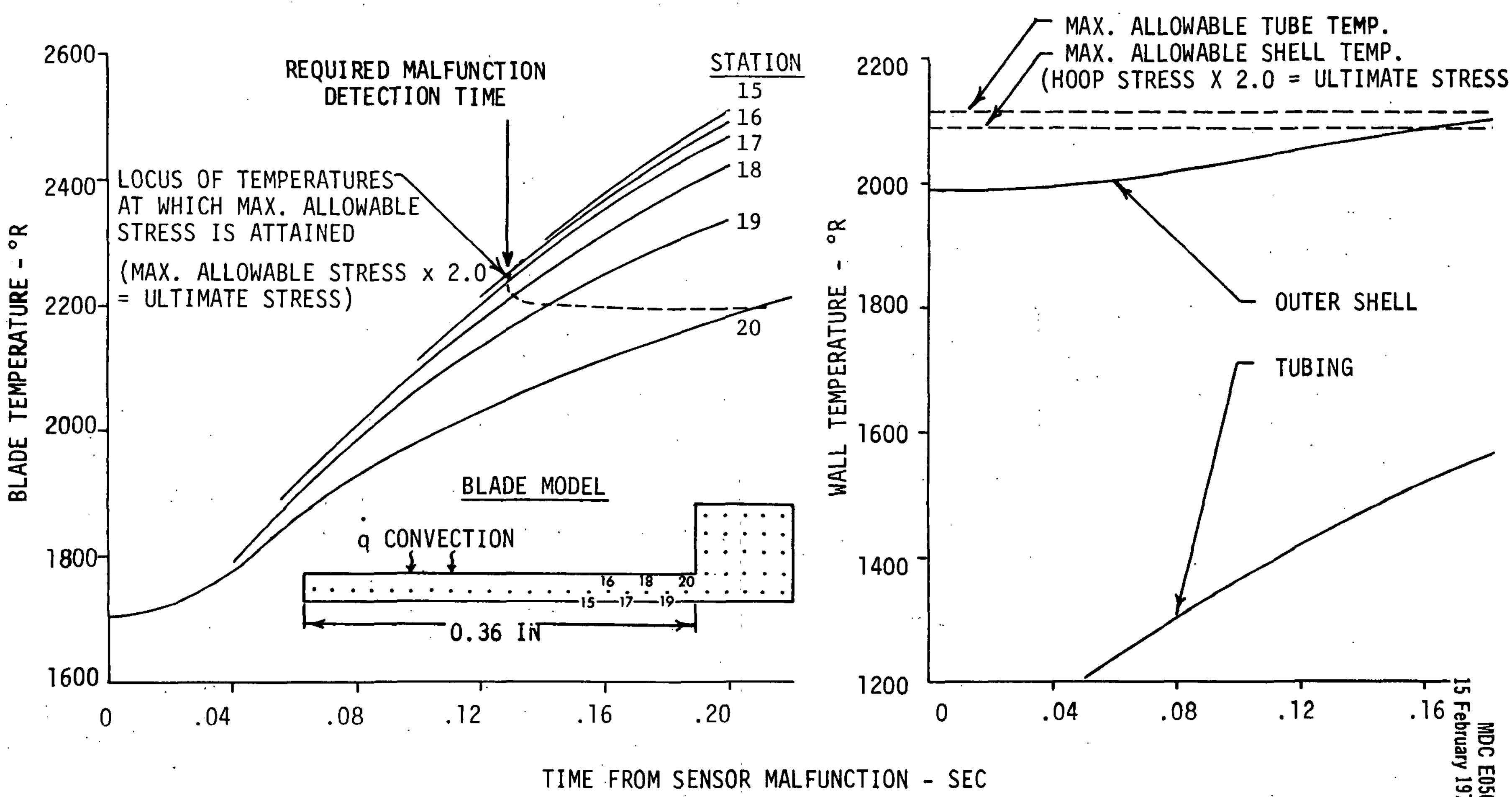


Figure 4-31

and injected into the accumulator. Unless this malfunction is detected and isolated promptly, the accumulator pressure and temperature could decay to unacceptable levels. This malfunction is simulated in the example of Figure 4-32 for both the H_2 and O_2 conditioner circuits. The assumed sequence is: (1) conditioner startup is commanded at the accumulator switching pressure; (2) the turbopump gas generator ignites but the heat exchanger gas generator does not; (3) nominal liquid propellant flow is injected into the accumulator for a 0.5 second time interval, during which time the malfunction is detected by thermocouples and the conditioner is shutdown; and (4) during the following 0.5 second interval, the accumulator continues to blow down until the back-up conditioner is started and delivers conditioned propellant flow to the accumulator. Based on these assumptions, the H_2 and O_2 accumulator pressures decay to minimum levels of 526 and 493 lbf/in^{2}a, respectively. Subsequent to this, back-up conditioner output is sufficient to maintain a nearly constant accumulator pressure level. These examples show that acceptable accumulator pressures and temperatures can be maintained in the parallel RCS following an ignition malfunction in the heat exchanger gas generator.

Whereas the above analyses were restricted to quantitative consideration of selected critical malfunction modes, a complete qualitative failure mode and effects analysis is presented in Appendix E which is applicable to all three RCS concepts.

4.5 Final System Design - For the selected controls of Section 4.2, final RCS designs were developed. The baseline schematic is shown in Figure 4-33, and detailed illustrations of the conditioner assemblies are presented in Figures 4-34 and 4-35 for the series and parallel RCS, respectively. These latter two figures identify the sensors required for conditioner operational control and malfunction detection. The types and number of system components are summarized in Figure 4-36, and details of the propellant distribution network and thruster locations are provided in Figure 4-37. The final design points developed for these system configurations are summarized in Figure 4-38, and corresponding pressure/temperature/flow balances are shown in Figures 4-39 through 4-41. The most significant differences between these data and the preliminary design points and balances presented in Appendix B are: (1) a reduced gas generator chamber pressure of 250 lbf/in^{2}a, (2) correspondingly lower optimum turbine pressure ratios for the parallel RCS (16:1 for both the H_2 and O_2 conditioner circuits), and (3) propellant/tankage weights which include allowances for mission variances in system specific impulse and mixture ratio.

SIMULATED PARALLEL RCS HEAT EXCHANGER GGA IGNITION FAILURE

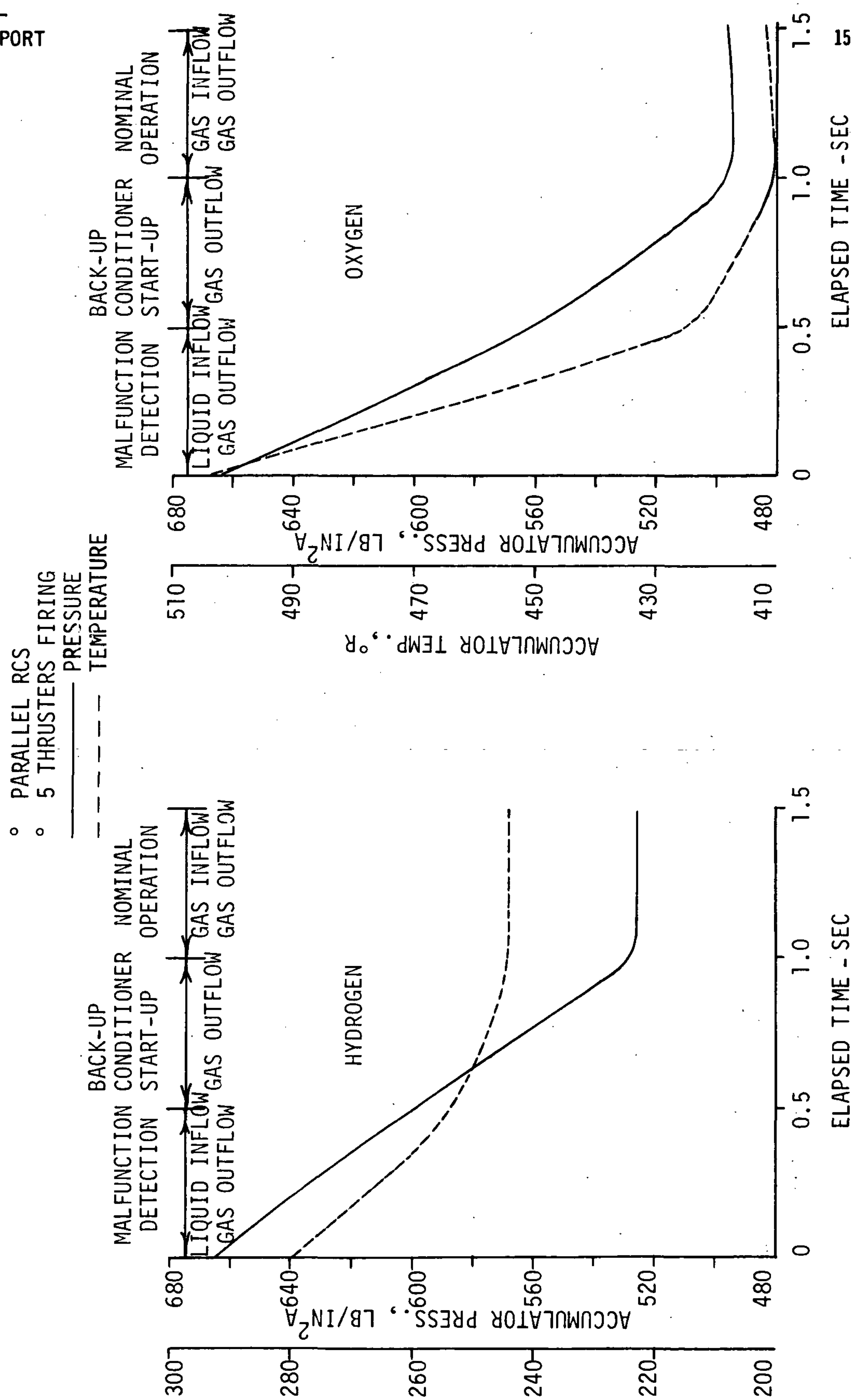


Figure 4-32

TYPICAL RCS SCHEMATIC
HYDROGEN SIDE OF BIPROPELLANT SYSTEM

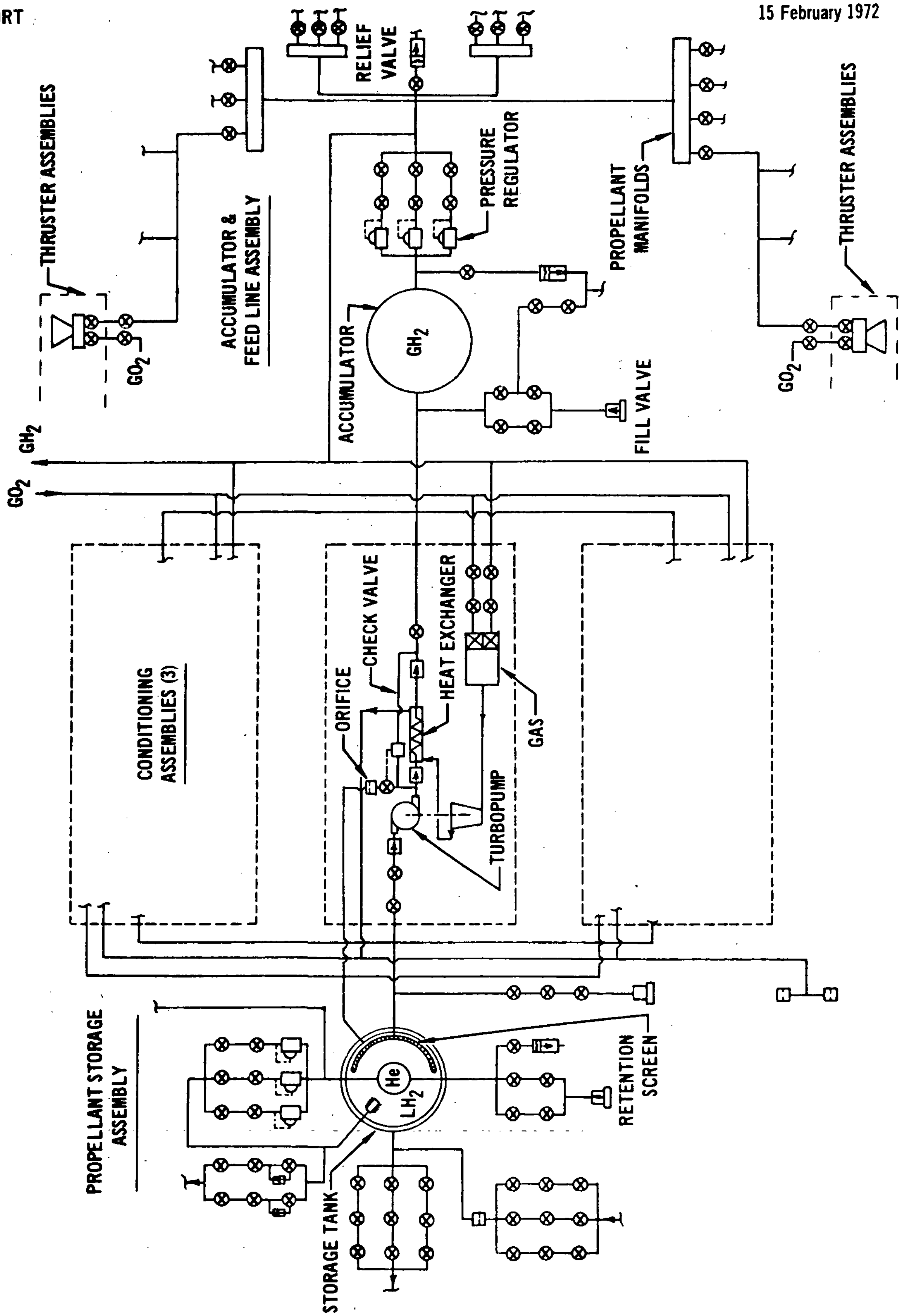


Figure 4-33

SERIES/TURBINE UPSTREAM
RCS INSTRUMENTATION REQUIREMENTS

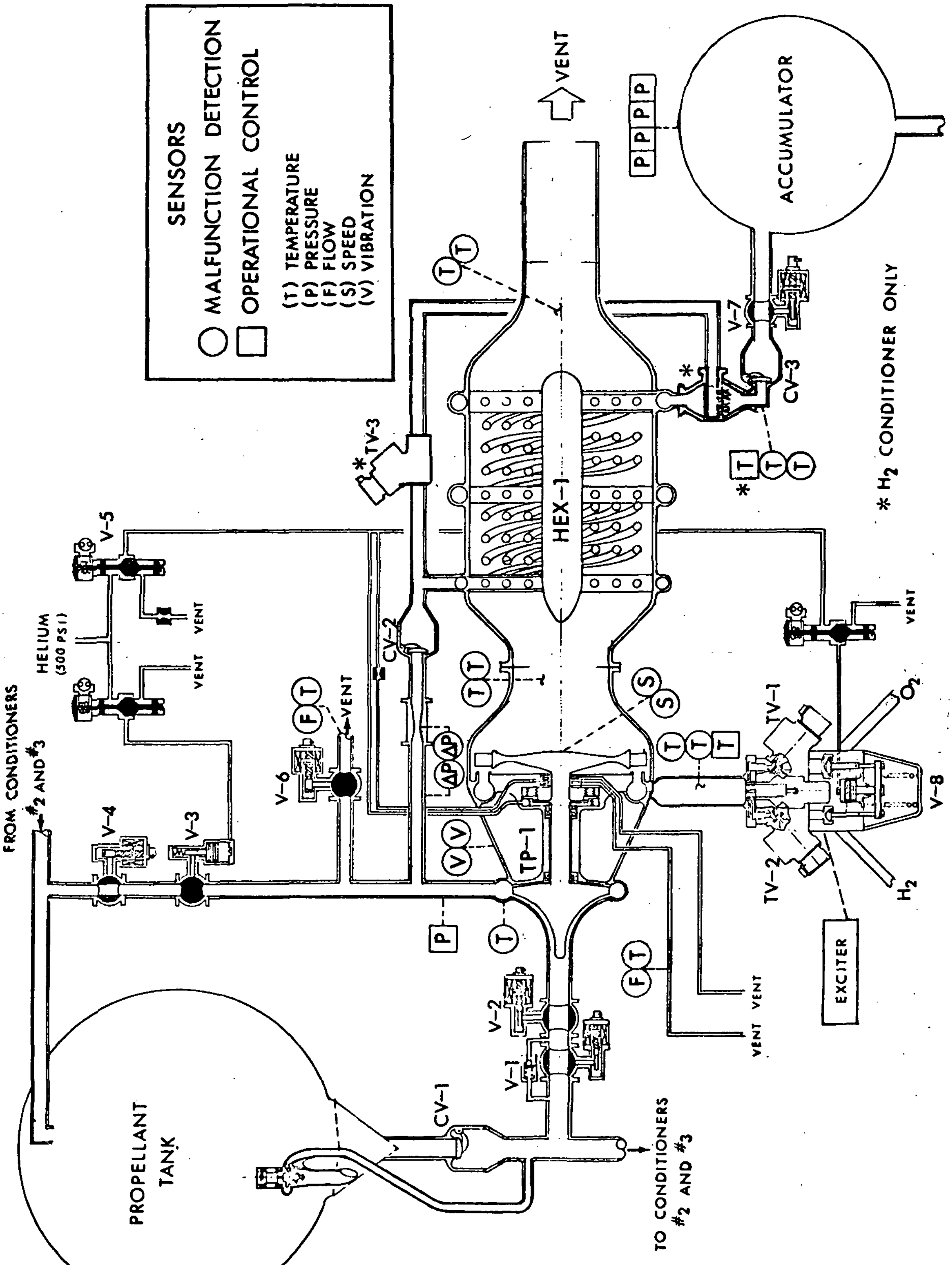


Figure 4-34

PARALLEL FLOW RCS INSTRUMENTATION REQUIREMENTS

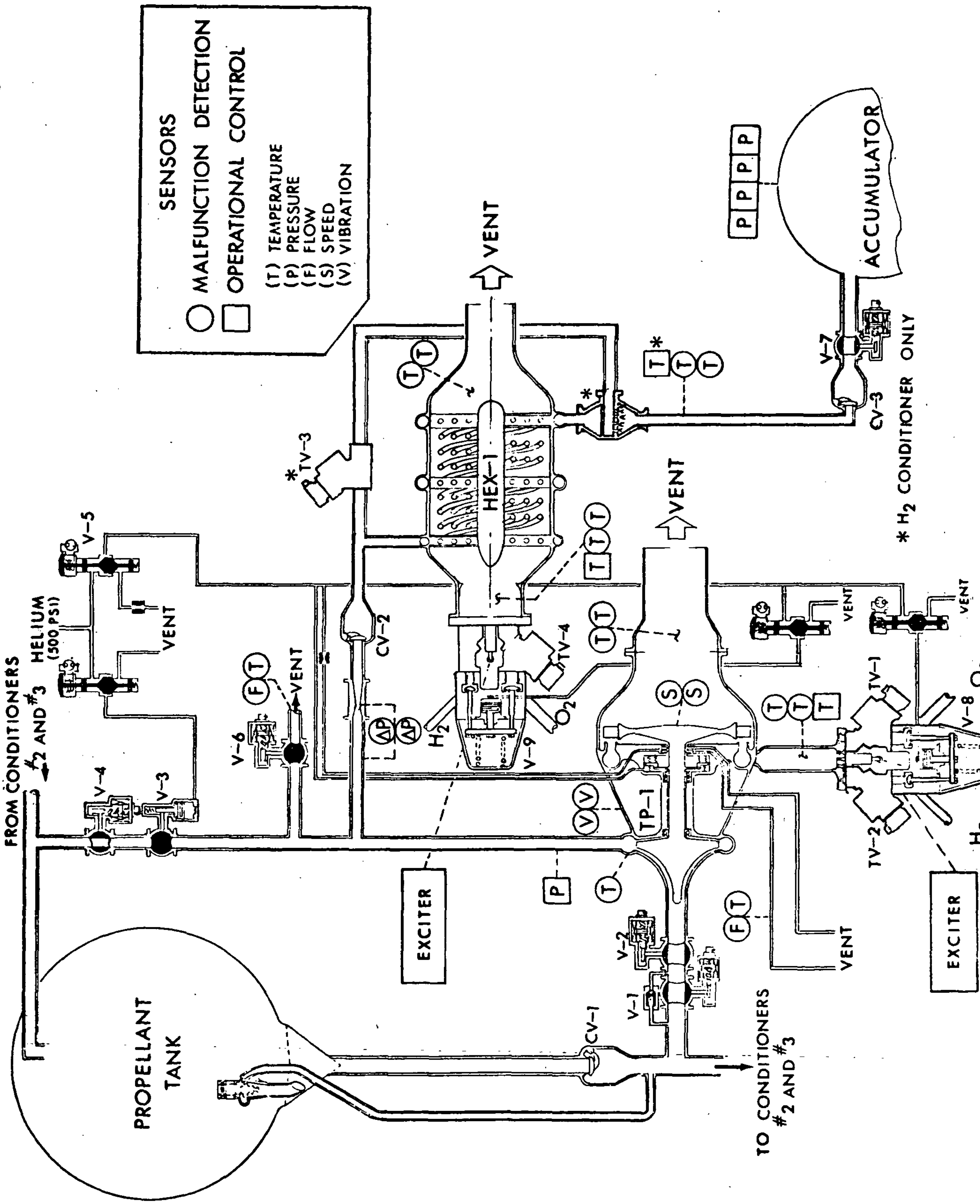


Figure 4-35

RCS COMPONENT SUMMARY

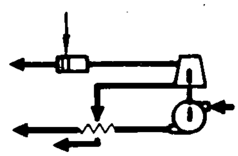
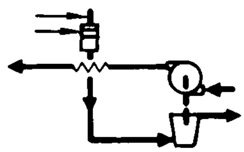
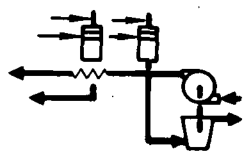
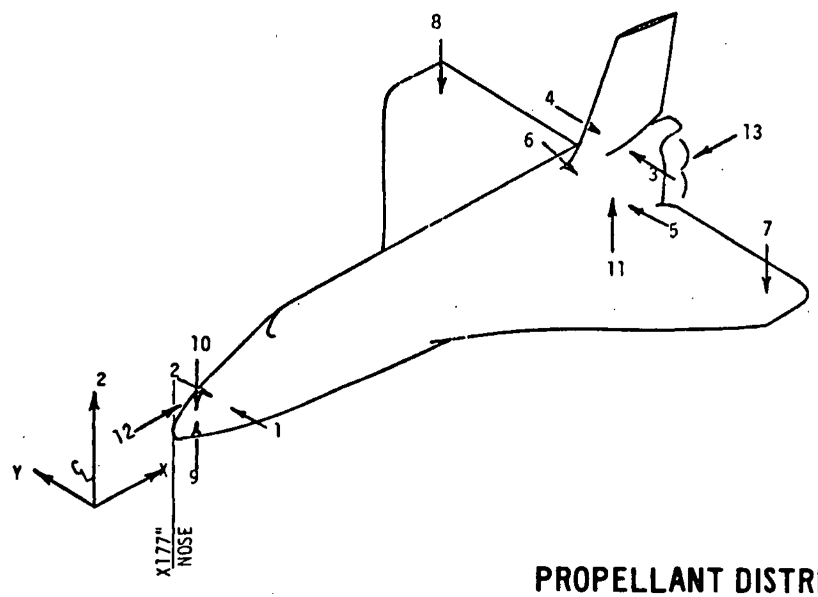
	SERIES: 	SERIES: 	PARALLEL: 
PROPELLANT STORAGE			
• NO. PRESSURANT/PROPELLANT TANKS	4	4	4
• NO. VALVES/REGUATORS	94	94	94
CONDITIONER ASSEMBLIES			
• NO. CONTROL VALVES	15	15	24
• NO. SENSORS	153	153	174
• NO. VALVES/GAS GENERATORS/PUMPS/ TURBINES/HEAT EXCHANGERS	126	126	156
PROPELLANT DISTRIBUTION AND THRUSTERS			
• NO. ACCUMULATORS	2	2	2
• NO. VALVES/REGULATORS	198	198	198
• NO. THRUSTERS	33	33	33
TOTAL NUMBER OF COMPONENTS (EXCLUDING SENSORS)	472	472	511

Figure 4-36

ORBITER THRUSTER LOCATIONS



LOCATION	NO. TCA'S	FUNCTION	X	Y	Z
1	2	+YAW, +Y	474	-93	281
2	2	-YAW, -Y	474	+93	281
3	2	-YAW, +Y, +ROLL	2029	-106	400
4	2	+YAW, -Y, -ROLL	2029	+106	400
5	2	-YAW, +Y, -ROLL	2029	-142	226
6	2	+YAW, -Y, +ROLL	2029	+142	226
7	3	+PITCH,-Z,-ROLL	1965	-530	235
8	3	+PITCH,-Z,+ROLL	1965	+530	235
9	2	+PITCH, +Z	368	0	175
10	3	-PITCH, -Z	368	0	294
11	2	-PITCH, +Z	1954	0	140
12	4	+X	474	0	307
13	4	-X	2100	0	307

PROPELLANT DISTRIBUTION

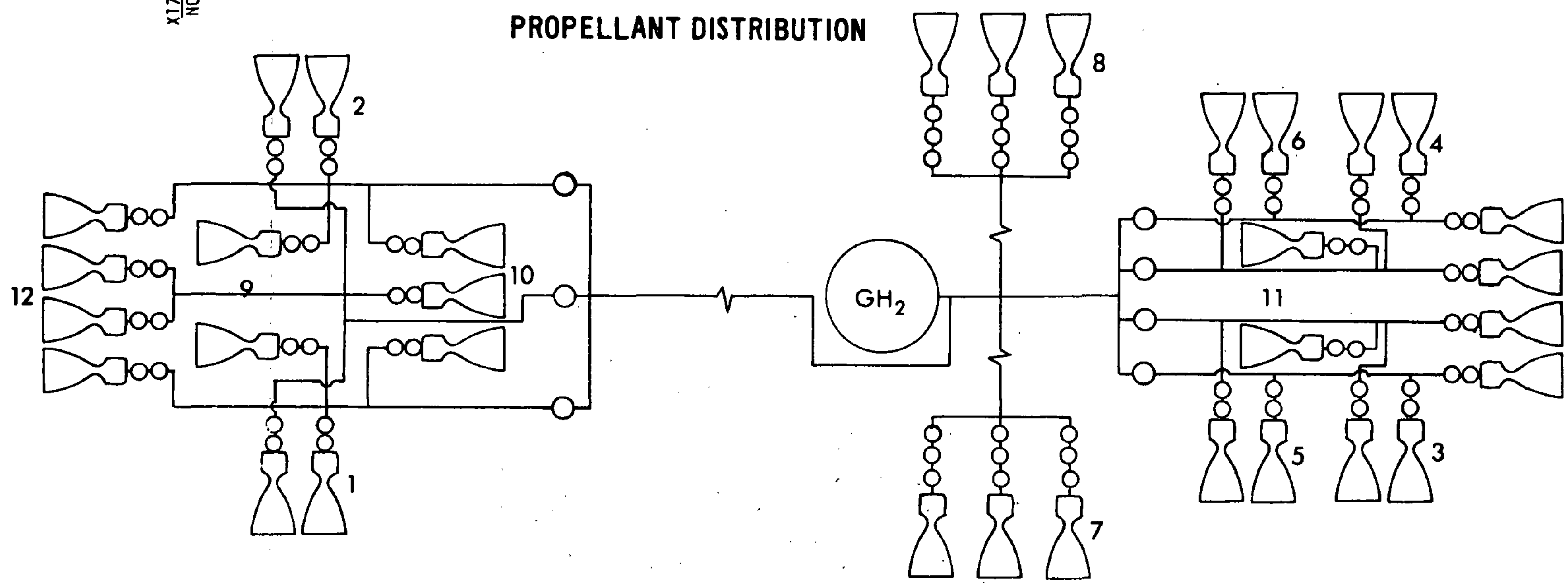


Figure 4-37

RCS FINAL DESIGN SUMMARY

• ORBITER RCS WITH SELECTED CONTROLS

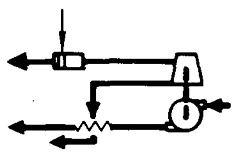
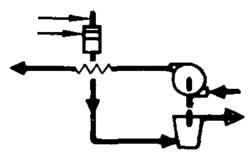
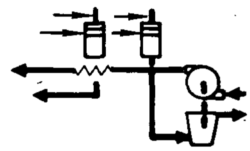
	SERIES:		SERIES:		PARALLEL:	
						
SYSTEM						
TOTAL IMPULSE, LBF-SEC	2.23 M		2.23 M		2.23 M	
SPECIFIC IMPULSE, SEC	367		366		348	
MIXTURE RATIO	3.12		3.11		2.91	
THRUSTER						
THRUST LEVEL, LBF	1150		1150		1150	
SPECIFIC IMPULSE, SEC	428		428		428	
MIXTURE RATIO	4		4		4	
CHAMBER PRESSURE, LBF/IN^{2}A	300		300		300	
GAS GENERATOR	H_2	O_2	H_2	O_2	H_2	O_2
COMBUSTION TEMPERATURE, ^{o}R	2000	2000	2000	2000	2000	2000
CHAMBER PRESSURE, LBF/IN^{2}A	250	250	250	250	250	250
MIXTURE RATIO	1	1	1	1	1	1
HEAT EXCHANGER						
HOT SIDE EXIT TEMPERATURE, ^{o}R	800	800	930	870	800	800
COLD SIDE EXIT TEMPERATURE, ^{o}R	263	503	253	506	245	466
TURBOPUMP						
FLOW RATE, LBM/SEC	3.80	11.86	3.82	11.88	4.23	12.30
DISCHARGE PRESSURE, LBF/IN^{2}A	1550	1750	1330	1850	1190	1570
TURBINE PRESSURE RATIO	2.60	2.10	8.00	2.90	16.7	16.7
ACCUMULATORS						
NUMBER OF CYCLES	50	50	50	50	50	50
VOLUME, FT^3	42.5	14.1	44.8	14.9	49.5	15.1
MINIMUM PRESSURE, LBF/IN^{2}A	571	571	571	571	571	571
SWITCH PRESSURE, LBF/IN^{2}A	665	666	665	666	660	661
MAXIMUM PRESSURE, LBF/IN^{2}A	1310	1555	1275	1545	1200	1450

Figure 4-38

FINAL CONDITIONER PRESSURE, TEMPERATURE AND FLOW BALANCE

o SERIES - UPSTREAM TURBINE RCS

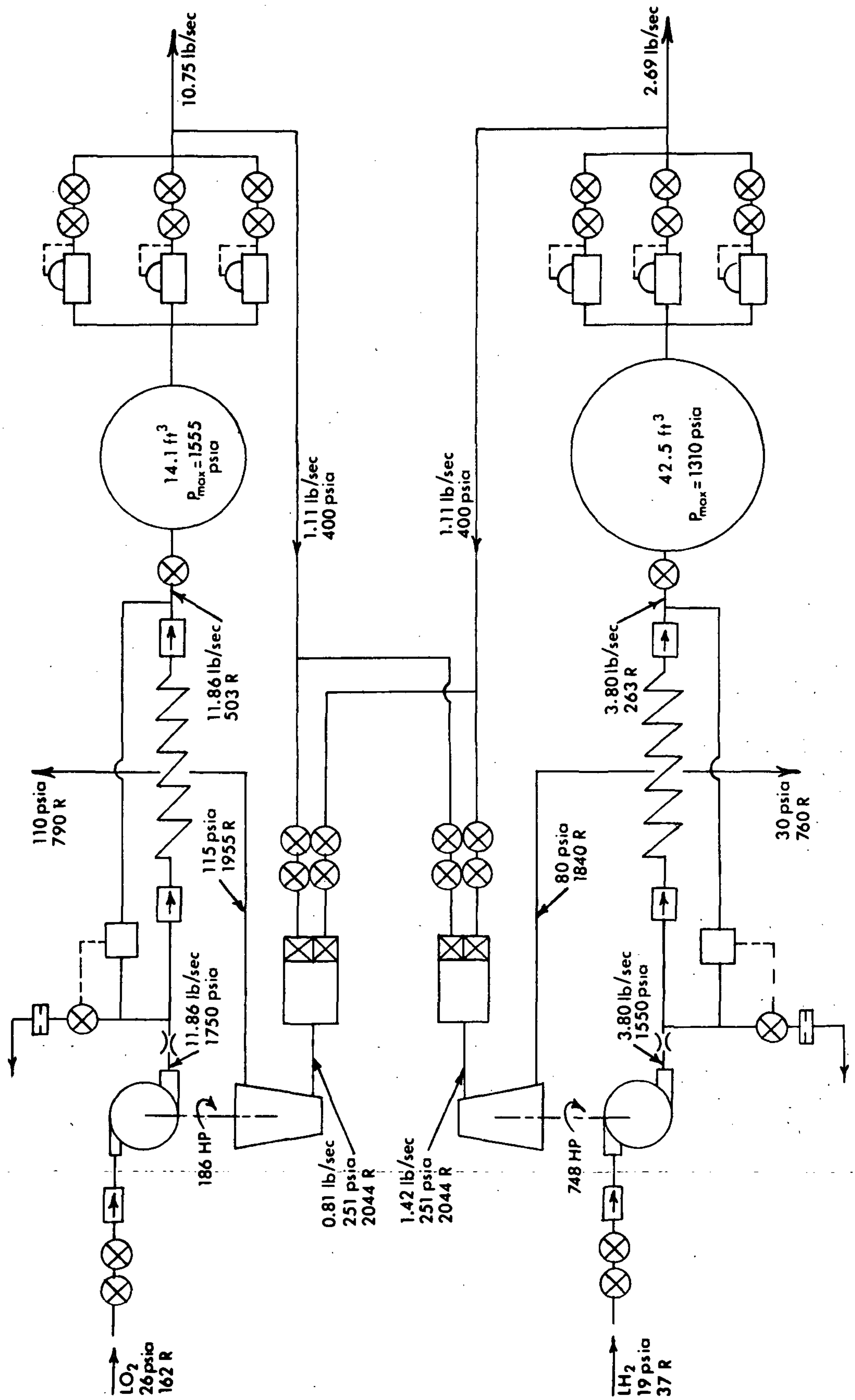


Figure 4-39

FINAL CONDITIONER PRESSURE, TEMPERATURE AND FLOW BALANCE

o SERIES - DOWNSTREAM TURBINE RCS

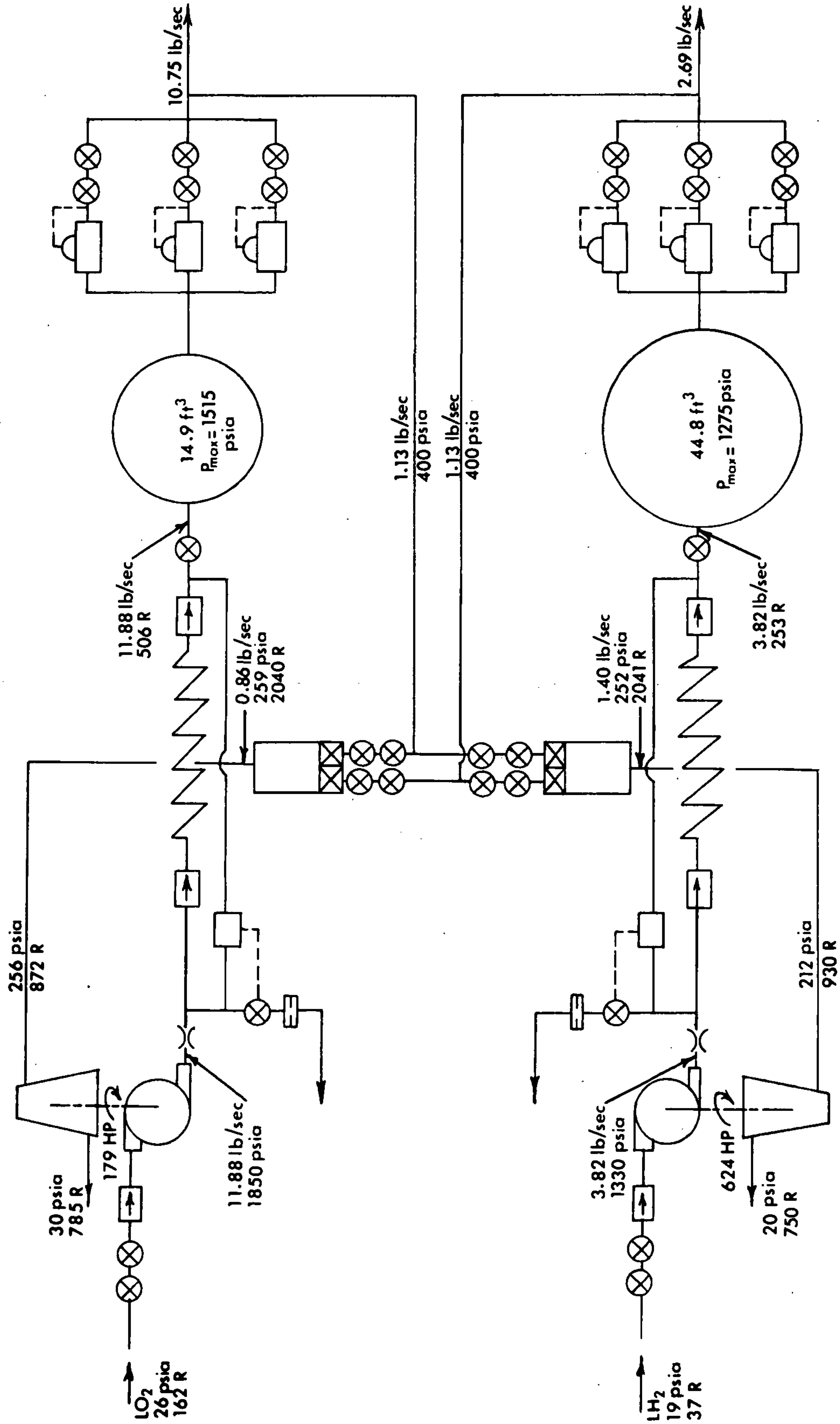


Figure 4-40

FINAL CONDITIONER PRESSURE, TEMPERATURE AND FLOW BALANCE

o PARALLEL RCS

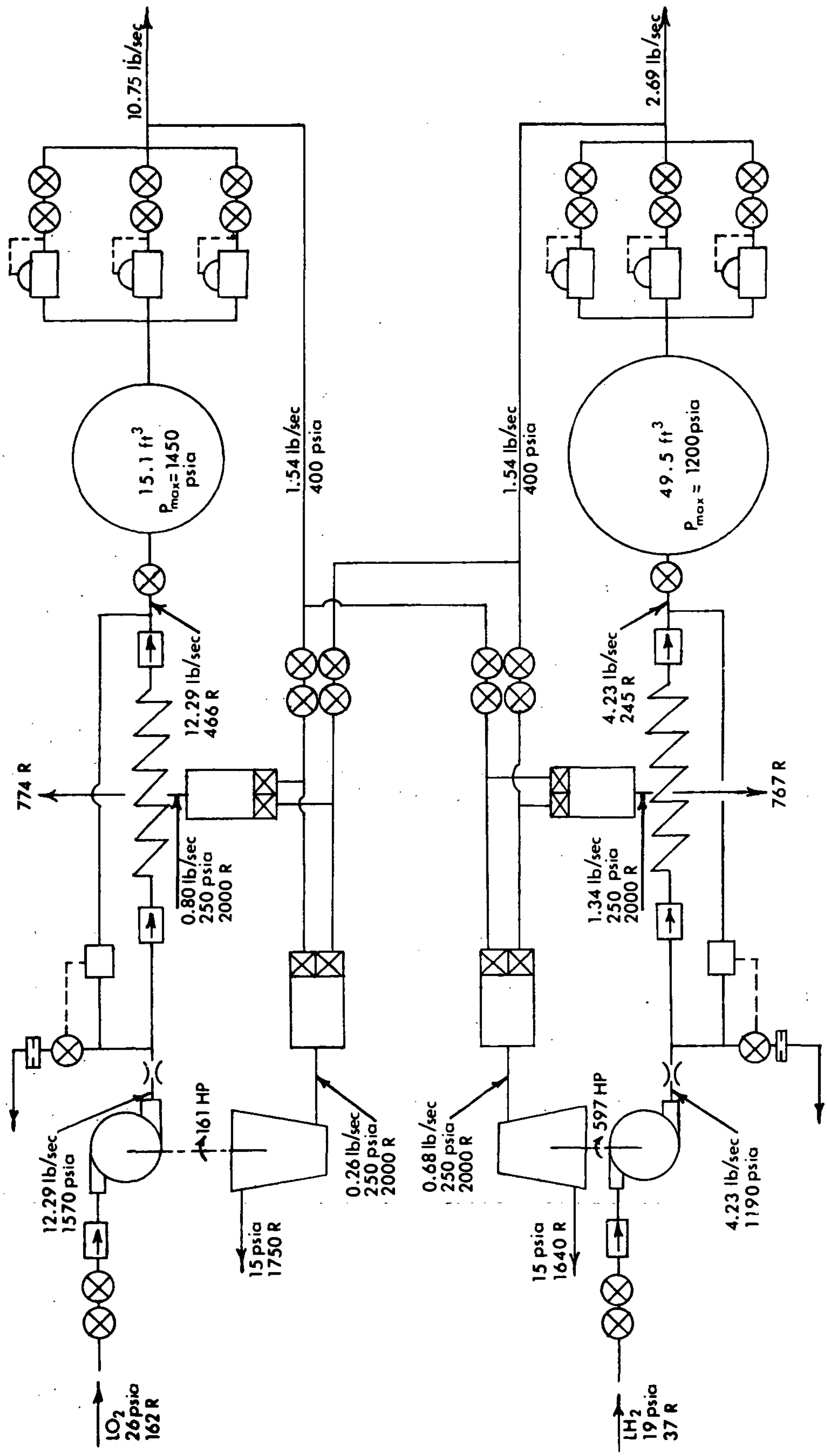


Figure 4-41

Detailed system weight breakdowns for the three RCS concepts are presented in Figure 4-42, and corresponding system weight sensitivities to variations in system design parameters are shown in Figures 4-43 through 4-45. These results parallel the preliminary sensitivity data of Appendix B with exception of lower optimum thruster chamber pressures. As shown in Figures 4-43 through 4-45, the optimum chamber pressure for all three RCS concepts is approximately 300 $lbf/in.^2a$, corresponding to the selected design value. This lower optimum chamber pressure results primarily from conditioned temperature and pump flow rate control bands which necessitate accumulators larger than those for the preliminary design points where the effect of conditioner operating bands was not considered.

The final system designs defined above and the selected controls of Section 4.2 formed the basis for final concept comparison and rating. These comparisons and ratings are presented in Section 5 with the primary study conclusions.

RCS FINAL WEIGHT SUMMARY

- ORBITER RCS WITH SELECTED CONTROLS
- TOTAL IMPULSE = 2.23 M LB-SEC
- 50 CONDITIONER CYCLES
- INCLUDES PROPELLANT ALLOWANCE FOR MIXTURE RATIO VARIATION

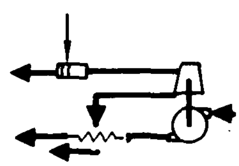
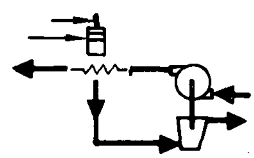
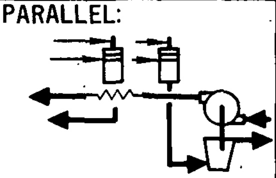
WEIGHT, LBM	SERIES: 	SERIES: 	PARALLEL: 
PROPELLANT	6,366	6,437	6,738
TANKAGE	502	515	532
PRESSURIZATION	165	166	185
INSULATION	157	157	166
CONDITIONER ASSEMBLIES	466	464	491
ACCUMULATORS	819	844	863
VENTS	145	159	320
LINES/VALVES/REGULATORS	620	614	620
THRUSTERS	992	992	992
TOTAL SYSTEM WEIGHT, LBM	10,232	10,348	10,907

Figure 4-42

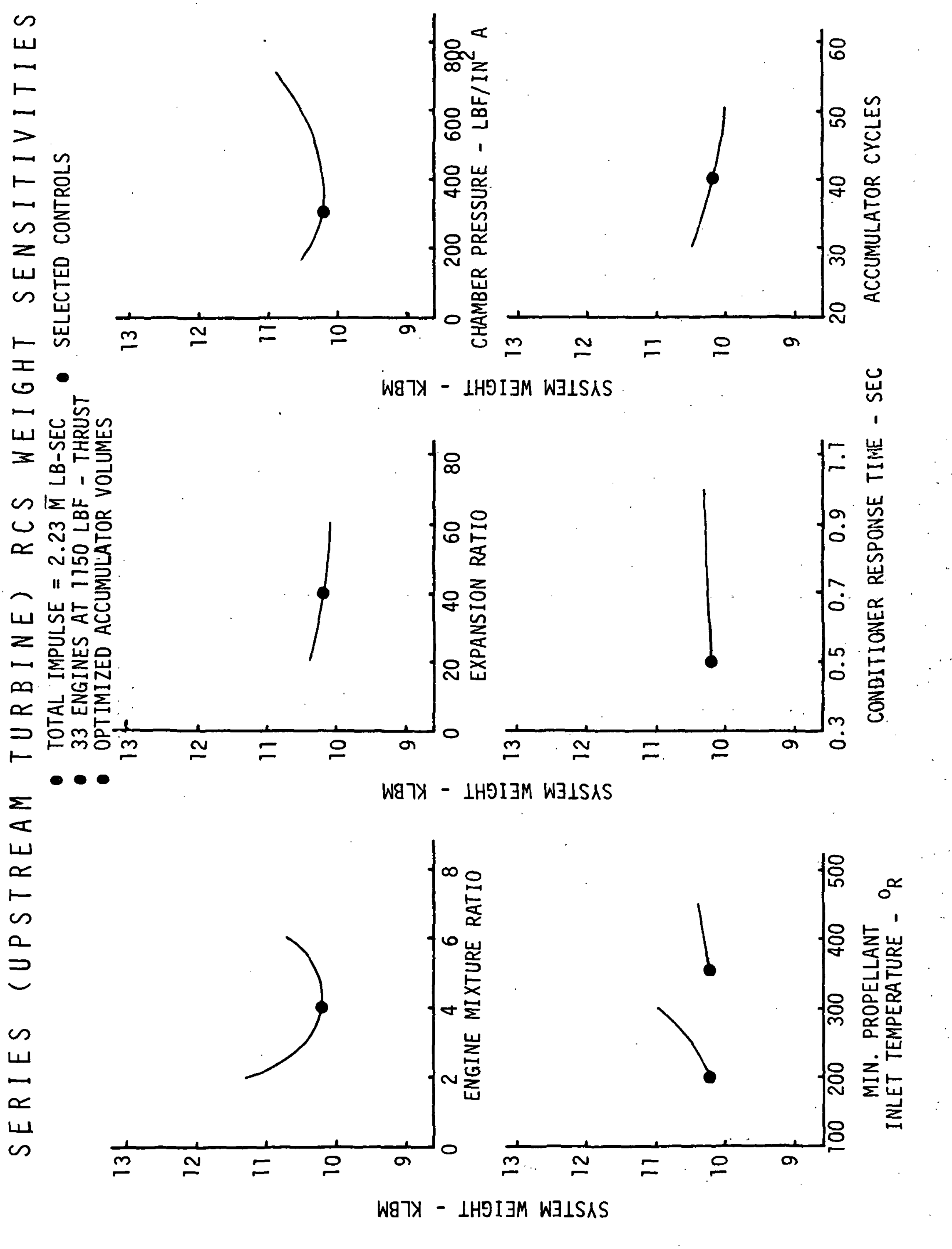
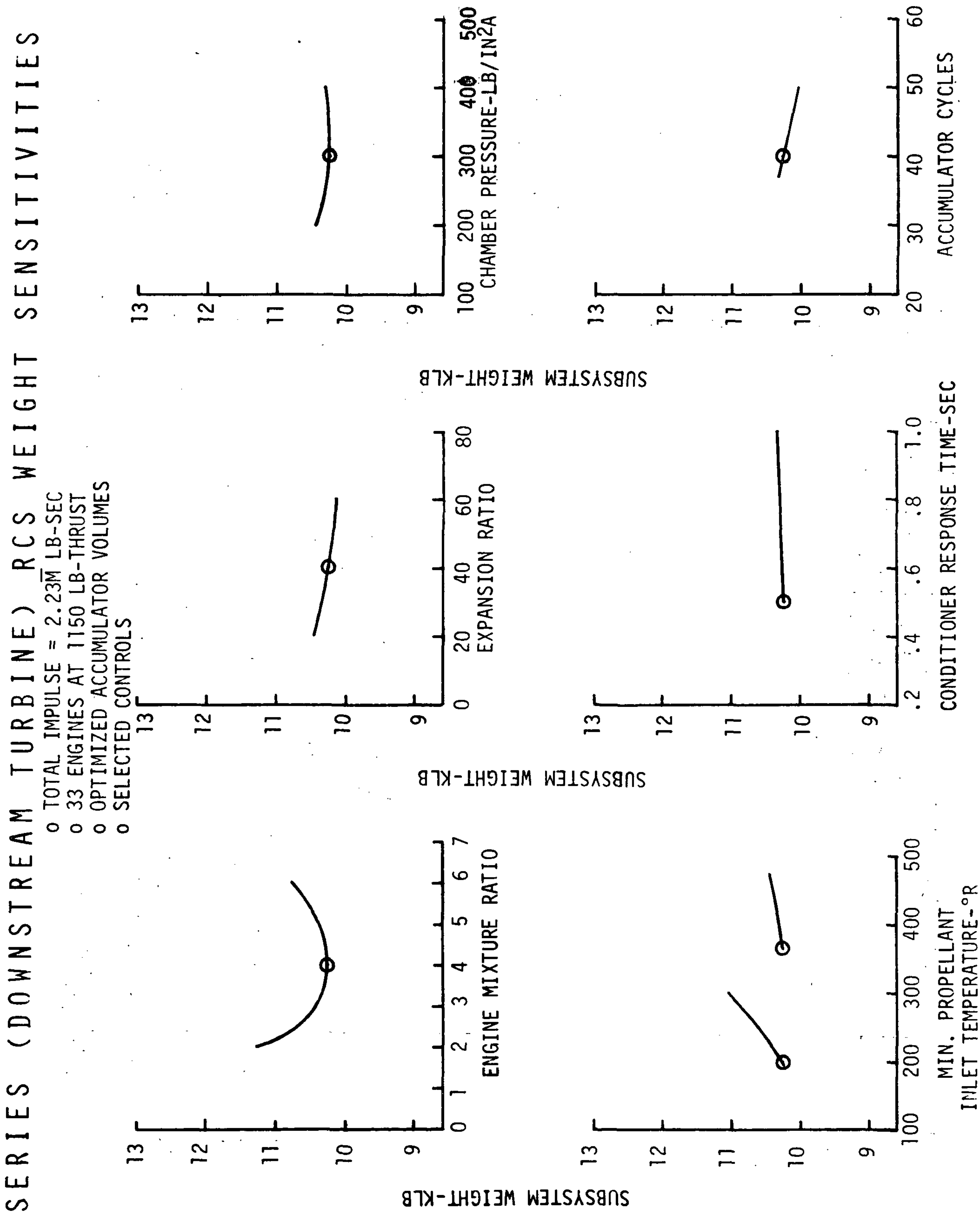

SERIES (UPSTREAM TURBINE) RCS WEIGHT SENSITIVITIES
• TOTAL IMPULSE = 2.23 M̄ LB-SEC
• 33 ENGINES AT 1150 LBF - THRUST
• OPTIMIZED ACCUMULATOR VOLUMES
• SELECTED CONTROLS
SYSTEM WEIGHT - KLBM
ENGINE MIXTURE RATIO
EXPANSION RATIO
CHAMBER PRESSURE - LBF/IN2 A
MIN. PROPELLANT INLET TEMPERATURE - °R
CONDITIONER RESPONSE TIME - SEC
ACCUMULATOR CYCLES

Figure 4-43


SERIES (DOWNSTREAM TURBINE) RCS WEIGHT SENSITIVITIES
o TOTAL IMPULSE = 2.23M LB-SEC
o 33 ENGINES AT 1150 LB-THRUST
o OPTIMIZED ACCUMULATOR VOLUMES
o SELECTED CONTROLS
SUBSYSTEM WEIGHT-KLB
ENGINE MIXTURE RATIO
EXPANSION RATIO
CHAMBER PRESSURE-LB/IN2A
MIN. PROPELLANT
INLET TEMPERATURE-°R
CONDITIONER RESPONSE TIME-SEC
ACCUMULATOR CYCLES

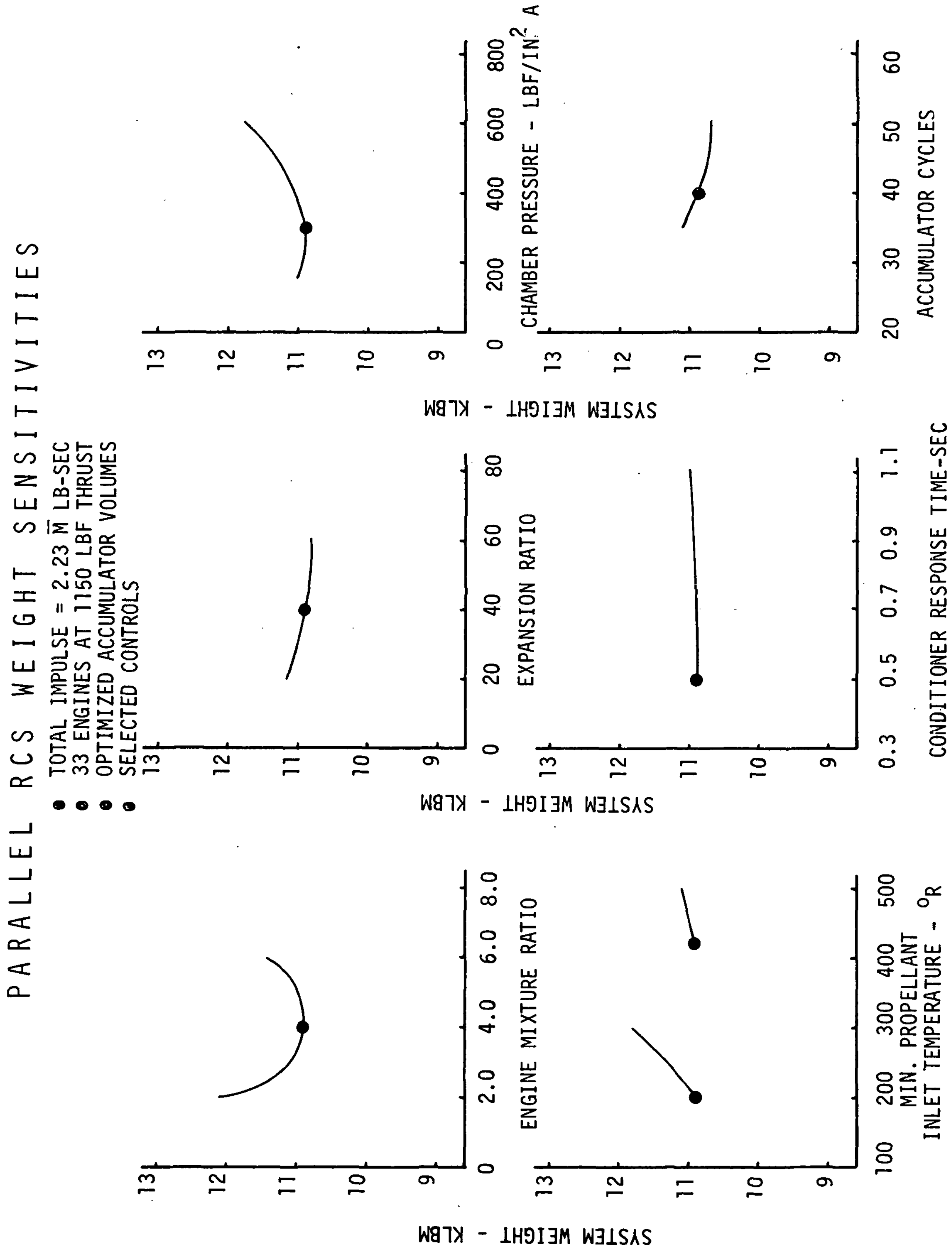

PARALLEL RCS WEIGHT SENSITIVITIES
• TOTAL IMPULSE = 2.23 M̄ LB-SEC
• 33 ENGINES AT 1150 LBF THRUST
• OPTIMIZED ACCUMULATOR VOLUMES
• SELECTED CONTROLS
SYSTEM WEIGHT - KLBM
CHAMBER PRESSURE - LBF/IN² A
ACCUMULATOR CYCLES
EXPANSION RATIO
CONDITIONER RESPONSE TIME-SEC
ENGINE MIXTURE RATIO
MIN. PROPELLANT
INLET TEMPERATURE - °R

Figure 4-45

5. RCS CONCEPT COMPARISON AND STUDY CONCLUSIONS

The three candidate RCS concepts were compared and rated applying the selection criteria and weighting factors presented in Figure 5-1. The detailed evaluations are summarized in Figures 5-2 through 5-4, and final system ratings for all three concepts are summarized in Figure 5-5. As shown, the two series concepts are rated even (scoring 77 out of a possible 100 points), ranking higher than the parallel RCS (63 points) in all categories except flexibility and required technology. A choice between the two series concepts depends on Shuttle development philosophy. The series-upstream turbine RCS affords the lowest weight and volume, but to take advantage of this performance benefit, system requirements must be firmly established at program outset. Attempts to improve system performance at a later date would lead to a complete conditioner redesign due to turbine inlet temperature restrictions. To a lesser extent, this is also true of the series-downstream turbine RCS. However, in this case, some performance uprating can be accomplished by increasing gas generator combustion temperature and reconfiguring only the heat exchanger for a higher hot side inlet temperature. Turbine inlet temperature would remain within acceptable limits (less than or equal to 2000°R) but turbine pressure ratios would increase necessitating a larger vent system. Parametric analyses presented in Figure 5-6 show that for a gas generator combustion temperature of 4000°R, a 20 second gain in system specific impulse could be realized for the series-downstream turbine RCS. This corresponds to a 340 lbm system weight reduction for the vehicle impulse requirements defined in Figure 5-6.

The performance of the parallel RCS is lower than either of the series concepts because of the high waste enthalpy (high exhaust temperature) at the turbine vents. However, as shown in Figure 5-7 much of this performance deficit could be overcome by increasing the heat exchanger inlet temperature to 4000°R (maintaining turbine gas generator combustion temperature at 2000°R). The parallel RCS offers the greatest flexibility of the candidate concepts because it can be modified without turbopump changes. However, it is most attractive if a phased, Block I - Block II Shuttle development is planned. In this case, the parallel gaseous O_2/H_2 RCS would be a strong candidate for the earlier vehicle because of commonality with subsequent advanced system concepts. Specifically, the parallel RCS hydrogen turbopump has design requirements similar to those of the turbopump on a liquid RCS (Reference J) and could be used in a cryogenic liquid RCS with little or no modification.

Based on the controls evaluation, it was found that all three RCS concepts can be controlled within tight operational limits. Gas generator combustion temperature

RCS SELECTION CRITERIA AND WEIGHTING FACTORS

SELECTION CRITERIA	WEIGHTING (TOTAL POINTS)	RATIONALE
SYSTEM WEIGHT AND VOLUME	25	Weighting based on absolute system weight and tankage/accumulator volume. System weight accounts for 15 pts., and volume accounts for 10 pts. Lowest system weight and volume receive maximum point award. Using the lowest system weight and volume as a reference, one point is deducted for each 100 lb in excess of the reference weight, and one point is deducted for each 10 ft^3 in excess of the reference volume.
CONDITIONER COMPLEXITY	15	Weighting based on consideration of: (1) total number of components - 4 pts.; (2) number of controls - 6 pts.; (3) number of sensors - 3 pts.; and (4) number of pump/turbine stages and vents - 2 pts. Unit numbers included in each of these categories include the required redundancy to satisfy the fail-operational, fail-safe criterion. The system having the fewest number of components in each category receives the maximum point award. Lower point awards are based on engineering judgement.
FLEXIBILITY	25	Weighting based on system weight sensitivity to: (1) mission total impulse - 7 pts.; (2) thrust level - 4 pts.; (3) number of conditioner cycles - 3 pts.; (4) maximum number of engines firing at one time - 3 pts; and (5) potential for improved conditioner performance - 8 pts. System having lowest sensitivity in each category receives maximum point award. Lower point awards are based on engineering judgement.
CONDITIONER RELIABILITY/SAFETY	15	Weighting based on number of potential catastrophic malfunction modes (10 pts.) and number of potential non-catastrophic malfunction modes (5 pts.). System having no catastrophic malfunction modes would receive maximum 10 point award. 2 points are deducted for each catastrophic malfunction mode. System having fewest non-catastrophic malfunction modes receives maximum 5 point award. Lower point awards are based on engineering judgement.
CONDITIONER TECHNOLOGY CONSIDERATIONS	20	Weighting based on engineering judgement of development risk. System having no critical technology areas would receive maximum point award. Up to a maximum of 3 points is deducted for each critical technology area.

Figure 5-1

SERIES (UPSTREAM TURBINE) RCS EVALUATION

- TOTAL IMPULSE = 2.23 $\overline{M}$ LB-SEC
- 33 THRUSTERS AT 1150 LB-THRUST
- SELECTED CONTROLS

CATEGORY	EVALUATION	POINTS
SYSTEM WEIGHT AND VOLUME (25 PTS.)	o TOTAL SYSTEM WEIGHT = 10,232 (Includes propellant allowance for mixture ratio variations)	15
	o TOTAL ACCUMULATOR/TANKAGE VOLUME = 531 FT^3	10
		SUBTOTAL=25
CONDITIONER COMPLEXITY (15 PTS.)	o NO. OF TOTAL SYSTEM COMPONENTS = 472 o NO. OF CONDITIONER COMPONENTS = 126 (Valves, pumps, gas generators, etc.)	4
	o NO. OF CONTROL VALVES = 15	6
	o NO. OF SENSORS (Pressure, temperature and flow)=153	3
	o NO. OF O_2 PUMP STAGES = 1 o NO. OF H_2 PUMP STAGES = 1 o NO. OF O_2 TURBINE STATES = 1 o NO. OF H_2 TURBINE STAGES = 2 o NO. OF EXHAUST VENTS = 6	2
		SUBTOTAL=15
FLEXIBILITY (25 PTS.)	o TOTAL IMPULSE SENSITIVITY ($\Delta W/\Delta I$)= 0.00338 LB_M/LB-SEC	7
	o THRUST LEVEL SENSITIVITY ($\Delta W/\Delta F$) = 0.5960 LB_M/LB	3
	o CONDITIONER CYCLE SENSITIVITY ($\Delta W/\Delta N_C$) = -24.6 LB_M/CYCLE	2
	o SENSITIVITY TO MAX. NO. OF THRUSTERS FIRING ($\Delta W/\Delta N_F$) = 62.0 LB_M/THRUSTER	2
	o PTOENTIAL FOR IMPROVED CONDITIONER PERFORMANCE	3
		SUBTOTAL=17
CONDITIONER RELIABILITY/ SAFETY (15 PTS.)	o NO. OF POTENTIAL CATASTROPHIC MALFUNCTION MODE=3 (1) Excessive O_2 leakage into fuel rich hot side heat exchanger flow. (2) Loss of GGA O_2 throttle valve control coupled with isolation valve failure causing excessive gas temperature and failure of turbine rotor blades. (3) Propellant flow cavitation causing pump overspeed	4
	o NO. OF POTENTIAL NON-CATASTROPHIC MALFUNCTION MODES = 61	5
		SUBTOTAL=9
CONDITIONER TECHNOLOGY CONSIDERATION (20 PTS.)	o RAPID TURBOPUMP SPIN-UP (Required rotational acceleration is approximately 200,000 RPM/sec-H_2)	(-3)
	o HIGH TURBOPUMP/HEAT EXCHANGER CYCLE LIFE (5000 CYCLES)	(-2)
	o POTENTIAL H_2O CONDENSATION/ICING ON HEAT EXCHANGER HOT SIDE TUBE WALLS	(-2)
	o POTENTIAL HEAT EXCHANGER FLOW INSTABILITY	(-2)
		SUBTOTAL=11
	TOTAL POINTS	77

Figure 5-2

SERIES (DOWNSTREAM TURBINE) RCS EVALUATION

- TOTAL IMPULSE = 2.23 M LB-SEC
- 33 THRUSTERS AT 1150 LB-THRUST
- SELECTED CONTROLS

CATEGORY	EVALUATION	POINTS
SYSTEM WEIGHT AND VOLUME (25 PTS.)	o TOTAL SYSTEM WEIGHT = 10,348 LB_M (Includes propellant allowance for mixture ratio variations)	14
	o TOTAL ACCUMULATOR/TANKAGE VOLUME = 538 FT^3	10
		SUBTOTAL=24
CONDITIONER COMPLEXITY (15 PTS.)	o NO. OF TOTAL SYSTEM COMPONENTS = 472 o NO. OF CONDITIONER COMPONENTS = 126 (Valves, pumps, gas generators, etc.)	4
	o NO. OF CONTROL VALVES = 15	6
	o NO. OF SENSORS (Pressure, temperature and flow) = 153	3
	o NO. OF O_2 PUMP STAGES = 1 o NO. OF H_2 PUMP STAGES = 1 o NO. OF O_2 TURBINE STAGES = 2 o NO. OF H_2 TURBINE STAGES = 2 o NO. OF EXHAUST VENTS = 6	1
		SUBTOTAL=14
FLEXIBILITY (25 PTS.)	o TOTAL IMPULSE SENSITIVITY ($\Delta W/\Delta I$) = 0.00340 LB_M/LB-SEC	6
	o THRUST LEVEL SENSITIVITY ($\Delta W/\Delta F$) = 0.780 LB_M/LB-SEC	2
	o CONDITIONER CYCLE SENSITIVITY ($\Delta W/\Delta N_C$) = -22.2 LB_M/CYCLE	3
	o SENSITIVITY TO MAX. NO. OF THRUSTERS FIRING ($\Delta W/\Delta N_F$) = 70.5 LB_M/THRUSTER	1
	o POTENTIAL FOR IMPROVED CONDITIONER PERFORMANCE	6
		SUBTOTAL=18
CONDITIONER RELIABILITY/ SAFETY (15 PTS.)	o NO. OF POTENTIAL CATASTROPHIC MALFUNCTION MODES = 3 (1) Excessive O_2 leakage into fuel rich hot side heat exchanger flow. (2) Loss of GGA O_2 throttle valve control coupled with isolation valve failure causing excessive gas temperature and failure of turbine rotor blades. (3) Propellant flow cavitation causing pump overspeed	4
	o NO. OF POTENTIAL NON-CATASTROPHIC MALFUNCTION MODES = 61	5
		SUBTOTAL=9
CONDITIONER TECHNOLOGY CONSIDERATION (20 PTS.)	o RAPID TURBOPUMP SPIN-UP (Required rotational acceleration is approximately 160,000 RPM/sec-H_2)	(-2)
	o HIGH TURBOPUMP/HEAT EXCHANGER CYCLE LIFE (5000 CYCLES)	(-2)
	o POTENTIAL H_2O CONDENSATION/ICING OF TURBINE BLADES	(-2)
	o POTENTIAL HEAT EXCHANGER FLOW INSTABILITY	(-2)
		SUBTOTAL=12
	TOTAL POINTS	77

Figure 5-3

PARALLEL RCS

- TOTAL IMPULSE = 2.23 $\overline{M}$ LB-SEC
- 33 THRUSTERS AT 1150 LB-THRUST
- SELECTED CONTROLS

CATEGORY	EVALUATION	POINTS
SYSTEM WEIGHT AND VOLUME (25 PTS.)	o TOTAL SYSTEM WEIGHT = 10,907 LB (Includes propellant allowance for mixture ratio variations)	8
	o TOTAL ACCUMULATOR/TANKAGE VOLUME = 580 FT^3	5
		SUBTOTAL=13
CONDITIONER COMPLEXITY (15 PTS.)	o NO. OF TOTAL SYSTEM COMPONENTS = 511 o NO. OF CONDITIONER COMPONENTS = 156 (Valves, pumps, gas generators, etc.)	3
	o NO. OF CONTROL VALVES = 24	4
	o NO. OF SENSORS (Pressure, temperature and flow) = 174	2
	o NO. OF O_2 PUMP STAGES = 1 o NO. OF H_2 PUMP STAGES = 1 o NO. OF O_2 TURBINE STAGES = 3 o NO. OF H_2 TURBINE STAGES = 3 o NO. OF EXHAUST VENTS = 12	0
		SUBTOTAL=9
FLEXIBILITY (25 PTS.)	o TOTAL IMPULSE SENSITIVITY ($\Delta W/\Delta I$) = 0.00362 LB_M/LB-SEC	4
	o THRUST LEVEL SENSITIVITY ($\Delta W/\Delta F$) = 0.540 LB_M/LB	4
	o CONDITIONER CYCLE SENSITIVITY ($\Delta W/\Delta N_C$) = -30.8 LB_M/CYCLE	1
	o SENSITIVITY TO MAX. NO. OF THRUSTERS FIRING ($\Delta W/\Delta N_F$) = 56.5 LB_M/THRUSTERS	3
	o POTENTIAL FOR IMPROVED CONDITIONER PERFORMANCE	8
		SUBTOTAL=20
CONDITIONER RELIABILITY/ SAFETY (15 PTS.)	o NO. OF POTENTIAL CATASTROPHIC MALFUNCTION MODES = 3 (1) Excessive O_2 leakage into fuel rich hot side heat exchanger flow. (2) Loss of GGA O_2 throttle valve control coupled with isolation valve failure causing excessive gas temperature and failure of turbine rotor blades. (3) Propellant flow cavitation causing pump overspeed	4
	o NO. OF POTENTIAL NON-CATASTROPHIC MALFUNCTION MODES = 75	4
		SUBTOTAL=8
CONDITIONER TECHNOLOGY CONSIDERATION (20 PTS.)	o RAPID TURBOPUMP SPIN-UP (Required rotational acceleration is approximately 110,000 RMP/sec-H_2)	(-1)
	o HIGH TURBOPUMP/HEAT EXCHANGER CYCLE LIFE (5000 CYCLES)	(-2)
	o POTENTIAL H_2O CONDENSATION/ICING ON HEAT EXCHANGER HOT SIDE TUBE WALLS	(-2)
	o POTENTIAL HEAT EXCHANGER FLOW INSTABILITY	(-2)
		SUBTOTAL=13
	TOTAL POINTS	63

Figure 5-4

RCS CONCEPT COMPARISON

CRITERION	WEIGHTING (TOTAL POINTS)	RATIONALE	POINTS AWARDED: SERIES RCS (TURBINE UPSTREAM)	POINTS AWARDED: SERIES RCS (TURBINE DOWNSTREAM)	POINTS AWARDED: PARALLEL RCS
SYSTEM WEIGHT AND VOLUME	25	ABSOLUTE SYSTEM WEIGHT	15	14	8
		VOLUME	10	10	5
		SUBTOTAL	25	24	13
CONDITIONER COMPLEXITY	15	NO. OF COMPONENTS	4	4	3
		NO. OF CONTROLS	6	6	4
		NO. OF SENSORS	3	3	2
		NO. OF PUMP/TURBINE STAGES AND VENTS	2	1	0
		SUBTOTAL	15	14	9
FLEXIBILITY	25	SYSTEM WEIGHT SENSITIVITY TO –			
		MISSION TOTAL IMPULSE	7	6	4
		THRUST LEVEL	3	2	4
		NO. CONDITIONER CYCLES	2	3	1
		MAX. NO. THRUSTERS FIRING	2	1	3
		POTENTIAL FOR IMPROVED PERFORMANCE	3	6	8
		SUBTOTAL	17	18	20
CONDITIONER RELIABILITY/ SAFETY	15	NO. OF POTENTIAL CATASTROPHIC MALFUNCTION MODES	4	4	4
		NO. OF POTENTIAL NON-CATASTROPHIC MALFUNCTION MODES	5	5	4
		SUBTOTAL	9	9	8
CONDITIONER TECHNOLOGY CONSIDERATIONS	20	ENGINEERING JUDGEMENT OF DEVELOPMENT RISK	11	12	13
		SUBTOTAL	11	12	13
		TOTAL POINTS	77	77	63

Figure 5-5

RCS PERFORMANCE PARAMETRIC

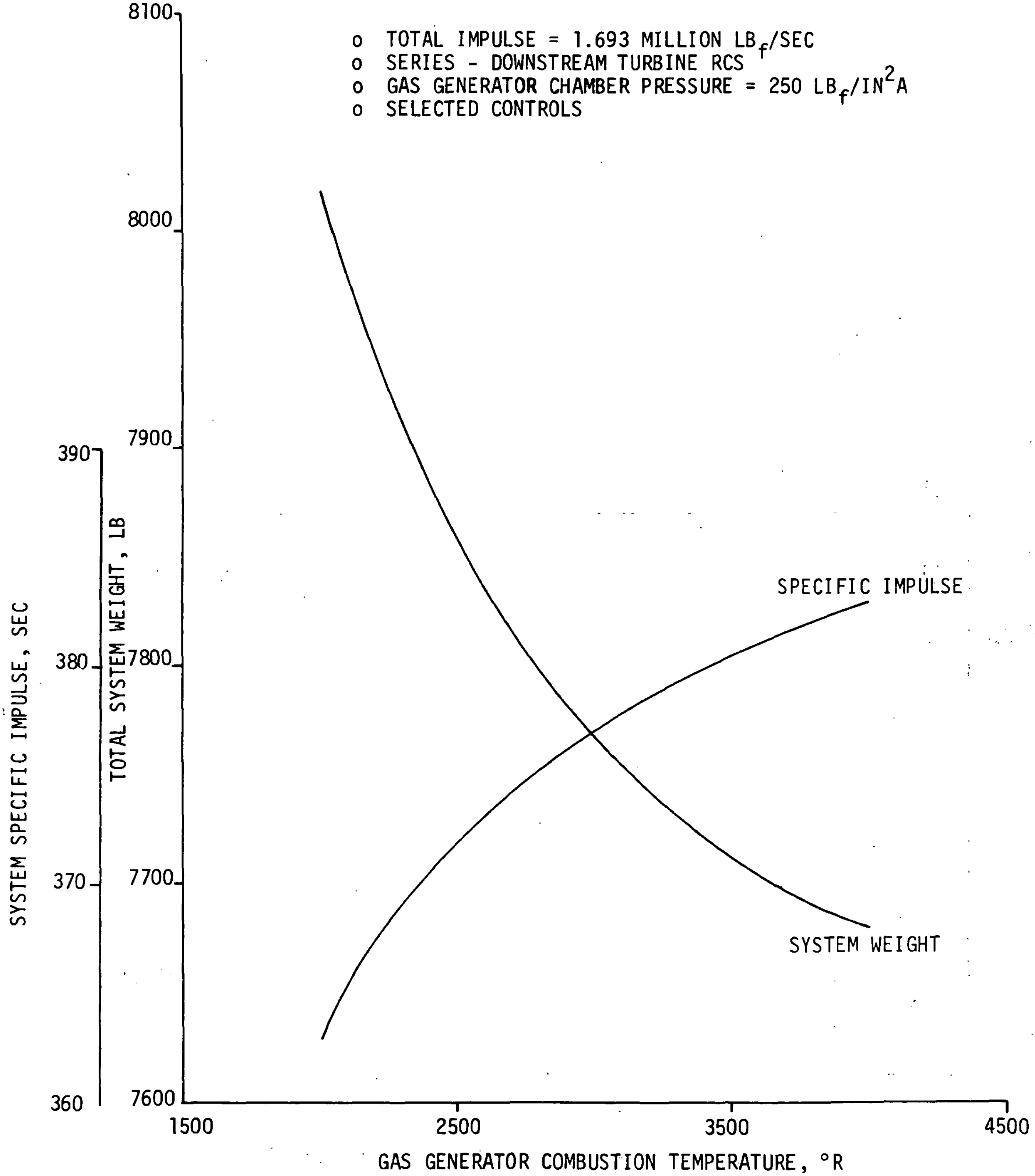


Figure 5-6

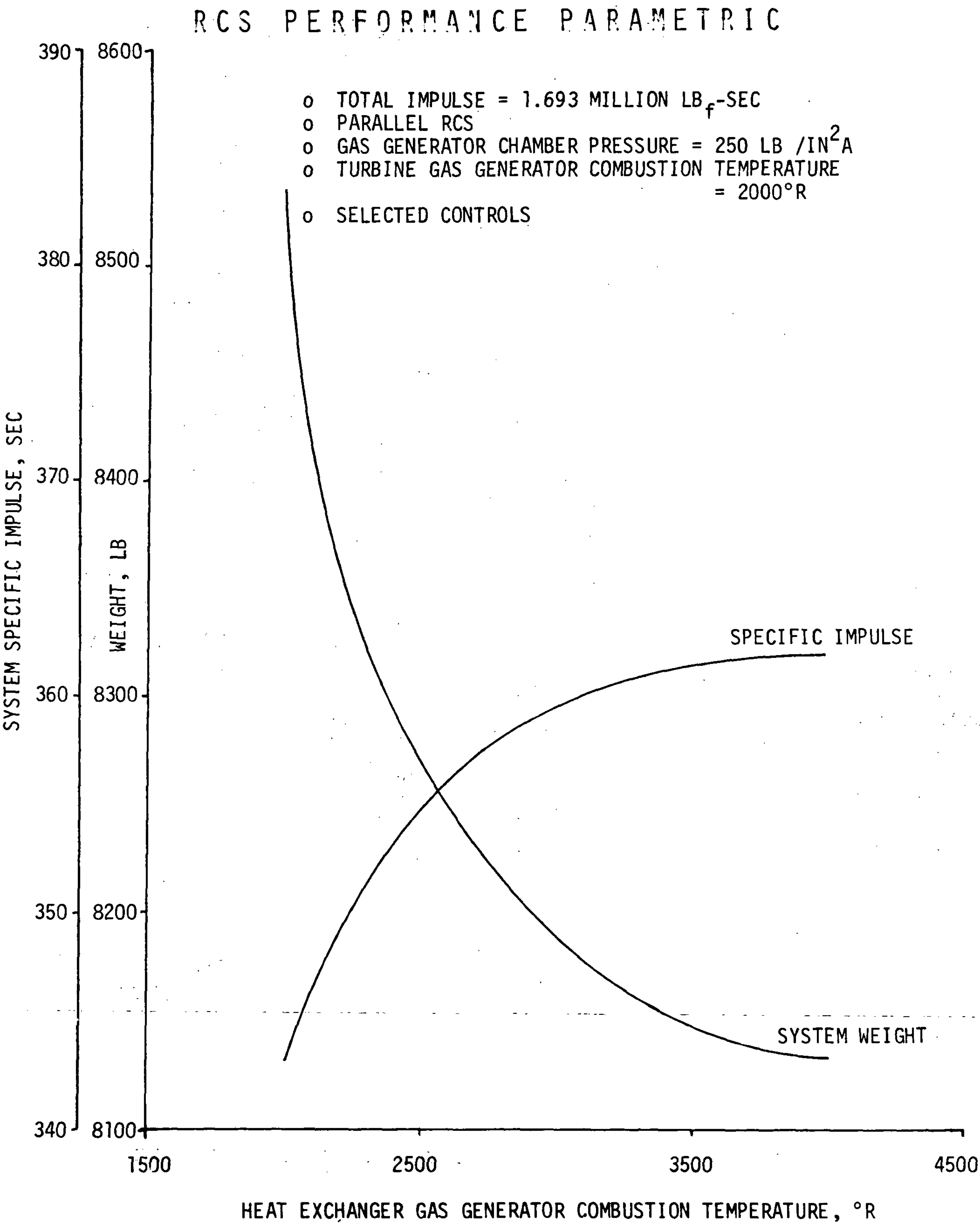

RCS PERFORMANCE PARAMETRIC
o TOTAL IMPULSE = 1.693 MILLION LB$_f$-SEC
o PARALLEL RCS
o GAS GENERATOR CHAMBER PRESSURE = 250 LB /IN^{2}A
o TURBINE GAS GENERATOR COMBUSTION TEMPERATURE = 2000°R
o SELECTED CONTROLS
SPECIFIC IMPULSE
SYSTEM WEIGHT
SYSTEM SPECIFIC IMPULSE, SEC
WEIGHT, LB
390
380
370
360
350
340
8600
8500
8400
8300
8200
8100
1500
2500
3500
4500
HEAT EXCHANGER GAS GENERATOR COMBUSTION TEMPERATURE, °R

Figure 5-7

control was found to be necessary to avoid excessive turbine/heat exchanger gas inlet temperatures, and was best achieved through modulation of the gas generator oxygen valve. This control provided a large system weight reduction and reduced the operating bands of other critical system parameters such as conditioned propellant temperature and pump discharge pressure (flow rate). Control of hydrogen conditioned temperature in the two series concepts and both hydrogen and oxygen conditioned temperature in the parallel RCS was selected to provide additional system weight reductions. This control was best achieved by modulating the amount of heat exchanger cold side bypass flow. This heat exchanger cold side bypass was required in the series-upstream turbine RCS (hydrogen conditioner) and parallel RCS (both hydrogen and oxygen conditioners) to preclude H_2O condensation and icing on the hot side tube walls. It was incorporated in the series-downstream turbine RCS (hydrogen conditioner) for the purpose of conditioned temperature control, only. The final system control provided for modulation of the gas generator H_2 valve in response to pump discharge pressure. Whereas this control effected only a modest system weight benefit (100-150 lbm), it provided excellent control of heat exchanger cold side inlet conditions and turbopump power, minimizing the development risk associated with these assemblies. Heat exchanger cold side flow instability has been encountered in previous development programs, and its potential for occurrence is reduced with tight control of inlet conditions.

Several technology concerns were identified during the course of the study. Among these was the high conditioner cycle life requirement of 5000 cycles (50 cycles per mission for 100 missions) which is a significant extension over the demonstrated life capabilities of current turbopump and heat exchanger designs. In addition, transient conditioner startup analyses showed that turbopump shaft accelerations in the order of 165,000-200,000 RPM/sec can be expected. Since current experience with propellant-cooled bearings is approximately 40,000 RPM/sec, pump bearing design must be regarded as a critical technology area.

6. REFERENCES

A. Kendall, A. S., McKee, H. B., and Orton, G. F., "Space Shuttle Low Pressure Auxiliary Propulsion Subsystem Definition - Subtask A Report", McDonnell Douglas Report No. MDC E0303, 29 January 1971.

B. Green, W. M., and Patten, T. C., "Space Shuttle Low Pressure Auxiliary Propulsion Subsystem Definition - Subtask B. Report", McDonnell Douglas Report No. MDC E0302, 29 January 1971.

C. Anglim, D. D., Baumann, T. L., and Ebbesmeyer, L. H., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Subtask A Report", McDonnell Douglas Report No. MDC E0299, 12 February 1971.

D. Gaines, R. D., Goldford, A. I., and Kaemming, T. A., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Subtask B Report", McDonnell Douglas Report No. MDC E0298, 12 February 1971.

E. Kelly, P. J., "Space Shuttle Auxiliary Propulsion System Design Study - Executive Summary", McDonnell Douglas Report No. MDC E0674, 29 December 1972.

F. Kelly, P. J., "Space Shuttle Auxiliary Propulsion System Design Study - Program Plan", McDonnell Douglas Report No. MDC E0436, 15 July 1971, 6 December 1971.

G. Orton, G. F. and Schweickert, T. F., "Space Shuttle Auxiliary Propulsion System Design Study - Phase A Requirements Definition", McDonnell Douglas Report No. MDC E0603, 15 February 1972.

H. Bruns, A. E., and Regnier, W. W., "Space Shuttle Auxiliary Propulsion System Design Study - Phase C Oxygen-Hydrogen RCS/OMS Integration", McDonnell Douglas Report No. MDC E0436, 15 June 1972.

I. Anglim, D. D., Bruns, A. E., Perryman, D. C., and Wieland, D. L., "Space Shuttle Auxiliary Propulsion System Design Study - Phase C, Earth Storable RCS/OMS/APU Integration and Phase E System Performance Analysis", McDonnell Douglas Report MDC No. E0708, 29 December 1972.

J. Baumann, T. L., Patten, T. C., and McKee, H. B., "Space Shuttle Auxiliary Propulsion System Design Study - Phase D Special RCS Studies", McDonnell Douglas Report No. MDC E0615, 15 June 1972.

K. Goldford, A. E., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Operational Performance Computer Program", McDonnell Douglas Report No. MDC E0344, 21 May 1971.

L. Herm, T. S., and Kaemming, T. A., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Conditioning Assembly Transient Computer Program", McDonnell Douglas Report No. MDC E0345, 25 June 1971.

M. Kaemming, T. A., "Space Shuttle High Pressure Auxiliary Propulsion Subsystem Definition Study - Design and Sizing Computer Program", McDonnell Douglas Report No. MDC E0343, 21 May 1971.

N. Kaemming, T. A., "Space Shuttle Auxiliary Propulsion System Design Study (NAS 9-12013) - High Pressure APS Design and Sizing Computer Program", McDonnell Douglas Report No. MDC E0451, 1 September 1971.

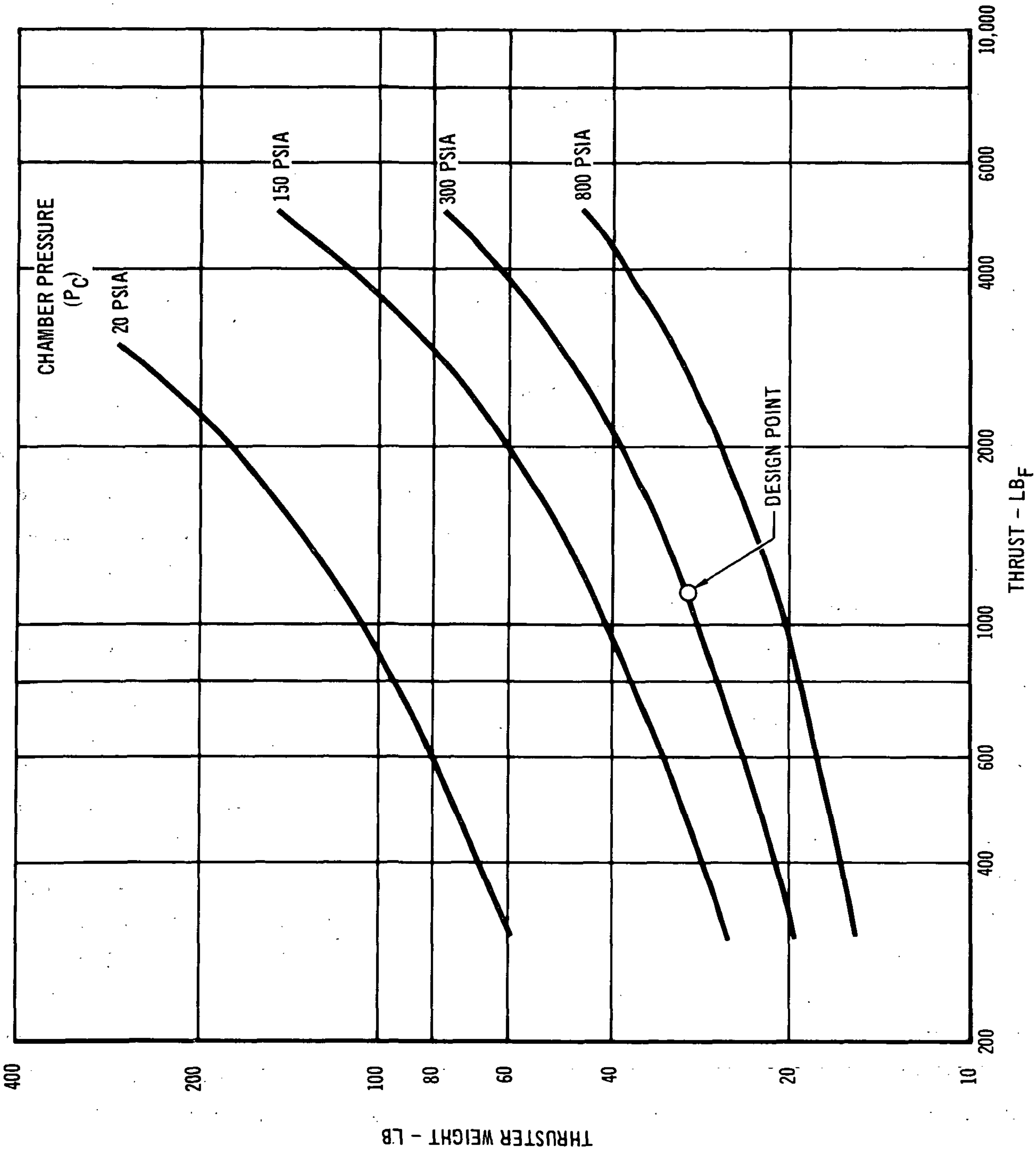
APPENDIX A
COMPONENT MODELS

All component models developed under previous studies (Contract NAS 8-26248) and defined in References C and D were reviewed to determine their suitability over the range of conditions to be investigated in this study. In particular, analytical models for the thruster, gas generator and turbopump were reviewed by Aerojet Liquid Rocket Company (ALRC) and, where required, were updated to reflect more recent data from concurrent NASA component technology programs. MDAC-E effort was devoted to a review of propellant tankage, heat exchanger, turbopump cooling, accumulator and vent line models. The gas generator and propellant tankage models were found to be suitable. All other component models from the above references were revised as discussed below.

A1. Thruster - A film cooled thruster concept was selected for the Phase B RCS study, replacing the regeneratively cooled concept defined in Reference D. This change was made because of the reduction in total impulse requirements from 11×10^6 to 2.3×10^6 lbf-sec which minimized the significance of the regenerative engine's performance advantage, and because of the desire to minimize development risk. Weight of the film-cooled thruster as a function of both thrust and chamber pressure is presented in Figure A-1. The performance model is based on test data obtained by ALRC under contract to NASA-Lewis. Delivered vacuum specific impulse of the film-cooled thruster is shown in Figure A-2.

A2. Turbopump - The turbopump models used for Reference D studies were based on specific designs, and provided no option for varying number of turbine and pump stages. In addition, turbopump weight equations were based only on flow rate, pressure rise and net positive suction pressure (NPSP), and weight was extremely sensitive to NPSP. In the subject study, the model was updated by Aerojet to allow evaluation of alternate numbers of turbine/pump stages using generalized efficiency curves and normalized pump equations.

Turbopump weight calculations were subdivided into turbine, power transmission and pump weights as functions of impeller diameter, specific speed, number of stages and shaft horsepower. The detailed weight equations are presented in Reference N, and results of parametric calculations employing these equations are shown in Figure A-3 (turbine weight) and Figure A-4 (pump and power transmission weights). Shaft speed, which is a primary parameter in the turbine weight model, is a function of propellant flow rate and NPSP, and is determined by applying the



FILM-COOLED THRUSTER WEIGHT

Figure A-1

FILM COOLED THRUSTER PERFORMANCE

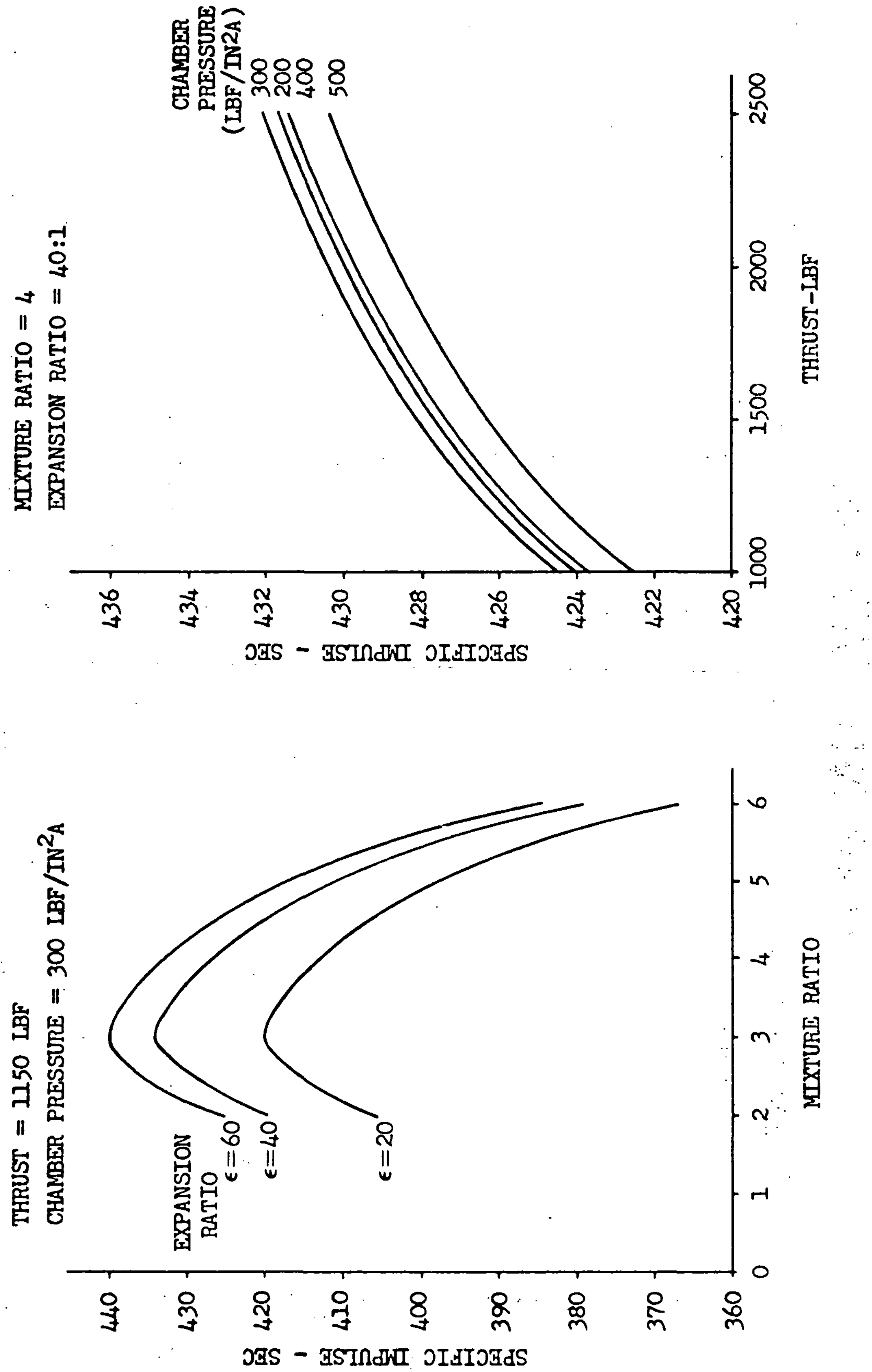


Figure A-2

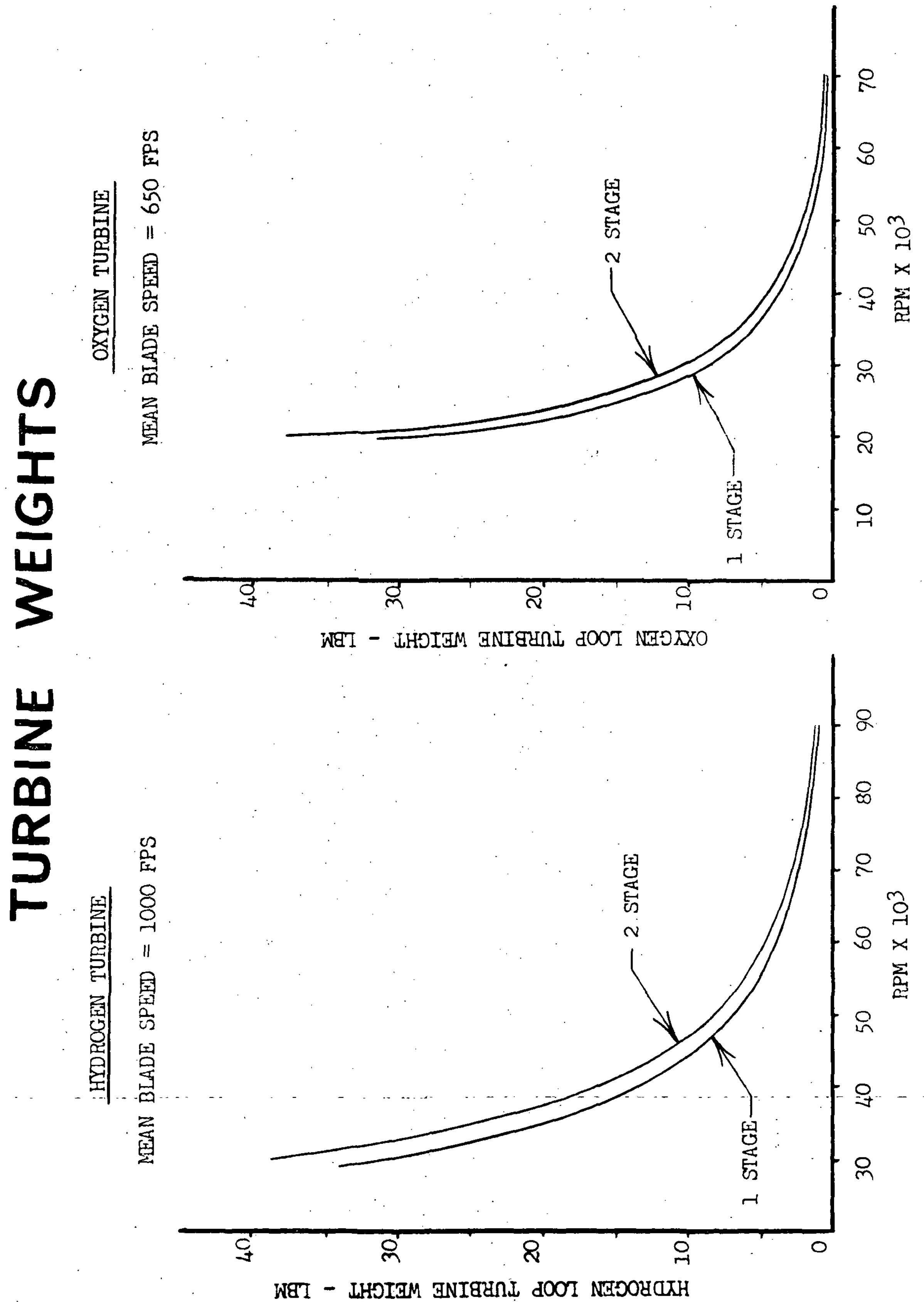

TURBINE WEIGHTS
OXYGEN TURBINE
MEAN BLADE SPEED = 650 FPS
2 STAGE
1 STAGE
OXYGEN LOOP TURBINE WEIGHT - LBM
RPM X 10^3
0
10
20
30
40
50
60
70
HYDROGEN TURBINE
MEAN BLADE SPEED = 1000 FPS
2 STAGE
1 STAGE
HYDROGEN LOOP TURBINE WEIGHT - LBM
RPM X 10^3
80
90

Figure A-3

PUMP AND POWER TRANSMISSION WEIGHTS

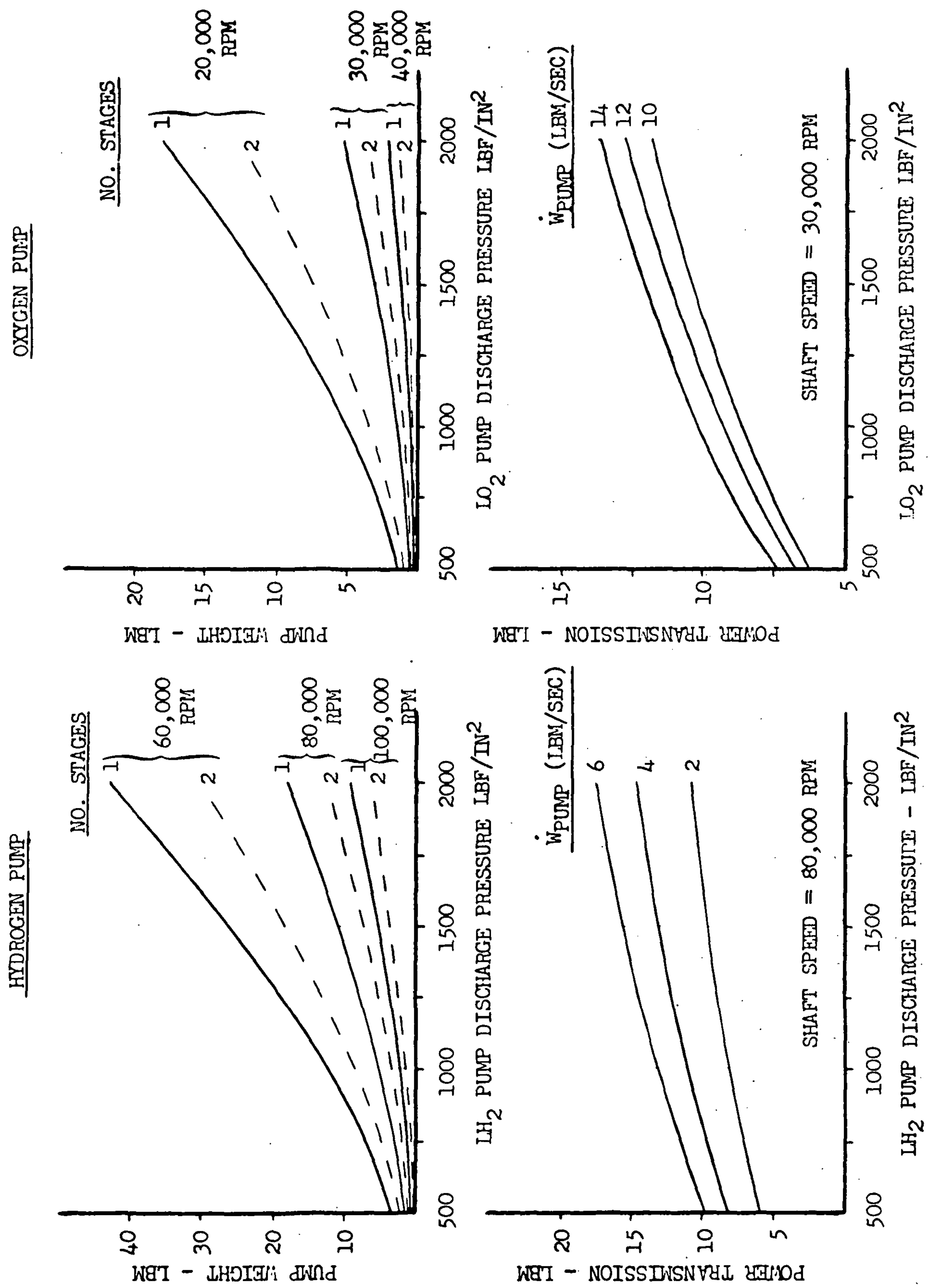


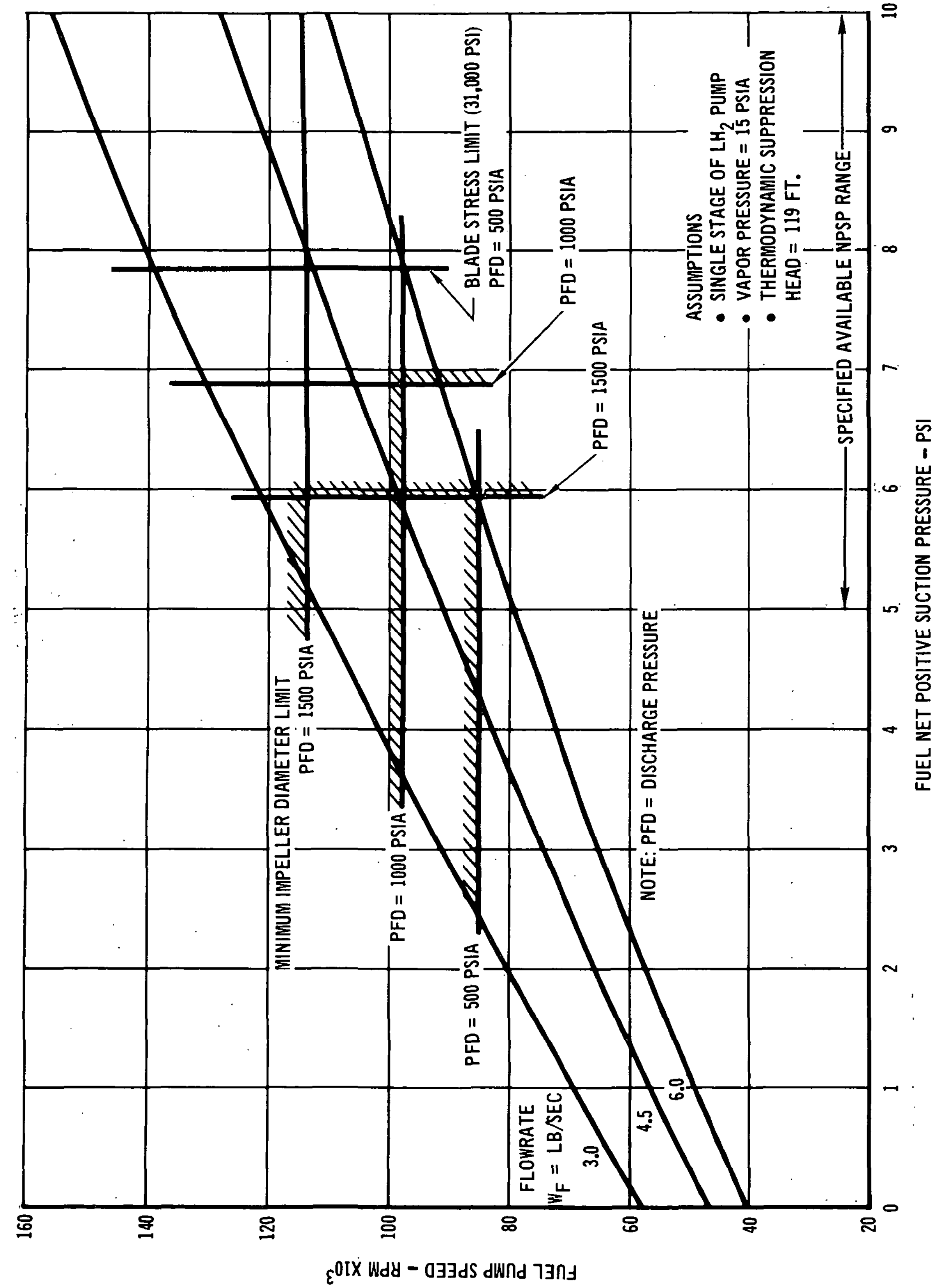
Figure A-4

curves of Figures A-5 and A-6. Pump and turbine efficiency curves are presented in Figures A-7 and A-8, respectively.

A3. Heat Exchanger - The previous heat exchanger concept defined in Reference D was a reburn, platelet configuration designed to yield maximum performance at the expense of increased technology and development risk. The concept was most attractive for applications where a gaseous O_2/H_2 Auxiliary Propulsion System is employed for all on-orbit maneuvers. For the subject study in which all X axis maneuvers equal to or greater than 20 fps are performed by a separate Orbit Maneuvering System, emphasis was redirected toward using a conventional tube and shell heat exchanger for the RCS, as maximum performance was not a dominant criterion. This concept provided acceptable weight and performance, and was considered a practical approach for low impulse (1 to 3 million lbf-sec) missions which provided limited potential for reducing RCS weight with increased specific impulse. The tube and shell heat exchanger concept considered in this study is illustrated in Figure A-9, and weight equations for the concept are defined in Reference N. Typical parametric results applying these equations are presented in Figure A-10 for varying cold side exit conditions and a fixed hot side temperature drop of 1100°R. In addition, a hot side inlet pressure of 300 $lbf/in.^2a$, and a hot side exit temperature of 800°R were assumed.

The principal operating constraints imposed on the heat exhcanger were: (1) sonic conditions shall not be achieved in the hot side flow; (2) cold side pressure drop shall be minimized; and (3) H_2O condensation and freezing shall not occur on the hot side tube walls. Based on these constraints, preliminary heat exchanger design characteristics were developed and are tabulated in Figures A-11 and A-12 for the candidate RCS concepts. Preliminary heat exchanger performance variations about the nominal design point are illustrated in Figures A-13 and A-14 for the parallel RCS. As shown, these heat exchanger designs offer little margin in operating conditions before freezing occurs. However, their weight characteristics were considered appropriate for establishment of initial system design points. Subsequent analyses of conditioner operation with worst case pressure, temperature and flow area tolerances showed that water vapor in the gas generator exhaust would condense and freeze on the heat exhanger hot side tube walls. This was evident for the series-upstream turbine RCS (H_2 side, only) and the parallel RCS (both H_2 and O_2 sides). Therefore, the heat exchangers were reconfigured for final RCS designs to preclude this problem. The revised design characteristics and operating performance maps are presented in Appendix D.

HYDROGEN TURBOPUMP DESIGN SPEED LIMITATION



OXYGEN TURBOPUMP DESIGN SPEED LIMITATION

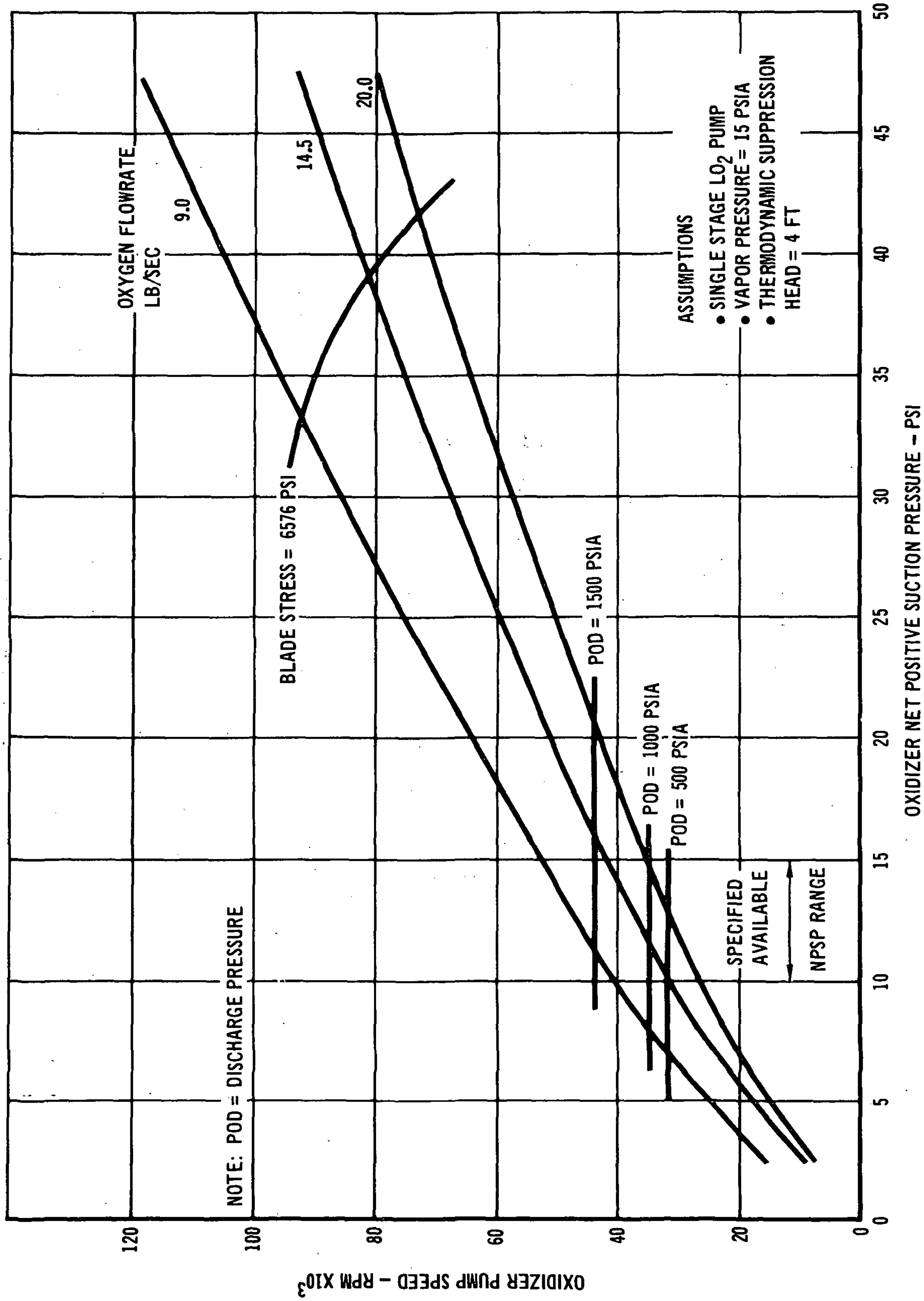


Figure A-6

PUMP EFFICIENCY VS SPECIFIC SPEED

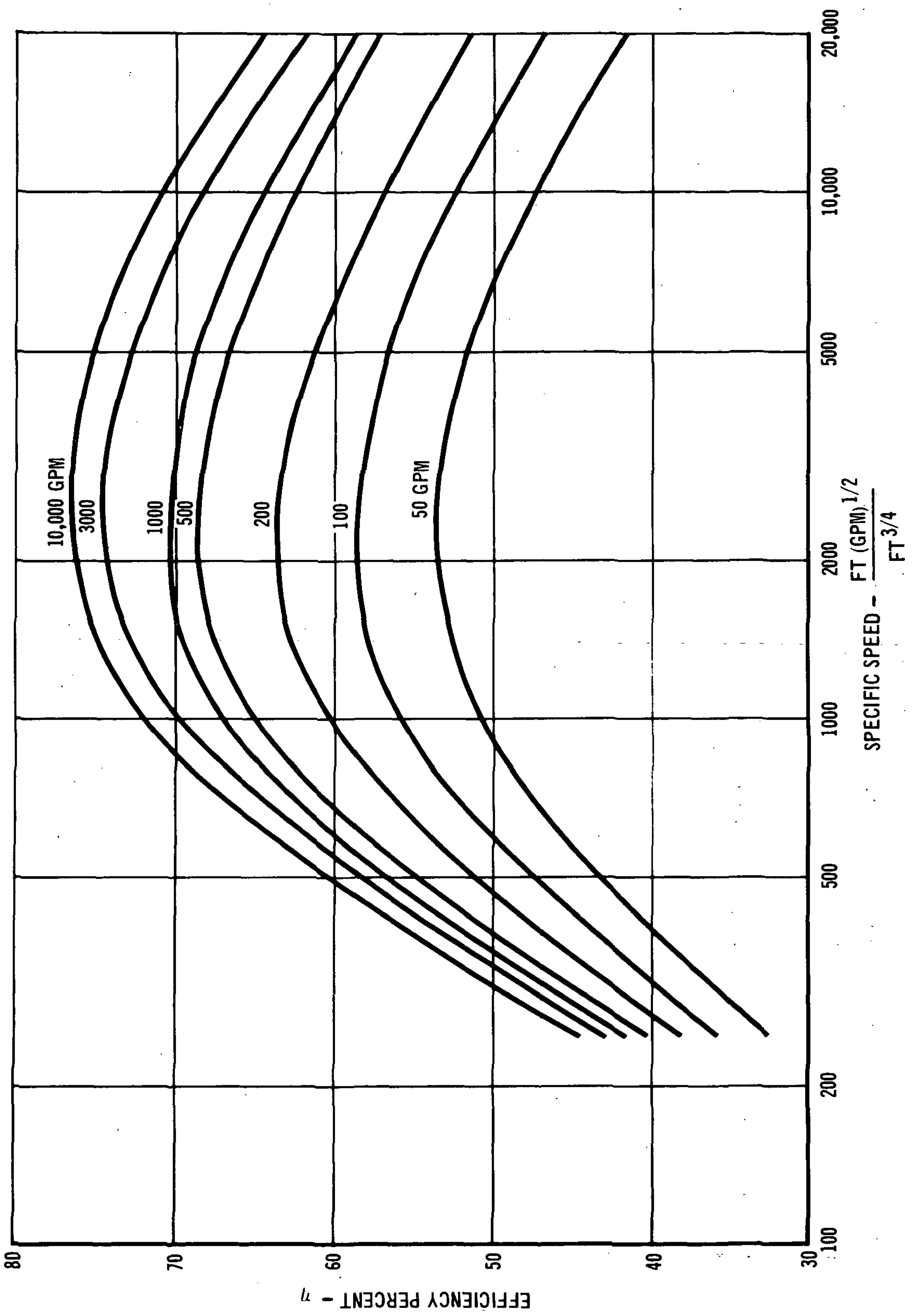


Figure A-7

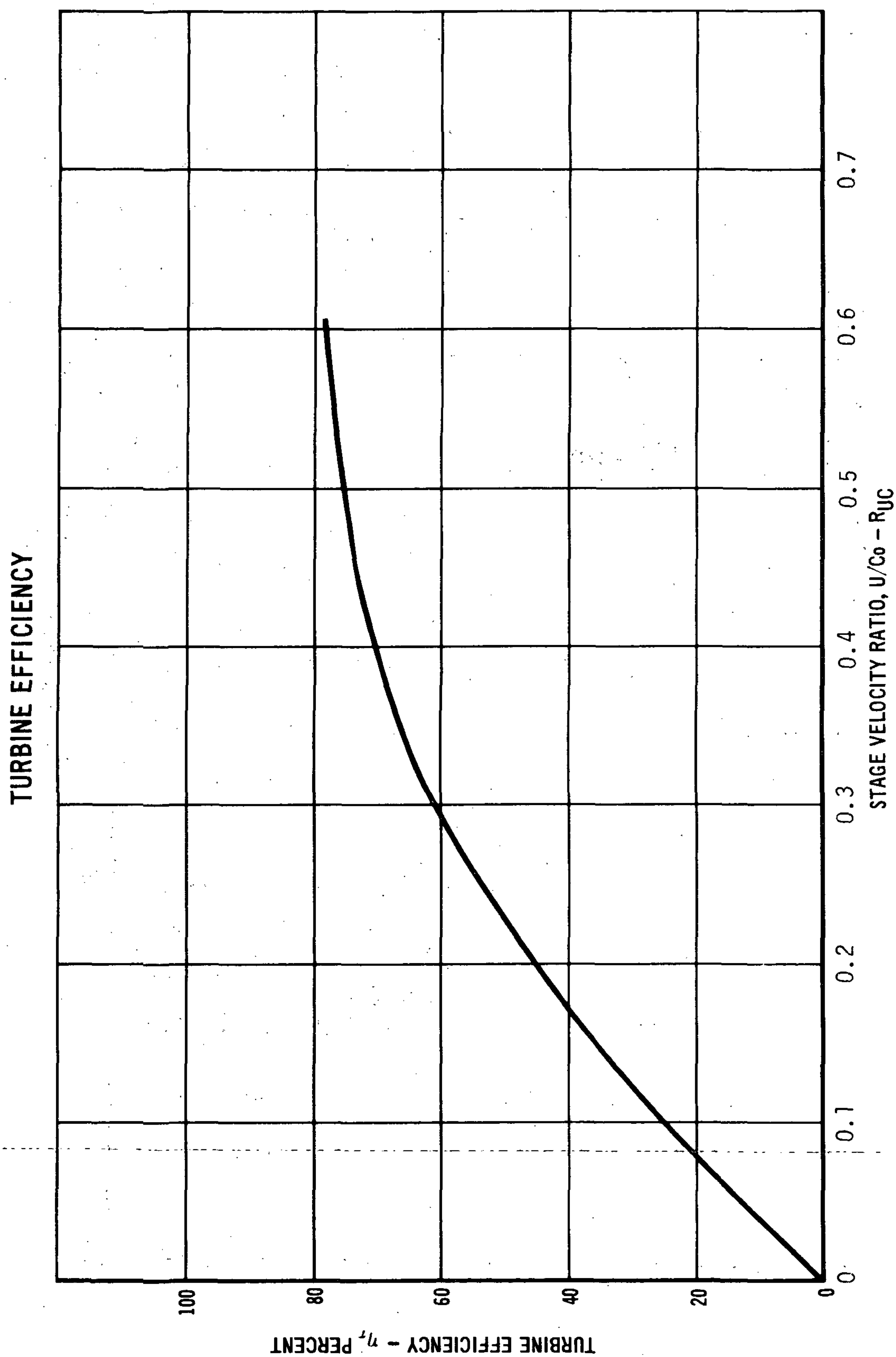
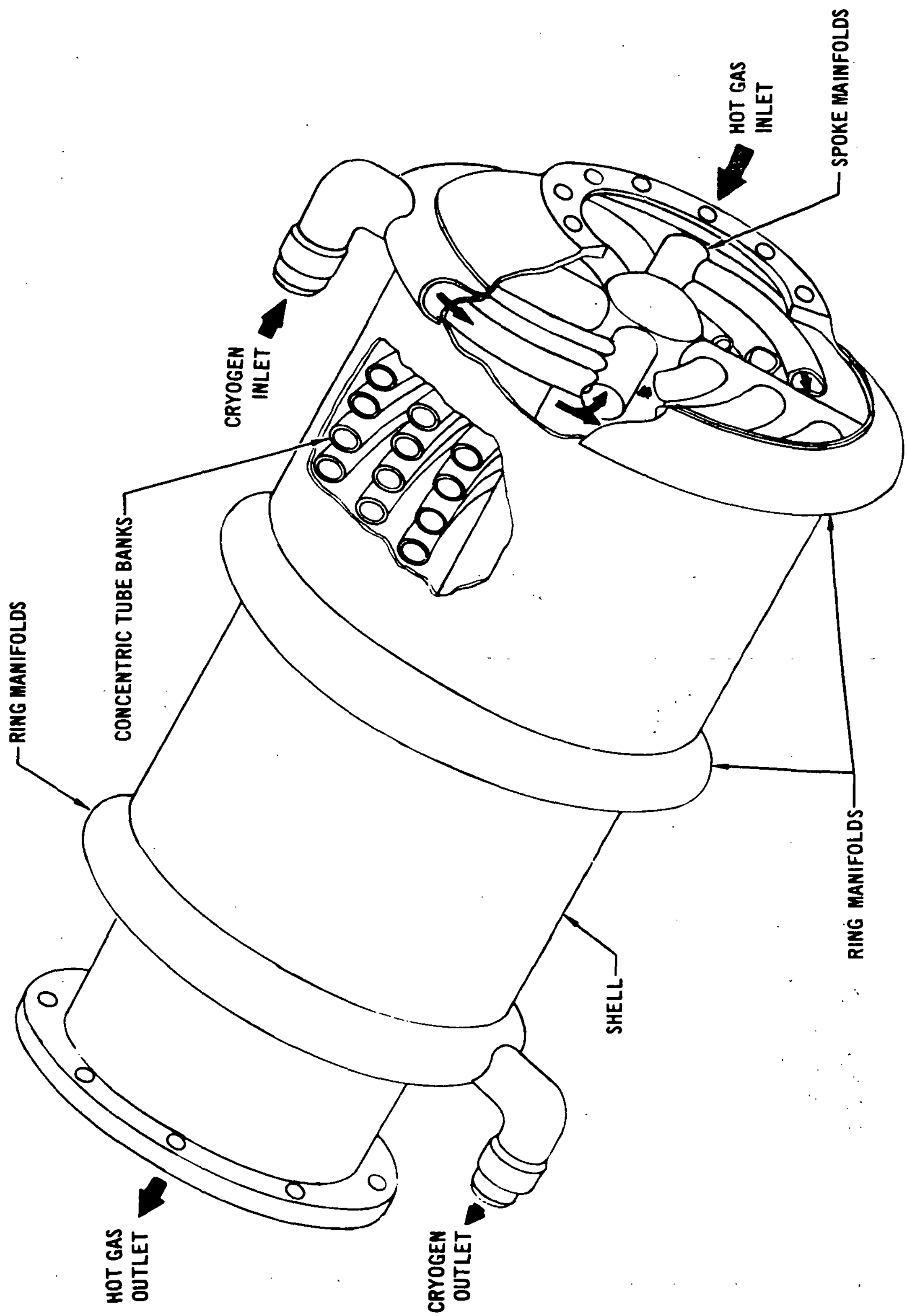

TURBINE EFFICIENCY
STAGE VELOCITY RATIO, U/Co – R_{UC}
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
TURBINE EFFICIENCY – η_T, PERCENT
0
20
40
60
80
100

Figure A-8

CONCENTRIC TUBE AND SHELL HEAT EXCHANGER MODEL



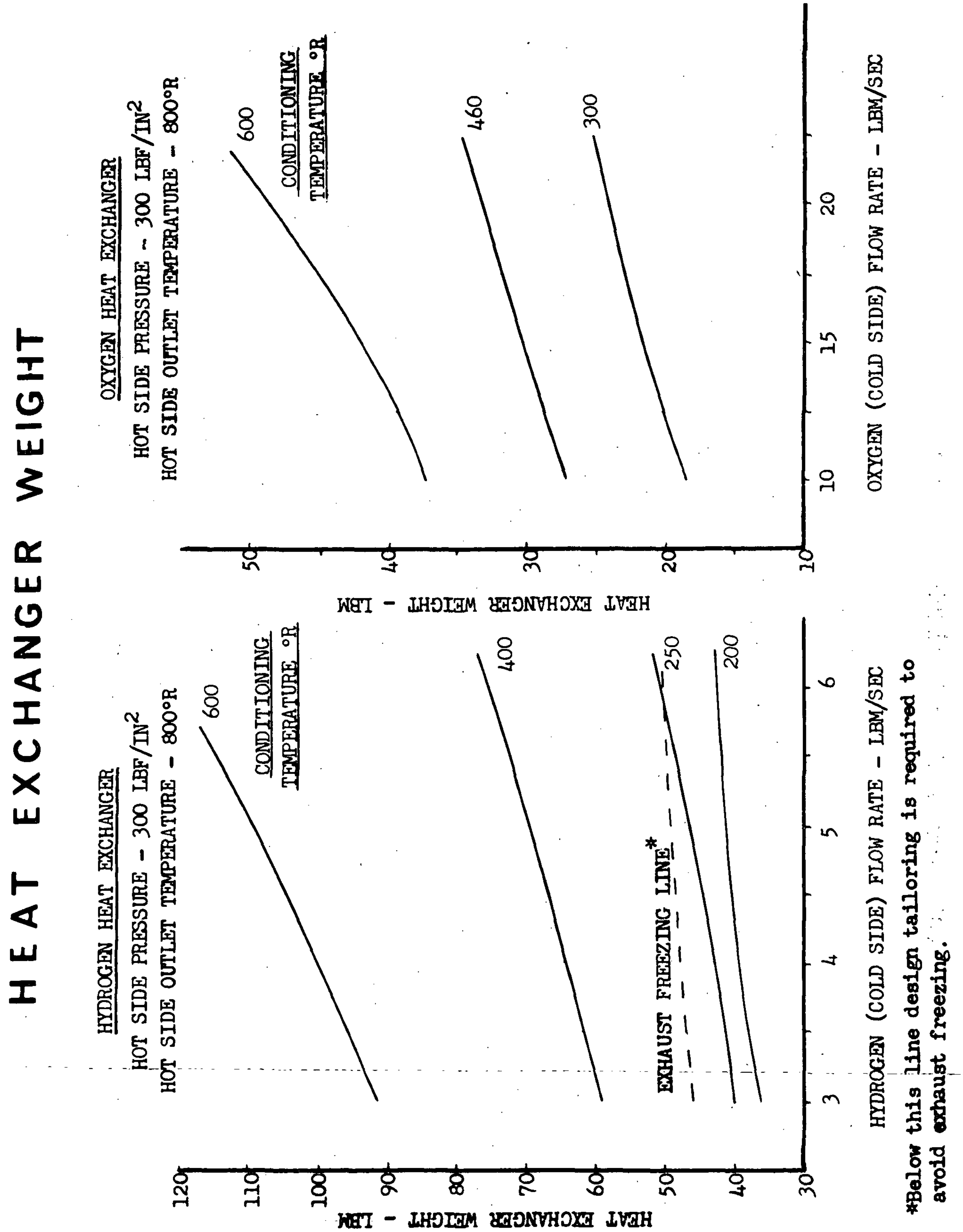

HEAT EXCHANGER WEIGHT
HYDROGEN HEAT EXCHANGER
HOT SIDE PRESSURE - 300 LBF/IN²
HOT SIDE OUTLET TEMPERATURE - 800°R
CONDITIONING TEMPERATURE °R
600
400
250
200
EXHAUST FREEZING LINE*
HEAT EXCHANGER WEIGHT - LBM
120
110
100
90
80
70
60
50
40
30
3
4
5
6
HYDROGEN (COLD SIDE) FLOW RATE - LBM/SEC
*Below this line design tailoring is required to avoid exhaust freezing.
OXYGEN HEAT EXCHANGER
HOT SIDE PRESSURE - 300 LBF/IN²
HOT SIDE OUTLET TEMPERATURE - 800°R
CONDITIONING TEMPERATURE °R
600
460
300
HEAT EXCHANGER WEIGHT - LBM
50
40
30
20
10
10
15
20
OXYGEN (COLD SIDE) FLOW RATE - LBM/SEC

Figure A-10

HYDROGEN HEAT EXCHANGER DESIGN

	Series Turbine Upstream	Series Turbine Downstream	Parallel Flow GGA'S
Design Parameter			
LH_2 Inlet Pressure (psia)/Temp. (°R)	1348/64.5	1118/64.5	1108/64.4
Hot Gas Inlet Pressure/Temp.	111/1833	300/2000	300/2000
$\dot{w}$ (lbm/sec) LH_2/Hot Gas	3.77/1.42	3.75/1.32	4.01/1.27
Configuration (Upstream/Downstream)			
No. Concentric Rings	5/5	5/4	5/4
No. Spokes	12	12	12
Tube O.D. (in.)	.3125/.4325	.3125/.4325	.3125/.4325
Tube Wall Thickness (in.)	.016/.036	.016/.036	.016/.036
Radial Gap (in.)	.100	.150	.100
Calculated Parameter			
Length (in.)	19.7	16.7	19.4
Wt (lbs)	31.8	34.2	33.8
GH_2 Outlet Pressure/Temp.	1341/252.6	1112/249.6	1101/246.4
Hot Gas Outlet Pressure/Temp.	64.4/808	296/992	289/825

Figure A-11

OXYGEN HEAT EXCHANGER DESIGN

	Series Turbine Upstream	Series Turbine Downstream	Parallel Flow GGA'S
Design Parameter			
LO_2 Inlet Pressure (psia)/Temp. (°R)	1597/179	1578/177	1378/178
Hot Gas Inlet Pressure/Temp.	169/1925	300/2000	300/2000
$\dot{w}$ (lbm/sec) LOX/Hot Gas	11.73/0.806	11.71/.804	11.97/.775
Configuration (Upstream/Downstream)			
No. Concentric Rings	5/5	5/5	5/5
No. Spokes	12	12	12
Tube O.D. (in.)	.250/.250	.250/.250	.250/.250
Tube Wall Thickness (in.)	.016/.016	.016/.016	.016/.016
Radial Gap (in.)	.150	.150	.150
Calculated Parameter			
Length (in.)	21.3	17.4	19.5
Wt (lbs)	24.2	22.7	24.3
GO_2 Outlet Pressure/Temp.	1567/497	1552/478	1345/471
Hot Gas Outlet Pressure/Temp.	166/781	298/898	298/831

Figure A-12

HYDROGEN HEAT EXCHANGER MAP

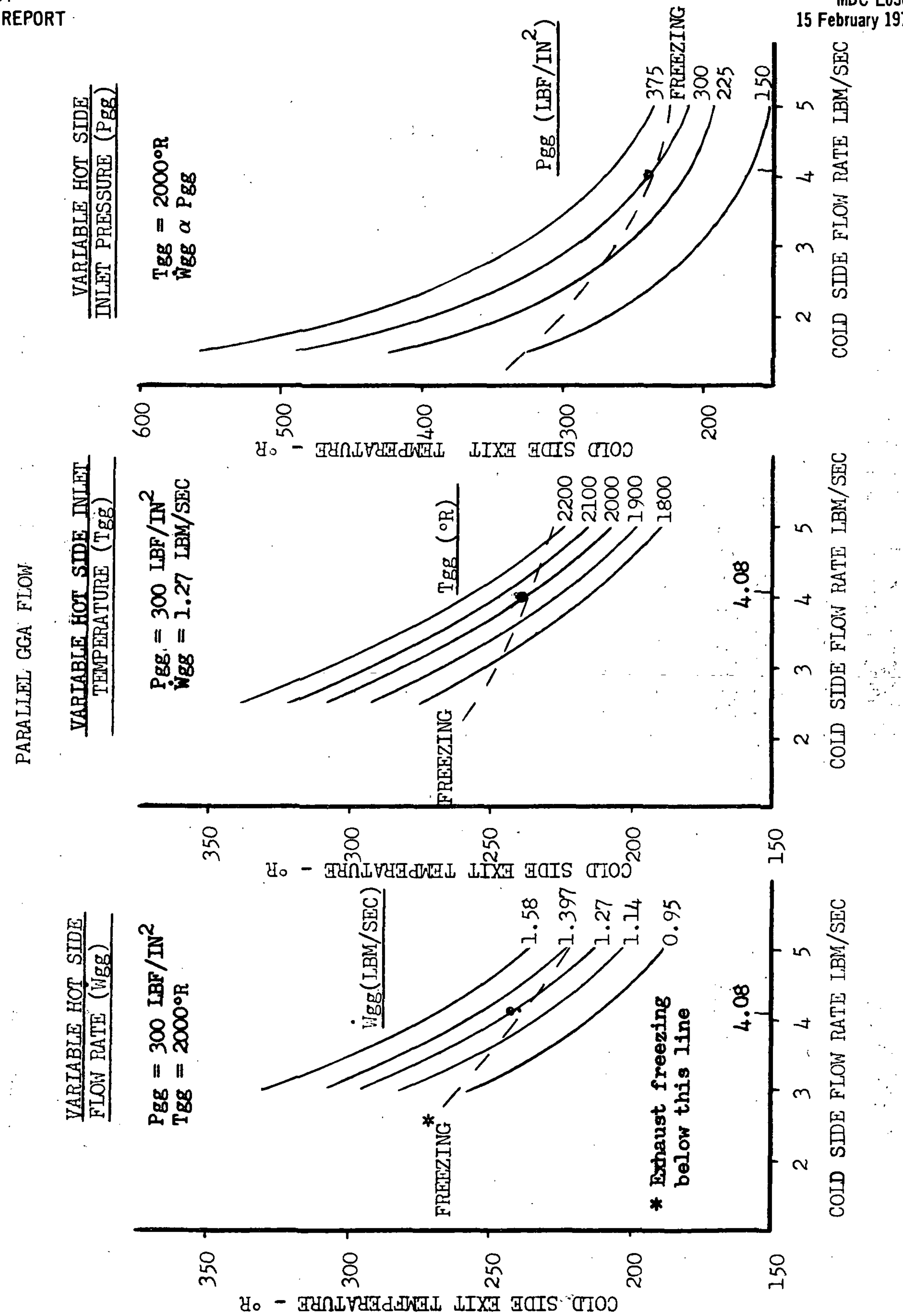


Figure A-13

OXYGEN HEAT EXCHANGER MAP

PARALLEL GGA FLOW

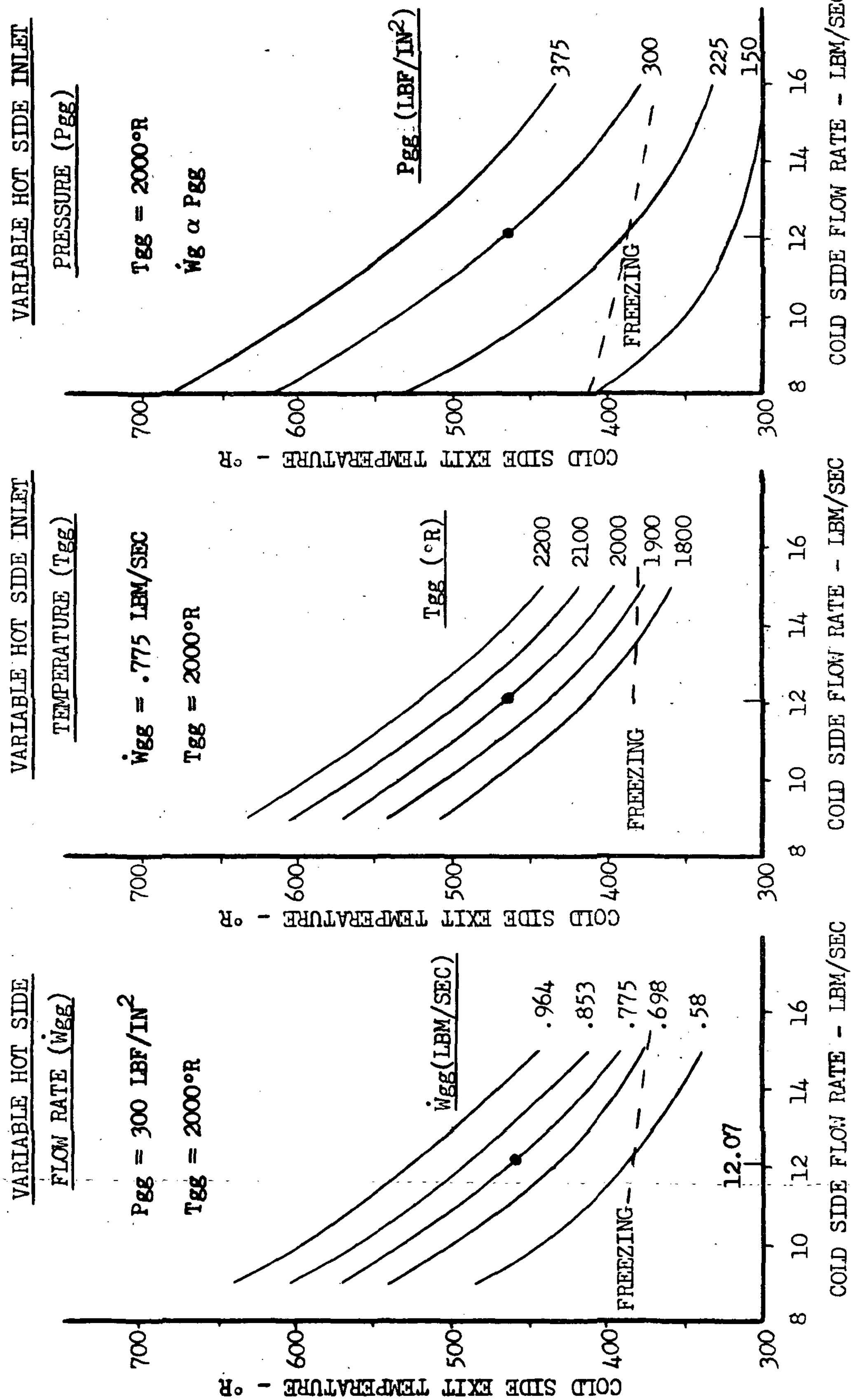


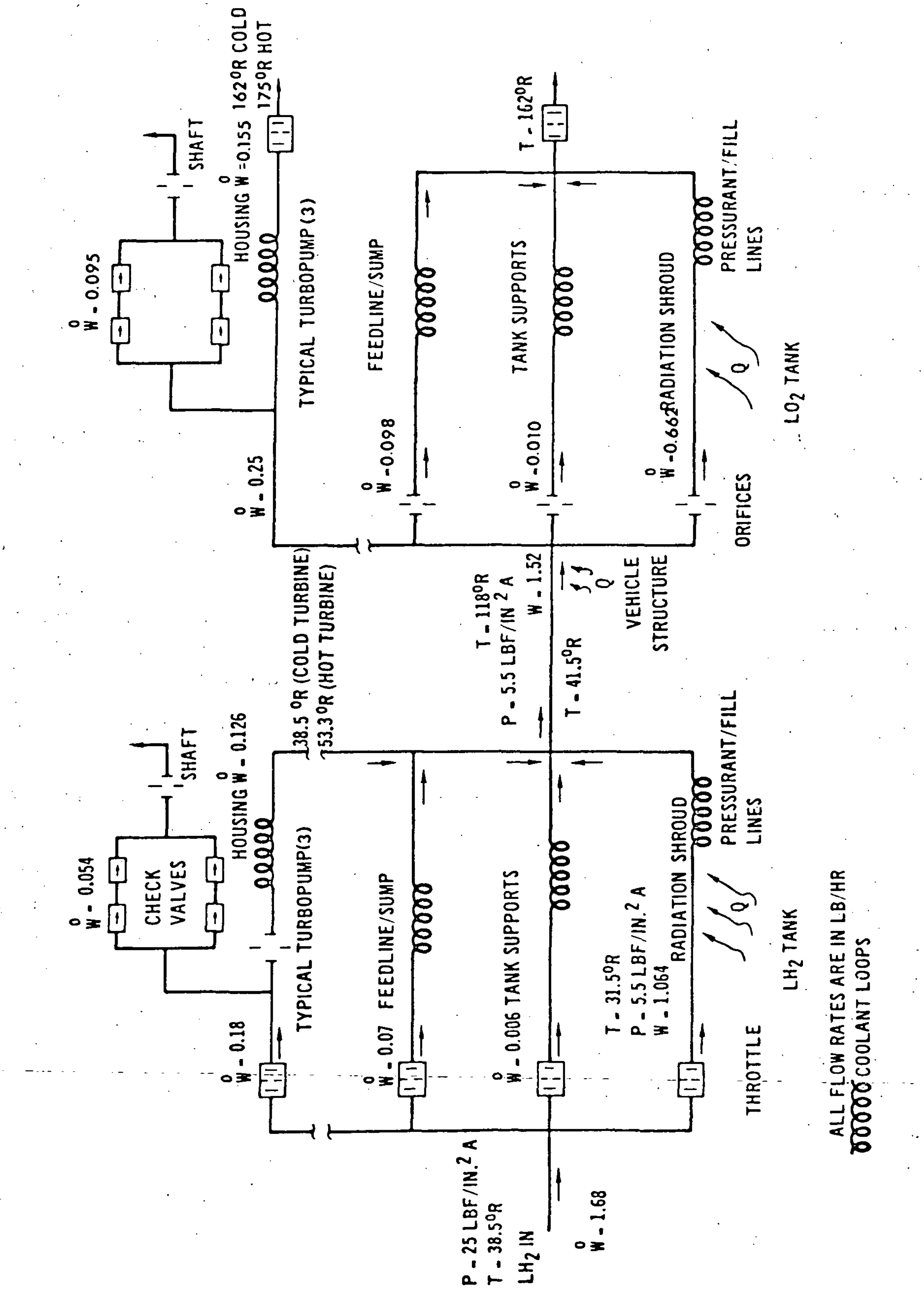
Figure A-14

A4. Turbopump Cooling - The thermodynamic vent system used for propellant tank and turbopump cooling in Reference D studies is shown schematically in Figure A-15. In this system,a small quantity of LH_2 is expanded through a throttle valve to provide coolant at a temperature below the LH_2 storage temperature. The coolant is then circulated successively around the propellant tanks, support structure and turbopumps, and finally vented overboard. The difficulty with this concept is that the turbopump cooling coils are wrapped around the shaft housing, providing only indirect cooling of pump impeller surfaces. To provide direct cooling, a pumped recirculation system was evaluated. In this concept, motor driven pumps in the propellant tank sump circulate propellant through the turbopumps and back to the tank as shown in Figure A-16. The additional heat added to the tanked propellant is removed by a thermodynamic vent system retained for cooling the propellant tank and its support structure. The schematic of this combined recirculation/thermodynamic vent cooling system is presented in Figure A-17. The weight of this combined system is 10.4 lbm heavier than the previous thermodynamic vent system (Figure A-18). However, since it provides direct cooling of the pump impeller surfaces it was selected as the baseline concept. Two recirculation pumps in each propellant tank provide fail-operational capability. Fail-safe operation is achieved by dump childown. In this emergency mode, a vent valve (shown in Figure A-16) is opened and tank pressure is used to establish pump cooling flow. It is estimated that a four minute dump chilldown requires 3 lbm of LH_2 and 10 lbm of LO_2.

A5. Accumulators - System weight sensitivity to accumulator design variables is illustrated by the examples of Figure A-19. Two examples are shown: the first assumes a conditioner startup response of 0.5 sec., while the second assumes a startup response of 1.0 sec. As shown by these examples, only a very limited system weight benefit can be achieved by designing the accumulators for more than 50 cycles.

As defined in Section 3 of this report, the design philosophy followed in this study was to restrict the number of conditioner start cycles to 50 per mission. Based on a required life of 100 missions before major system overhaul or refurbishment, this represents a useful conditioner life requirement of 5000 cycles. This was accomplished by assigning 40 of the accumulator cycles to satisfy mission multi-axis attitude control impulse requirements. An additional 10 cycles were alotted for vernier translation burns (less than 20 fps). As shown in Figure A-19

VENT SUBASSEMBLY SCHEMATIC



SERIES/TURBINE UPSTREAM RCS CONDITIONER SCHEMATIC

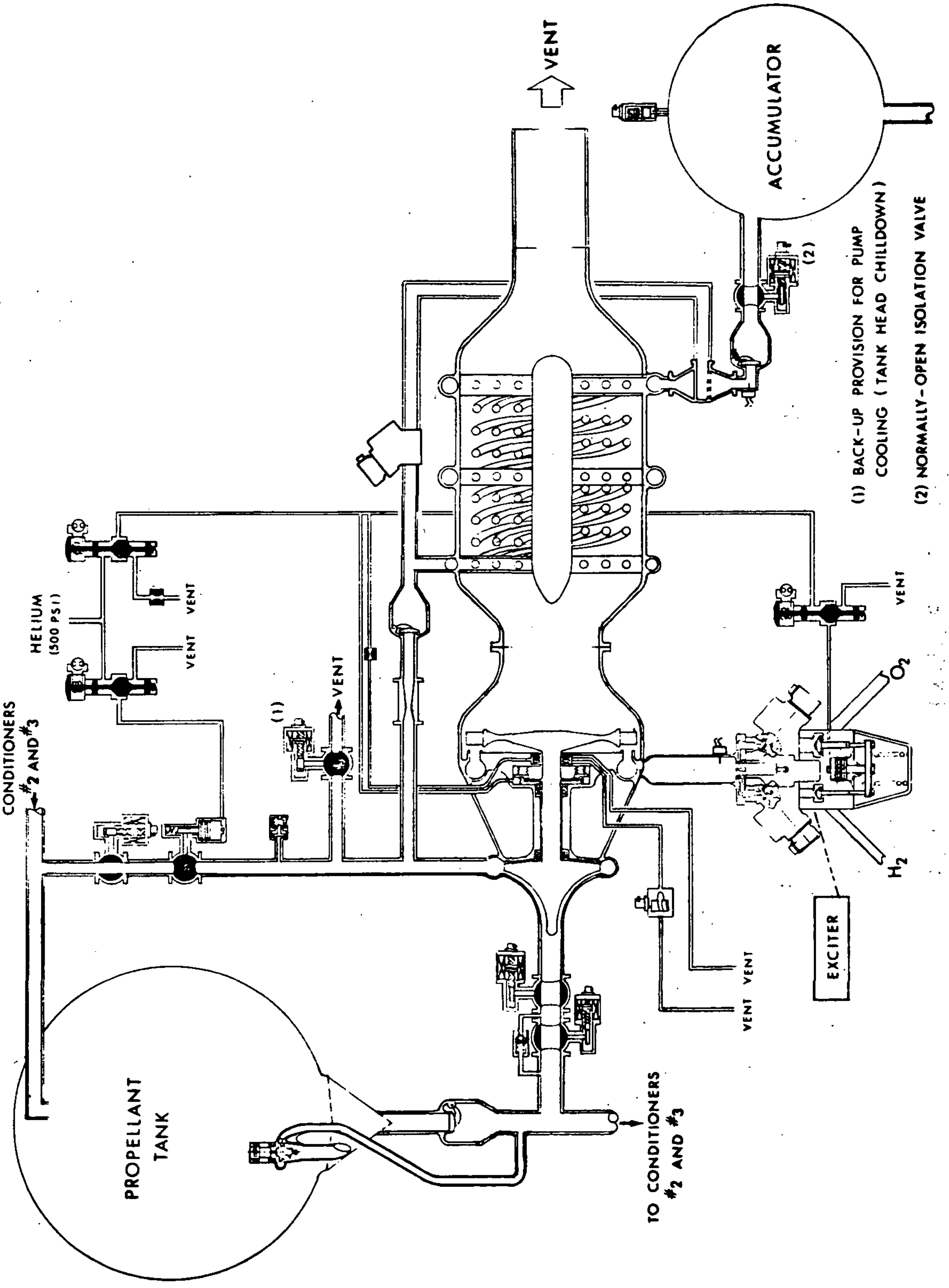


Figure A-16

RECIRCULATION SYSTEM WITH DUMP VENT

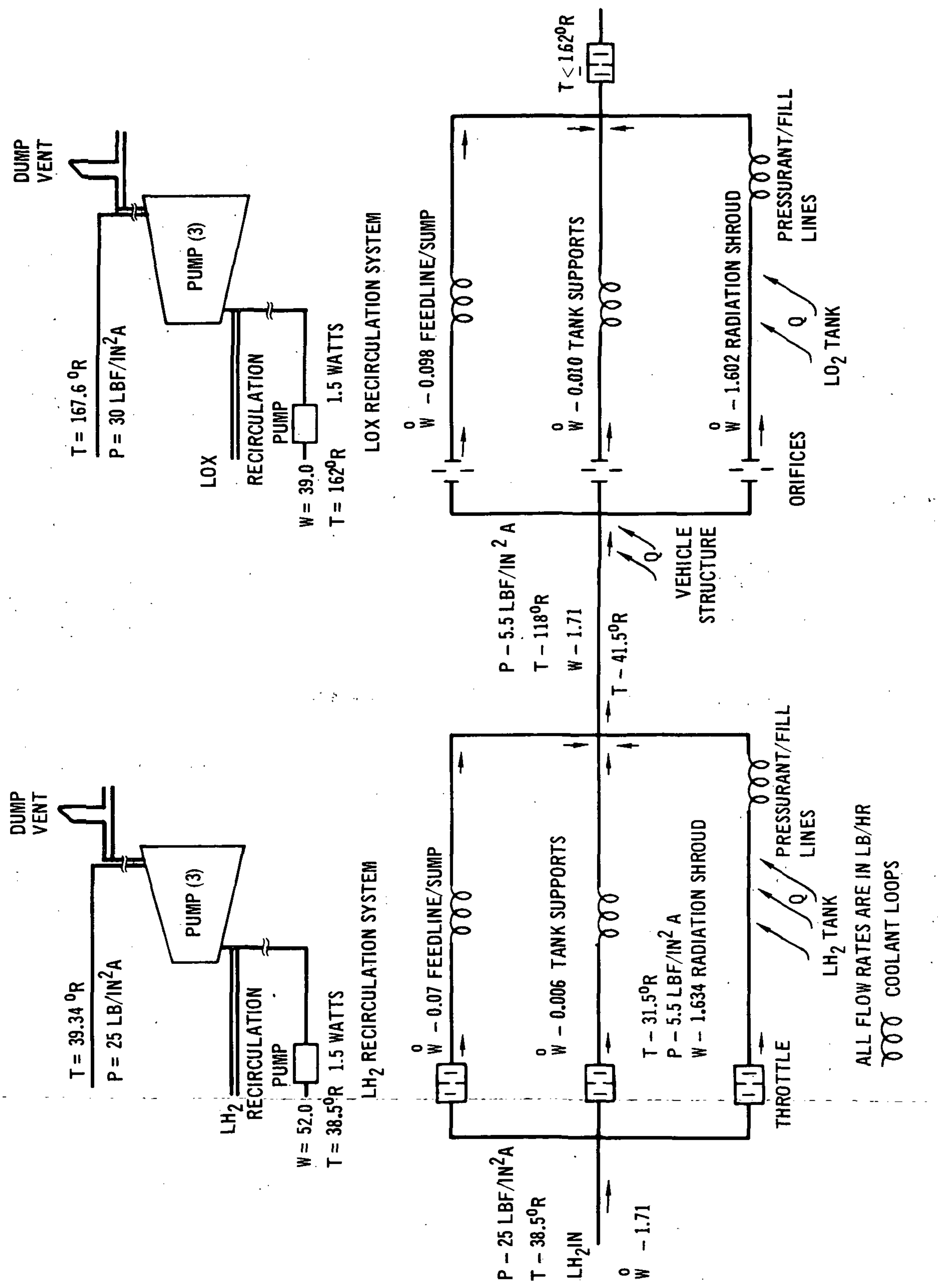


Figure A-17

RECIRCULATION WEIGHT PENALTY

- 164 Hour Mission
- Redundant Recirculation Pumps

COMPONENT	WEIGHT	REFERENCE
Motor (Hydrogen; stator and rotor only)	0.2 lbm	
Motor (Oxygen; sep. rotor incl. housing)	0.3 lbm	
Pump (Hydrogen)	0.3 lbm	Pesco Products Bulletin 0-5, 28 July 1967
Pump (Oxygen)	0.3 lbm	Pesco Products Bulletin 0-5, 28 July 1967
Case (Oxygen)	0.1 lbm	Pesco Products Bulletin 0-5, 28 July 1967
Case (Hydrogen)	0.1 lbm	Pesco Products Bulletin 0-5, 28 July 1967
Total circulator pump/motor wt.	1.3 x 2 = 2.6 lbm	
Associated Fuel Cell		
Power @ 286 lbm/kw	0.858 lbm	MDC Memo No. SSPO-E454-593, "Incremental Weight Penalty of Electrical Power and Energy", 16 Sept. 1971.
Energy @ 1.98 lbm/kwh	0.975 lbm	MDC Memo No. SSPO-E454-593, "Incremental Weight Penalty of Electrical Power and Energy", 16 Sept. 1971.
Increased tank coolant (0.03 lbm/hr x 164 hr)	4.92 lbm	MDC Memo No. SSPO-E454-593, "Incremental Weight Penalty of Electrical Power and Energy", 16 Sept. 1971.
Recirculation electronic controls	1 lbm	Estimated
Total Weight Penalty*	10.353 lbm	

* Relative to thermodynamic vent cooling

Figure A-18

RCS WEIGHT SENSITIVITY TO ACCUMULATOR PRESSURE RATIOS AND CYCLES

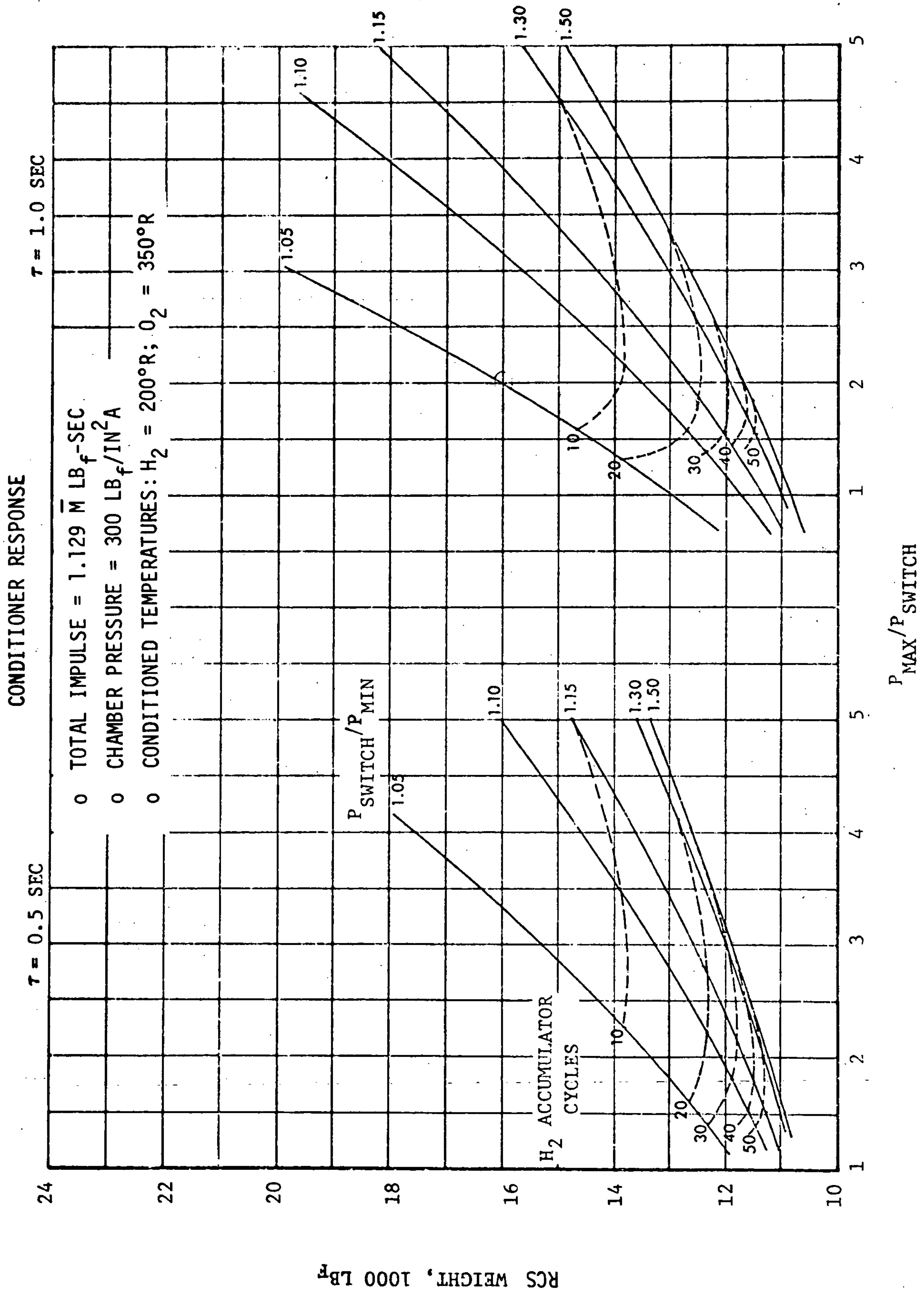


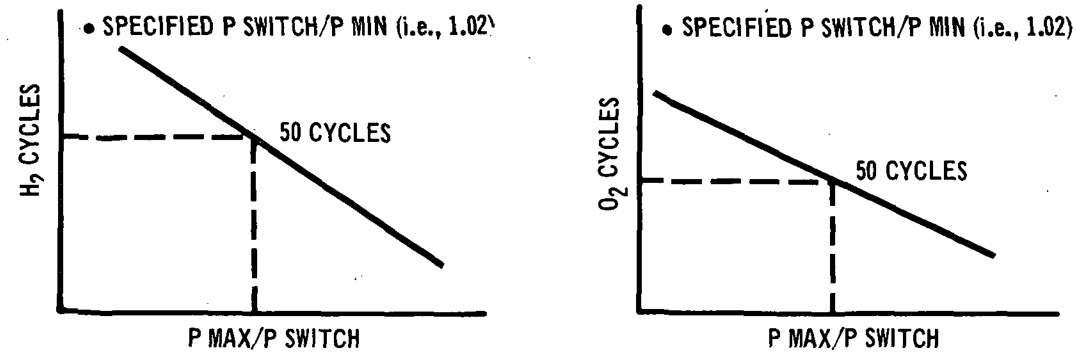
Figure A-19

for a conditioner response of 0.5 sec. and 40 accumulator cycles for attitude control, optimum system weight is achieved at accumulator pressure ratios of $P_{max}/P_{switch} \simeq 2.0$, and $P_{switch}/P_{min} \simeq 1.2$. To facilitate accumulator optimizations for varying system requirements (i.e., conditioner response time, thruster chamber pressure, attitude control total impulse, conditioned propellant temperatures, etc.), the design and sizing techniques described in Reference M, were modified by incorporating an optional accumulator optimization technique. This optimization technique is illustrated in Figure A-20. As shown in Step 1, accumulator switch/min pressure ratios are initialized at arbitrarily low values (i.e., 1.02 for both O_2 and H_2), and the required max/switch pressure ratios which satisfy the specified number of accumulator cycles are calculated iteratively. When the required maximum pressure is determined, total system weight is calculated. Then, holding O_2 switch/min pressure ratio constant (1.02) H_2 switch/min pressure is incremented upward with system weight calculated for each step. When minimum system weight is found as shown in Step 2 of Figure A-20, H_2 switch/min pressure ratio is reset at 1.02, and O_2 switch/min pressure ratio is stepped upward. The preceeding system weight optimization for varying H_2 switch/min pressure ratios is then repeated as shown in Step 3. From the resulting locus of optimums determined in Step 3, the minimum system weight (for the specified number of accumulator cycles) and optimum switch/min pressure ratios (both H_2 and O_2) are found in Step 4.. Typical results from these calculations are illustrated in the three dimensional plots of Figure A-21. Shown are accumulator pressure ratio optimizations for the series-upstream turbine RCS at conditioner response times of 0.5 and 1.0 seconds. (50 accumulator cycles were specified for this example.) As shown, if the switch/min pressure ratio is too small, system weight is excessive because of the large accumulator volumes required to satisfy system mass flow demands during conditioner startup. Also, if the switch/min pressure ratio is too large, accumulator volumes are small, but accumulator max/switch pressure ratio is large (to satisfy conditioner cycle constraints) causing excessive system weight. Comparing the accumulator design optimizations for both response times of Figure A-21, it is seen that with proper sizing, conditioner response (startup) time could increase during hardware development without significant impact. This is because minimum system weight could be maintained by a slight increase in the accumulator switch/min pressure ratio.

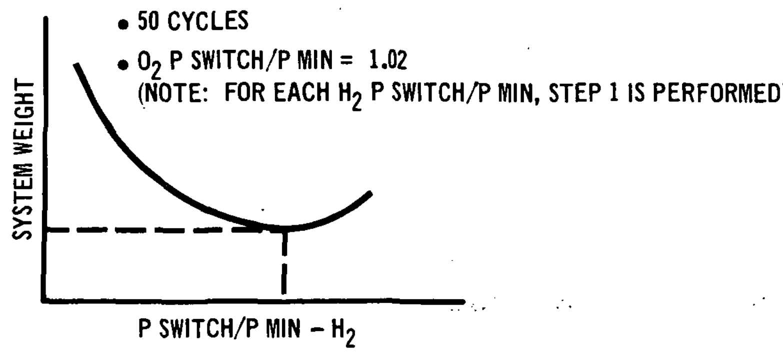
A6. Vent - Because of the large on-orbit maneuver requirements of previous studies (Contract NAS 8-26248), propulsive conditioner vents were employed to reduce

ACCUMULATOR OPTIMIZATION TECHNIQUE

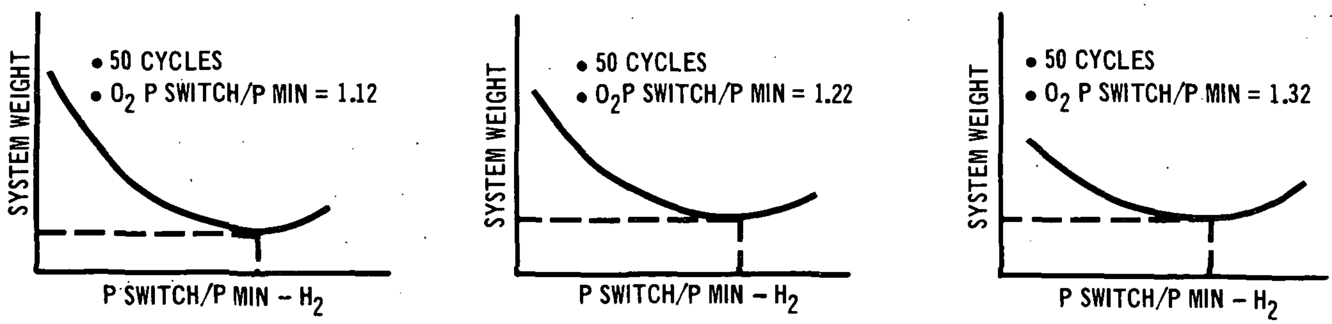
Step 1. Determine Required Max/Switch Pressure Ratio to Satisfy Cycle Constraint



Step 2. Determine Optimum H_2 Switch/Min Pressure Ratio for Specified O_2 Switch/Min Pressure Ratio



Step 3. Repeat Step 2 for Various O_2 Switch/Min Pressure Ratios



Step 4. Determine Optimum O_2 Switch/Min Pressure Ratio

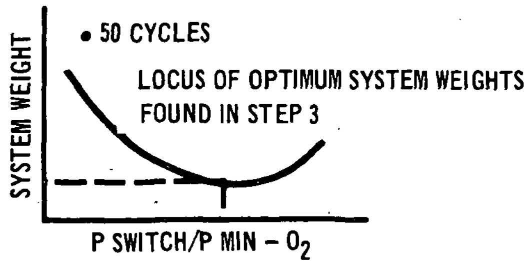


Figure A-20

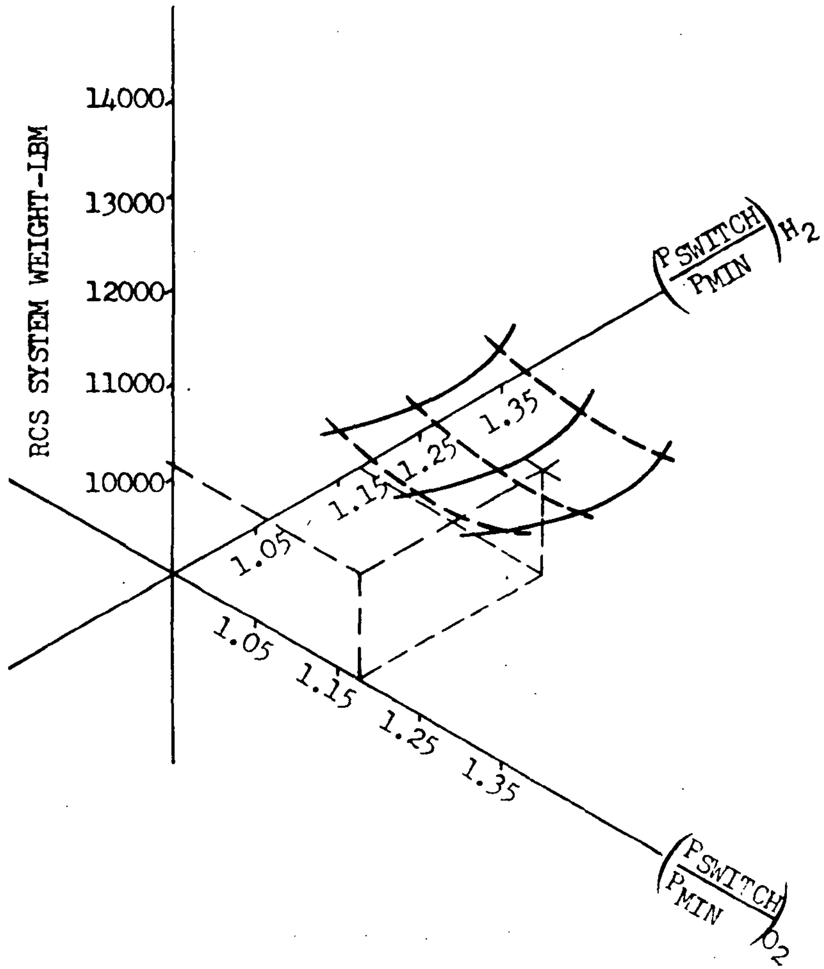
EXAMPLE ACCUMULATOR OPTIMIZATION

- o SERIES-UPSTREAM TURBINE RCS
- o 50 CYCLES

- ° PESPONSE TIME = 0.5 SEC
- ° MINIMUM WT = 10155 LBM@

$$\left(\frac{P_{SWITCH}}{P_{MIN}}\right)_{H_2} = 1.173$$

$$\left(\frac{P_{SWITCH}}{P_{MIN}}\right)_{O_2} = 1.175$$



- ° RESPONSE TIME = 1.0 SEC
- ° MINIMUM WT = 10235 LBM@

$$\left(\frac{P_{SWITCH}}{P_{MIN}}\right)_{H_2} = 1.254$$

$$\left(\frac{P_{SWITCH}}{P_{MIN}}\right)_{O_2} = 1.275$$

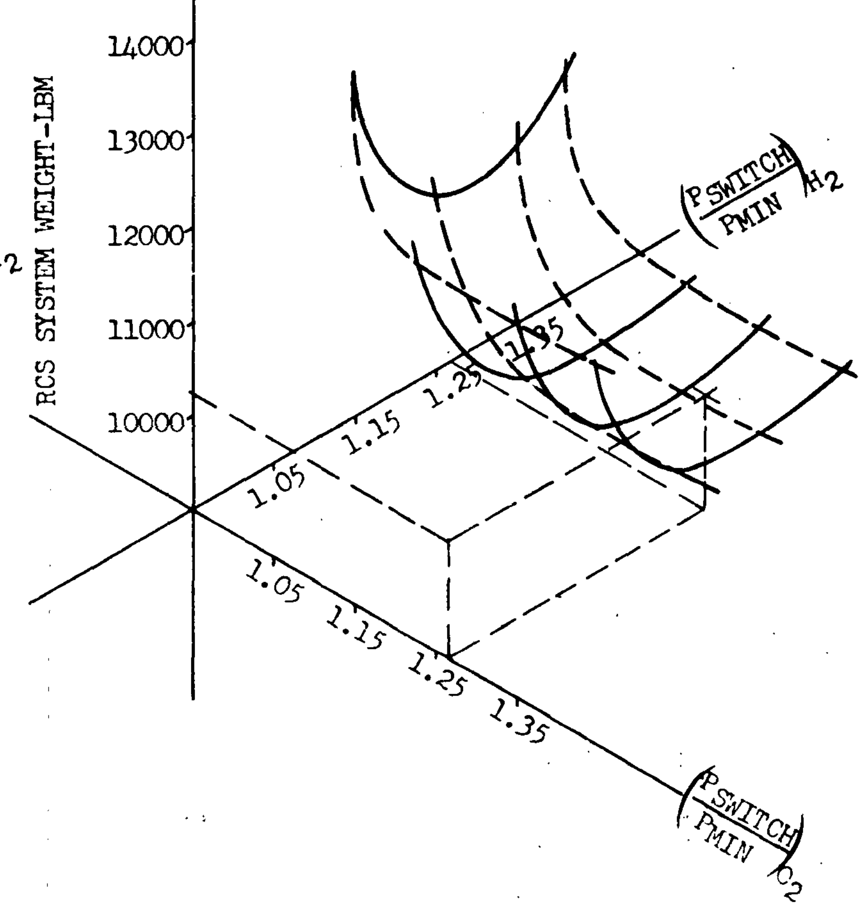


Figure A-21

orbiter RCS impulse expenditure during major maneuver burns. However, for the smaller RCS maneuver requirements of the current study, this impulse benefit was offset by the weight penalty associated with the long lines required to duct the gas generator exhaust flow to the tail of the vehicle. As a result, the weight model for the conditioner vent system was modified to simulate non-propulsive vehicle side vents. The revised model described in Reference N is based on the use of aluminum lines, stainless steel linear compensators, and stainless steel angulation joints.

APPENDIX B

PRELIMINARY SYSTEM ANALYSES

Applying the requirements of Section 3, preliminary system design points were established which formed the basis for subsequent controls screening and control concept comparisons. These design points, which are summarized in Figure B-1, were developed applying the design and sizing techniques described in References M and N, and the system model shown in Figure B-2. As shown in the schematic of Figure B-2, complete fail-operational/fail-safe component redundancy was employed, except in relation to propellant tankage, accumulators, and supply lines. The primary component models employed for the design point evaluations are defined in Appendix A.

Conditioner pressure, temperature and flow balances at the preliminary design points are presented in Figures B-3 through B-5 for the series-upstream turbine, series-downstream turbine, and parallel RCS, respectively. The system weight sensitivities to pertinent design and operating parameters are shown in Figures B-6 through B-8. As seen in these figures, the design points reflect selection of a non-optimum thruster chamber pressure of 300 lbf/in^2a. Based on the low weight penalty (approximately 150 lbm), this selection was made to afford better utilization of data from complementing component technology programs. Two approaches were used to define the weight sensitivities of Figures B-6 through B-8: (1) development of sensitivities assuming constant accumulator pressure ratios ($P_{switch}/P_{min} \simeq 1.2$ and $P_{max}/P_{switch} \simeq 2.0$) which were found to be near optimum in previous Auxiliary Propulsion System Studies (NAS 8-26248), and (2) development of sensitivities based on accumulator re-optimization for each design parameter change. The most significant difference between the two approaches is seen in the system weight sensitivity to conditioner startup response. The weight penalty associated with increased response time is appreciably reduced through accumulator optimization. The accumulator optimization technique employed in these analyses is described in Appendix A, Section A5.

PRELIMINARY RCS DESIGN POINTS

	SERIES RCS UPSTREAM TURBINE		SERIES RCS DOWNSTREAM TURBINE		PARALLEL RCS	
SYSTEM						
TOTAL IMPULSE, LBF-SEC	2.23M		2.23M		2.23M	
MIXTURE RATIO	3.11		3.15		2.95	
SPECIFIC IMPULSE, SEC	371		373		355	
THRUSTER						
THRUST LEVEL, LBF	1150		1150		1150	
MIXTURE RATIO	4.0		4.0		4.0	
CHAMBER PRESSURE, LBF/IN^2.	300		300.		300.	
EXPANSION RATIO	40:1		40:1		40:1	
SPECIFIC IMPULSE, SEC	433		433		433	
GAS GENERATOR	O_2	H_2	O_2	H_2	O_2	H_2
COMBUSTION TEMPERATURE, °R	2061	2060	2061	2058	1997	2000
FLOW RATE, LBM/SEC	.811	1.42	.821	1.33	.227 - TPA .770 - HEX	.605 - TPA 1.27 - HEX
HEAT EXCHANGER						
HOT SIDE INLET TEMP, °R	1970	1851	2061	2058	1997	2000
COLD SIDE EXIT TEMP, °R	506	263	517	254	473	246
TURBOPUMP ASSEMBLY						
FLOWRATE, LBM/SEC	11.73	3.77	11.71	3.71	12.11	4.03
DISCHARGE PRESSURE, LBF/IN^2.	1894	1545	1846	1327	1556	1201
SHAFT HORSEPOWER	133	528	129	439	150	560
TURBINE PRESSURE RATIO	2.09	2.59	2.87	7.98	20	20
ACCUMULATOR						
NO. CYCLES	50.	50	50	50.	50.	50.
VOLUME, FT^3	13.6	41.5	13.6	43.2	14.7	47.7
PRESSURE, LBF/IN^2 - MAX	1550	1300	1551	1265	1448	1195
SWITCH	666	665	666	660	660	659
MIN	571	571	571	571	571	571
WEIGHT	10,168		10,184		10,763	

* BASELINE FOR STUDY

Figure B-1

TYPICAL RCS SCHEMATIC
HYDROGEN SIDE OF BIPROPELLANT SYSTEM

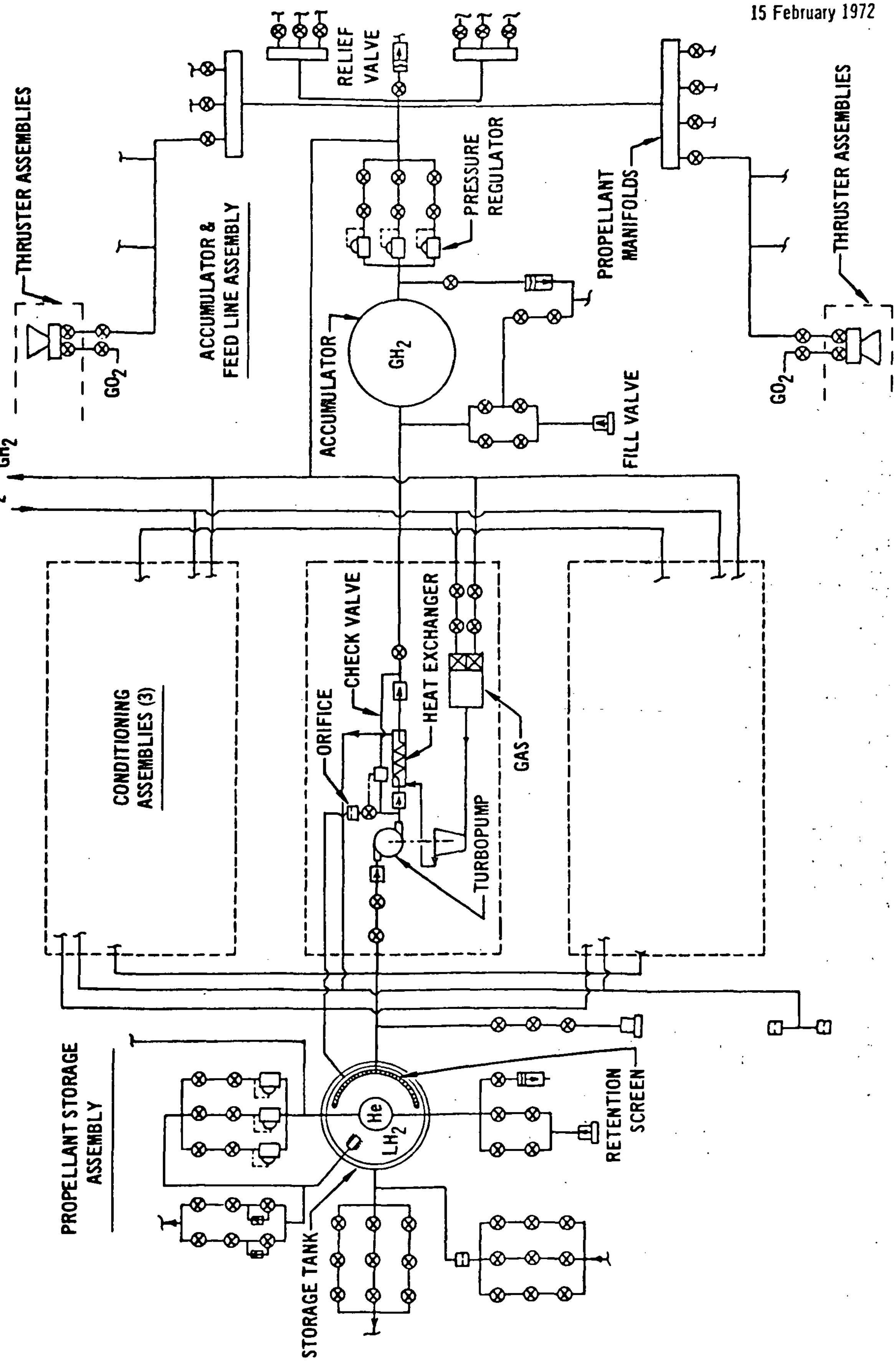


Figure B-2

CONDITIONER PRESSURE, TEMPERATURE, AND FLOW BALANCE

SERIES GGA FLOW (TURBINE UPSTREAM)

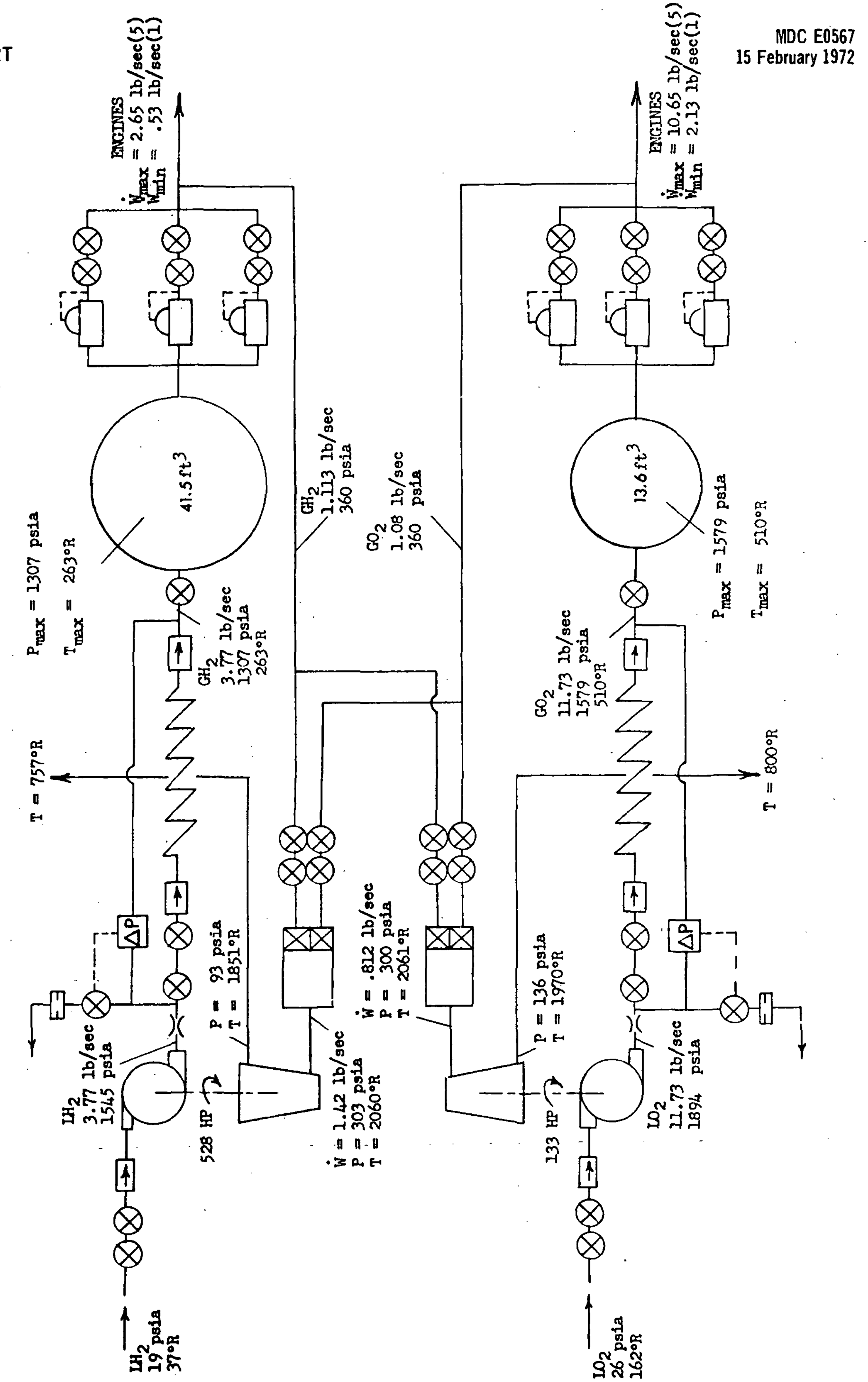


Figure B-3

CONDITIONER PRESSURE, TEMPERATURE, AND FLOW BALANCE

SERIES GGA FLOW (TURBINE DOWNSTREAM)

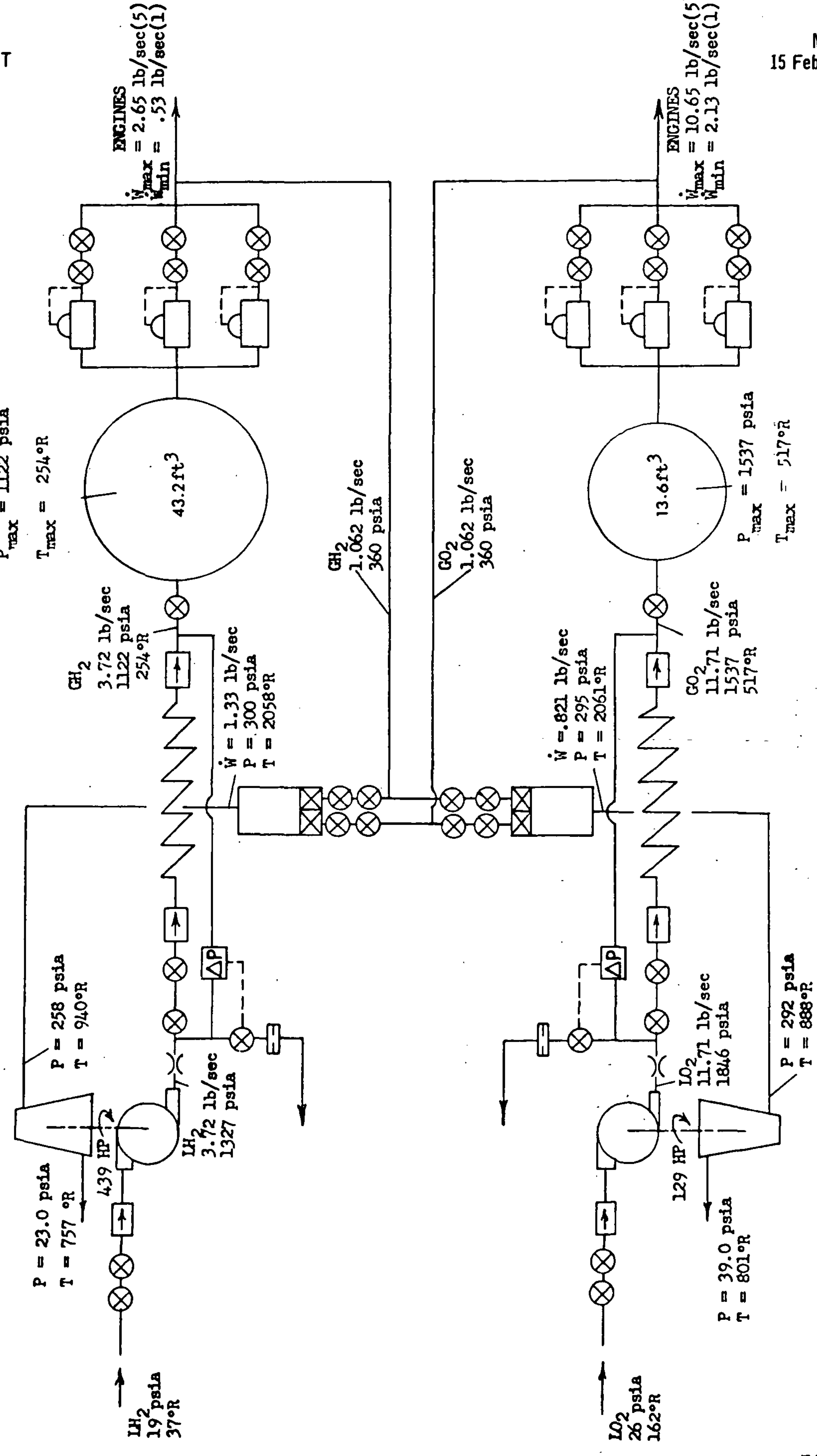


Figure B-4

CONDITIONER PRESSURE, TEMPERATURE, AND FLOW BALANCE

PARALLEL GGA FLOW

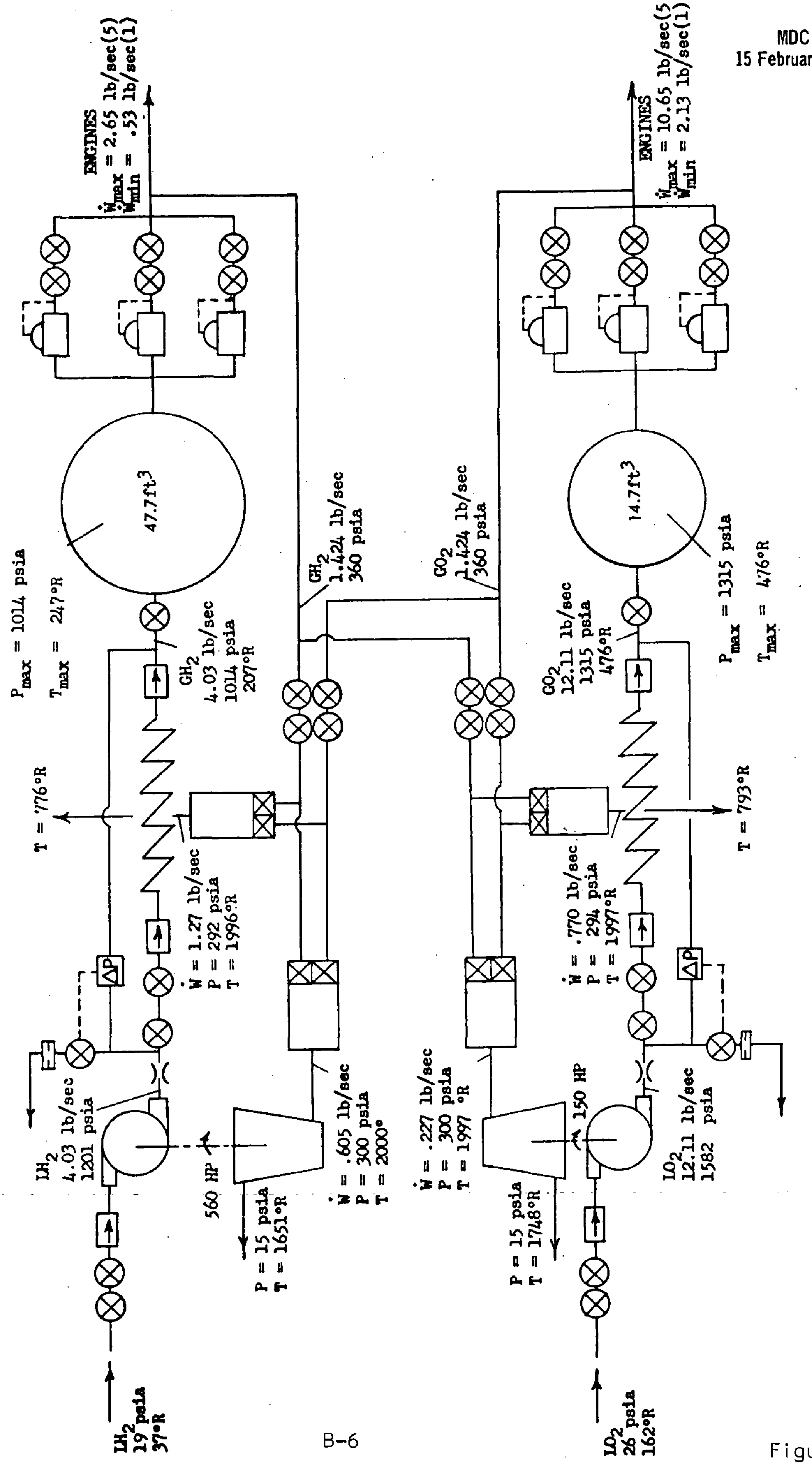


Figure B-5

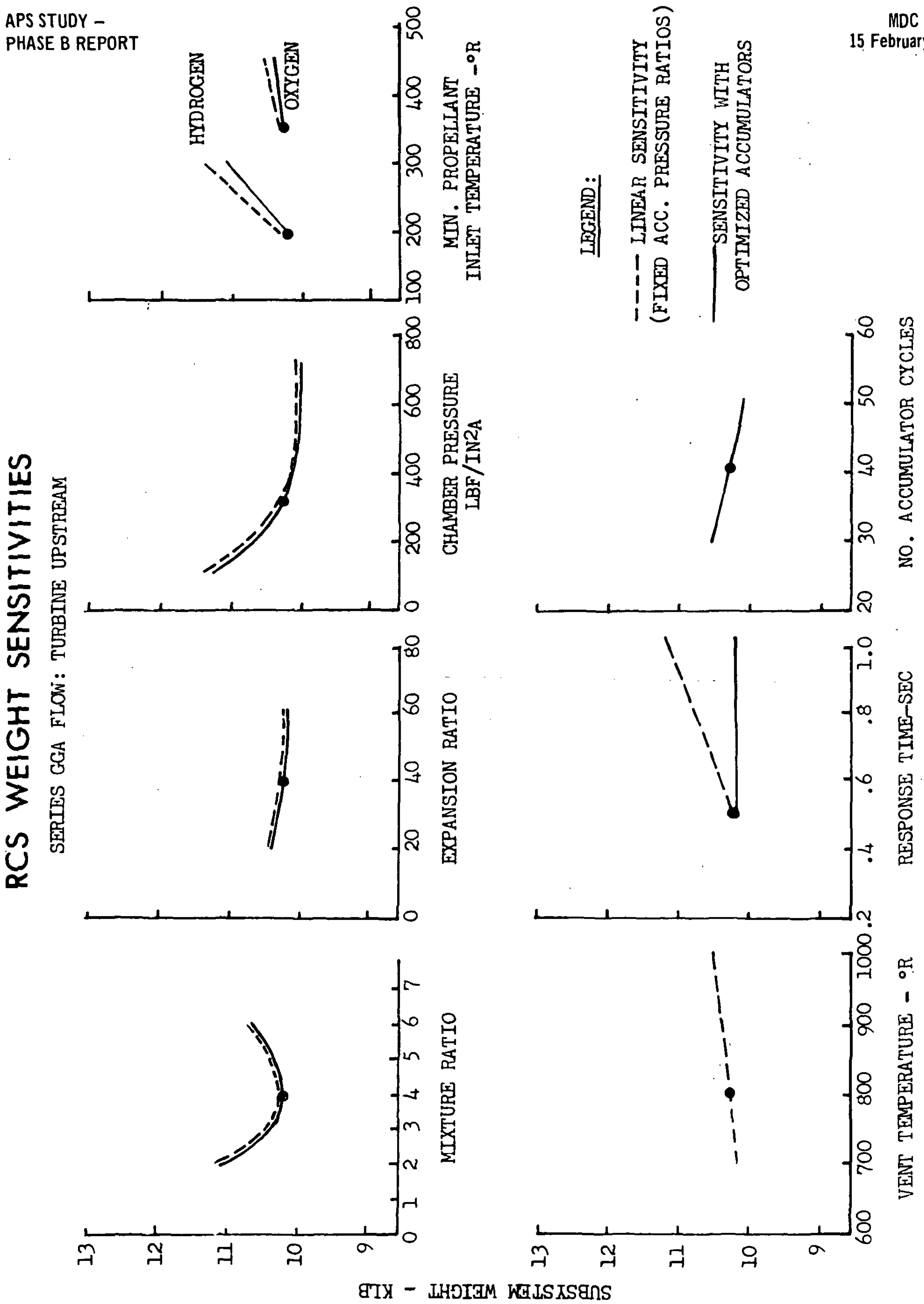
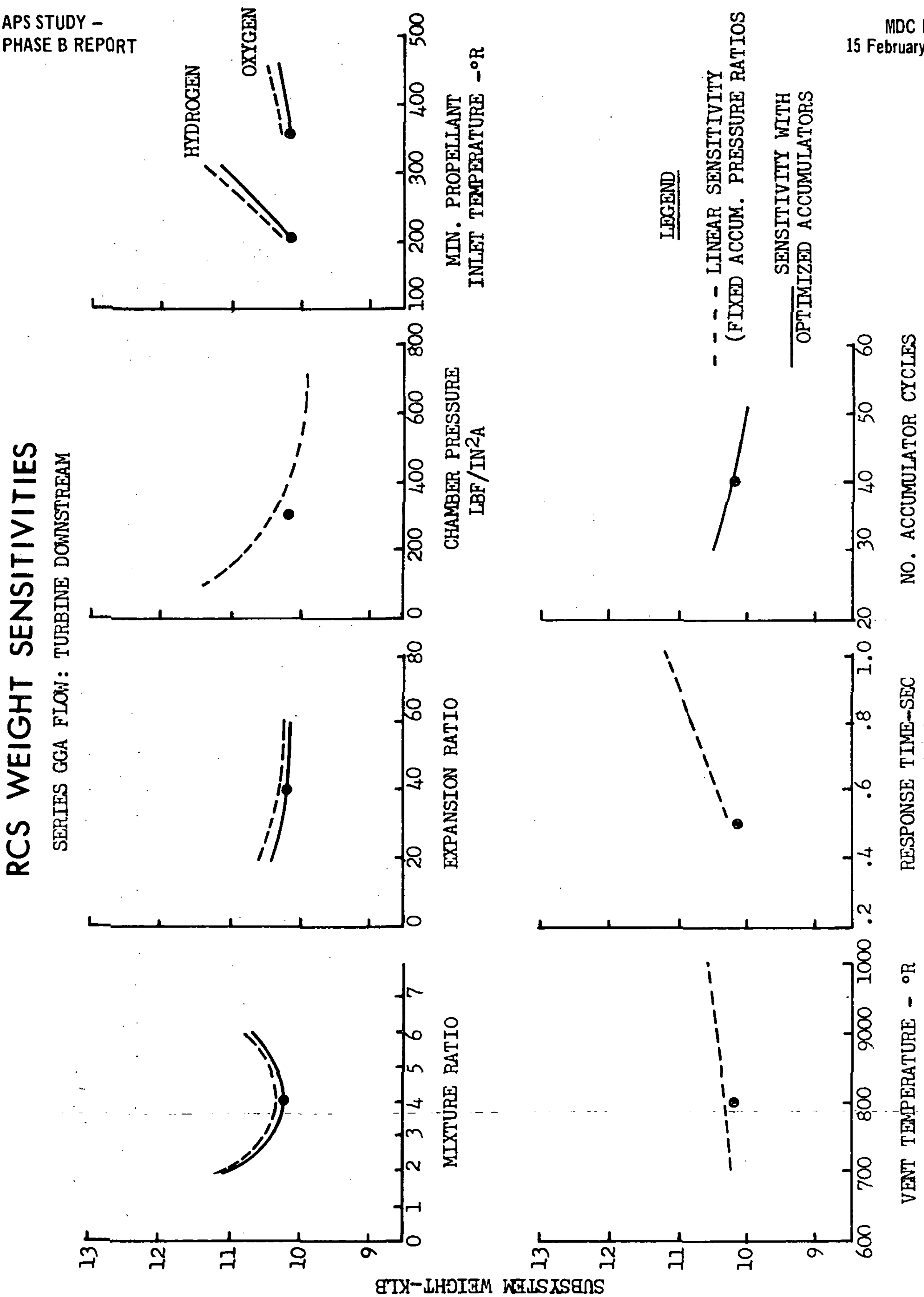

RCS WEIGHT SENSITIVITIES
SERIES GGA FLOW: TURBINE UPSTREAM
SUBSYSTEM WEIGHT - KLB
MIXTURE RATIO
EXPANSION RATIO
CHAMBER PRESSURE LBF/IN2A
MIN. PROPELLANT INLET TEMPERATURE -°R
HYDROGEN
OXYGEN
VENT TEMPERATURE - °R
RESPONSE TIME–SEC
NO. ACCUMULATOR CYCLES
LEGEND:
LINEAR SENSITIVITY (FIXED ACC. PRESSURE RATIOS)
SENSITIVITY WITH OPTIMIZED ACCUMULATORS

Figure B-6


RCS WEIGHT SENSITIVITIES
SERIES GGA FLOW: TURBINE DOWNSTREAM
SUBSYSTEM WEIGHT-KLB
MIXTURE RATIO
EXPANSION RATIO
CHAMBER PRESSURE LBF/IN²A
MIN. PROPELLANT INLET TEMPERATURE -°R
HYDROGEN
OXYGEN
VENT TEMPERATURE - °R
RESPONSE TIME-SEC
NO. ACCUMULATOR CYCLES
LEGEND
LINEAR SENSITIVITY (FIXED ACCUM. PRESSURE RATIOS)
SENSITIVITY WITH OPTIMIZED ACCUMULATORS

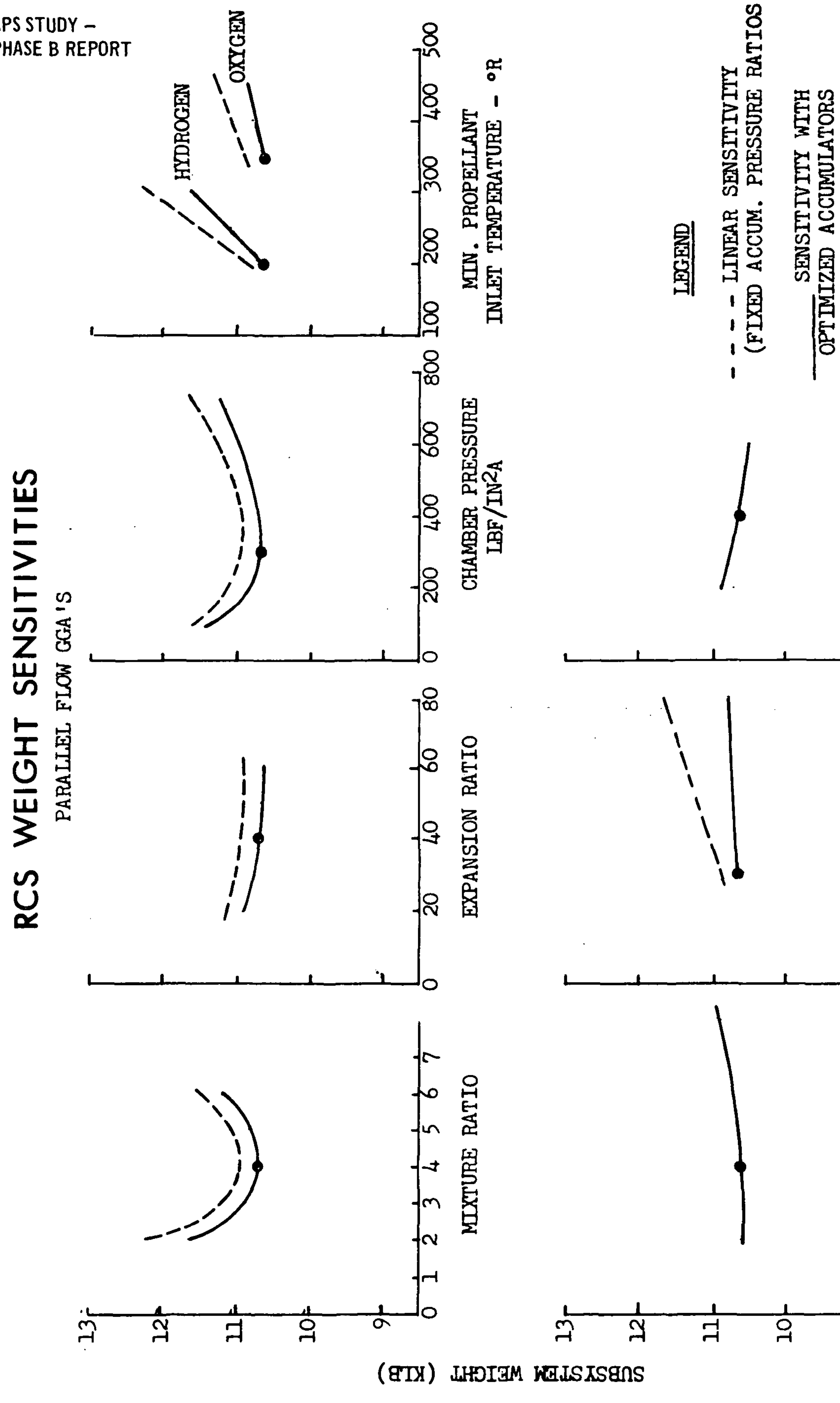

RCS WEIGHT SENSITIVITIES
PARALLEL FLOW GGA'S
SUBSYSTEM WEIGHT (KLB)
MIXTURE RATIO
EXPANSION RATIO
CHAMBER PRESSURE
LBF/IN²A
MIN. PROPELLANT
INLET TEMPERATURE - °R
HYDROGEN
OXYGEN
VENT TEMPERATURE - °R
RESPONSE TIME-SEC
ACCUMULATOR CYCLES
LEGEND
LINEAR SENSITIVITY
(FIXED ACCUM. PRESSURE RATIOS
SENSITIVITY WITH
OPTIMIZED ACCUMULATORS

Figure B-8

APPENDIX C

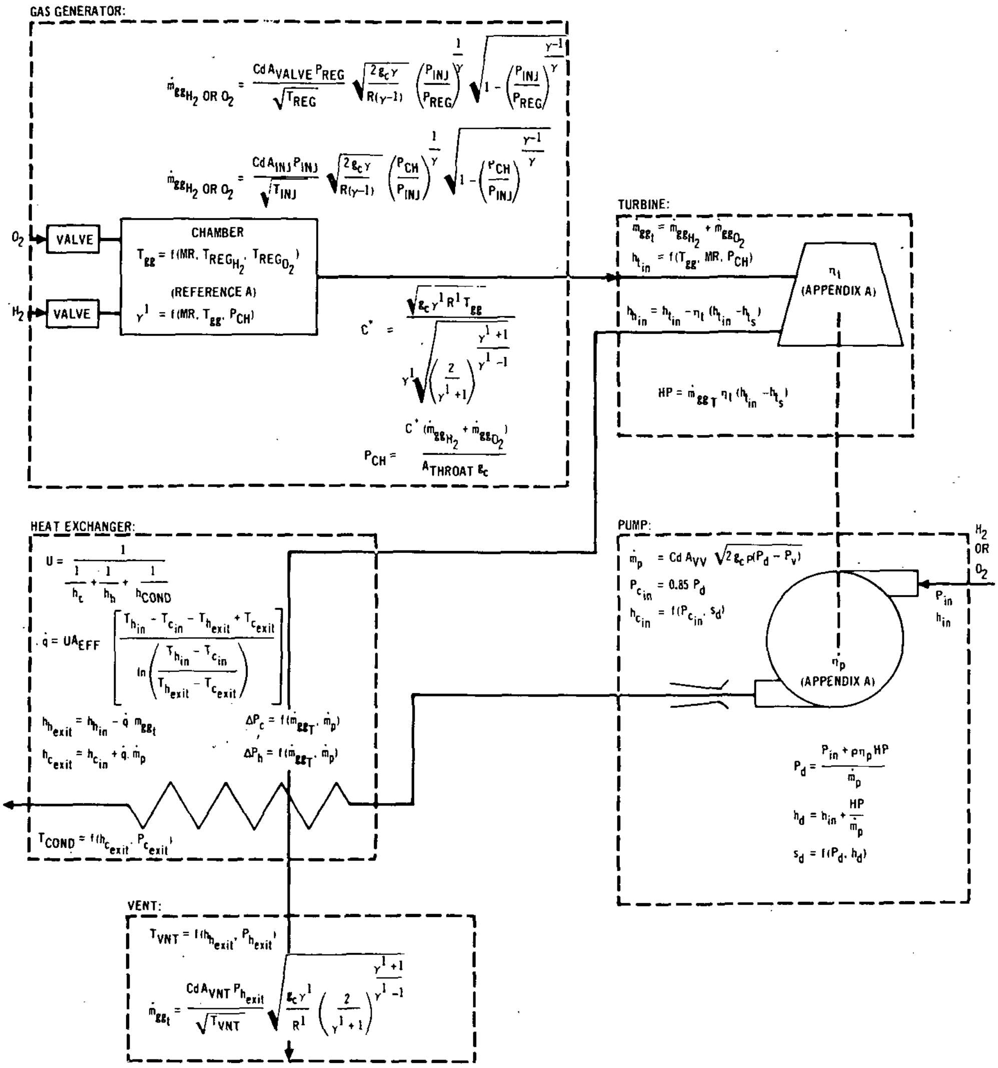
CONDITIONER CONTROLS EVALUATIONS

Conditioner operating analyses were conducted at the design point pressure, temperature, and flow balances defined in Appendix B to determine the effectiveness of alternate control concepts. Initially, all pertinent valve and component flow areas were sized to obtain steady-state pressure, temperature, and flow balances at the design point. Complete RCS mission duty cycles were then simulated using the operational performance program described in Reference K to define gas generator inlet temperature bands (expected temperature variation during the mission). These operating temperature bands, in conjunction with component tolerance data, were applied to determine open-loop (no controls) conditioner performance for each candidate RCS concept. Various control concepts were then evaluated applying the mass flow and energy balance equations shown in Figure C-1 to determine their effectiveness in controlling critical conditioner parameters (e.g., gas generator combustion temperature, conditioned propellant temperature, pump discharge pressure and pump flow rate). The control concepts were evaluated in three levels of complexity: (1) evaluation of passive control devices, (2) evaluation of active controls in conjunction with a passive flow control device upstream of the gas generator, and (3) evaluation of active controls, only. Effectiveness for the various control concepts was measured in terms of total system weight at the extremes in conditioned temperature, pump discharge pressure and flow rate, using the design and sizing techniques of Reference N. Preferred control concepts were then selected for each RCS concept to provide the best compromise between system weight and design complexity. Results of these analyses and selection of preferred control concepts for each RCS are contained in the following paragraphs.

C1. <u>Valve/Component Flow Areas</u> - Pertinent valve/component flow areas are shown in Figure C-2 for each of the three RCS concepts. These areas were calculated for the design point pressure, temperature and flow balances presented in Appendix B. As shown in Figure C-2, a cavitating venturi is employed at the pump outlet to decouple pump flow rate from variations in downstream accumulator pressure.

C2. <u>Gas Generator Inlet Temperature Bands</u> - Accumulator (gas generator inlet) temperature variations were evaluated for environmental temperatures of 300 and 560°R, by simulating each of the three RCS missions defined in Reference G using the operational performance program described in Reference K. Figures C-3 and C-4 show results for the resupply mission. The temperature histories of Figure C-3

RCS CONDITIONER MASS FLOW AND ENERGY BALANCES



NOMENCLATURE

- A – FLOW AREA
- Cd – VALVE DISCHARGE COEFFICIENT
- C^* – CHARACTERISTIC VELOCITY
- f – FUNCTIONAL NOTATION
- g_c – DIMENSIONAL CONSTANT
- h – ENTHALPY
- $\bar{h}$ – CONVECTION HEAT TRANSFER COEFFICIENT
- h_{COND} – CONDUCTION HEAT TRANSFER COEFFICIENT
- HP – TURBINE DELIVERED SHAFT POWER
- $\dot{m}$ – MASS FLOW RATE
- MR – MIXTURE RATIO (OXYGEN HYDROGEN)
- P – PRESSURE
- $\dot{q}$ – HEAT TRANSFER RATE
- R – SPECIFIC GAS CONSTANT (OXYGEN OR HYDROGEN)
- R^1 – SPECIFIC GAS CONSTANT (COMBUSTION PRODUCTS)
- s – ENTROPY
- T – TEMPERATURE
- U – OVERALL HEAT TRANSFER COEFFICIENT
- γ – RATIO OF SPECIFIC HEATS (OXYGEN OR HYDROGEN)
- γ^1 – RATIO OF SPECIFIC HEATS (COMBUSTION PRODUCTS)
- η – EFFICIENCY
- ρ – DENSITY

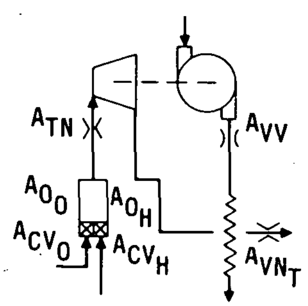
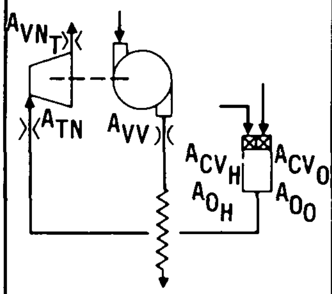
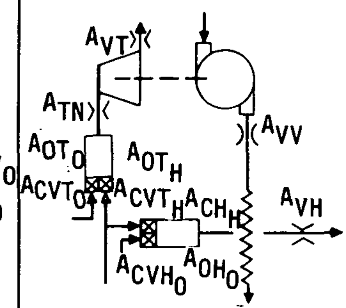
SUBSCRIPTS

- H_2 – HYDROGEN; O_2 – OXYGEN
- C – HEAT EXCHANGER COLD SIDE FLOW
- CH – CHAMBER
- COND – CONDITIONED PARAMETER
- d – PUMP DISCHARGE
- EFF – EFFECTIVE
- gg – GAS GENERATOR
- h – HEAT EXCHANGER HOT SIDE FLOW
- INJ – INJECTOR
- P – PUMP
- REG – PRESSURE REGULATED CONDITION
- s – ISENTROPIC EXPANSION
- t – TURBINE
- T – TOTAL
- v – VAPOR
- VNT – VENT
- VV – CAVITATING VENTURI

Figure C-1

SYSTEM VALVE/COMPONENT DESIGN AREAS

• GAS GENERATOR CHAMBER PRESSURE = 300 $LB_f/IN.^2A$

	SERIES TURBINE UPSTREAM	SERIES TURBINE DOWNSTREAM	PARALLEL
			
•GAS GENERATOR CONTROL VALVE AREA, $IN.^2$	H_2 , O_2 / H_2 , O_2	H_2 , O_2 / H_2 , O_2	H_2 , O_2 / H_2 , O_2
A_{CV_O}/A_{CV_H}	0.1142, 0.0655/0.3420, 0.1960	0.1070, 0.0652/0.3200, 0.1950	–
A_{CVT_O}/A_{CVT_H}	–	–	0.0505, 0.0190/0.1400, 0.0529
A_{CVH_O}/A_{CVH_H}	–	–	0.1000, 0.0620/0.2800, 0.1730
INJECTOR ORIFICE AREA, $IN.^2$			
A_{O_O}/A_{O_H}	0.1012, 0.0580/0.3020, 0.1730	0.0947, 0.0577/0.2830, 0.1720	–
A_{OT_O}/A_{OT_H}	–	–	0.0370, 0.1381/0.1175, 0.0439
A_{OH_O}/A_{OH_H}	–	–	0.0769, 0.0466/0.2446, 0.1481
•TURBINE ADMISSION AREA – ATN, $IN.^2$	1.060 , 0.6080	0.7950, 0.4220	0.4607, 0.1722
•PUMP CAVITATING VENTURI AREA – A_{VV}, $IN.^2$	0.0835, 0.0496	0.0864, 0.0501	0.0950, 0.0555
•VENT AREA, $IN.^2$			
A_{VN_T}	4.100 , 0.8680	8.000 , 3.000	–
A_{VT}	–	–	8.321 , 3.215
A_{VH}	–	–	0.6960, 0.3758

ACCUMULATOR PROPELLANT TEMPERATURE HISTORIES

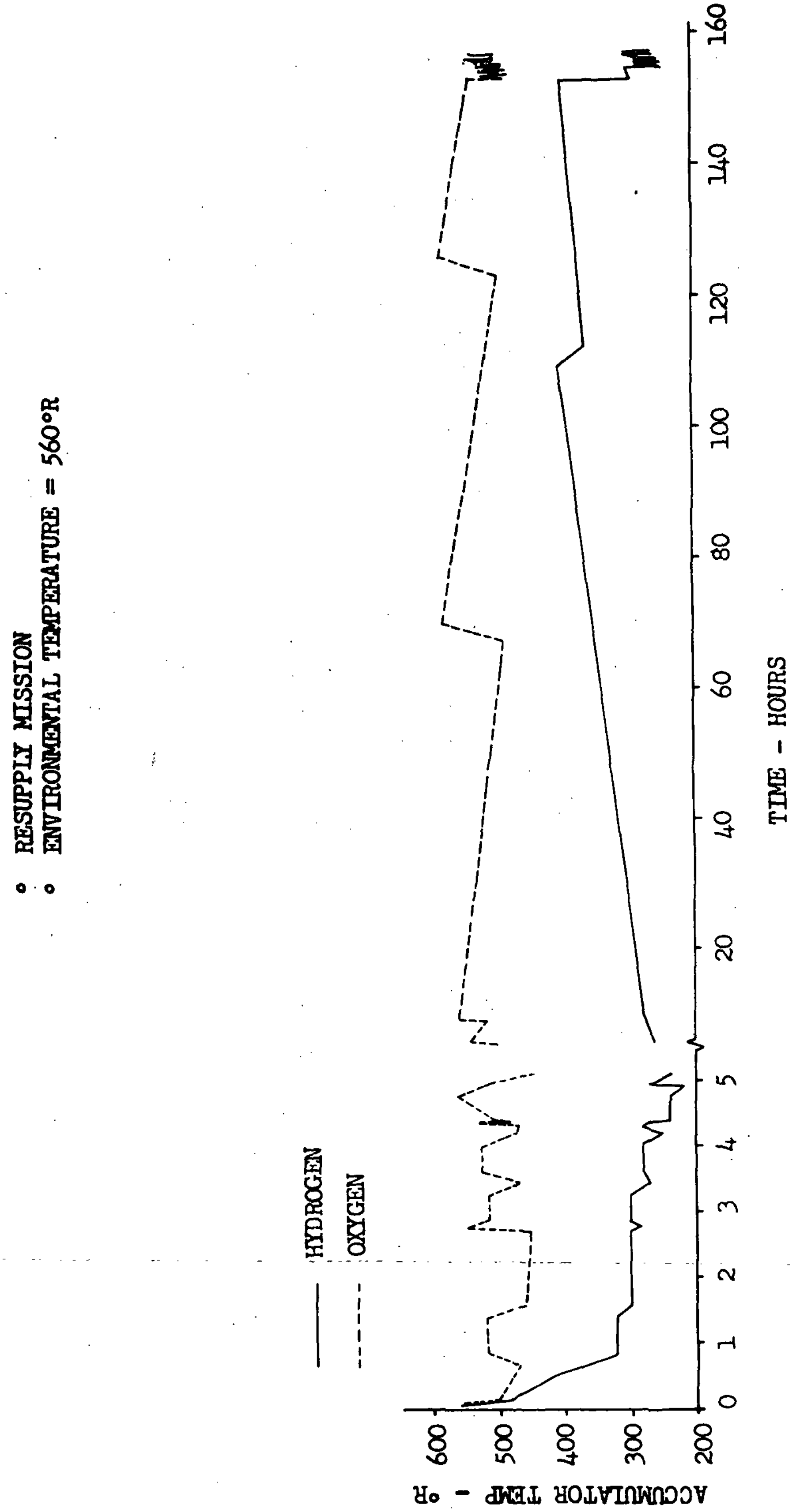


Figure C-3

EFFECT OF ENVIRONMENT ON ACCUMULATOR TEMPERATURE

° RESUPPLY MISSION

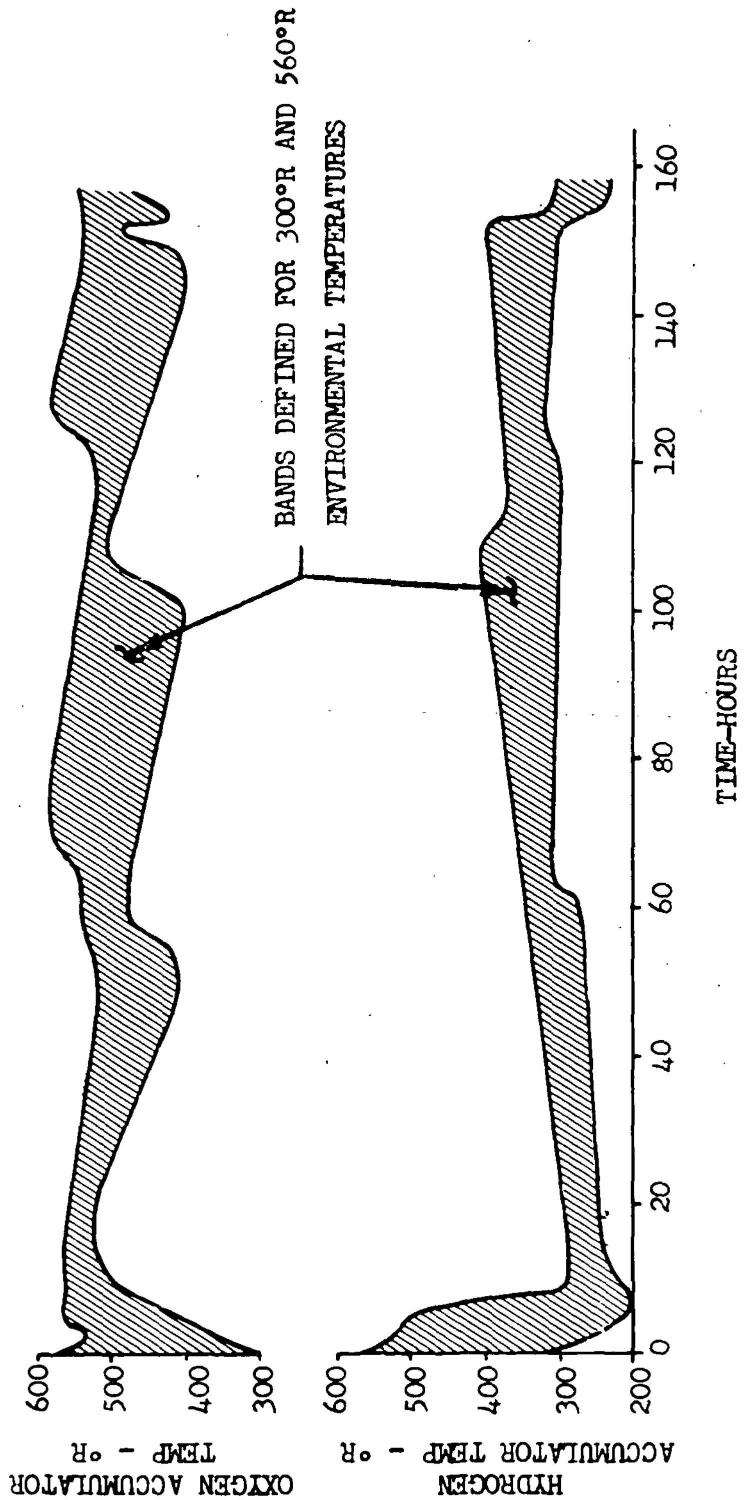


Figure C-4

show that the oxygen and hydrogen accumulator blowdown/recharge cycles can get out of phase, creating large gas generator inlet temperature differentials during conditioner operation. The propellant temperatures may be nearly equal, as at the start of the mission, or they may be driven toward opposite extremes as a result of compression during accumulator recharge, and expansion during blowdown. Figure C-4 shows the operating temperature envelopes as defined by the environmental temperature extremes. These envelopes define propellant inlet temperature ranges for the gas generators.

C3. Component Tolerances - Requests for technical information were submitted to forty component and propulsion system contractors to obtain historical data for component performance tolerances and sensor accuracies. Responses to these requests are summarized in Figure C-5, and contractor sources are identified in Figure C-6. Run-to-run component tolerances were preferred for subsequent conditioner open-loop performance and controls evaluations. However, unit-to-unit tolerances were used in certain instances because of the lack of sufficient data samples for a given unit, or an inability to differentiate between run-to-run and unit-to-unit data samples. An example of this latter case is illustrated in Figure C-7, which shows the computed standard deviation in pump efficiency as a function of flow coefficient. These data were obtained from tests of twenty-four Mark 15 (J-2 engine) hydrogen pumps. Averaging the data of Figure C-5 provided the component tolerances summarized in Figure C-8. These tolerances formed the basis for conditioner open-loop performance and controls evaluations.

C4. Conditioner Open-Loop Performance - Applying the gas generator inlet temperature bands and component tolerances defined in the preceeding paragraphs, conditioner open-loop performance was evaluated using the equations of Figure C-1. To provide maximum understanding of how these component and temperature tolerances affect conditioner performance, each tolerance was evaluated independently to determine its singular effect on the performance of the conditioner assemblies (i.e., gas generator, turbopump, and heat exchanger). Defined in Figure C-9 are the tolerances employed in this evaluation as well as the nomenclature used in subsequent figures. Example open-loop performance evaluations are shown in Figures C-10 through C-12 for the hydrogen side of the series-upstream turbine RCS. These figures show the progressive effect of each component tolerance on: (1) gas generator performance, (2) turbopump performance, and (3) heat exchanger performance. The effects of positive tolerances (areas or parameters greater than the design

PRELIMINARY COMPONENT TOLERANCE SUMMARY

COMPONENT	OPERATING VARIABLE	TOLERANCE RANGE	TOLERANCE TYPE	TOLERANCE SOURCE	REMARKS
TURBINE, HOT GAS	EFFICIENCY	$\pm$2.0%	DATA	7	PART TO PART VARIATION
		$\pm$8.0%	DATA	8	PART TO PART VARIATION
		$\pm$1.68%	DATA	9	RUN TO RUN VARIATION
		$\pm$4.53%	DATA	9	PART TO PART VARIATION
	PRESSURE RATIO	$\pm$1.0%	DATA	7	PART TO PART VARIATION
		$\pm$2.0%	SPEC	1	
PUMP, CRYOGENIC O_2 OR H_2	EFFICIENCY	$\pm$5.0%	DATA	2	PART TO PART VARIATION
		$\pm$1.6%	DATA	7	PART TO PART VARIATION, FUEL
		$\pm$ .9%	DATA	7	PART TO PART VARIATION, OXID
		$\pm$6.0%	DATA	8	PART TO PART VARIATION, FUEL
		$\pm$.81%	DATA	9	RUN TO RUN VARIATION
		$\pm$2.58%	DATA	9	PART TO PART VARIATION
	PRESSURE RISE	$\pm$3.0%	SPEC	1	
		$\pm$2.5%	DATA	7	PART TO PART VARIATION
		$\pm$3.1%	DATA	8	PART TO PART VARIATION, OXID
		$\pm$6.3%	DATA	8	PART TO PART VARIATION, FUEL
		$\pm$1.65%	DATA	9	RUN TO RUN VARIATION
		$\pm$5.22%	DATA	9	PART TO PART VARIATION
HEAT EXCHANGER	OVERALL HEAT TRANSFER RATE	$\pm$5.6%	DATA	10	PART TO PART VARIATION
	HOT SIDE ΔP	$\pm$1.0%	DATA	7	PART TO PART VARIATION
	COLD SIDE ΔP	$\pm$9.0%	DATA	7	PART TO PART VARIATION
GAS GENERATOR	COMBUSTION EFFICIENCY	<$\pm$1.0%	-	ESTIMATE	

PRELIMINARY COMPONENT TOLERANCE SUMMARY
(Continued)

COMPONENT	OPERATING VARIABLE	TOLERANCE RANGE	TOLERANCE TYPE	TOLERANCE SOURCE	REMARKS
VALVE, SHUTOFF	FLOW AREA	±0.8%	DATA	7	PART TO PART VARIATION
VALVE, THROTTLING	REPEATABILITY	±1.4% of THROTTLE RANGE	DATA	12	RUN TO RUN VARIATION
VALVE, CHECK	CRACKING PRESSURE	±50%	SPEC	6	
		±10%	SPEC	13	
	FLOW AREA	<±1.0%	-	ESTIMATE	
VENTURI, CAVITATING	PRESSURE RECOVERY	±1.0%	DATA	12	PART TO PART VARIATION
REGULATOR, PRESSURE	REGULATED PRESSURE	±0.8% to ±5.7%	SPEC	3	
		±1.5%	SPEC	6	
		±2.5%	DATA	11	RUN TO RUN VARIATION
		±2.0%	DATA	11	RUN TO RUN VARIATION
SENSOR, PRESSURE	ACCURACY	±1.5%FS	SPEC	4	
		±0.05%FS to ±1.0%FS	SPEC	14	BASED ON RUN TO RUN DATA
		±0.1%FS to ±0.5%FS	SPEC	15	BASED ON RUN TO RUN DATA

Figure C-5 Continued

PRELIMINARY COMPONENT TOLERANCE SUMMARY
(Continued)

COMPONENT	OPERATING VARIABLE	TOLERANCE RANGE	TOLERANCE TYPE	TOLERANCE SOURCE	REMARKS
SENSOR, TEMPERATURE (a) THERMOCOUPLES	ACCURACY	±0.5% to ±1.0%	SPEC	5	
(b) RESISTANCE THERMOMETERS		±0.05% to ±0.4%	SPEC	5	
SENSOR, FLOW	ACCURACY	±0.1% to ±3.0%	SPEC	5	

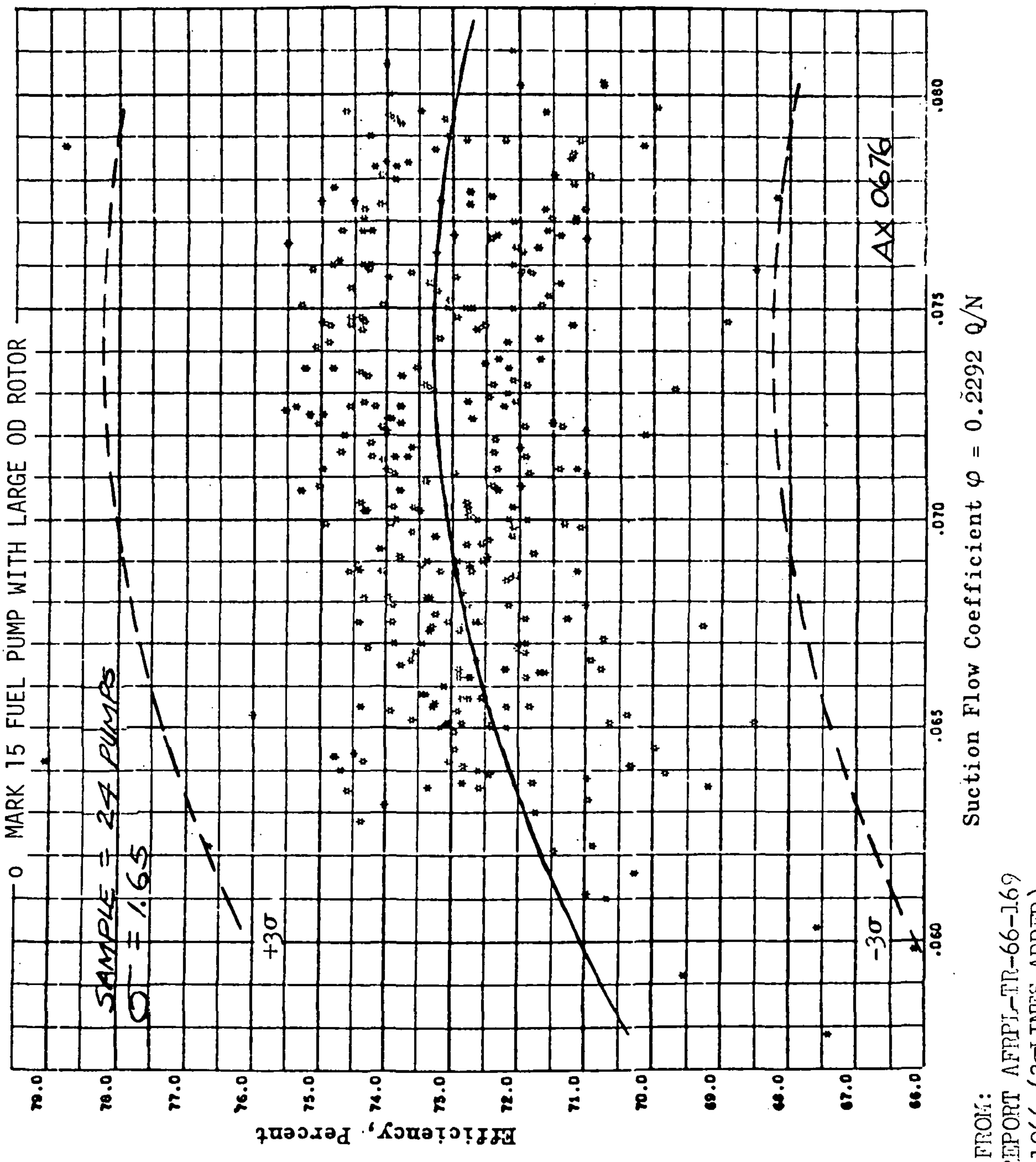
Figure C-5 Continued

COMPONENT TOLERANCE SUMMARY

TOLERANCE DATA SOURCE REFERENCE	
1. TECHNICAL REPORT AFRPL-TR-66-169, JULY 1966.	10. MDAC-EAST SKYLAB AND GEMINI TEST DATA.
2. TECHNICAL REPORT USAF (RPL) TDR-64-25, FEBRUARY 1970.	11. FOX VALVE DEVELOPMENT CO., INC., LETTER DATED 16 AUGUST 1971.
3. NASA REPORT SP-4-28-D, 20 SEPTEMBER 1966.	12. JAMES, POND & CLARK, INC. DIVISION CIRCLE SEAL CORPORATION - CATALOG.
4. TECHNICAL REPORT AFRPL-TR-71-1 VOL. III, JANUARY 1971.	13. STATHAM INSTRUMENTS INC., PRODUCT BULLETIN AM117.
5. TRW SYSTEMS REPORT NO. 12411-6012-R000, JULY 1970.	14. CONSOLIDATED CONTROLS CORPORATION SPECIFICATIONS.
6. ROCKETDYNE UNPUBLISHED DATA PER TELECON.	15. ROCKETDYNE LETTER 71RC6622 DATED 7 SEPTEMBER 1971.
7. PRATT & WHITNEY AIRCRAFT, SPACE SHUTTLE COORDINATION SHEET NO. 3X-58, 20 APRIL 1971.	16. VALCOR ENGINEERING CORPORATION LETTER DATED 2 SEPTEMBER 1971.
8. AEROJET INFORMAL DATA.	
9. AIRESEARCH MFG. CO. LETTER DATED 24 AUGUST 1971.	

Figure C-6

REPRESENTATIVE EFFICIENCY SCATTER FOR
PRODUCTION HYDROGEN PUMPS



REPRODUCED FROM:
TECHNICAL REPORT AFRPL-TR-66-169
DATED JULY 1966 (3σ LINES ADDED)

COMPONENT TOLERANCES USED FOR CONTROLS STUDY

COMPONENT	OPERATING VARIABLE	TOLERANCE RANGE (±%)	REFERENCE SOURCE *
VENTURI, CAVITATING	AREA	1.0	11
REGULATOR, PRESSURE	REGULATED PRESSURE	2.45	2,5,10
SENSOR, TEMPERATURE	ACCURACY	1.0	4
SENSOR, FLOW	ACCURACY	2.58	4,15
TURBINE, HOT GAS	EFFICIENCY	3.08	7,8,15
PUMP, CRYOGENIC	EFFICIENCY	0.82	1,7,8,15
HEAT EXCHANGER	OVERALL HEAT TRANSFER RATE	5.6	9
GAS GENERATOR	COMBUSTION EFFICIENCY	1.0	15
VALVE, SHUTOFF	FLOW AREA	1.0	6,16
VALVE, THROTTLING	REPEATABILITY	1.0	11
SENSOR, PRESSURE	ACCURACY	1.0	3,13,14

Figure C-8

TOLERANCE EFFECTS ON SYSTEM PERFORMANCE

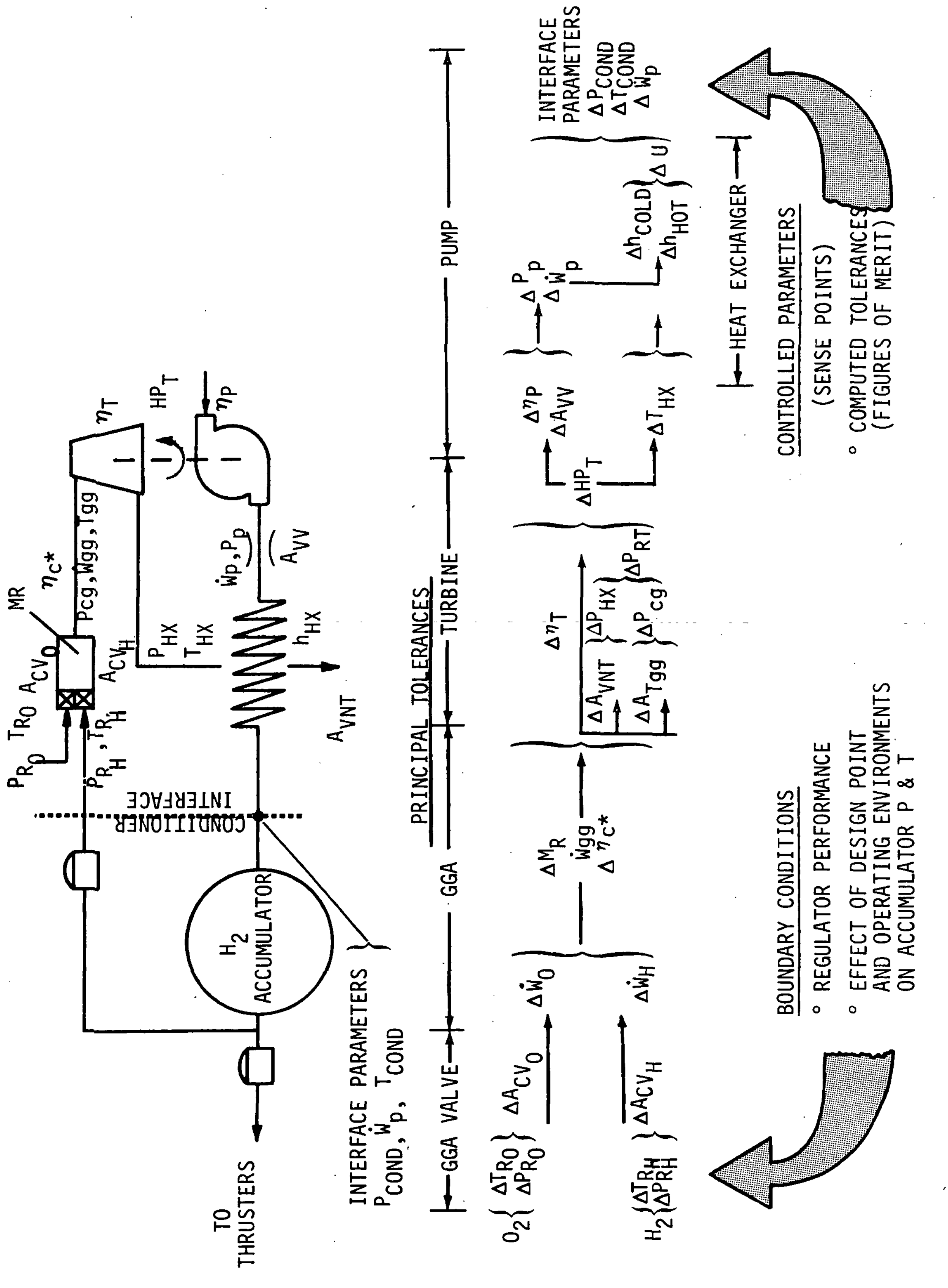


Figure C-9

TOLERANCE EFFECTS ON GGA AND CONDITIONER PERFORMANCE (HYDROGEN)

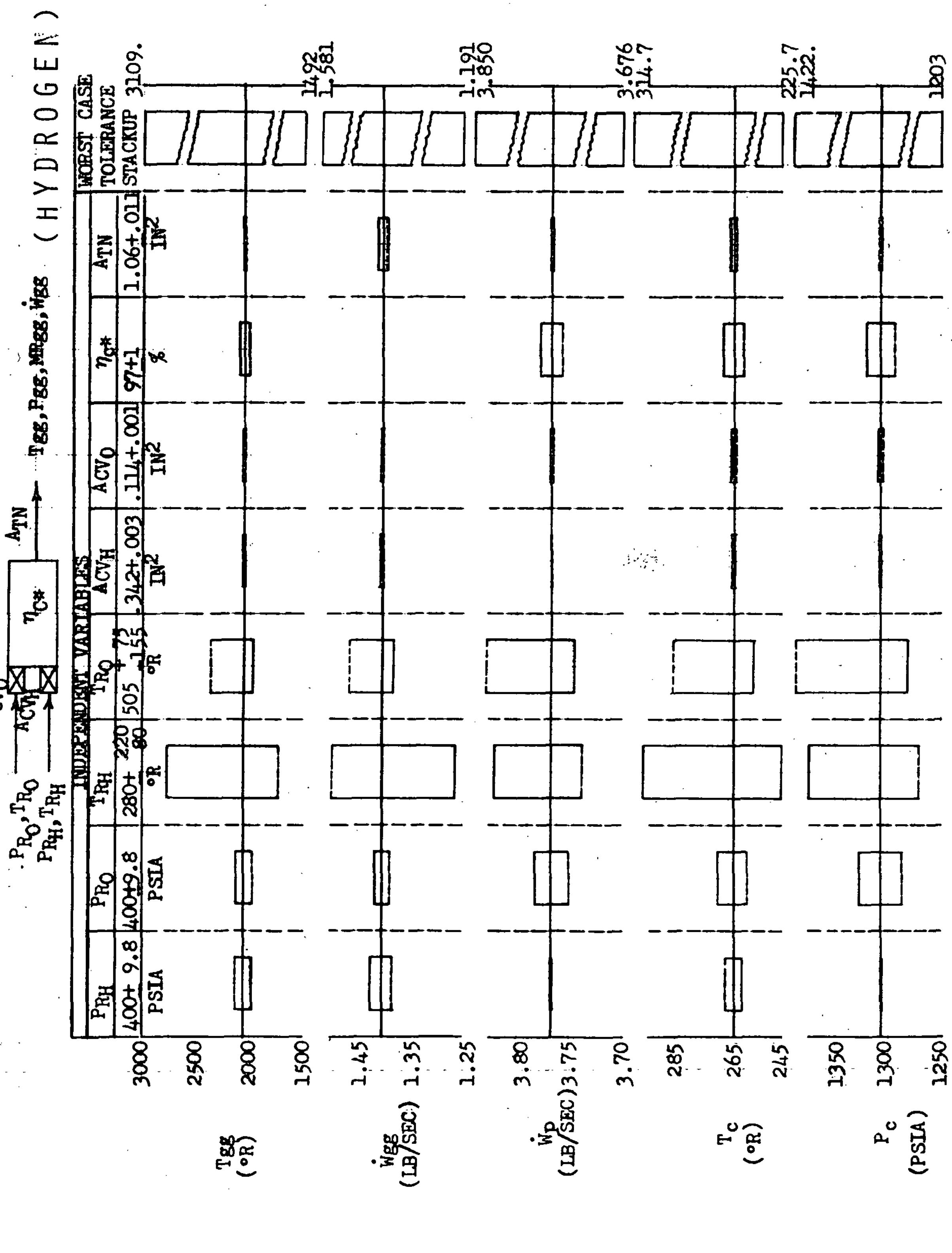


Figure C-10

TOLERANCE EFFECTS ON TPA PERFORMANCE (HYDROGEN)

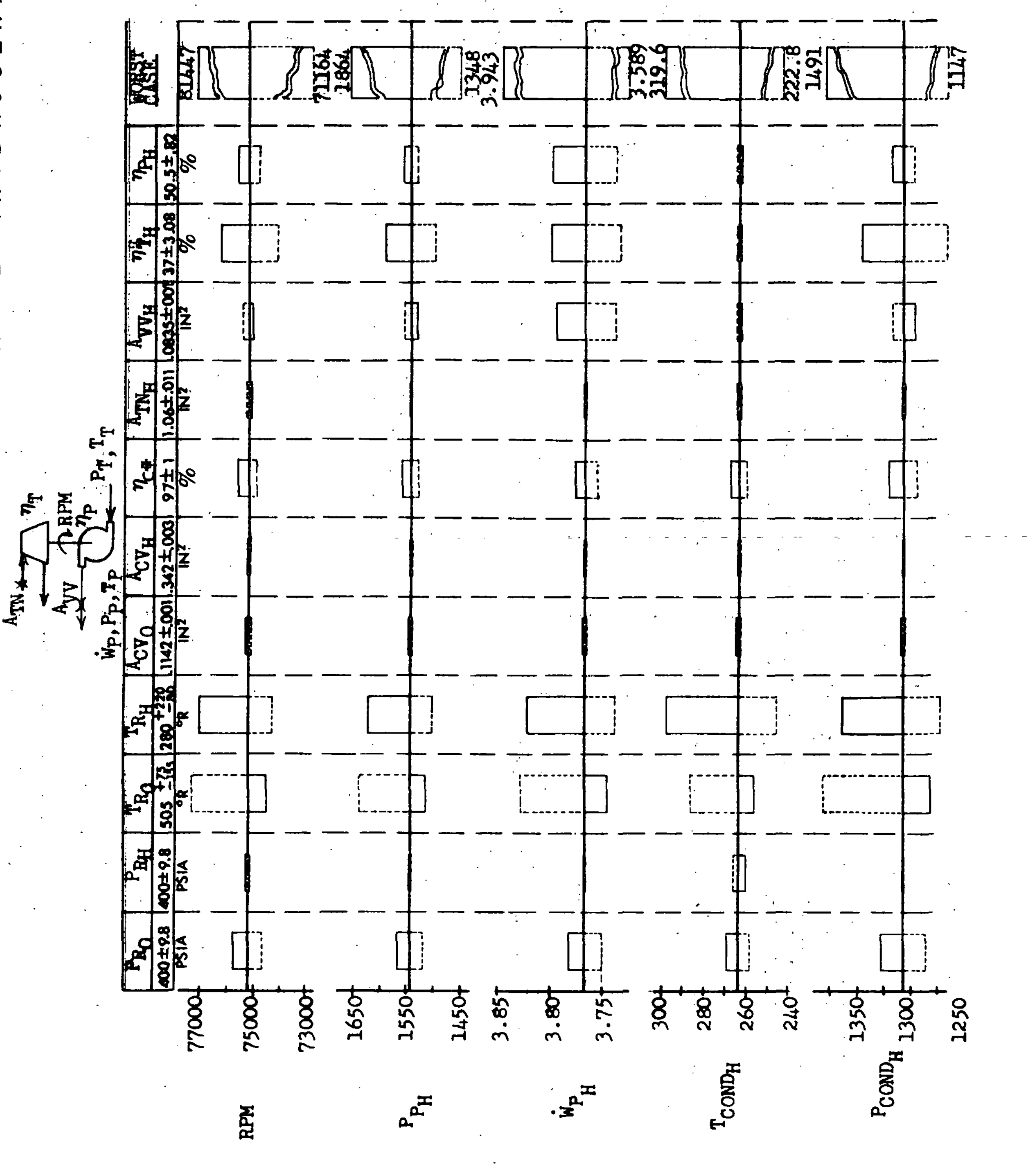
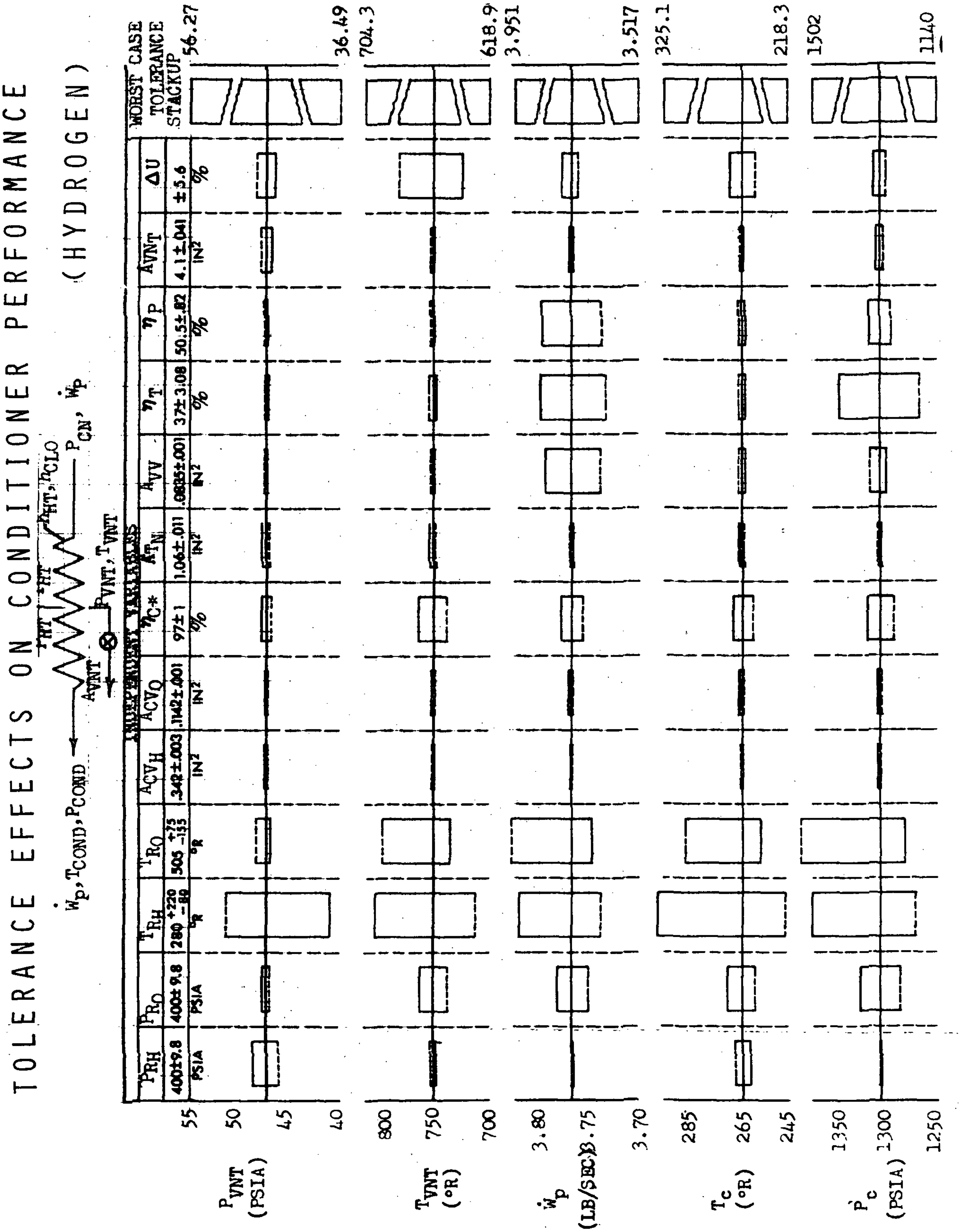


Figure C-11



TOLERANCE EFFECTS ON CONDITIONER PERFORMANCE (HYDROGEN)

Figure C-12

value) are indicated by the solid bars in each figure, and the effects of negative tolerances are indicated by the dashed bars. Examination of these figures shows that gas generator inlet temperatures have the greatest impact on overall conditioner performance.

Considering the gas generator performance example of Figure C-10, it is seen that the oxygen and hydrogen temperatures at the gas generator inlet affect gas generator flow rate in the same manner. However, they have an opposite effect on combustion temperature and other conditioner interface parameters (i.e., pump flow rate, conditioned temperature and conditioned pressure). This is because positive hydrogen inlet temperature tolerances decrease hydrogen density, and thus reduce hydrogen and total gas generator flow rate. The reduction in hydrogen flow rate causes a corresponding increase in reactant mixture ratio giving rise to increased gas generator combustion temperature. The effect of increased oxygen inlet temperature on flow rate is the same, but the decreased oxygen flow rate causes a decrease in reactant mixture ratio, resulting in a lower gas generator combustion temperature.

The effect of gas generator inlet temperatures on propellant conditioning capability is shown in the lower half of Figures C-10 through C-12. Pump flow rate, conditioned temperature,and conditioned pressure are all increased with an increase in hydrogen gas generator inlet temperature, because combustion temperature and available gas generator power are increased. (In this case, the gain in combustion temperature overrides the decrease in flowrate causing an increase in total available power.) Conversely, the magnitudes of these conditioner interface parameters decrease with an increase in oxygen gas generator inlet temperature. This is due to a decrease in available gas generator power resulting from decreases in both combustion temperature and flow rate.

Considering the turbopump performance example of Figure C-11, it is seen that the effects of gas generator inlet temperatures on shaft speed and pump discharge pressure are similar to those for the conditioner interface parameters. This is because shaft speed and pump discharge pressure are also directly related to gas generator exhaust power.

From the heat exchanger performance example of Figure C-12, it is seen that an increase in hydrogen gas generator inlet temperature is reflected as a decrease in heat exchanger hot side flow rate, and an increase in heat exchanger hot side exit temperature (T_{VNT}). Since hot side exit pressure is proportional to flow rate and the square root of temperature, the decrease in gas generator flow rate overrides the increase in gas generator exhaust temperature, resulting in a lower hot side exit pressure (P_{VNT}). A decrease in heat exchanger hot side exit pressure also occurs with increased oxygen temperature at the gas generator inlet, because both the flow rate and temperature of the exhaust flow are decreased.

In the examples of Figures C-10 through C-12, each assembly is affected by upstream tolerances and its own tolerance contribution. The gas generator is decoupled from the turbopump and heat exchanger by choked turbine inlet nozzles and as such is affected only by the eight tolerances shown in Figure C-10. The turbopump is decoupled from the heat exchanger by a cavitating venturi at the pump outlet, and thus is affected by the same eight tolerances which affect the gas generator plus three additional tolerances inherent to itself. Finally, as shown in Figure C-12, the heat exchanger is affected by the eleven upstream tolerances plus its own tolerance on overall heat transfer coefficient and the tolerance on vent area.

For each of the examples of Figures C-10 through C-12, root-sum-square tolerance stackups were calculated, and are shown on the right-hand side of the figures. Other than gas generator inlet temperatures, the only tolerances of notable significance are the turbine and pump efficiencies of Figure C-11. Similar results were obtained for the oxygen conditioner of the series-upstream turbine concept, and both the hydrogen and oxygen conditioners of the series-downstream and parallel RCS concepts. Summaries of open-loop operating variations for all three RCS concepts (both oxygen and hydrogen) are presented in Figure C-13. It is apparent from these results that gas generator controls are required to: (1) prevent combustion temperatures from exceeding maximum design capabilities of the turbine and/or heat exchanger, or (2) prevent gas generator exhaust power from decaying to such a low value that required propellant conditioning cannot be achieved.

C5. Passive Controls Evaluations - One potential means of controlling gas generator combustion temperature is the mass flow controller which is being studied by the NASA (Contract NAS 9-11750) as a technology item. Installed in the gas generator inlet propellant lines, the mass flow controller adjusts regulated pressure as a function of propellant temperature to provide a nearly constant combustion temperature or flow rate. Figure C-14 shows mass flow controller characteristics required for constant combustion temperature control. The hydrogen pressure curve is biased slightly to account for inlet enthalpy effects. As shown by the tolerance bands of Figure C-14, accuracy of the mass flow controller was set at $\pm 5\%$. MDAC-E

RCS CONDITIONER OPEN-LOOP PERFORMANCE

- No Mass Flow Control
- Gas Generator Chamber Pressure = 300 lbf/in^2a

Hydrogen	Series RCS Turbine Upstream	Series RCS Turbine Downstream	Parallel RCS
Gas Generator Combustion Temp., °R	1502 - 3092 (2000)*	1485 - 3053 (2000)	1490 - 3367 (2000)
Pump Flow Rate, lbm/sec	3.58 - 3.95 (3.77)	3.43 - 3.97 (3.71)	3.71 - 4.32 (4.01)
Pump Discharge Press., lb/in^2a	1348 - 1774 (1545)	1088 - 1573 (1327)	1003 - 1411 (1187)
Conditioned Propellant Temp., °R	218 - 325 (263)	204 - 331 (255)	200 - 338 (246)

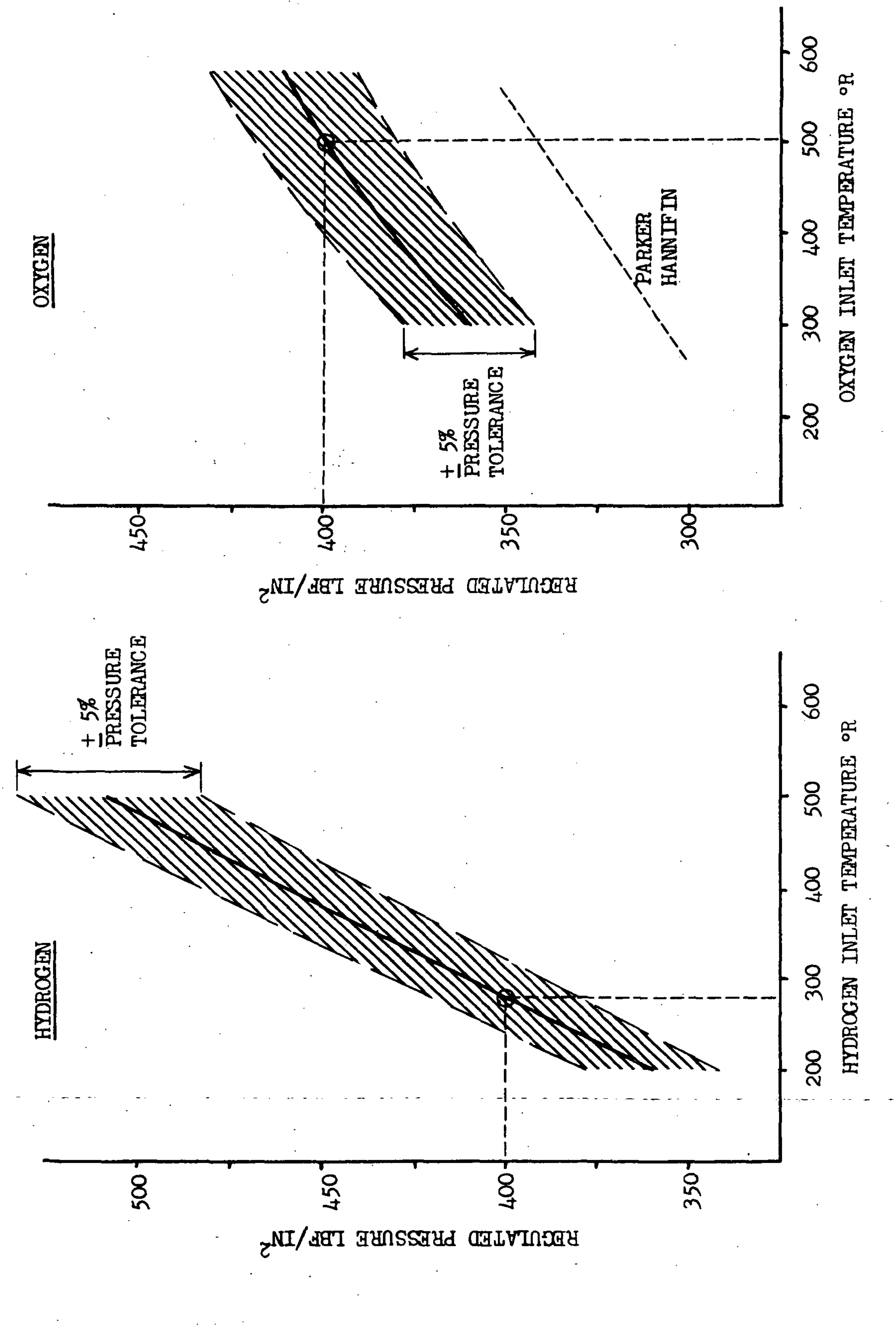
Oxygen	Series RCS Turbine Upstream	Series RCS Turbine Downstream	Parallel RCS
Gas Generator Combustion Temp., °R	1492 - 3109 (2000)	1491 - 3061 (2000)	1489 - 3364 (2000)
Pump Flow Rate, lbm/sec	11.00 - 12.89 (11.76)	10.75 - 12.89 (11.71)	11.15 - 13.09 (12.05)
Pump Discharge Press., lb/in^2a	1673 - 2298 (1901)	1529 - 2213 (1846)	1326 - 1829 (1565)
Conditioned Propellant Temp., °R	394 - 672 (506)	379 - 707 (517)	365 - 741 (473)

*(   ) - design value

Figure C-13

MASS FLOW CONTROLLER

ASSUMED OPERATING CHARACTERISTICS



experience with conventional pressure regulators showed that minimum tolerances of ±2% can be achieved with these devices, but a ±5% tolerance appeared more realistic for the mass flow controller because of its temperature compensation requirement. Conditioner performance with a mass flow controller at ±5% accuracy is presented in Figure C-15 for all three RCS concepts and similar data is presented in Figure C-16 for the series-upstream turbine RCS with a ±2% mass flow controller. Although gas generator combustion temperature and conditioner interface parameter control bands are narrowed from those of Figure C-13, the need for additional gas generator controls is evident.

An alternate passive control concept which combines an inter-propellant heat exchanger with the mass flow controller was also investigated. This concept is illustrated in Figure C-17. The heat exchanger is incorporated just upstream of the mass flow controller for the purpose of equalizing propellant inlet temperatures. Required heat exchanger design characteristics for the series-upstream turbine RCS are also shown in Figure C-17, and performance characteristics for the concept are presented in Figure C-18. Comparing results of Figure C-18 with those of Figure C-15, it is seen that combustion temperature and conditioner interface parameter control is not improved significantly with addition of the heat exchanger. This result can be understood by considering the propellant heat capacities and gas generator operational requirements. The oxygen heat capacity is very low compared to hydrogen, and thus very large oxygen temperature changes produce only small changes in hydrogen temperature. Although oxygen-hydrogen propellant exit temperatures could be made nearly equal by making the heat exchanger large enough, the oxygen temperature must be restricted to a minimum of 260°R to preclude O_2 condensation in the gas generator injector. Thus, when hydrogen inlet temperature is near its minimum value, thermal equalization of propellant flows cannot be achieved since the heat exchanger oxygen outlet temperature is constrained to be above 260°R. Since addition of the heat exchanger has a limited benefit to the range of gas generator inlet temperatures (reactant enthalpies) and, in turn, has limited impact on conditioner performance, this concept was not evaluated further.

To circumvent the above hardware limitations, emphasis was shifted to the evaluation of additional active control points in conjunction with the mass flow controller.

C6. <u>Active Controls Evaluation (With Mass Flow Control)</u> – Figure C-19 illustrates the active control points which were evaluated to determine the most effective way of controlling gas generator combustion temperature and the conditioner

RCS CONDITIONER PERFORMANCE

- *With Mass Flow Control* (*±5% Accuracy*)
- Gas Generator Chamber Pressure = 300 lbf/in^2a

Hydrogen	Series RCS Turbine Upstream	Series RCS Turbine Downstream	Parallel RCS
Gas Generator Combustion Temp., °R	1714 - 2631 (2000)*	1716 - 2678 (2000)	1692 - 2714 (2000)
Pump Flow Rate, lbm/sec	3.67 - 3.86 (3.77)	3.60 - 3.86 (3.71)	3.82 - 4.19 (4.01)
Pump Discharge Press., lb/in^2a	1465 - 1714 (1545)	1241 - 1452 (1327)	1074 - 1302 (1187)
Conditioned Propellant Temp., °R	230 - 312 (263)	219 - 311 (255)	213 - 303 (246)

Oxygen	Series RCS Turbine Upstream	Series RCS Turbine Downstream	Parallel RCS
Gas Generator Combustion Temp., °R	1716 - 2634 (2000)	1717 - 2697 (2000)	1700 - 2698 (2000)
Pump Flow Rate, lbm/sec	11.23 - 12.46 (11.76)	11.34 - 12.33 (11.71)	11.48 - 12.58 (12.05)
Pump Discharge Press., lb/in^2a	1748 - 2138 (1901)	1731 - 2050 (1846)	1423 - 1705 (1565)
Conditioned Propellant Temp., °R	425 - 624 (506)	410 - 653 (517)	393 - 627 (473)

*(   ) - design value

Figure C-15

RCS CONDITIONER PERFORMANCE

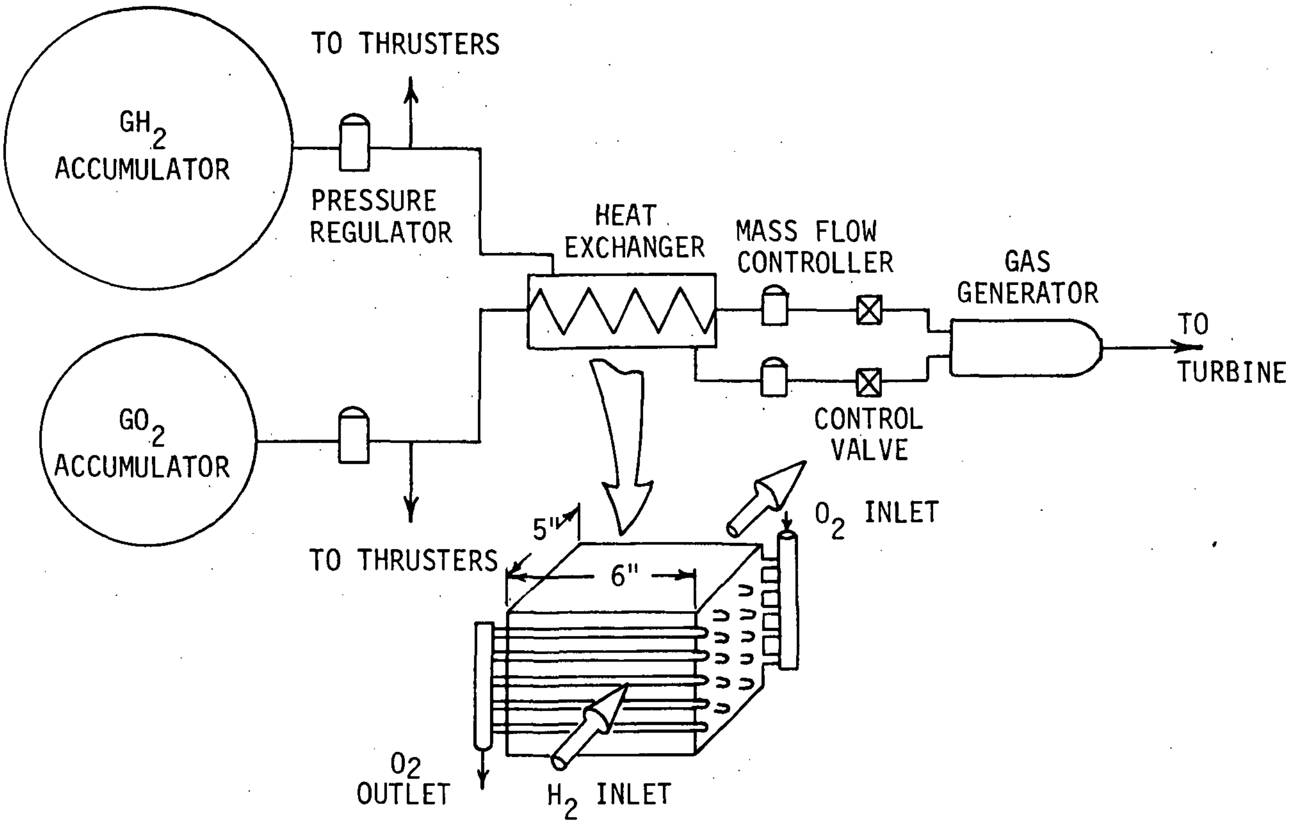
- o Series-Upstream Turbine RCS
- o With Mass Flow Control (±2% Accuracy)
- o Gas Generator Chamber Pressure = 300 $lbf/in.^2a$

	Hydrogen	Oxygen
Gas Generator Combustion Temp., °R	1835 - 2474 (2000)*	1737 - 2325 (2000)
Pump Flow Rate, lbm/sec	3.69 - 3.91 (3.77)	11.31 - 12.43 (11.76)
Pump Discharge Press., $lbf/in.^2a$	1508 - 1669 (1545)	1764 - 2056 (1901)
Conditioned Propellant Temp., °R	239 - 307 (263)	434 - 590 (506)

*( )-design value

PASSIVE HEAT EXCHANGER DESIGN CHARACTERISTICS

- o SERIES-UPSTREAM TURBINE RCS
- o GAS GENERATOR CHAMBER PRESS. = 300 LB_f/IN^2A



DESIGN PARAMETER	HYDROGEN CONDITIONER	OXYGEN CONDITIONER
H_2 INLET TEMPERATURE, °R	280	280
O_2 INLET TEMPERATURE, °R	505	505
H_2 INLET PRESSURE, LB_f/IN^2A	400	400
O_2 INLET PRESSURE, LB_f/IN^2A	400	400
NO. OF O_2 TUBES	21	11
O_2 TUBE OUTSIDE DIAMETER, IN.	.25	.25
O_2 TUBE WALL THICKNESS, IN.	.016	.016
CENTER-TO-CENTER O_2 TUBE SPACING, IN.	.35	.35
EFFECTIVE HEAT TRANSFER SURFACE AREA, IN^2	1410	740
DESIGN HEATING RATE, BTU/SEC	58	31

Figure C-17

RCS CONDITIONER PERFORMANCE

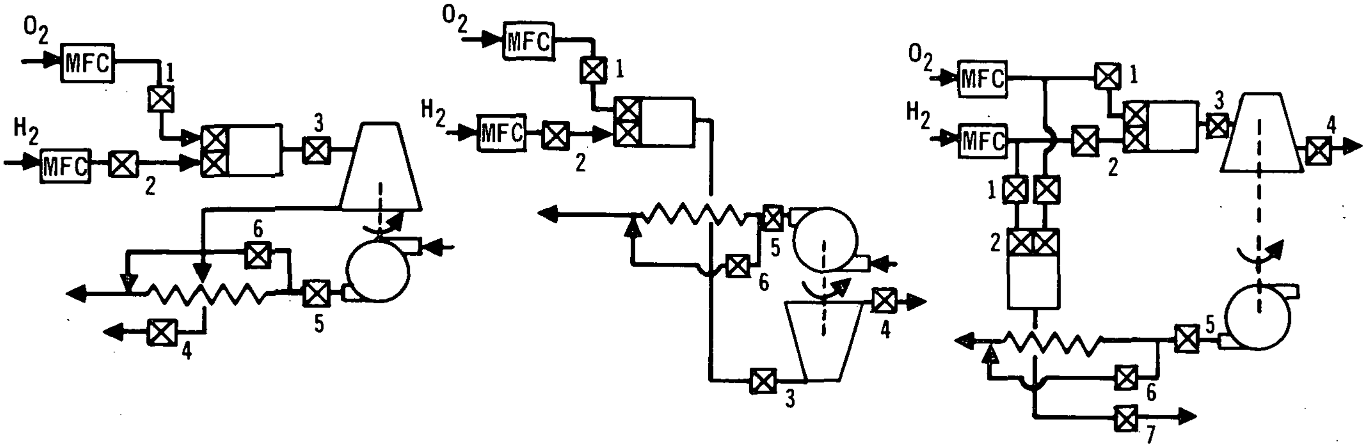
- With mass flow control ($\pm$5% accuracy) in conjunction with upstream heat exchanger for temperature equalization
- Gas generator chamber pressure = 300 $lbf/in.^2a$
- Series RCS/Upstream Turbine

	Hydrogen	Oxygen
Gas Generator Combustion Temp., °R	2082 - 2669 (2000)*	1966 - 2477 (2000)
Pump Flow Rate, lbm/sec	3.69 - 3.95 (3.77)	11.45 - 12.38 (11.76)
Pump Discharge Press., $lbf/in.^2a$	1517 - 1737 (1545)	1837 - 2071 (1901)
Conditioned Propellant Temp., °R	259 - 319 (263)	486 - 613 (506)

* (   )-design value

CANDIDATE CONTROL POINTS
(With Mass Flow Control)

Series RCS (Turbine Upstream) | Series RCS (Turbine Downstream) | Parallel RCS



ACTIVE CONTROL POINT

1. GAS GENERATOR OXYGEN VALVE
2. GAS GENERATOR HYDROGEN VALVE
3. GAS GENERATOR DISCHARGE VALVE
4. HOT GAS VENT VALVE
5. PUMP DISCHARGE VALVE
6. HEAT EXCHANGER COLD SIDE BYPASS VALVE
7. HEAT EXCHANGER VENT VALVE (PARALLEL RCS, ONLY)

INTERMEDIATE CONTROL FUNCTIONS

TURBINE INLET TEMPERATURE
TURBINE FLOW RATE
TURBINE PRESSURE RATIO
--- TURBINE POWER

HEAT EXCHANGER COLD SIDE FLOW RATE

HEAT EXCHANGER HOT SIDE FLOW RATE

CONTROLLED PARAMETER

--- GAS GENERATOR COMBUSTION TEMPERATURE

--- PUMP FLOW RATE
PUMP DISCHARGE PRESSURE

--- CONDITIONED PROPELLANT TEMPERATURE

interface parameters (conditioned propellant temperature, pump flow rate and pump discharge pressure). Also shown are the intermediate functions each control point influences. All the control points affect conditioned propellant temperature. All of the control points except the heat exchanger cold side bypass valve (point 6) and the parallel RCS heat exchanger vent valve (point 7) affect pump flow rate and discharge pressure. Gas generator combustion temperature is affected by the gas generator control valves, only.

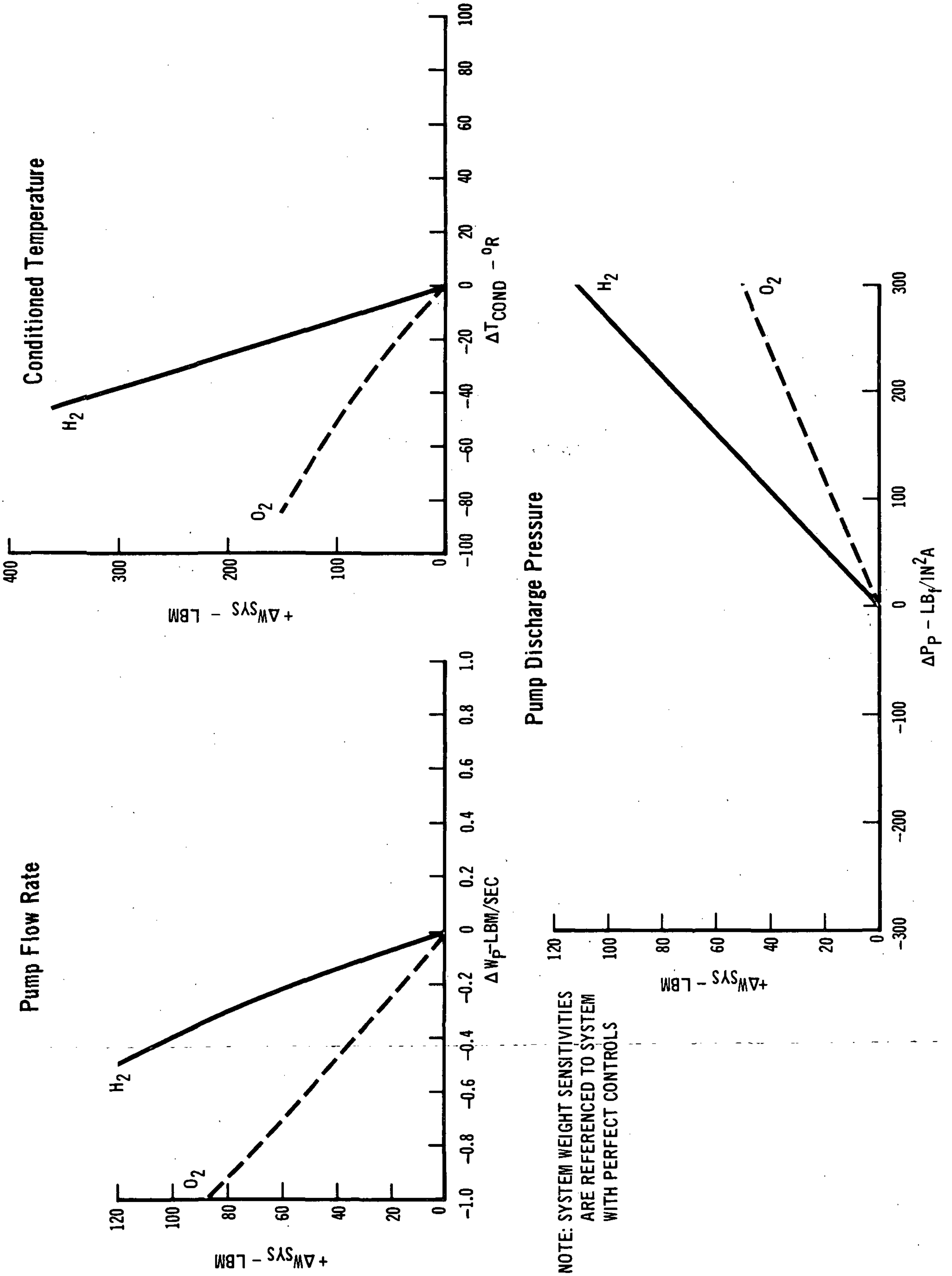
Effectiveness of each control valve was evaluated independently for the series-upstream turbine RCS. This allowed identification of a few high value control approaches which were then studied in-depth for the other RCS concepts. These controls evaluations were conducted in two steps: (1) the appropriate component and gas generator inlet temperature/pressure tolerances were applied to determine operating bands for combustion temperature, conditioned propellant temperature, pump flow rate, and discharge pressure, and (2) the conditioner operating bands found in Step (1) were applied to determine relative system weights using the sensitivity curves of Figures C-20 through C-22. These sensitivities,which were developed using the design and sizing techniques described in Reference N,include weight allocations for system hardware and propellant for steady-state system performance. They do not include additional propellant allowances due to variances in system mixture ratio during the mission. These additional mission propellant allowances are defined in Section 4, Paragraph 4.3 of this report and were included in the final active controls evaluations described in Paragraph C7.

Initial constraints imposed on the controls evaluations and guidelines employed in identification of high value approaches are identified in Figure C-23. Presented below are the detailed control evaluations for the series-upstream turbine RCS. These are followed by a summary of results for the series-downstream turbine and parallel RCS. Results were similar for all three RCS concepts.

C6.1 Combustion Temperature Control - Due to the limited combustion temperature control effectiveness with passive mass flow control (Figure C-15), active controls in conjunction with the mass flow-controller were investigated. This active control can be achieved only through modulation of gas generator valve areas (mixture ratio variation), and as such both the O_2 and H_2 valves were evaluated. As shown in Figure C-24, both valves are capable of narrowing combustion temperature operating bands. However, for the same percent area change, the O_2 valve is more effective in limiting the maximum combustion temperature, and thus is preferred.

SYSTEM WEIGHT SENSITIVITIES TO CONDITIONER INTERFACE PARAMETER TOLERANCES

- SERIES RCS (TURBINE UPSTREAM)



NOTE: SYSTEM WEIGHT SENSITIVITIES ARE REFERENCED TO SYSTEM WITH PERFECT CONTROLS

Figure C-20

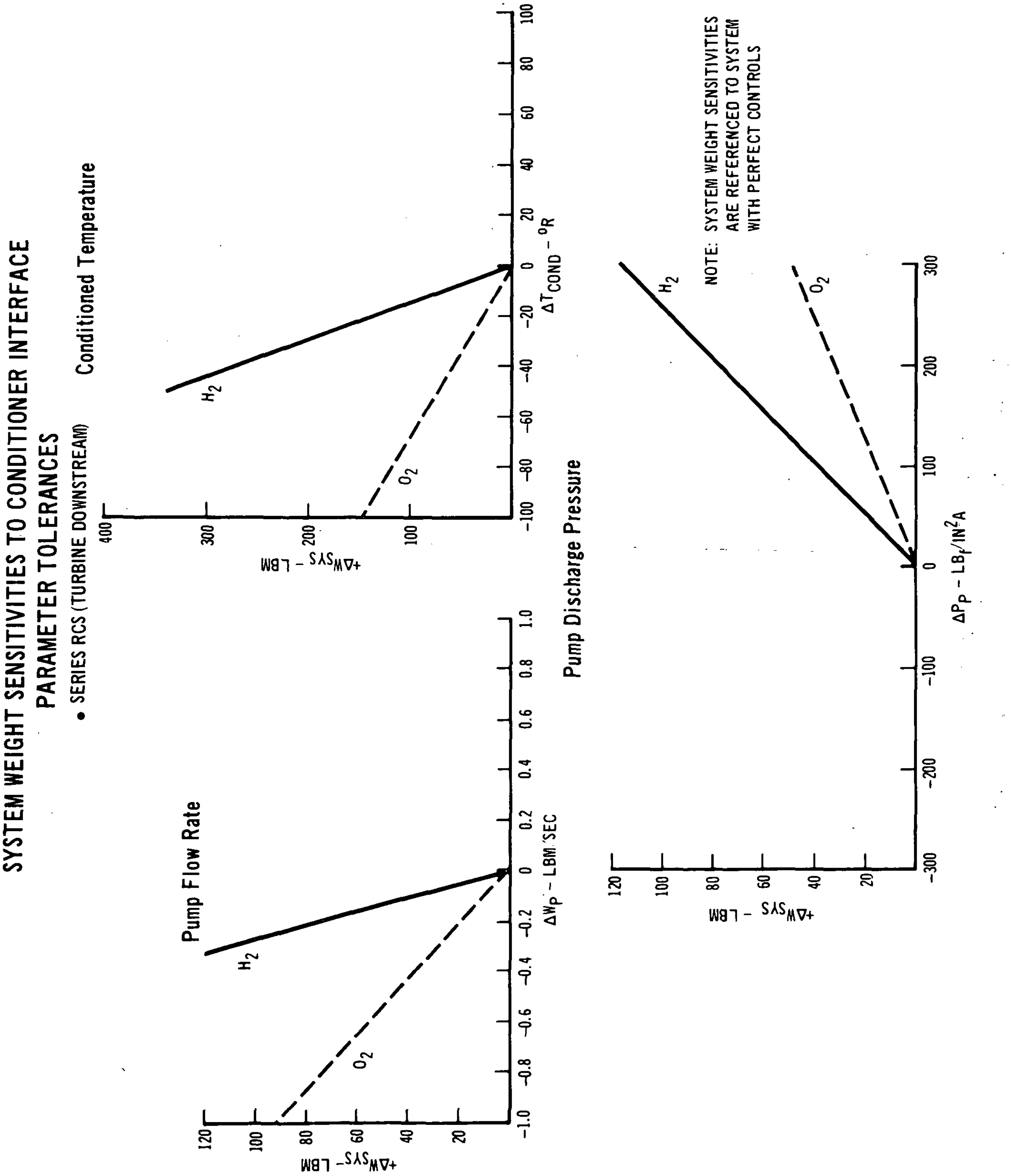
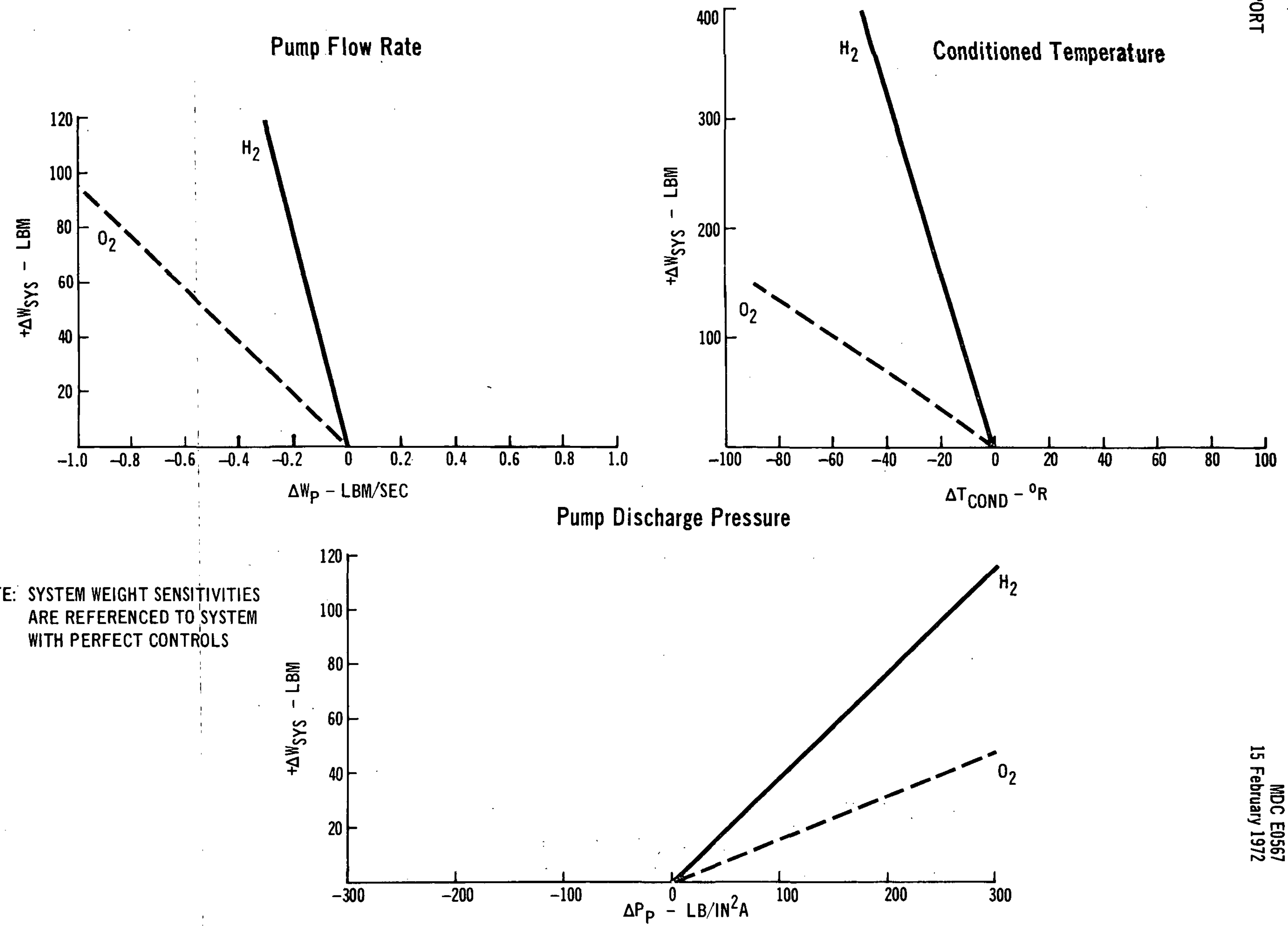

SYSTEM WEIGHT SENSITIVITIES TO CONDITIONER INTERFACE PARAMETER TOLERANCES
• SERIES RCS (TURBINE DOWNSTREAM)
Conditioned Temperature
$+\Delta W_{SYS}$ – LBM
400
300
200
100
ΔT_{COND} – °R
–100
–80
–60
–40
–20
0
20
40
60
80
100
H_2
O_2
Pump Flow Rate
$+\Delta W_{SYS}$ – LBM
120
100
80
60
40
20
ΔW_P – LBM/SEC
–1.0
–0.8
–0.6
–0.4
–0.2
0
0.2
0.4
0.6
0.8
1.0
H_2
O_2
Pump Discharge Pressure
NOTE: SYSTEM WEIGHT SENSITIVITIES ARE REFERENCED TO SYSTEM WITH PERFECT CONTROLS
$+\Delta W_{SYS}$ – LBM
120
100
80
60
40
20
ΔP_P – LB_f/IN^2A
–300
–200
–100
0
100
200
300
H_2
O_2

Figure C-21

SYSTEM WEIGHT SENSITIVITIES TO CONDITIONER INTERFACE PARAMETER TOLERANCES

• PARALLEL RCS



NOTE: SYSTEM WEIGHT SENSITIVITIES ARE REFERENCED TO SYSTEM WITH PERFECT CONTROLS

Figure C-22

IMPOSED CONTROL CONSTRAINTS AND SELECTION GUIDELINES

Constraints:

- Precision - Dictated by Sensor Accuracy
  - (a) Flow rate, ±2.58%
  - (b) Pressure, ±1.00%
  - (c) Temperature, ±1.00%
- Control Authority - Limited to ±50% valve area change to minimize controls coupling and impact on uncontrolled system variables.
- System Design - No redesign to improve control effectiveness.
- Same control concepts for both H_2 and O_2 conditioners.

Selection Guidelines:

Factor	Goal
o Weight	o Minimum system weight
o Precision and performance interactions	o Accurate control with minimum impact on uncontrolled variables
o Complexity	o Minimum computation requirements and controls coupling

GAS GENERATOR COMBUSTION TEMPERATURE CONTROL EFFECTIVENESS

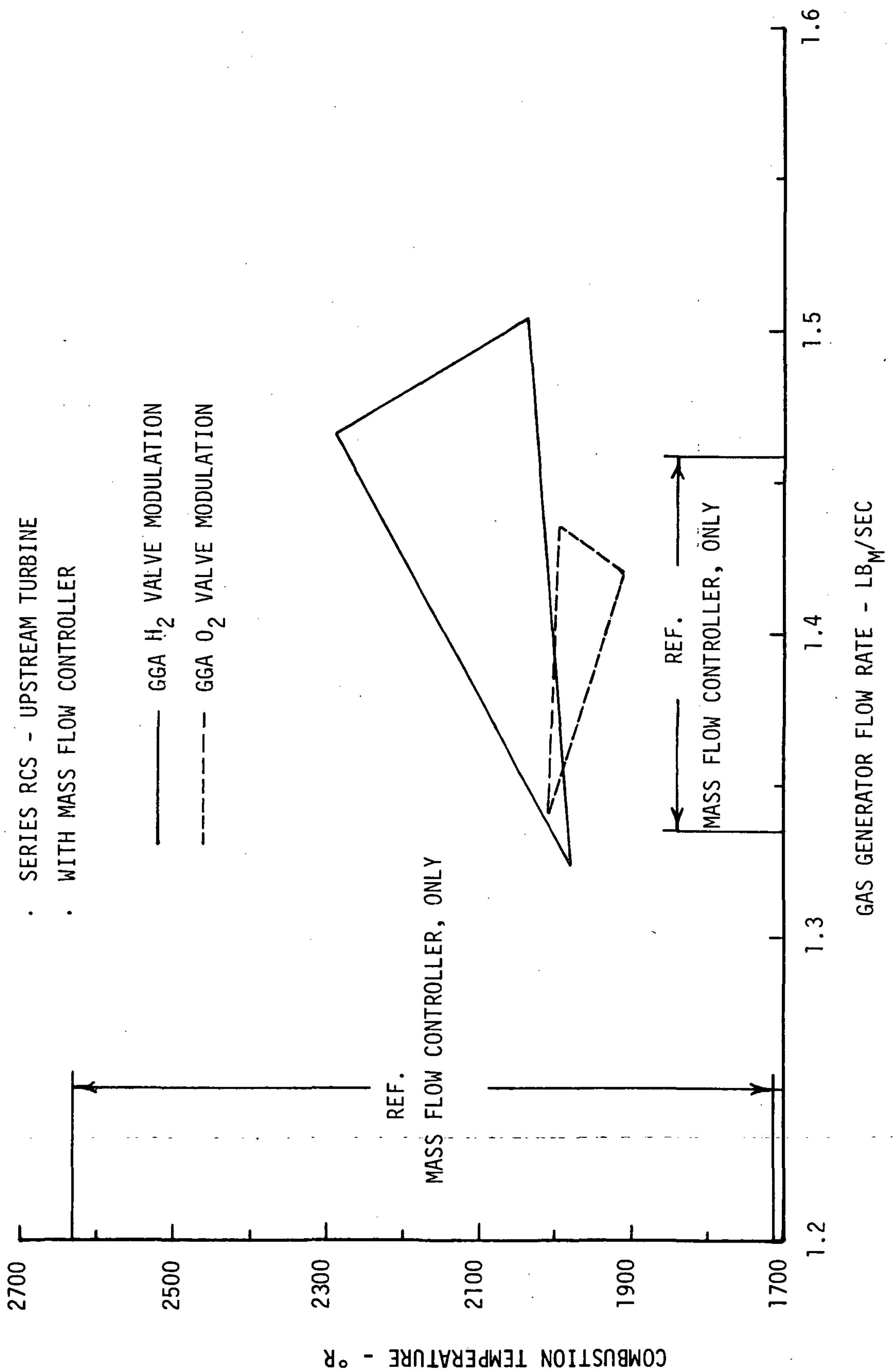


Figure C-24

The effect of active gas generator valve controls on the conditioner interface parameters is shown in Figures C-25 and C-26. The resulting impact on relative system weight is summarized in Figure C-27. As shown, O_2 valve modulation affords a slight weight advantage due to better control of conditioner temperature and maximum pump discharge pressure.

C6.2 Conditioned Propellant Temperature Control - Considering the sensitivities of Figures C-20 through C-22, it is seen that conditioned propellant temperature variations have the greatest impact on system weight. Altogether, seven active control concepts were evaluated for conditioned temperature control. These included the singular control (points 2 through 6 of Figure C-19), and two dual control options employing gas generator O_2 valve modulation for combustion temperature control. In the first dual control concept, the gas generator H_2 valve was also modulated for conditioned temperature control, while in the second, conditioned temperature control was effected by modulating the heat exchanger cold side bypass valve. Results of evaluations for all seven control options are presented in the conditioner operating maps of Figures C-28 through C-34 and the weight summary chart of Figure C-35.

Comparing the five singular control concepts of Figures C-28 through C-32, it is seen that the hot gas vent valve (Figure C-30) provides the poorest conditioned temperature control, and also widens the pump discharge pressure/flow rate operating bands beyond the open-loop limits. As such, this control was considered unattractive simply on the basis of system weight (Figure C-35). The gas generator discharge valve (Figure C-29) provides improved conditioned temperature control compared to the hot gas vent valve, and also narrows the operating bands on pump discharge pressure and flow rate. However, this control is not weight competitive with the remaining control concepts and requires hot gas (2000°R) valve modulation which was considered extremely undesirable. The pump discharge valve control (Figure C-31) provides the attractive features of liquid valve modulation coupled with good conditioned temperature control. However, these features are achieved through the sacrifice of increased operating bands on pump flow rate and discharge pressure. Because of the potential impact of cold side pressure variations on heat exchanger operating stability and the large weight penalty (Figure C-35), this control concept was not evaluated further. The above evaluations reduced the number of viable, single point propellant temperature control concepts to two: (1) gas generator H_2 valve modulation (Figure C-28), and (2) heat exchanger cold side bypass flow modulation (Figure C-32). The latter concept provides the best control of conditioned temperature, but requires use of a mixer downstream of the heat exchanger for the liquid bypass and heat exchanger exit flows.

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o COMBUSTION TEMPERATURE CONTROL
  (GAS GENERATOR H_2 VALVE MODULATION)

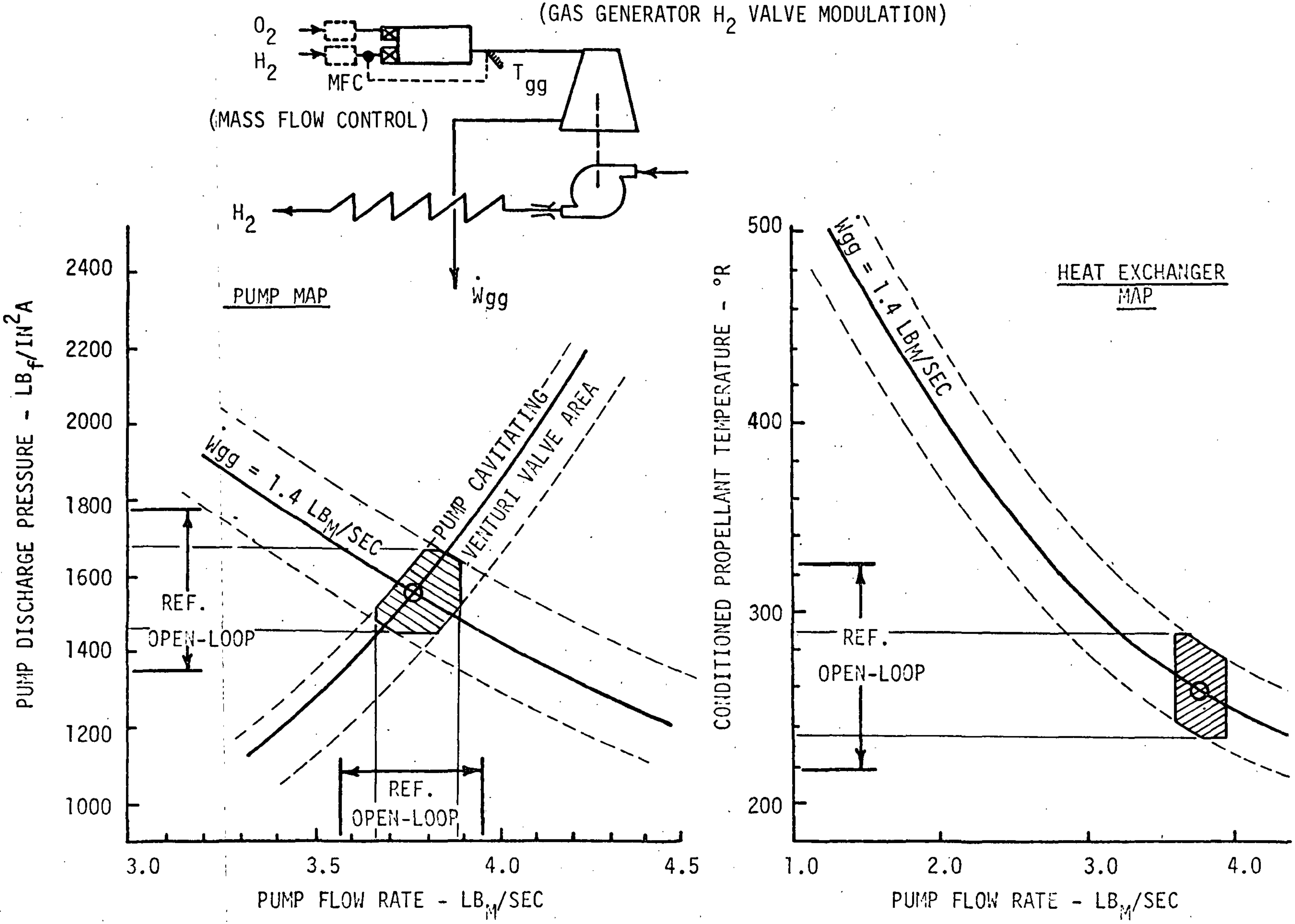


Figure C-25

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o COMBUSTION TEMPERATURE CONTROL (GAS GENERATOR O_2 VALVE MODULATION)

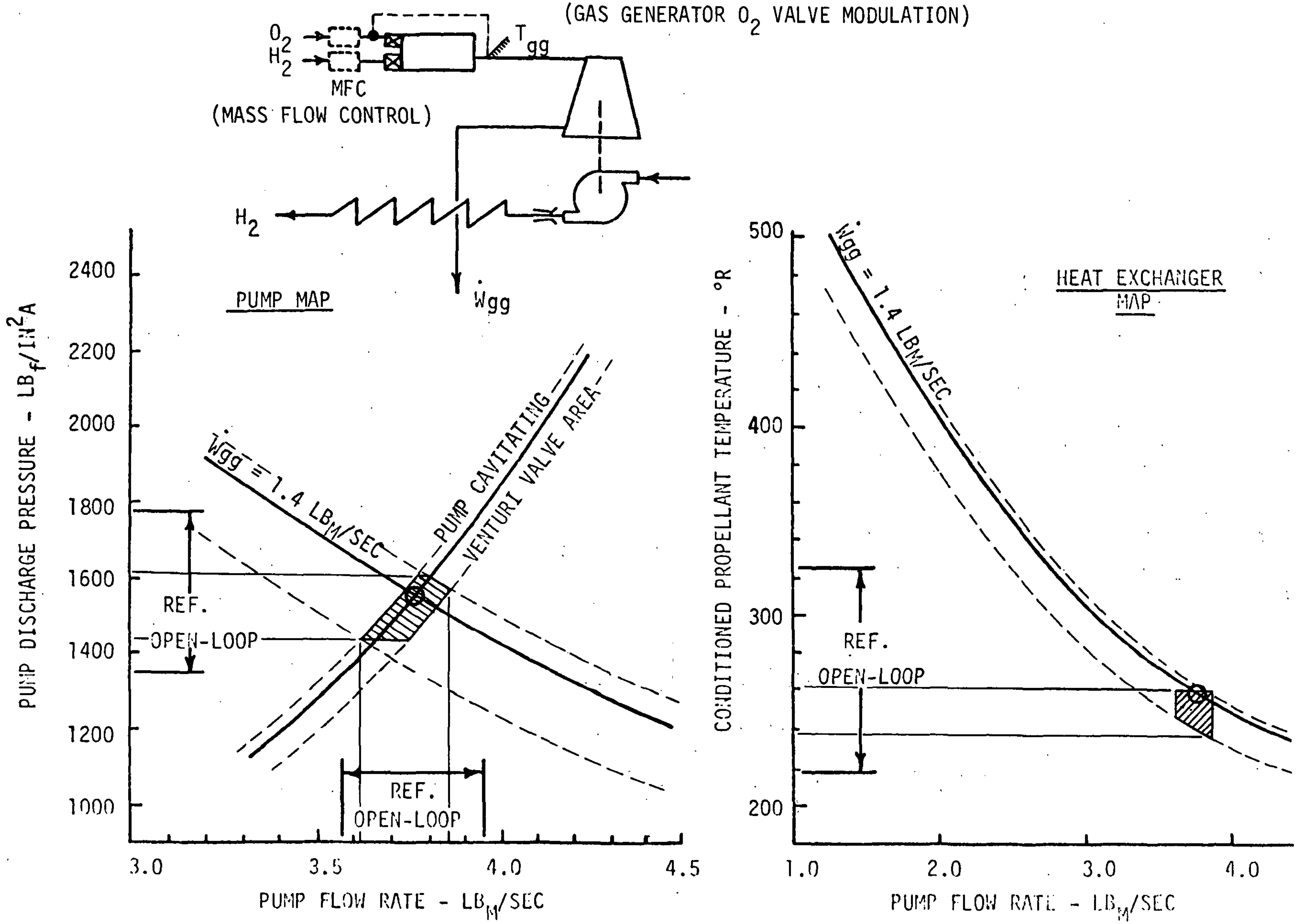


Figure C-26

WEIGHT SUMMARY CHART FOR COMBUSTION TEMPERATURE CONTROL

- Series- Upstream Turbine RCS
- With mass flow controller

Active Control Point	Prop	Nominal Control Value	Operating Band				ΔSystem Weight (lbm)*			
			$\dot{w}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{w}_p$-Tol	Pd-Tol	T_{cond}-Tol	Total
None	H_2		3.67-3.86	1465-1714	230-312	1714-2631	29	63	257	349 } 574
	O_2		11.23-12.46	1748-2138	425-624	1716-2634	43	39	143	225
GGA O_2 Valve	H_2	2000	3.62-3.85	1422-1605	238-263	1913-2008	42	23	195	260 } 423
	O_2	2000	11.11-12.15	1701-1964	455-511	1914-2009	54	11	98	163
GGA H_2 Valve	H_2	2000	3.66-3.89	1458-1665	236-287	1980-2289	32	45	210	287 } 470
	O_2	2000	11.26-12.42	1725-2085	447-547	1919-2291	41	30	112	183

*Referenced to System with Perfect Control

Figure C-27

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o CONDITIONED TEMPERATURE CONTROL (GAS GENERATOR H_2 VALVE MODULATION)

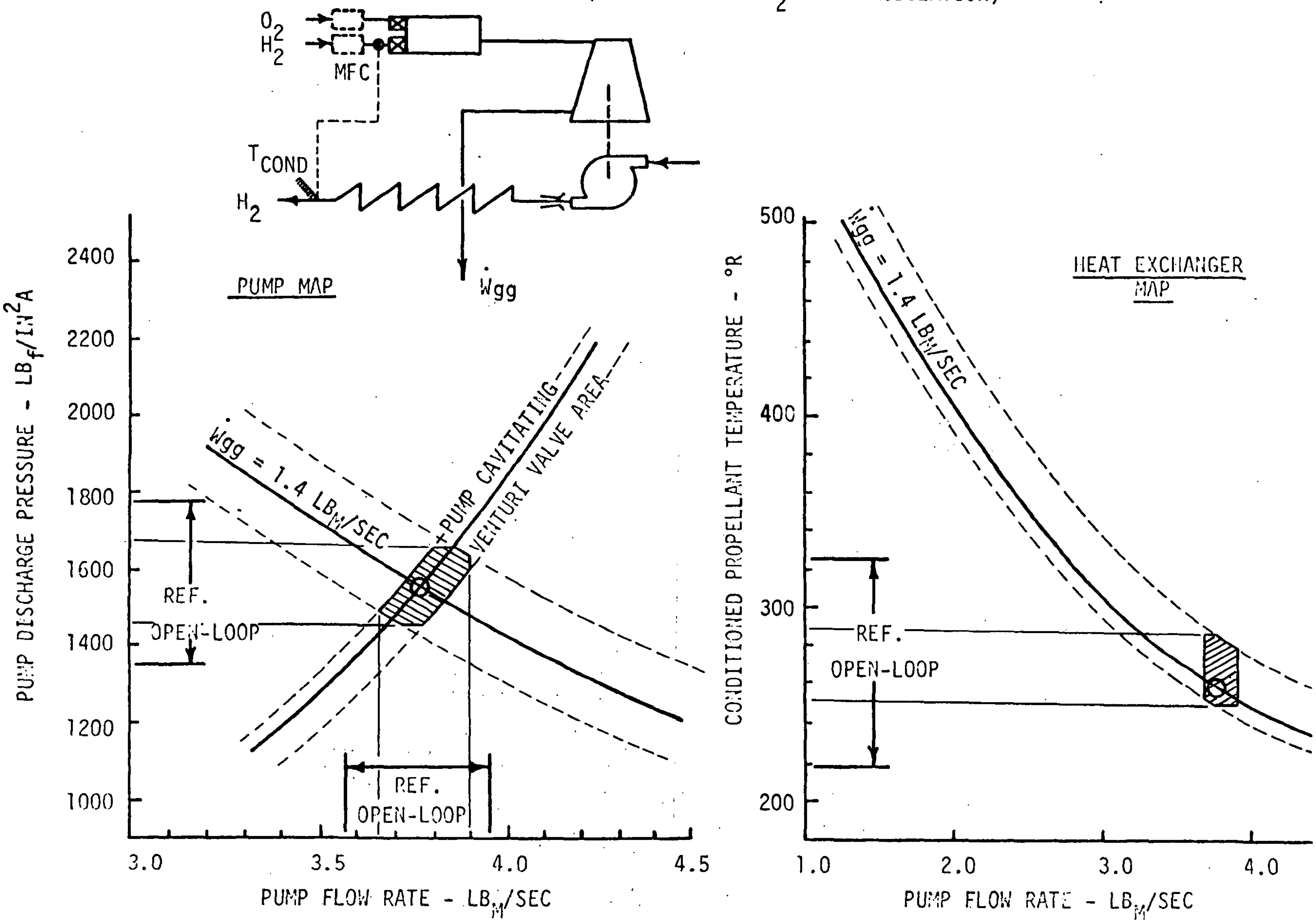


Figure C-28

CONDITIONER CONTROLS EVALUATION

- SERIES-UPSTREAM TURBINE RCS
- CONDITIONED TEMPERATURE CONTROL (GAS GENERATOR DISCHARGE VALVE MODULATION)

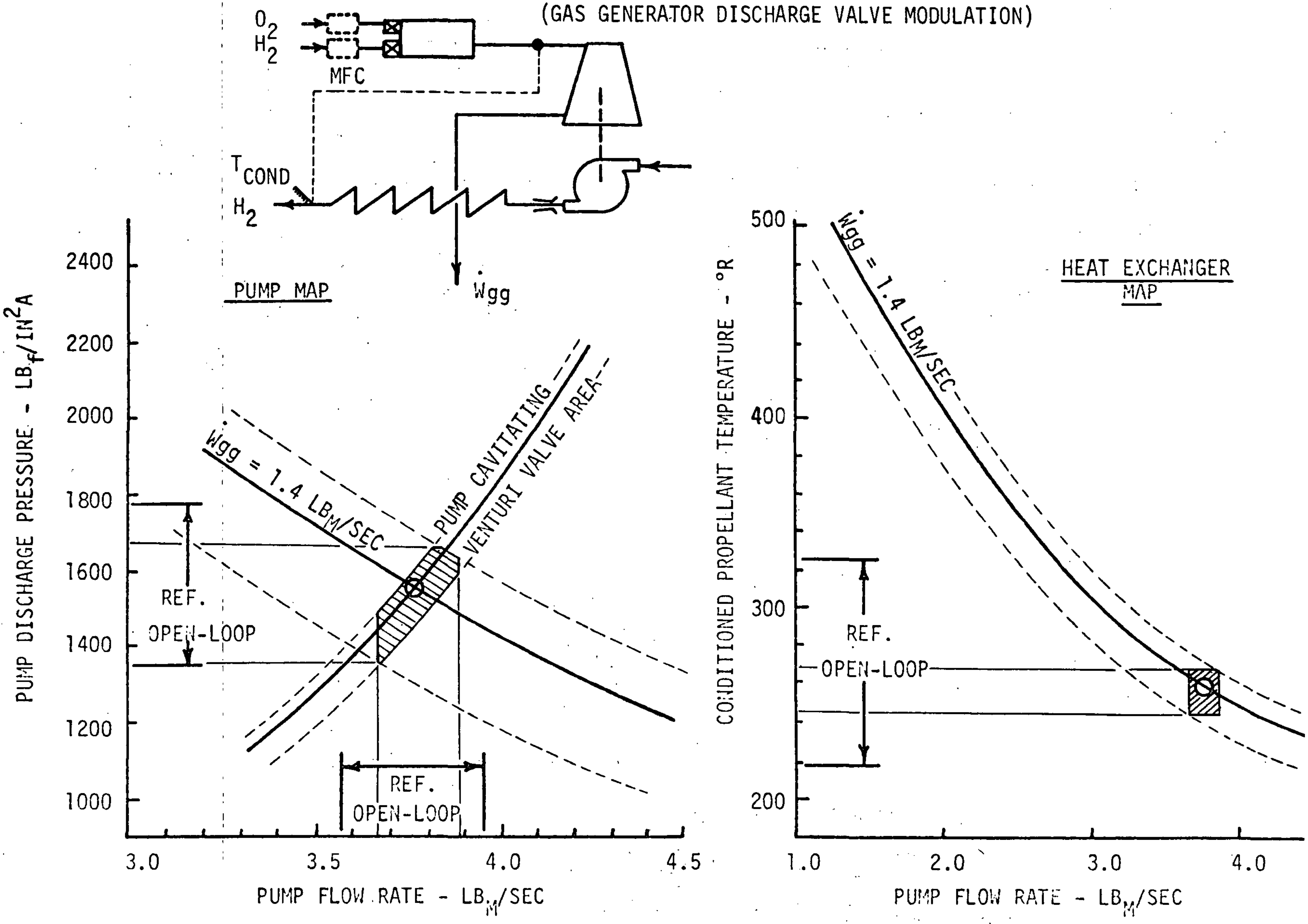


Figure C-29

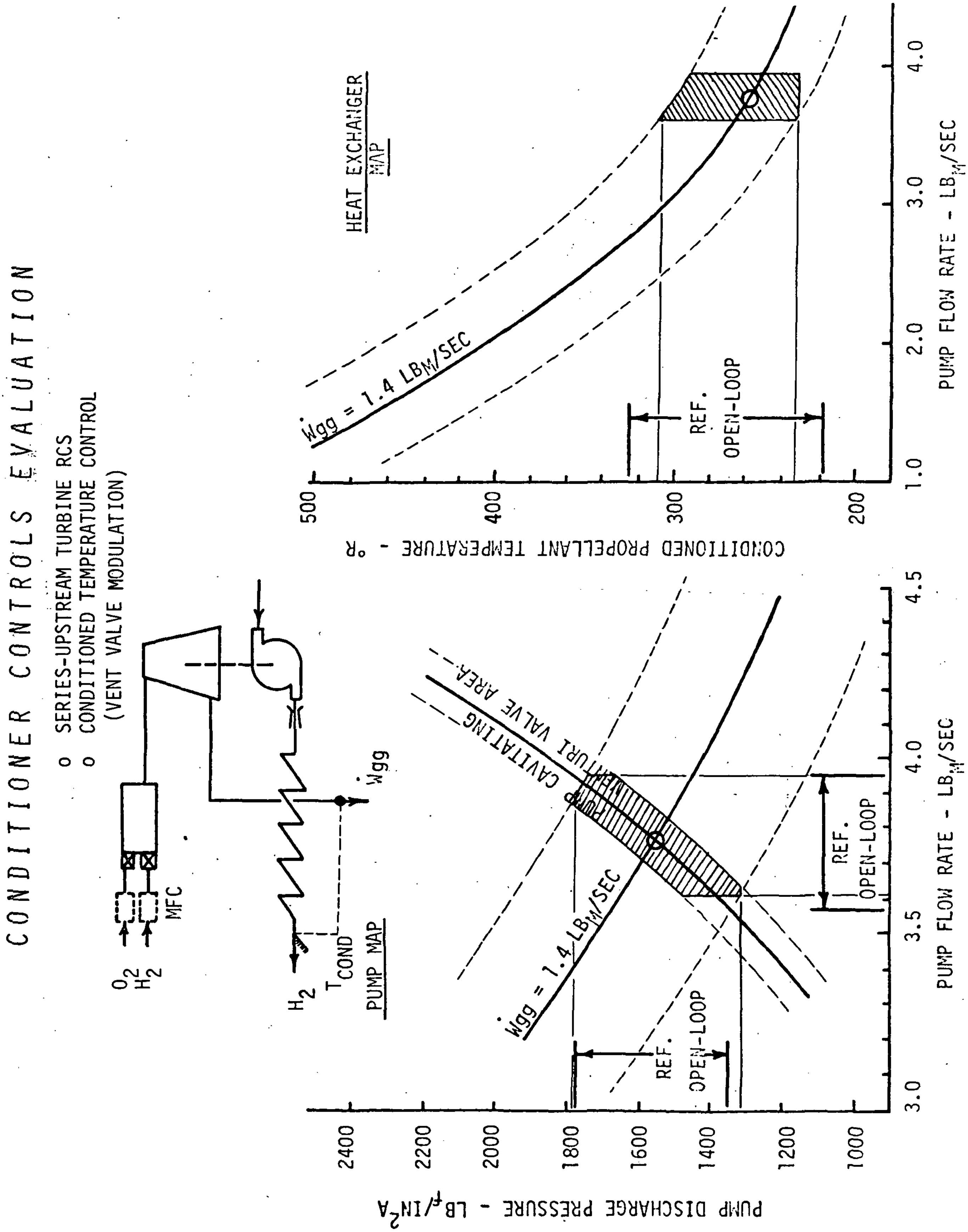

CONDITIONER CONTROLS EVALUATION
o SERIES-UPSTREAM TURBINE RCS
o CONDITIONED TEMPERATURE CONTROL
(VENT VALVE MODULATION)
O2
H2
MFC
Wgg
H2
TCOND
PUMP MAP
HEAT EXCHANGER MAP
Wgg = 1.4 LBM/SEC
REF.
OPEN-LOOP
PUMP FLOW RATE - LBM/SEC
CONDITIONED PROPELLANT TEMPERATURE - °R
PUMP CAVITATING VENTURI VALVE AREA
PUMP DISCHARGE PRESSURE - LBf/IN2A
1.0
2.0
3.0
4.0
500
400
300
200
3.0
3.5
4.0
4.5
2400
2200
2000
1800
1600
1400
1200
1000

Figure C-30

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o CONDITIONED TEMPERATURE CONTROL

(PUMP DISCHARGE VALVE MODULATION)

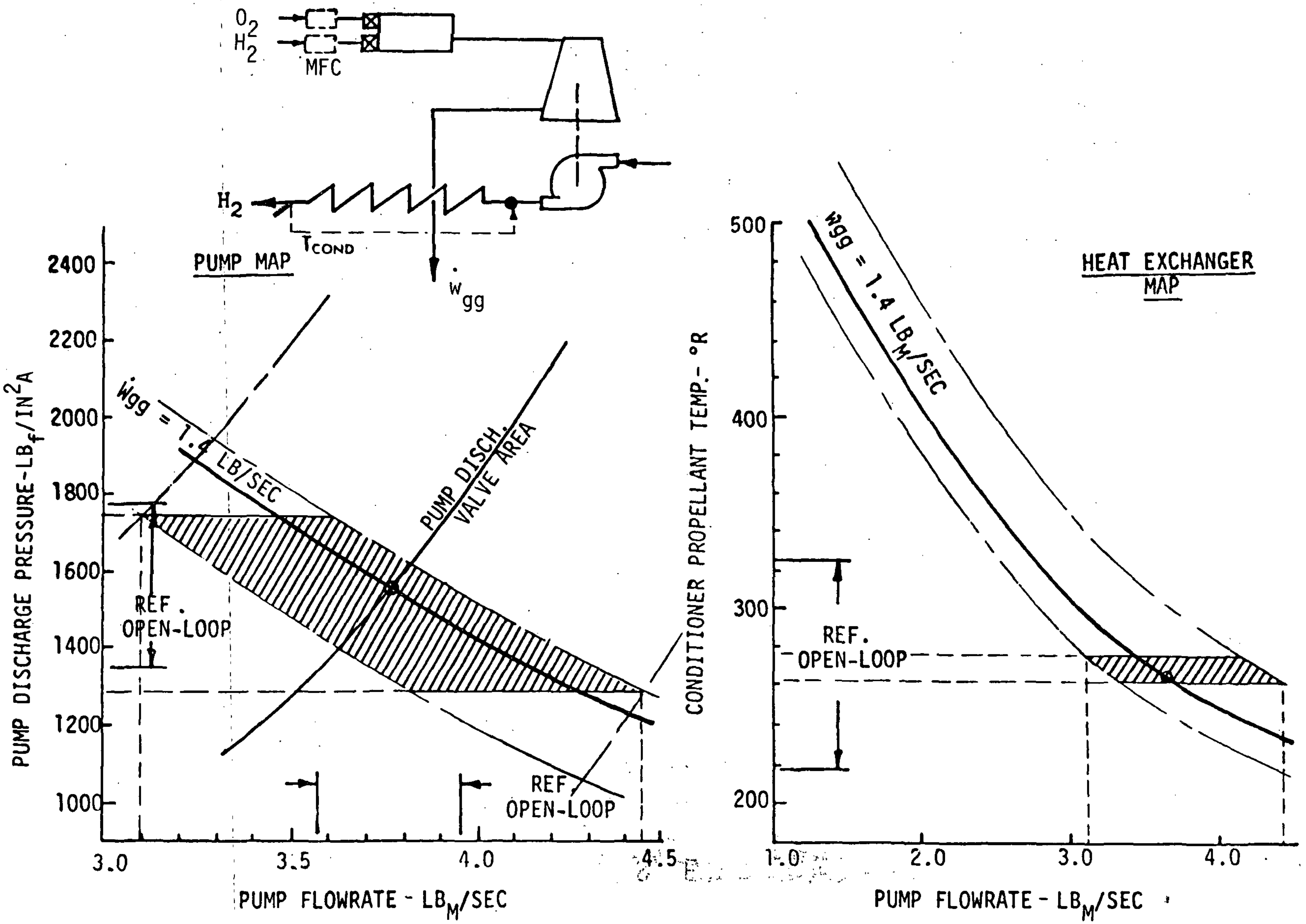


Figure C-31

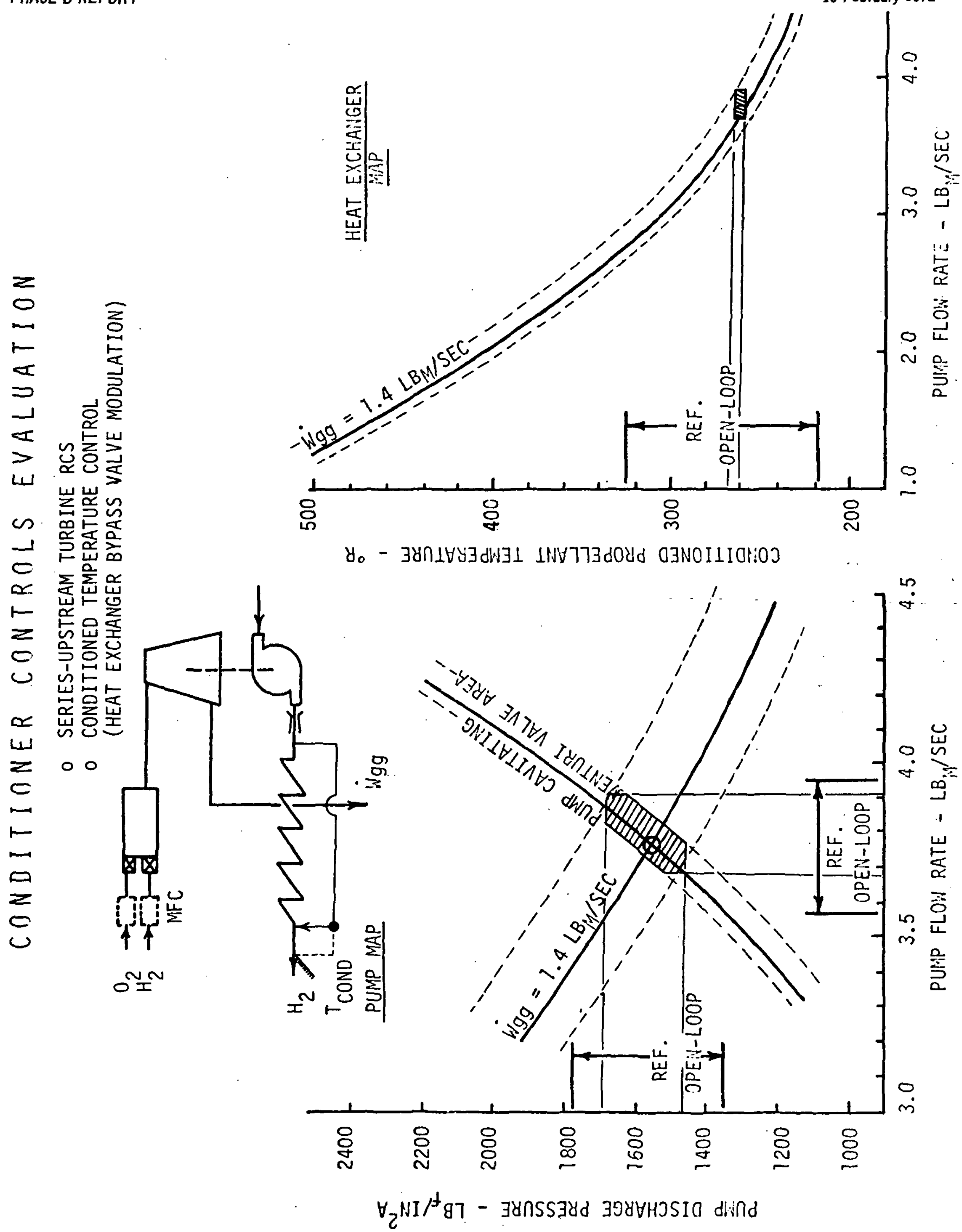
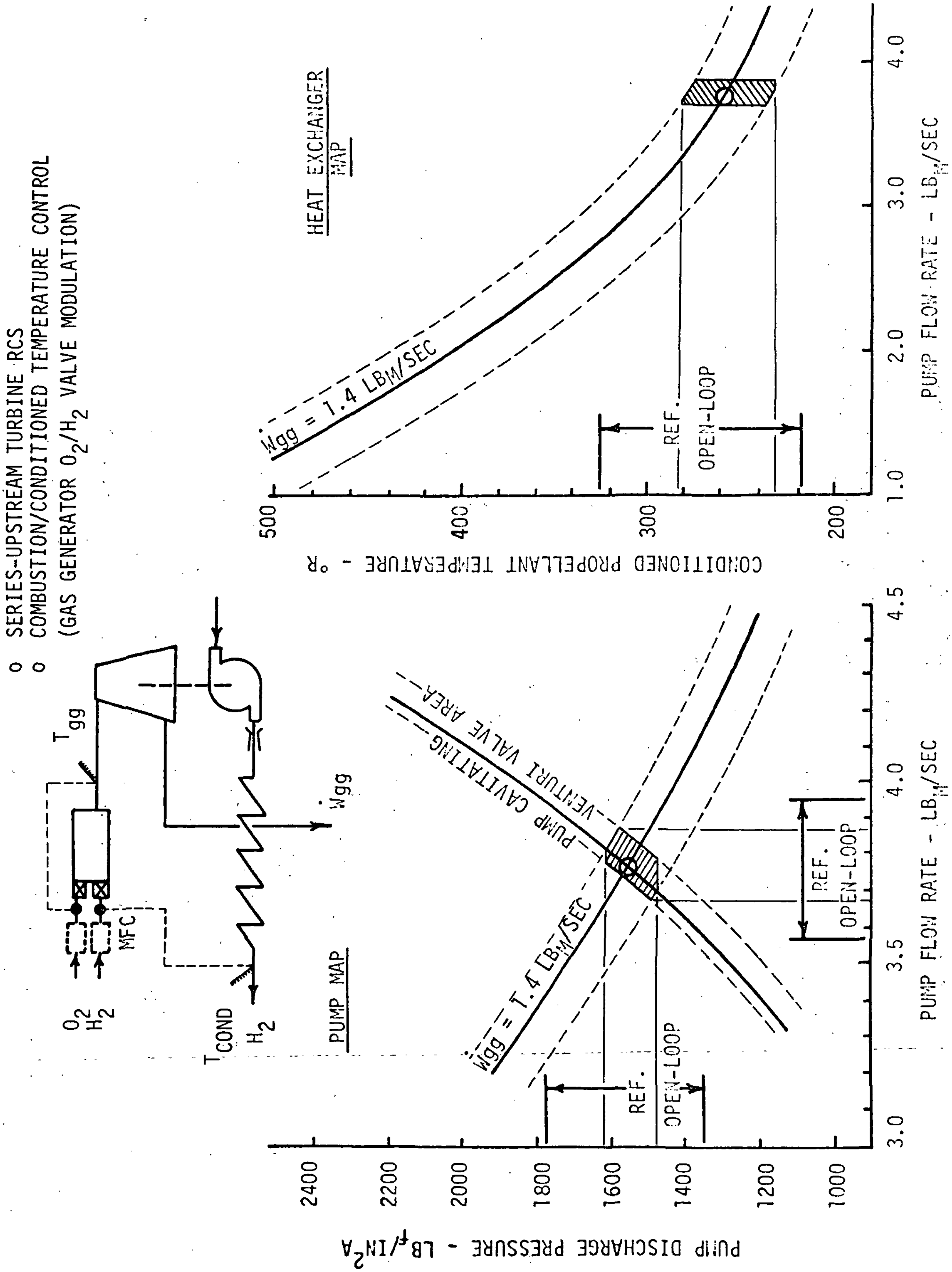

CONDITIONER CONTROLS EVALUATION
o SERIES-UPSTREAM TURBINE RCS
o CONDITIONED TEMPERATURE CONTROL
(HEAT EXCHANGER BYPASS VALVE MODULATION)
O2
H2
MFC
H2
TCOND
Wgg
PUMP MAP
HEAT EXCHANGER MAP
Wgg = 1.4 LBM/SEC
REF.
OPEN-LOOP
CONDITIONED PROPELLANT TEMPERATURE - °R
500
400
300
200
1.0
2.0
3.0
4.0
PUMP FLOW RATE - LBM/SEC
PUMP CAVITATING
VENTURI VALVE AREA
PUMP DISCHARGE PRESSURE - LBf/IN2A
2400
2200
2000
1800
1600
1400
1200
1000
3.0
3.5
4.0
4.5

Figure C-32

CONDITIONER CONTROLS EVALUATION



C-42

Figure C-33

CONDITIONER CONTROLS EVALUATION

- SERIES-UPSTREAM TURBINE RCS
- COMBUSTION/CONDITIONED TEMPERATURE CONTROL
  (GAS GENERATOR O_2 VALVE/HEAT EXCHANGER BYPASS MODULATION)

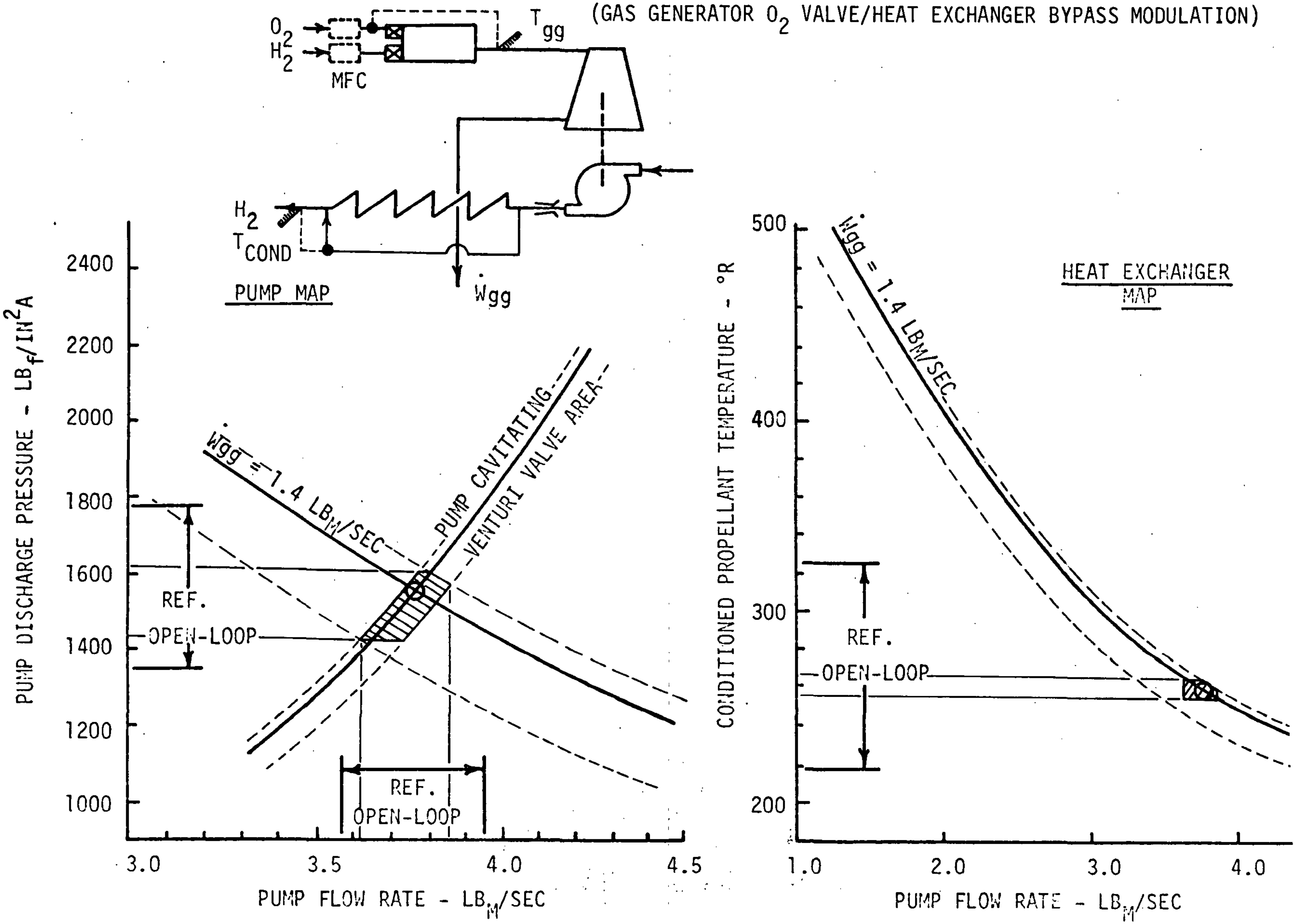


Figure C-34

WEIGHT SUMMARY CHART FOR CONDITIONED TEMPERATURE CONTROL

- o Series-Upstream Turbine RCS
- o With Mass Flow Controller

Control Point	Prop	Nominal Control Value, °R	Operating Band: $\dot{w}_p$ lbm/sec	Operating Band: P_d lbf/in.^{2}a	Operating Band: T_{cond} °R	Operating Band: T_{GG} °R	ΔSystem Weight (Lbm)*: $\dot{w}_p$-Tol	ΔSystem Weight (Lbm)*: P_d-Tol	ΔSystem Weight (Lbm)*: T_{cond}-Tol	ΔSystem Weight (Lbm)*: Total
None	H_2 O_2		3.67-3.86 11.23-12.46	1465-1714 1748-2138	230-312 425-624	1714-2631 1716-2634	29 43	63 39	257 143	349 225 } 574
GGA H_2 Valve	H_2 O_2	263 506	3.66-3.89 11.26-12.43	1453-1666 1724-2095	252-287 494-574	2061-2377 2046-2378	32 41	45 32	86 27	163 100 } 263
GGA Discharge Valve	H_2 O_2	263 506	3.66-3.89 11.20-12.41	1367-1668 1688-2109	244-266 451-513	1706-2374 1708-2379	32 46	46 35	150 105	228 186 } 414
Hot Gas Vent Valve	H_2 O_2	263 506	3.61-3.95 11.05-13.19	1318-1754 1597-2420	233-309 444-600	1714-2631 1716-2634	58 60	78 86	233 116	369 262 } 631
Pump Discharge Valve	H_2 O_2	263 506	3.10-4.45 9.35-13.99	1284-1745 1621-2237	261-276 501-539	1714-2631 1716-2634	140 220	75 56	20 10	235 286 } 521
Hex Cold Side By-pass Valve	H_2 O_2	263 506	3.67-3.91 11.23-12.46	1465-1714 1748-2138	261-266 501-511	1714-2631 1716-2634	29 43	63 39	20 10	112 92 } 204
GGA O_2 Valve + GGA H_2 Valve**	H_2	263	3.67-3.86 11.30-12.20	1466-1615 1737-2009	231-282 440-567	1807-2351 1808-2353	29 38	26 18	250 123	305 179 } 484
GGA O_2 Valve + Hex By-pass Valve	H_2	263	3.62-3.85 11.11-12.15	1422-1606 1701-1964	261-266 501-511	1913-2008 1914-2009	42 54	23 11	20 7	85 72 } 157

* Referenced to system with perfect control. **GGA O_2 valve used for combustion temperature control. (H_2 and O_2 valve area bands restricted to ±10% and ±30% of nominal, respectively, to preclude excessive combustion temperature variation.)

Due to the attractiveness of these last two singular control concepts, both were evaluated in conjunction with gas generator 0_2 valve modulation for combustion temperature control. Conditioner operating maps for these two dual control concepts are presented in Figures C-33 and C-34. It is noteworthy that when both gas generator propellant valves are modulated for temperature control (Figure C-33), the valve areas must be constrained to $\pm 10\%$ and $\pm 30\%$ of nominal for H_2 and 0_2, respectively. This is necessary to preclude excessive variations in gas generator combustion temperature. Comparing the results of Figures C-33 and C-34, it is seen that the heat exchanger bypass with gas generator 0_2 valve modulation provides the best control of conditioned temperature. Furthermore, from Figure C-35, it is seen that this control concept provides minimum system weight.

C6.3 Pump Discharge Pressure/Flow Rate Control - Control of pump discharge pressure and flow rate is desirable to preclude the possibility of heat exchanger flow instability resulting from variable cold side inlet conditions. Four singular control options (points 2 through 5 of Figure C-19) and one dual control option (point 2 in conjunction with control point 1) were evaluated to determine their effectiveness in controlling both pump flow rate and discharge pressure. Primary results of these evaluations are presented in the conditioner operating maps of Figures C-36 through C-43 and the weight summary charts of Figures C-44 and C-45.

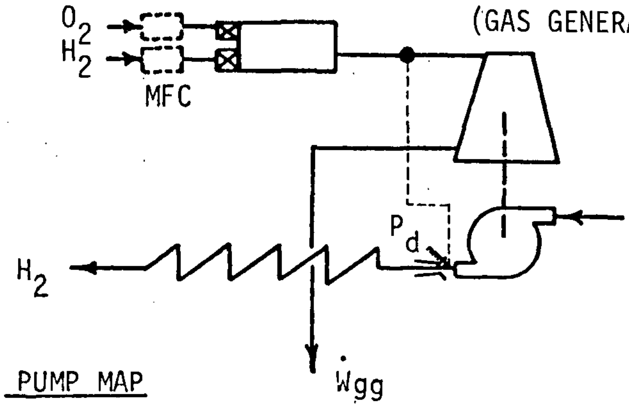
Modulation of the gas generator discharge valve (control point 3 shown in Figures C-36 and C-37) produces variations in both turbine flow rate and pressure ratio, which in turn varies shaft power delivered to the pump. If the valve is throttled to reduce pump flow rate or discharge pressure, less energy is available for thermally conditioning propellant flow in the heat exchanger. Hence, control of pump flow rate or discharge pressure is achieved at the expense of additional variation in conditioned propellant temperature. This characteristic is undesirable since overall system weight is extremely sensitive to conditioned temperature. Because of this characteristic and the further unattractiveness of modulating a 2000°R hot gas valve, the gas generator discharge valve was not considered further.

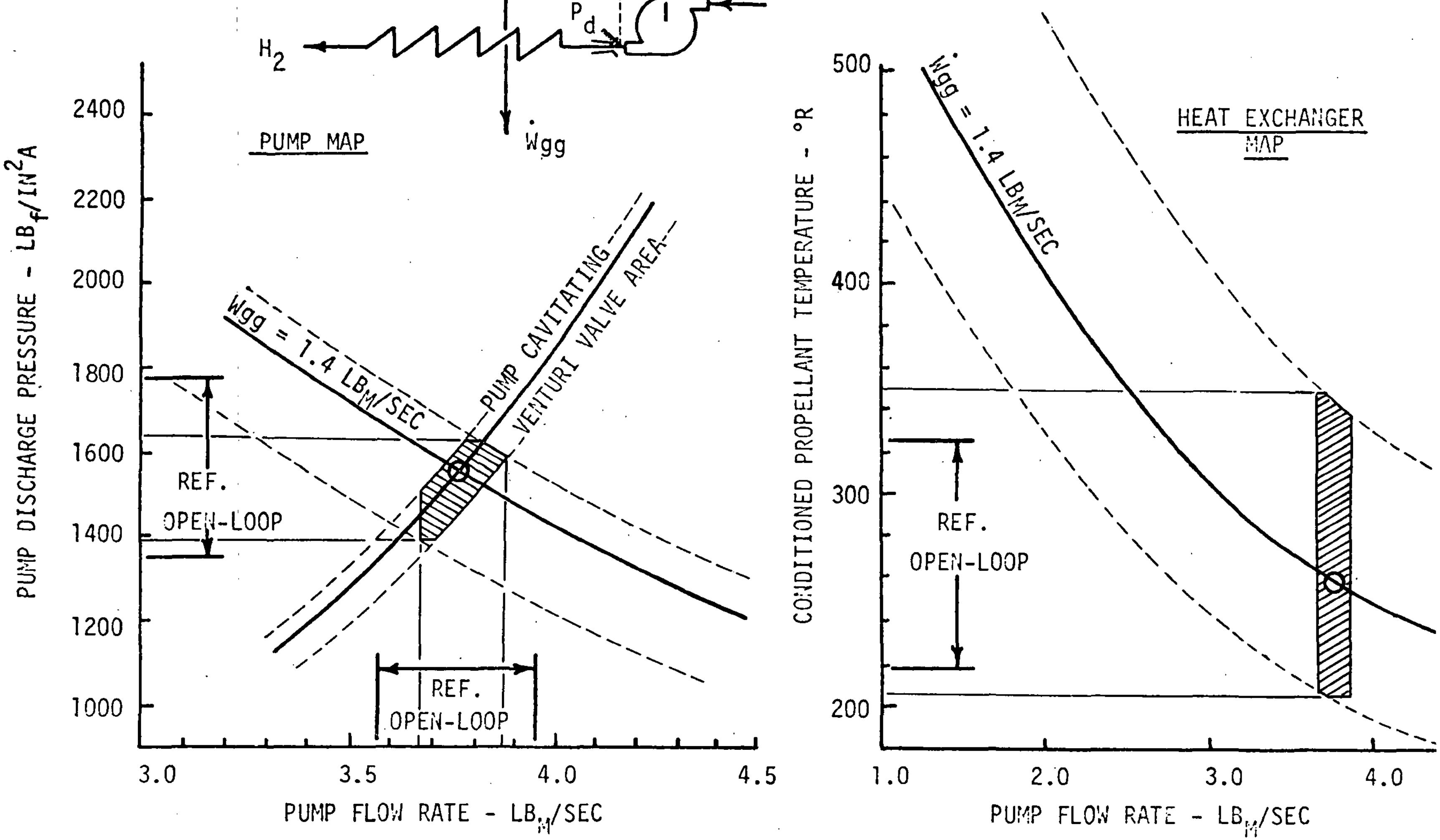
Use of the pump discharge valve (control point 5) for flow or pressure control is seemingly more desirable since it involves liquid throttling. However, as shown in Figures C-38 and C-39, this control is self-defeating. Since turbine delivered shaft power (pump flow rate and pressure ratio) is not effected by pump discharge valve modulation, pump hydraulic power remains constant. Therefore, decreasing discharge valve area for the purpose of reducing pump flow rate, causes a corresponding increase in discharge pressure. Because control of pump flow

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o CONDITIONED PRESSURE CONTROL

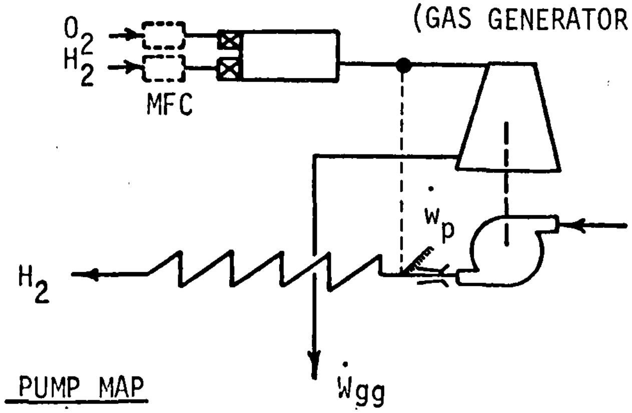
(GAS GENERATOR DISCHARGE VALVE MODULATION)

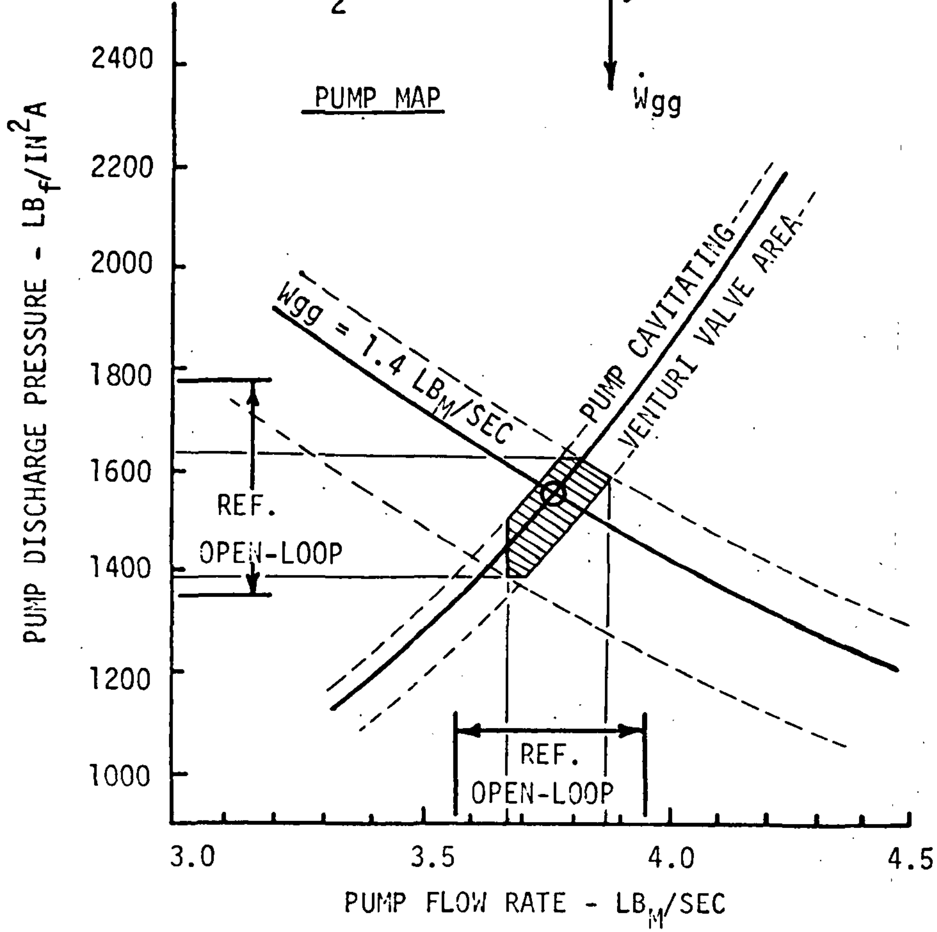


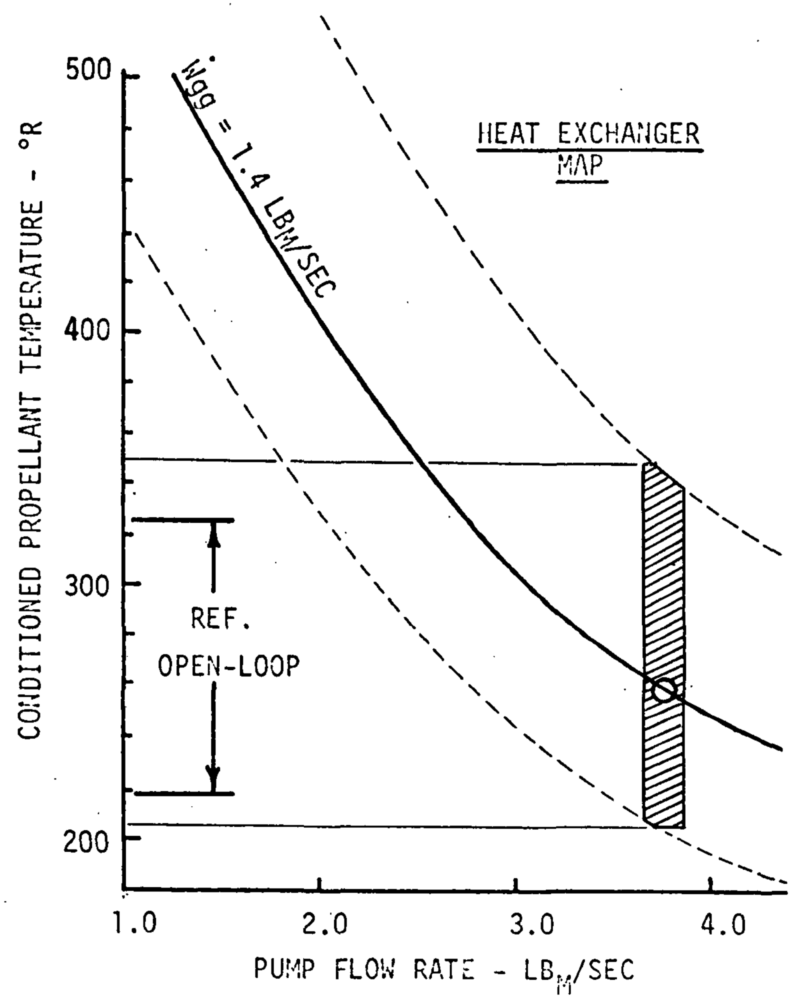


CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o PUMP FLOW RATE CONTROL
  (GAS GENERATOR DISCHARGE VALVE MODULATION)







CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o CONDITIONED PRESSURE CONTROL (PUMP DISCHARGE VALVE MODULATION)

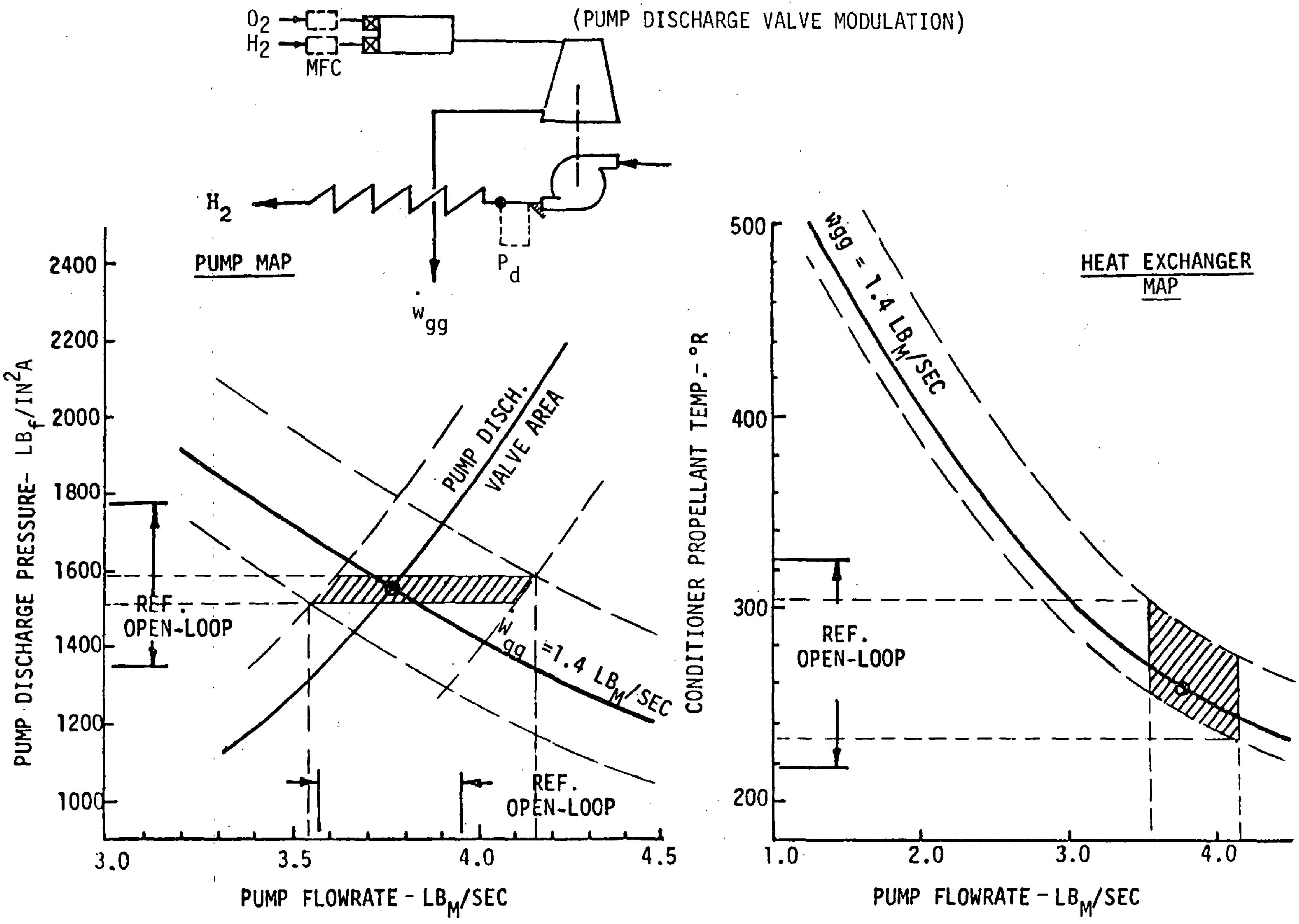


Figure C-38

CONDITIONER CONTROLS EVALUATION

- SERIES-UPSTREAM TURBINE RCS
- PUMP FLOW RATE CONTROL

(PUMP DISCHARGE VALVE MODULATION)

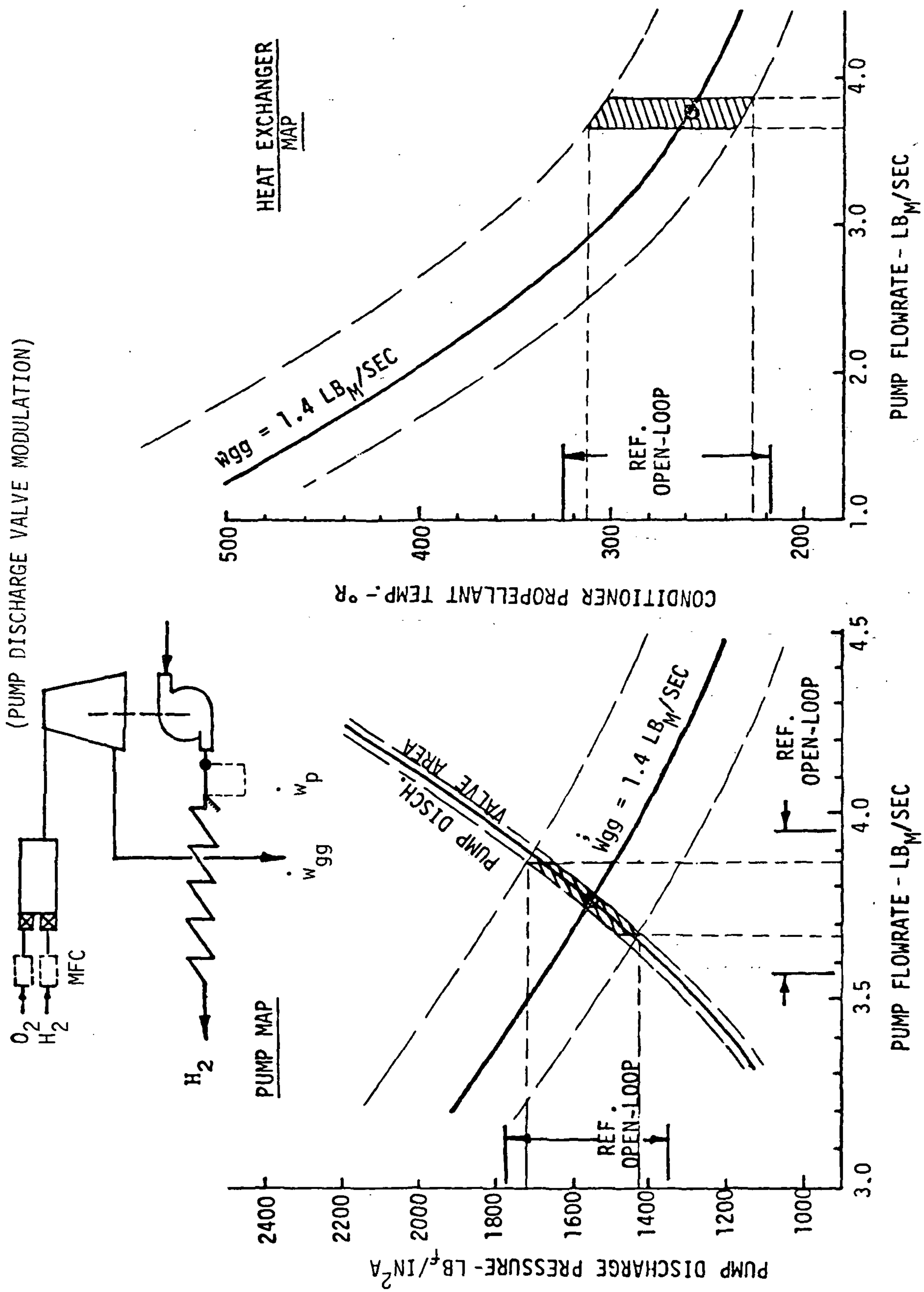
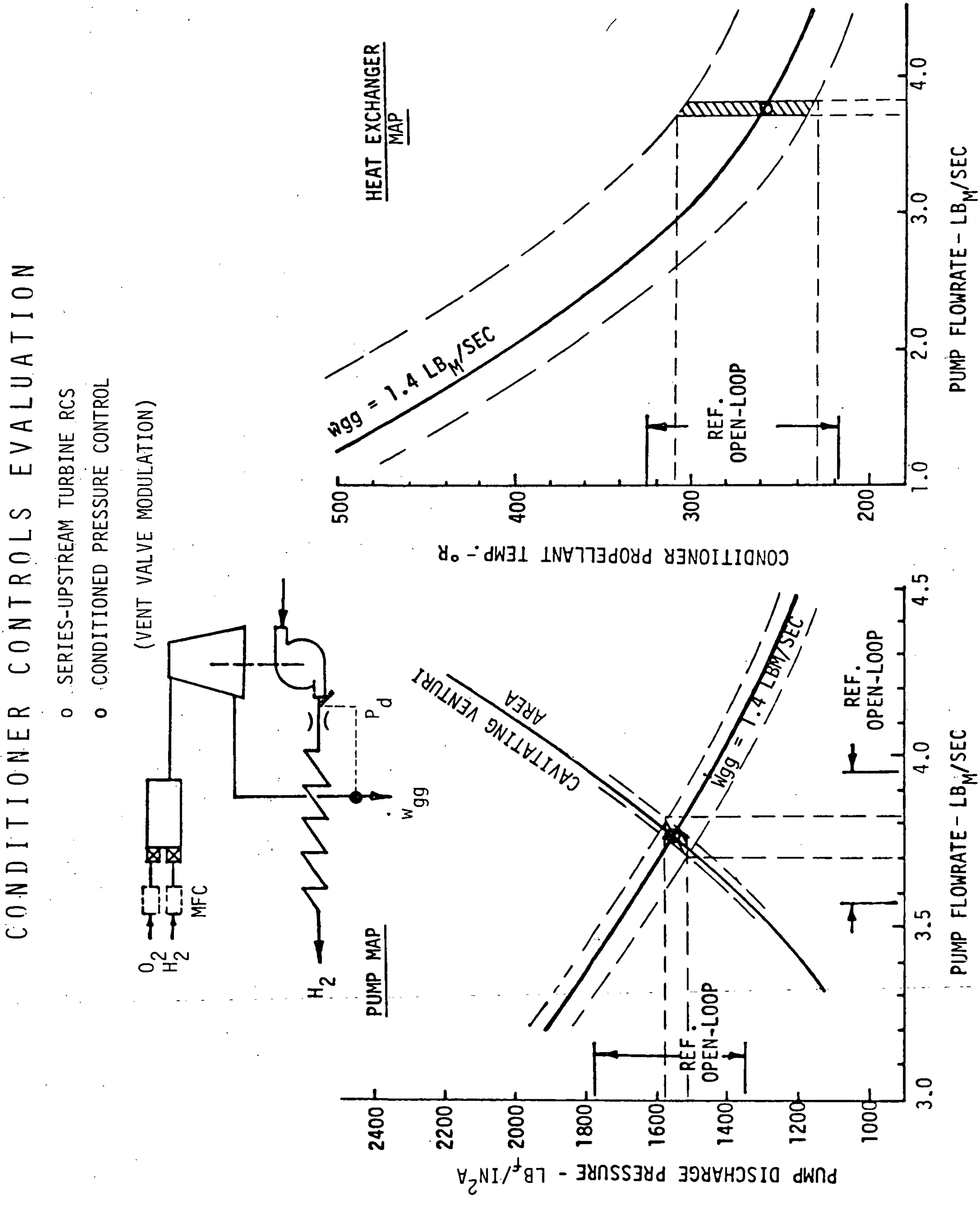


Figure C-39


CONDITIONER CONTROLS EVALUATION
o SERIES-UPSTREAM TURBINE RCS
o CONDITIONED PRESSURE CONTROL
(VENT VALVE MODULATION)
O2
H2
MFC
Pd
ẇgg
H2
PUMP MAP
CAVITATING VENTURI AREA
Wgg = 1.4 LBM/SEC
REF. OPEN-LOOP
REF. OPEN-LOOP
PUMP DISCHARGE PRESSURE - LBf/IN2A
2400
2200
2000
1800
1600
1400
1200
1000
3.0
3.5
4.0
4.5
PUMP FLOWRATE - LBM/SEC
HEAT EXCHANGER MAP
Wgg = 1.4 LBM/SEC
REF. OPEN-LOOP
CONDITIONER PROPELLANT TEMP - °R
500
400
300
200
1.0
2.0
3.0
4.0
PUMP FLOWRATE - LBM/SEC

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o PUMP FLOW RATE CONTROL

  (VENT VALVE MODULATION)

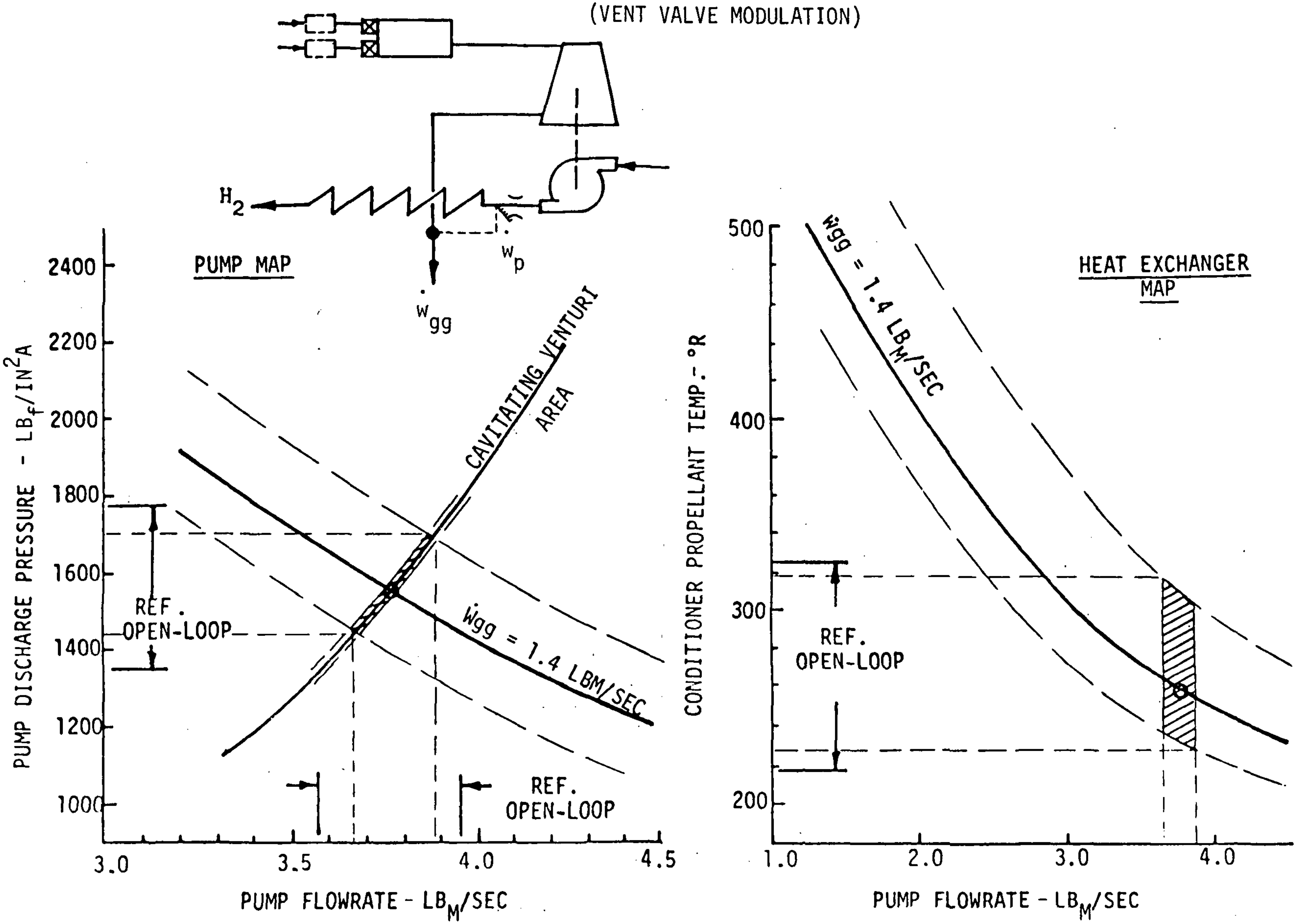
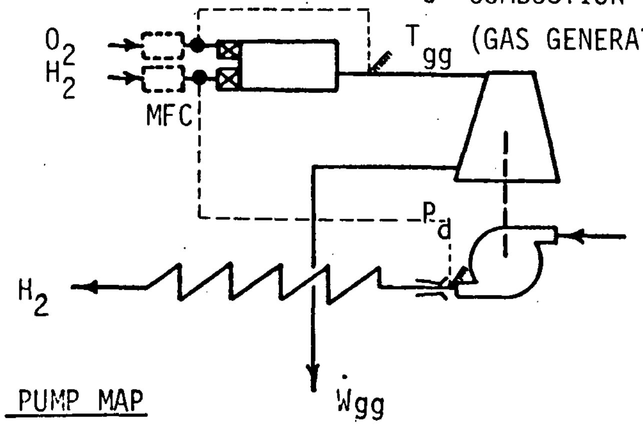


Figure C-41

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o COMBUSTION TEMPERATURE/CONDITIONED PRESSURE CONTROL (GAS GENERATOR O_2/H_2 VALVE MODULATION)



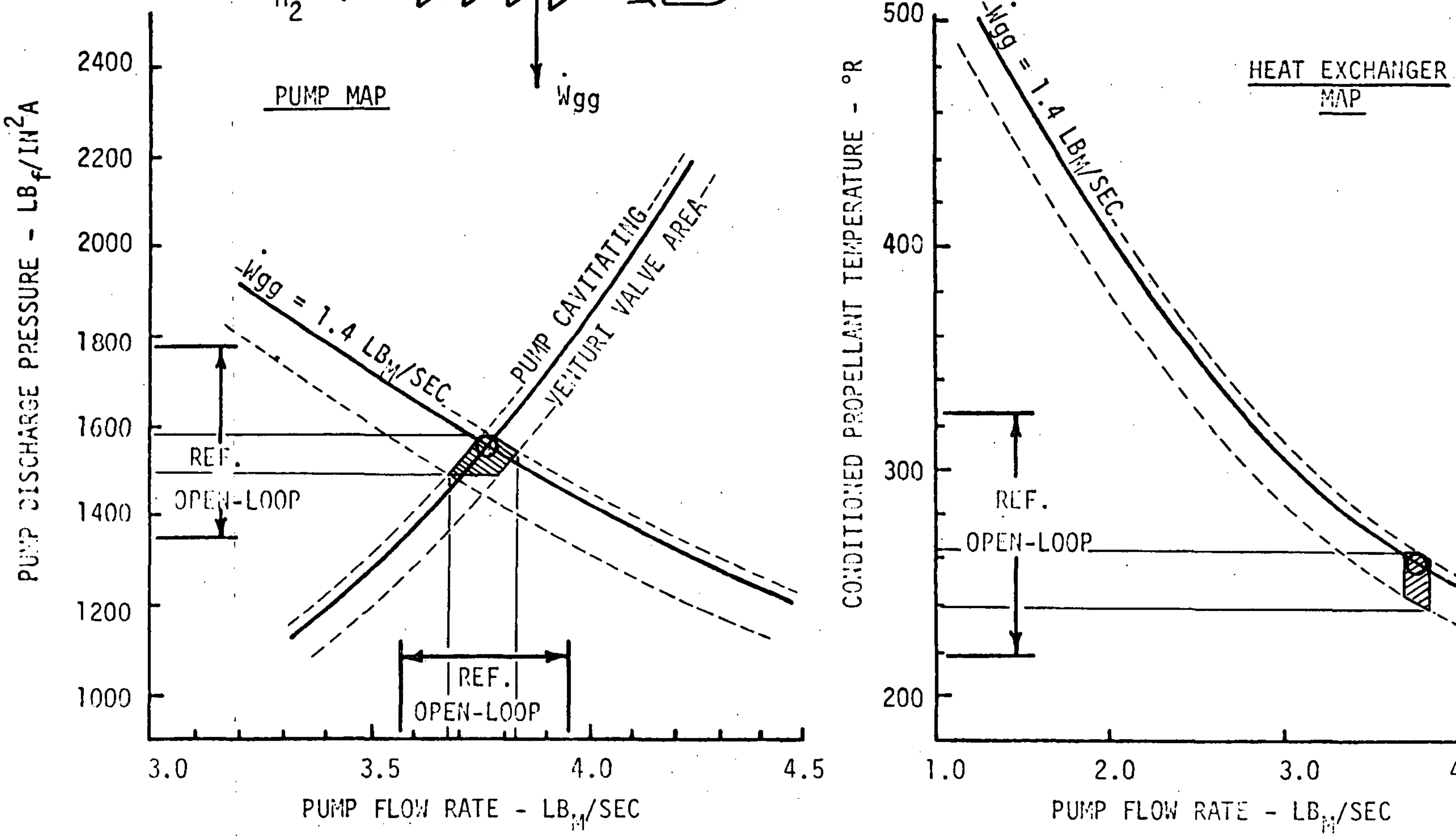


Figure C-42

CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o COMBUSTION TEMPERATURE/PUMP FLOW RATE CONTROL
  T_{gg} (GAS GENERATOR O_2/H_2 VALVE MODULATION)

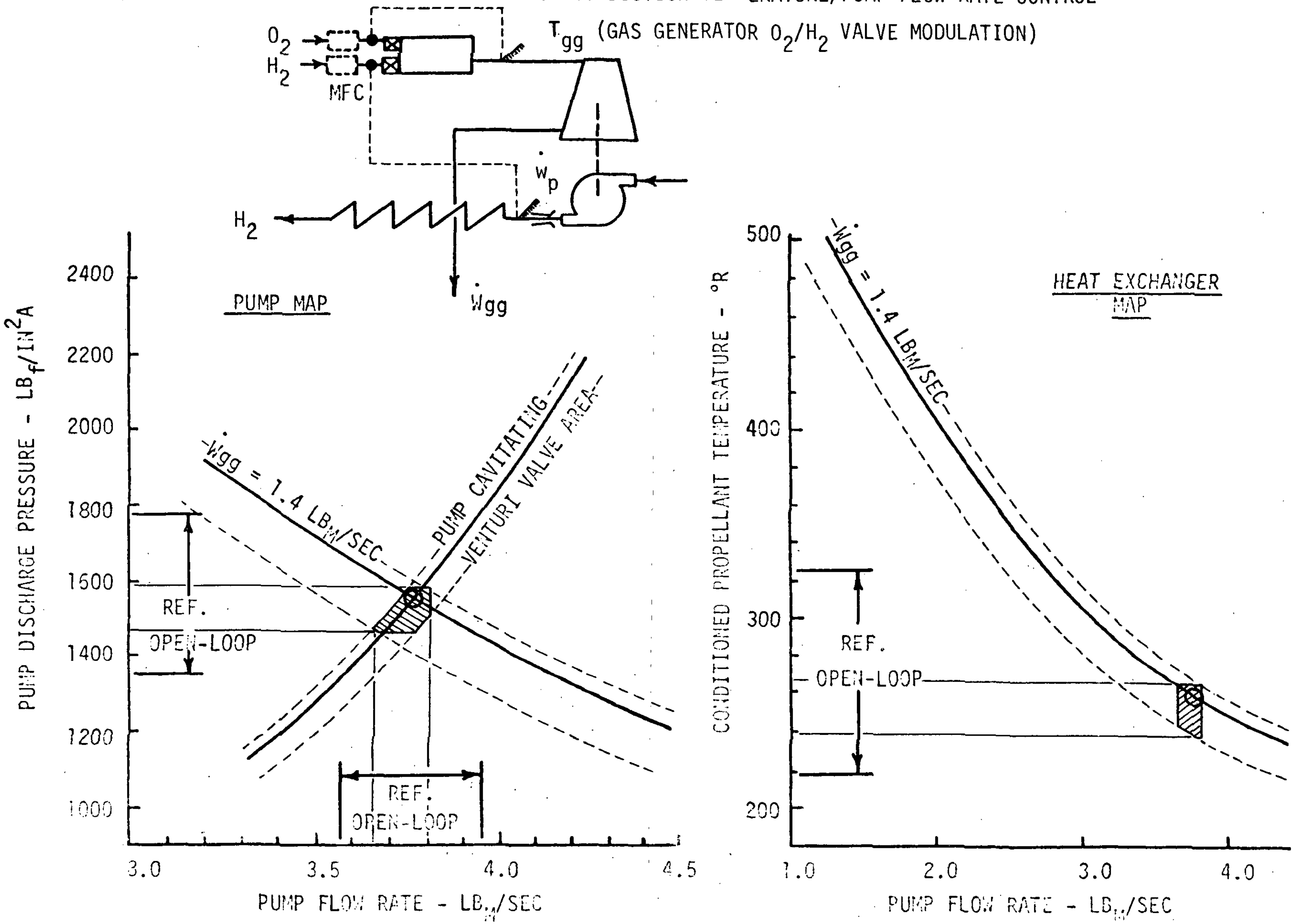


Figure C-43

WEIGHT SUMMARY CHART FOR PUMP DISCHARGE PRESSURE CONTROL

- o Series-Upstream Turbine RCS
- o With Mass Flow Controller

Control Point	Prop	Nominal Control Value	Operating Band				Δ System Weight (Lbm)*			
			$\dot{W}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{W}_p$-Tol	P_d-Tol	T_{cond}-Tol	Total
None	H_2 O_2		3.67-3.86 11.23-12.46	1465-1714 1748-2138	230-312 425-624	1714-2631 1716-2634	29 43	63 39	257 143	349 255 } 574
GGA Discharge Valve	H_2 O_2	1545 1901	3.66-3.86 11.12-12.32	1391-1624 1701-2050	207-348 382-710	1737-2776 1736-2749	32 54	30 25	540 191	602 270 } 872
Pump Discharge Valve	H_2 O_2	1545 1901	3.53-4.17 10.47-13.45	1529-1562 1875-1928	234-303 434-607	1714-2631 1716-2634	65 120	7 5	230 130	302 255 } 557
Hot Gas Vent Valve	H_2 O_2	1545 1901	3.70-3.83 11.58-11.95	1529-1561 1879-1923	227-314 421-636	1714-2631 1716-2634	20 14	6 4	280 147	306 165 } 471
GGA O_2 Valve + GGA H_2 Valve**	H_2 O_2	1545 1901	3.67-3.83 11.40-11.99	1486-1556 1793-1921	239-266 464-512	1936-1993 1793-1913	29 29	5 3	187 85	221 117 } 338

*Referenced to system with perfect control.
**GGA O_2 valve used for combustion temperature control.

Figure C-44

WEIGHT SUMMARY CHART FOR PUMP FLOW RATE CONTROL

- o Series-Upstream Turbine RCS
- o With Mass Flow Controller

Control Point	Prop	Nominal Control Value	Operating Band				△System Weight-Lbm*			
			$\dot{w}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{w}_p$-Tol	P_d-Tol	T_{cond}-Tol	Total
None	H_2 O_2		3.67-3.86 11.23-12.46	1465-1714 1748-2138	230-312 425-624	1714-2631 1716-2634	29 43	63 39	257 143	349 225 } 574
GGA Discharge Valve	H_2 O_2	3.77 11.76	3.66-3.84 11.12-12.32	1391-1624 1701-2054	207-348 401-699	1737-2776 1728-2737	32 54	30 26	540 171	602 251 } 853
Pump Discharge Valve	H_2 O_2	3.77 11.76	3.67-3.87 11.45-12.07	1420-1710 1762-2048	227-313 421-634	1714-2631 1716-2634	29 25	62 25	280 149	371 199 } 570
Hot Gas Vent Valve	H_2 O_2	3.77 11.76	3.67-3.87 11.45-12.07	1430-1710 1762-2048	228-313 422-634	1714-2631 1716-2634	29 25	62 25	275 147	366 197 } 563
GGA O_2 Valve + GGA H_2 Valve**	H_2 O_2	3.77 11.76	3.66-3.80 11.37-11.92	1460-1597 1766-1943	239-266 456-511	1962-2026 1954-2009	32 31	20 7	187 98	239 136 } 375

*Referenced to system with perfect control.
**GGA O_2 valve used for combustion temperature control.

Figure C-45

rate can be achieved only at the sacrifice of discharge pressure control and vice versa, pump discharge valve modulation was also discarded as a means of providing flow or pressure control.

Both hot gas vent valve (control point 4 in Figures C-40 and C-41) and gas generator H_2 valve modulation (control point 2) were also evaluated for this control function. In the former case, turbine delivered shaft power is affected through variation in turbine pressure ratio, only, since turbine flow rate is constant (choked conditions at the gas generator discharge port). This makes control simple and direct. Increased pump flow rate or discharge pressure is achieved by opening the vent valve area (increasing the ratio of turbine inlet to exit pressure).

Control is not so straight forward with the gas generator H_2 valve. Modulation of this valve affects not only turbine pressure ratio, but also turbine inlet temperature and flow rate. For system tolerances (pressures, temperatures, and areas) which produce either a maximum pump flow rate or discharge pressure, gas generator H_2 valve area must be increased to effect a reduction in flow rate and discharge pressure. This increases turbine flow rate, but reduces combustion temperature (lower mixture ratio), thus reducing the total energy of the exhaust flow (turbine delivered power). However, an anomaly is encountered for the system tolerances which produce minimum discharge pressure and flow rate. To effect an increase in pump flow rate or discharge pressure at these conditions, it is also necessary to increase H_2 valve area. In this case, the increase in turbine flow rate overrides the decrease in combustion temperature, thereby increasing the total energy of the exhaust flow. Hence, the control gains required for pump flow rate (discharge pressure) control with gas generator H_2 valve modulation are a function of system tolerances. Since the required gains could not be programmed without a prior knowledge of system tolerances, this control was not considered a viable candidate by itself.

This anomaly is not present if the gas generator O_2 valve (control point 1) is employed concurrently for combustion temperature control. In this dual control approach, evaluated in Figures C-42 and C-43, nearly constant combustion temperature is maintained with variations in H_2 valve area for pump discharge pressure and flow control.

Based on the above controls analyses, two high value approaches were identified for pump discharge pressure/flow control: (1) a singular control concept which employs modulation of the hot gas vent valve; and (2) a dual control concept in which the gas generator H_2 valve is modulated to control pump performance, and the O_2 valve is modulated to maintain nearly constant combustion temperature. Examination of Figures C-40 and C-41 for the hot gas vent valve, and Figures C-42 and C-43 for the gas generator valves, shows that valve modulation in response to pump discharge pressure produces the tightest operational bands on both discharge pressure and flow rate, and provides lower total system weight (Figure C-44). This results primarily from the lower sensor accuracy associated with a pressure transducer compared to a flow meter (Figure C-8).

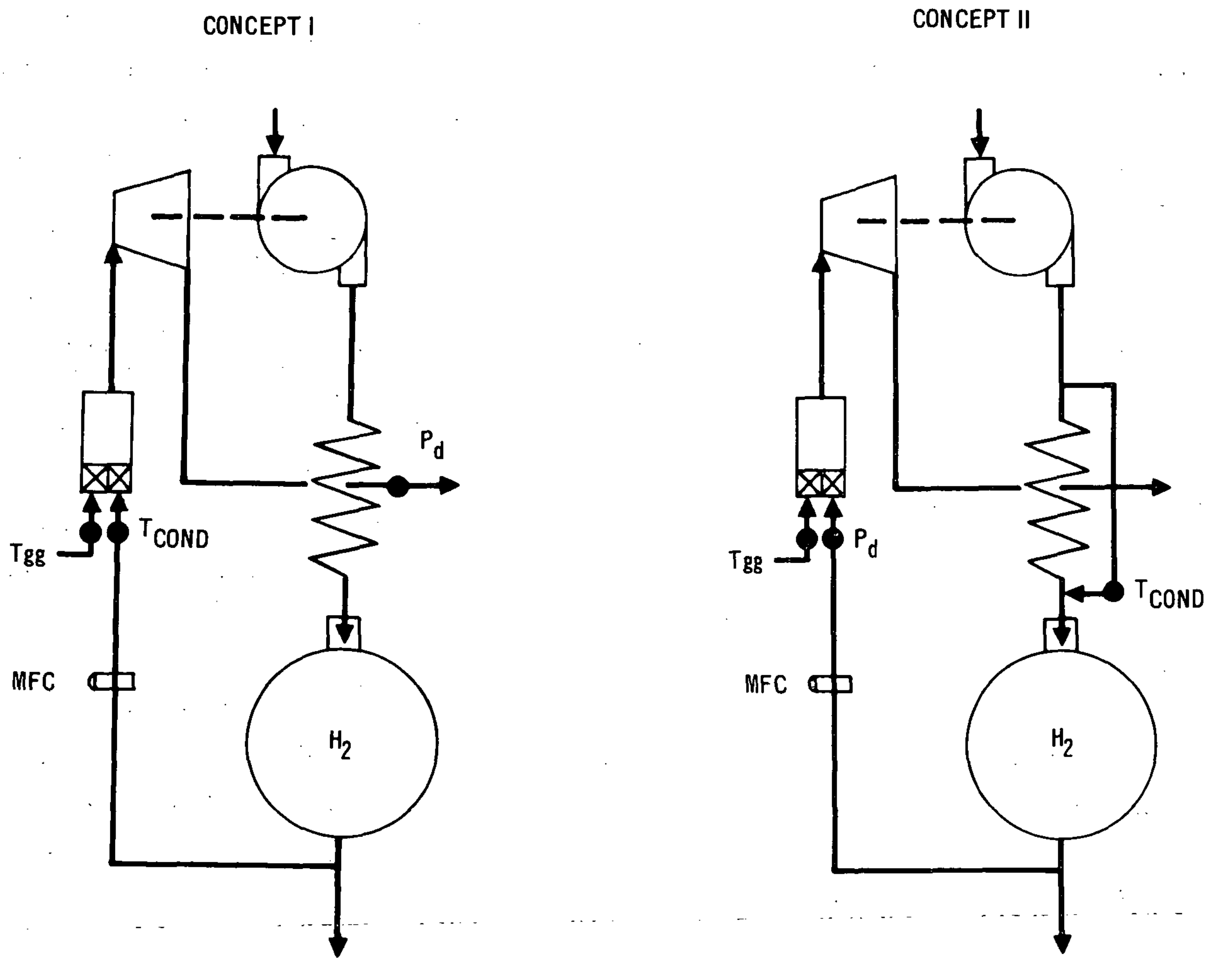
C6.4 Multiple Controls - Applying the most attractive control concepts identified above, multiple control approaches were investigated for the simultaneous control of: (1) gas generator combustion temperature to preclude the possibility of turbine rotor blade/heat exchanger failures, (2) conditioned propellant temperature to minimize system weight, and (3) pump discharge pressure (flow rate) to provide further weight reduction and reduce the potential for heat exchanger cold side flow instability. The investigations are summarized below for the three candidate RCS concepts.

C6.4.1 Series-Upstream Turbine RCS - Based on the detailed evaluations of the series-upstream turbine RCS, two high value multiple control approaches were investigated. These are shown in Figure C-46, and differ in the manner by which conditioned temperature and pressure controls are implemented. In the first, conditioned temperature is controlled with the gas generator H_2 valve, while pump discharge pressure (flow rate) is maintained with the hot gas vent valve. In the second approach, conditioned temperature is controlled with the heat exchanger cold side bypass valve, thus freeing the gas generator H_2 valve for pump discharge pressure (flow control). Conditioner performance maps for these two approaches are presented in Figures C-47 and C-48 and weight summaries are shown in Figure C-49. Comparing the results it is seen that the latter approach employing heat exchanger cold side bypass provides the greatest system weight benefit due to its excellent control of conditioned temperature.

C6.4.2 Series-Downstream Turbine RCS - Due to the similarity of the series RCS concepts, the same two control approaches identified above were investigated for the series-downstream turbine RCS. These are illustrated in Figure C-50. Conditioner performance maps for each approach are shown in Figures C-51 and C-52, and weight summaries are presented in Figure C-53.

MULTIPLE CONTROL CONCEPTS

- SERIES/UPSTREAM TURBINE RCS
- WITH MASS FLOW CONTROL



CONDITIONER CONTROLS EVALUATION

- o SERIES-UPSTREAM TURBINE RCS
- o MULTIPLE CONTROLS

(GAS GENERATOR O_2/H_2 VALVE AND VENT VALVE MODULATION)

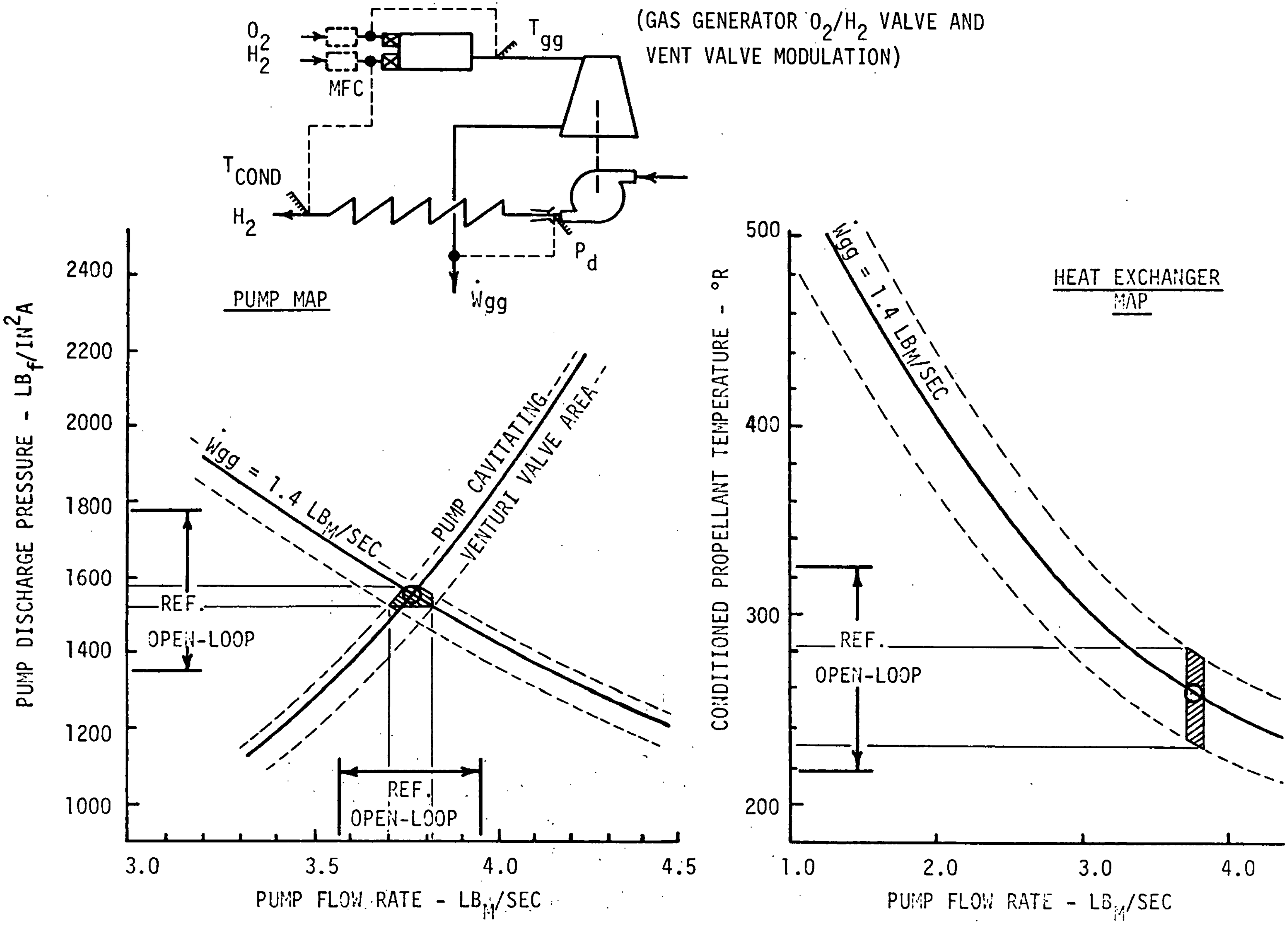


Figure C-47

CONDITIONER CONTROLS EVALUATIONS

- o SERIES-UPSTREAM TURBINE RCS
- o MULTIPLE CONTROLS

(GAS GENERATOR O_2/H_2 VALVE AND HEAT EXCHANGER BYPASS VALVE MODULATION)

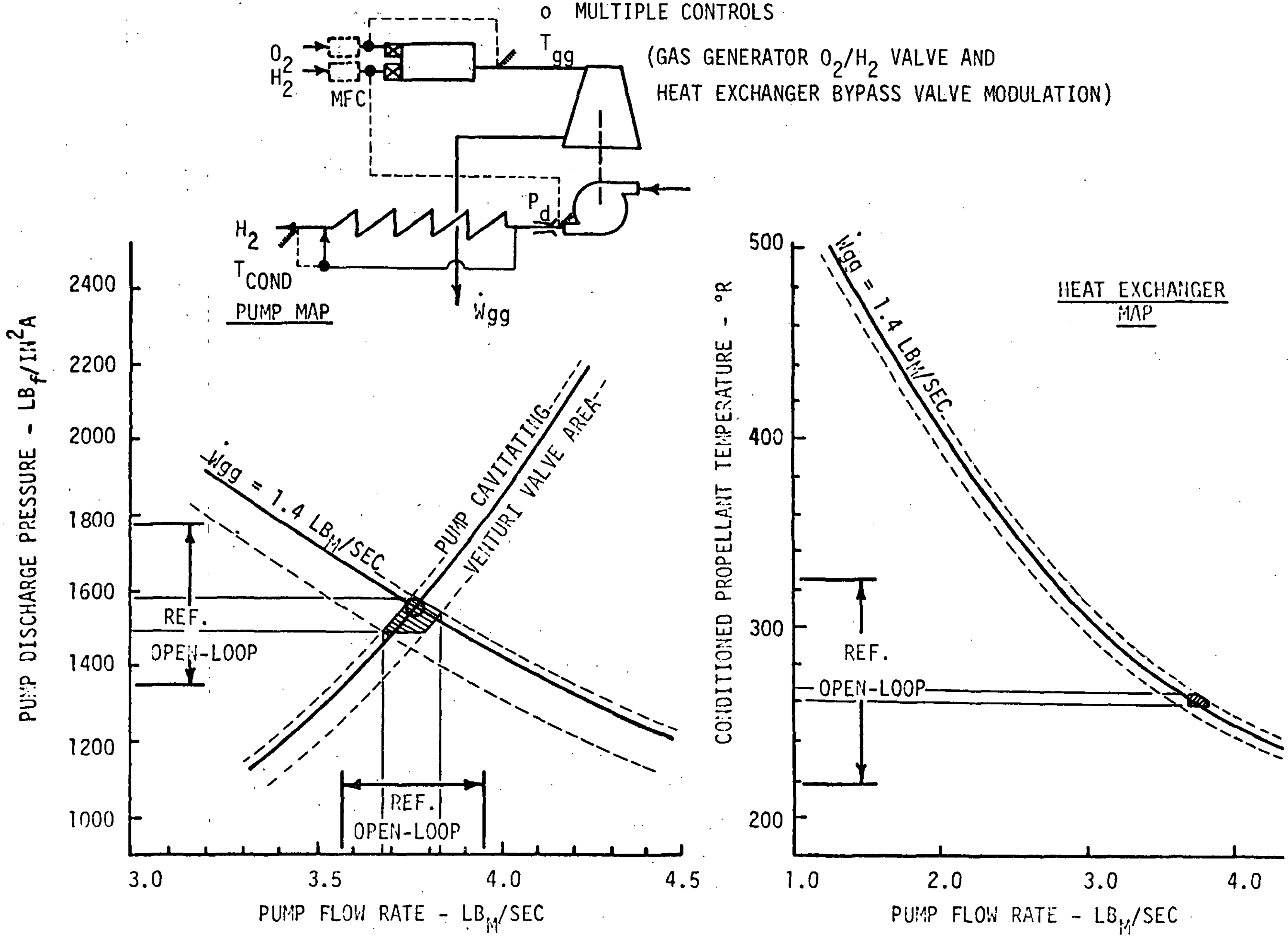


Figure C-48

WEIGHT SUMMARY CHART FOR MULTIPLE CONTROLS

- o Series-Upstream Turbine RCS
- o With Mass Flow Controller

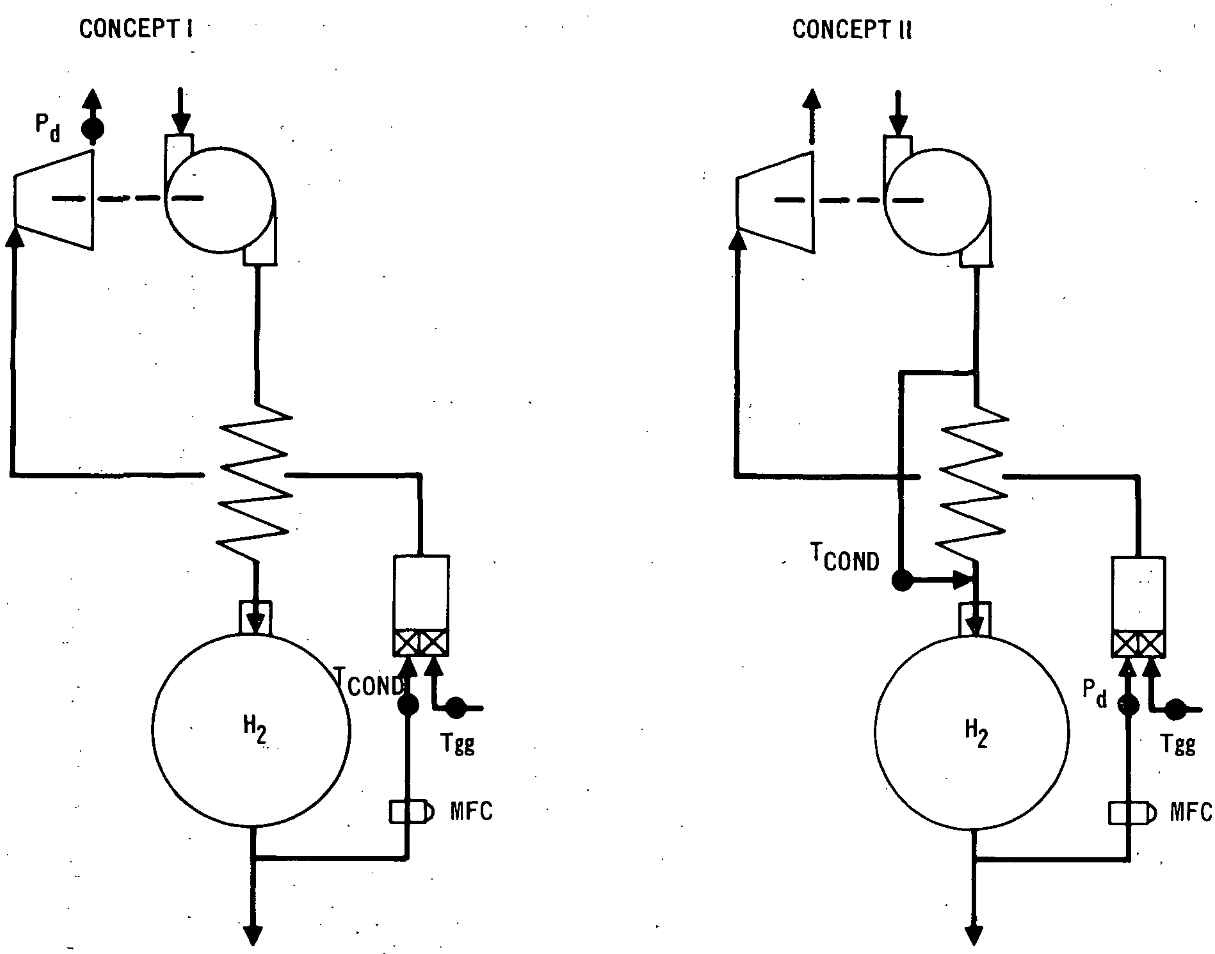
Controls	Prop	Nominal Control Value			Operating Band				ΔSystem Weight - Lbm*				
					$\dot{w}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{w}_p$-Tol	P_d-Tol	T_{cond}-Tol	Total	
None	H_2				3.67-3.86	1465-1714	230-312	1714-2631	29	63	257	349	574
	O_2				11.23-12.46	1748-2138	425-624	1716-2634	43	39	143	225	
Concept I		T_{gg}	T_{cond}	P_d									
GGA O_2 Valve+	H_2	2000	263	1545	3.70-3.80	1530-1561	231-282	1807-2351	21	6	250	277	412
GGA H_2 Valve+ Hot Gas Vent Valve	O_2	2000	506	1901	11.66-11.92	1882-1921	440-567	1808-2353	8	4	123	135	
Concept II		T_{gg}	T_{cond}	P_d									
GGA O_2 Valve+	H_2	2000	263	1545	3.67-3.83	1486-1561	260-266	1936-1993	29	6	17	52	92
GGA H_2 Valve+ Hex By-pass Valve	O_2	2000	506	1901	11.40-11.99	1793-1921	501-511	1793-1913	29	4	7	40	

*Referenced to system with perfect control.

Figure C-49

MULTIPLE CONTROL CONCEPTS

- SERIES/DOWNSTREAM TURBINE RCS
- WITH MASS FLOW CONTROL



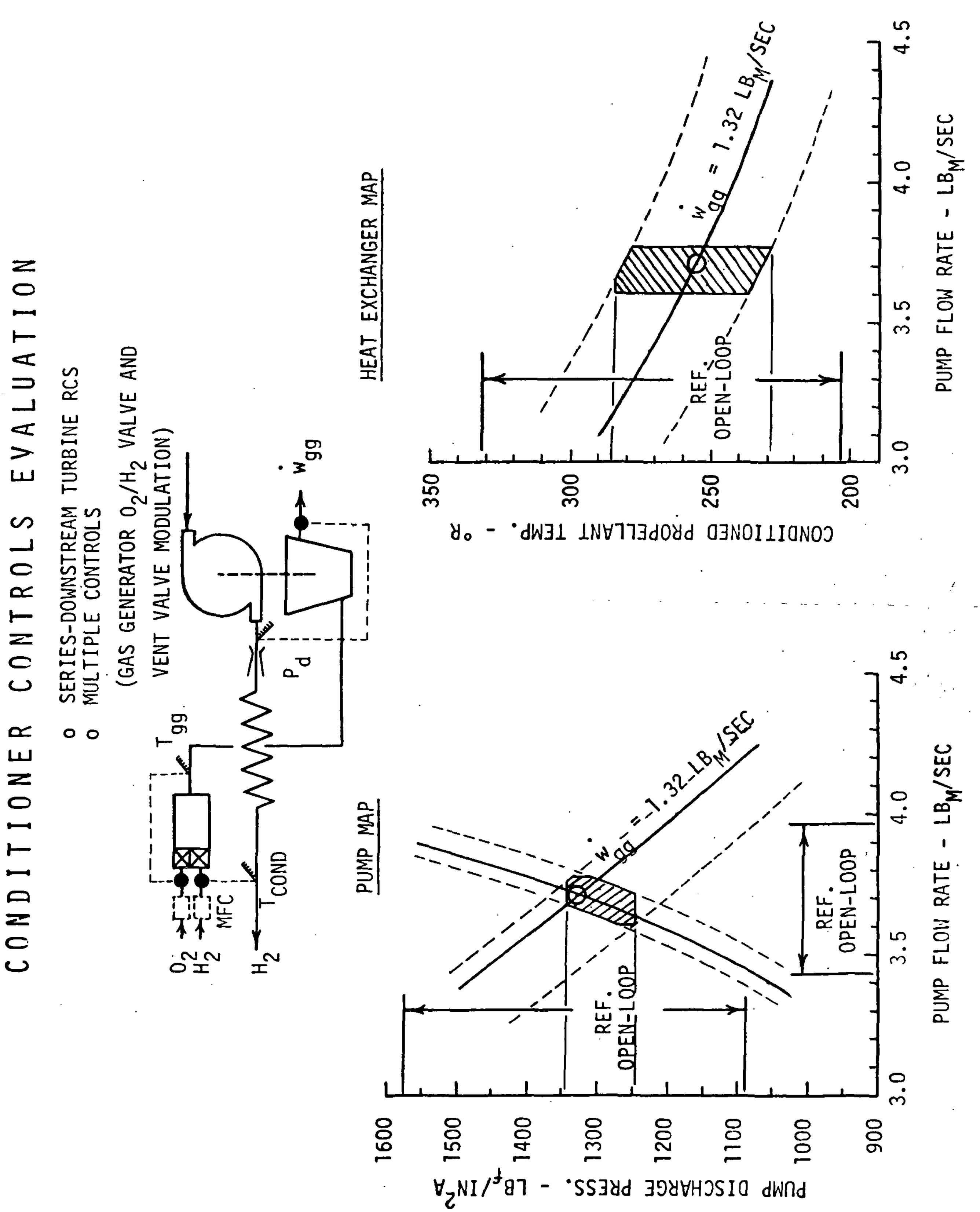

CONDITIONER CONTROLS EVALUATION
o SERIES-DOWNSTREAM TURBINE RCS
o MULTIPLE CONTROLS
(GAS GENERATOR O_2/H_2 VALVE AND
VENT VALVE MODULATION)
O_2
H_2
MFC
T_{gg}
T_{COND}
H_2
P_d
$\dot{w}_{gg}$
HEAT EXCHANGER MAP
$\dot{w}_{gg}$ = 1.32 LB_M/SEC
REF. OPEN-LOOP
CONDITIONED PROPELLANT TEMP. - °R
350
300
250
200
3.0
3.5
4.0
4.5
PUMP FLOW RATE - LB_M/SEC
PUMP MAP
$\dot{w}_{gg}$ = 1.32 LB_M/SEC
REF. OPEN-LOOP
REF. OPEN-LOOP
PUMP DISCHARGE PRESS. - LB_f/IN^2A
1600
1500
1400
1300
1200
1100
1000
900
3.0
3.5
4.0
4.5
PUMP FLOW RATE - LB_M/SEC

Figure C-51

CONDITIONER CONTROLS EVALUATION

- o SERIES-DOWNSTREAM TURBINE RCS
- o MULTIPLE CONTROLS

  (GAS GENERATOR O_2/H_2 VALVE AND HEAT EXCHANGER BYPASS VALVE MODULATION)

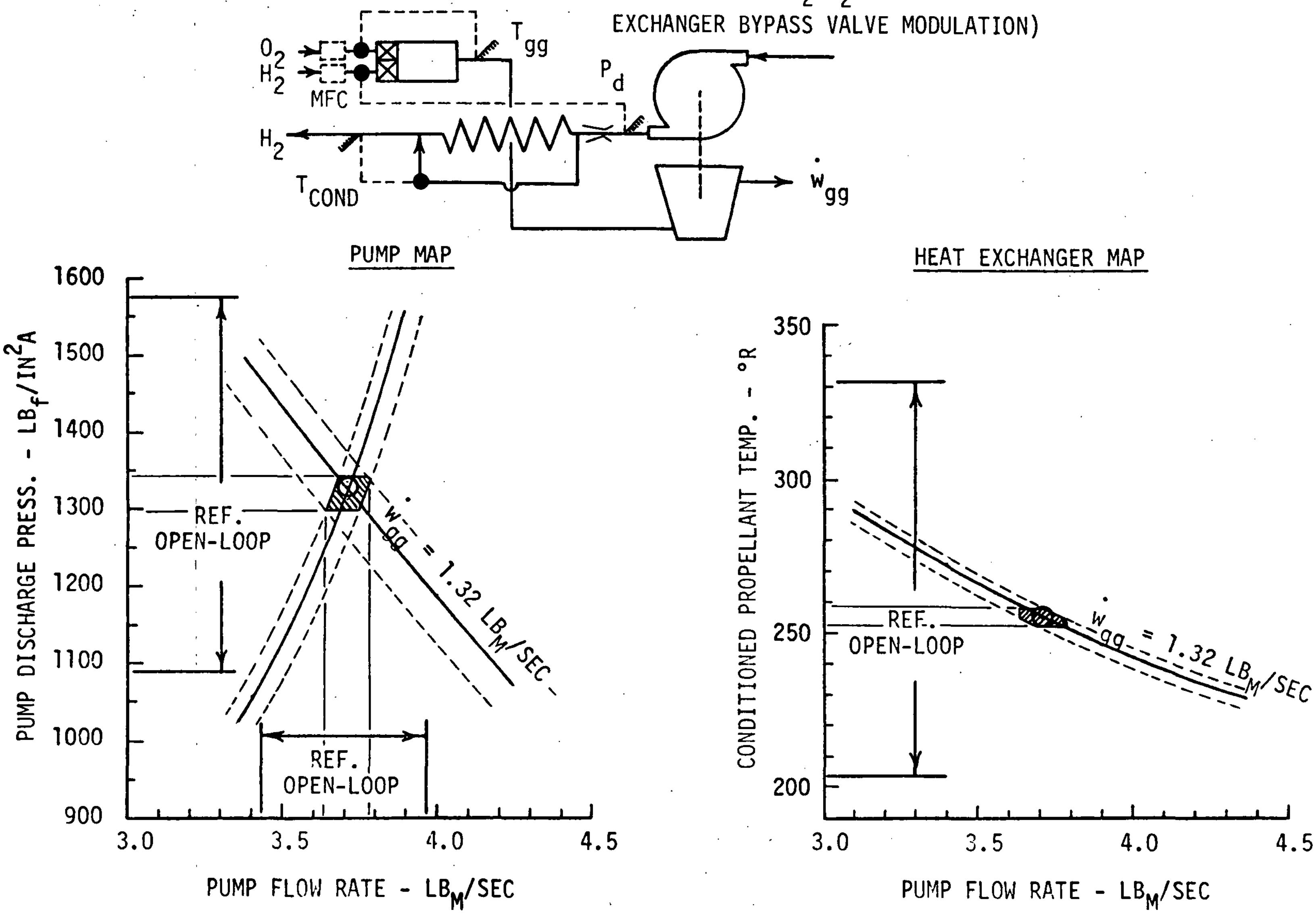


Figure C-52

WEIGHT SUMMARY CHART FOR MULTIPLE CONTROLS

- o Series-Downstream Turbine RCS
- o With Mass Flow Controller

Controls	Prop	Nominal Control Value			Operating Band				△System Weight – Lbm*				
					$\dot{w}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{w}_p$-Tol	P_d-Tol	T_{cond}-Tol	Total	
None	H_2				3.60-3.86	1241-1452	219-311	1716-2678	40	49	234	323	551
	O_2				11.34-12.33	1731-2050	410-653	1717-2697	34	33	161	228	
Concept I		T_{gg}	T_{cond}	P_d									
GGA O_2 Valve +	H_2	2000	255	1327	3.63-3.76	1290-1340	228-285	1807-2374	29	5	173	207	345
GGA H_2 Valve + Hot Gas Vent Valve	O_2	2000	517	1846	11.61-11.87	1826-1864	431-573	1808-2379	10	3	125	138	
Concept II		T_{gg}	T_{cond}	P_d									
GGA O_2 Valve +	H_2	2000	255	1327	3.63-3.78	1292-1340	252-258	1972-2011	29	5	17	51	78
GGA H_2 Valve + Hex By-pass Valve	O_2	2000	517	1846	11.53-11.96	1800-1864	512-521	1971-2014	17	3	7	27	

*Referenced to system with perfect controls.

Figure C-53

Examining the pump performance map of Figure C-51, it is noteworthy that good pressure/flow control is achieved on the high-side of nominal, but poor control is achieved on the low-side. This unidirectional control effectiveness of the hot gas vent valve results from the fact that the turbine design point is near the knee of the power-pressure ratio curve shown in Figure C-54. Hence, increases in turbine back pressure (decreases in turbine pressure ratio) result in significant changes in delivered shaft power, whereas, decreases in turbine back pressure provide only minimal shaft power gains. This tendency is not present if the gas generator H_2 valve is modulated to control pump discharge pressure and flow (Figure C-52). Here, delivered shaft power is controlled directly by turbine flow rate, with turbine pressure ratio providing only a secondary effect.

The performance map of the multiple control scheme which uses heat exchanger cold side bypass (Figure C-52) is very similar to the corresponding map for the series-upstream turbine concept. As for the series-upstream turbine concept, minimum system weight is achieved using the heat exchanger cold side bypass loop for conditioned temperature control.

C6.4.3 Parallel RCS - The limited control effectiveness of the turbine vent valve for controlling pump performance in the series-downstream turbine RCS was even more pronounced in the parallel RCS. This control proved completely ineffective in the parallel RCS since the turbine design point was well to the right of the knee of the power-pressure ratio curve (Figure C-55). Because of this characteristic and the high temperature of the turbine exhaust flow (1600°R), the only acceptable approach for pump discharge pressure (flow) control was modulation of the turbine gas generator H_2 valve. Applying this result, three high value multiple control approaches were identified for the parallel RCS, and are shown in Figure C-56. The three approaches differ only in the manner of conditioned temperature control: (1) heat exchanger gas generator H_2 valve modulation, (2) heat exchanger cold side bypass flow modulation, and (3) heat exchanger vent valve modulation. Conditioner performance maps for the three approaches are presented in Figures C-57 through C-59 and weight summaries are presented in Figure C-60. Comparing these figures it is seen that both the heat exchanger cold side bypass valve and the vent valve for conditioned temperature control yield similar results and provide the greatest system weight benefit.

C7. Active Controls Evaluations (No Mass Flow Control) - The significant conclusions derived from the preceeding analyses were:

(1) Even with mass flow controllers, active controls are required to provide acceptable conditioner performance. In the interest of system simplicity

TURBINE POWER - PRESSURE RATIO CURVE

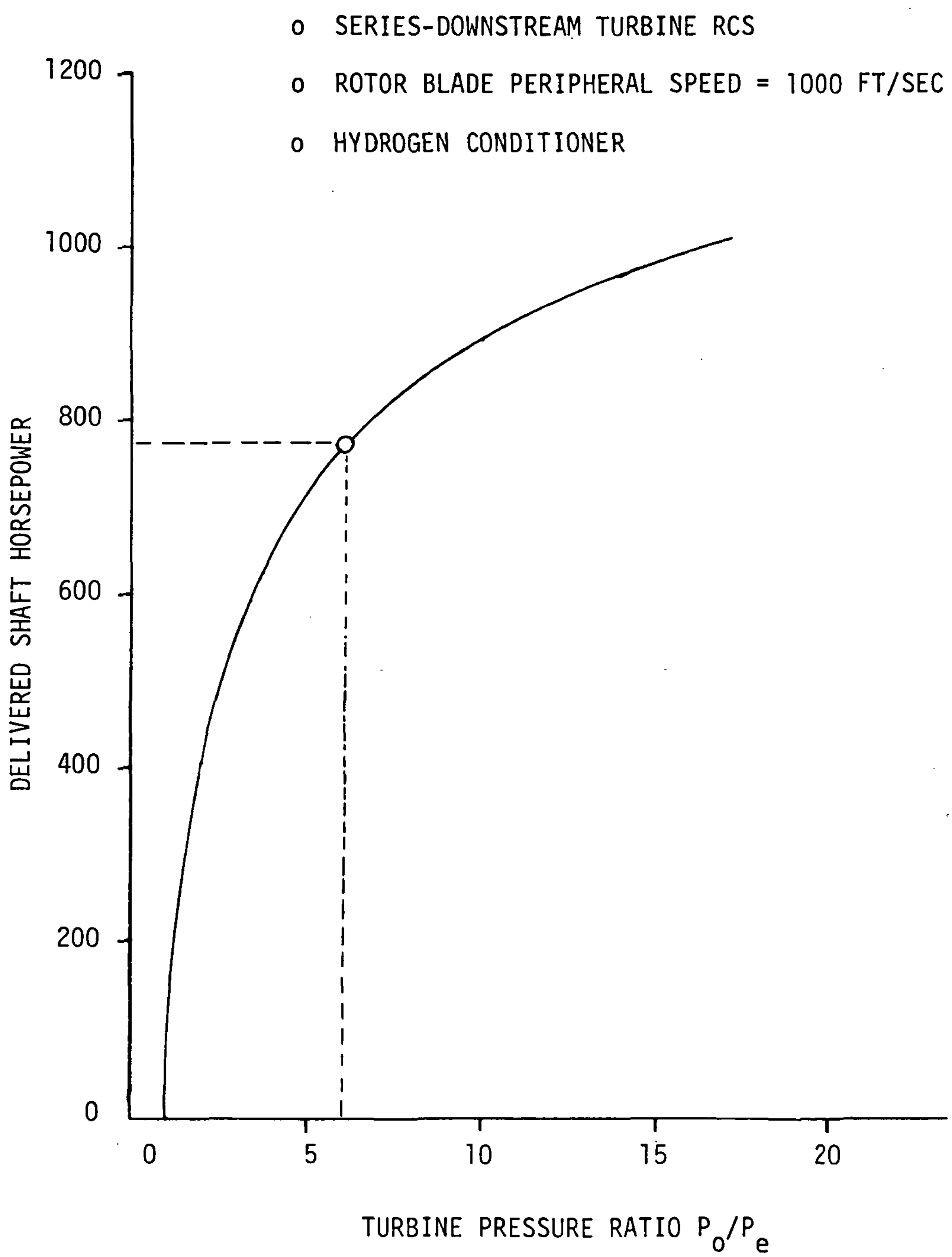
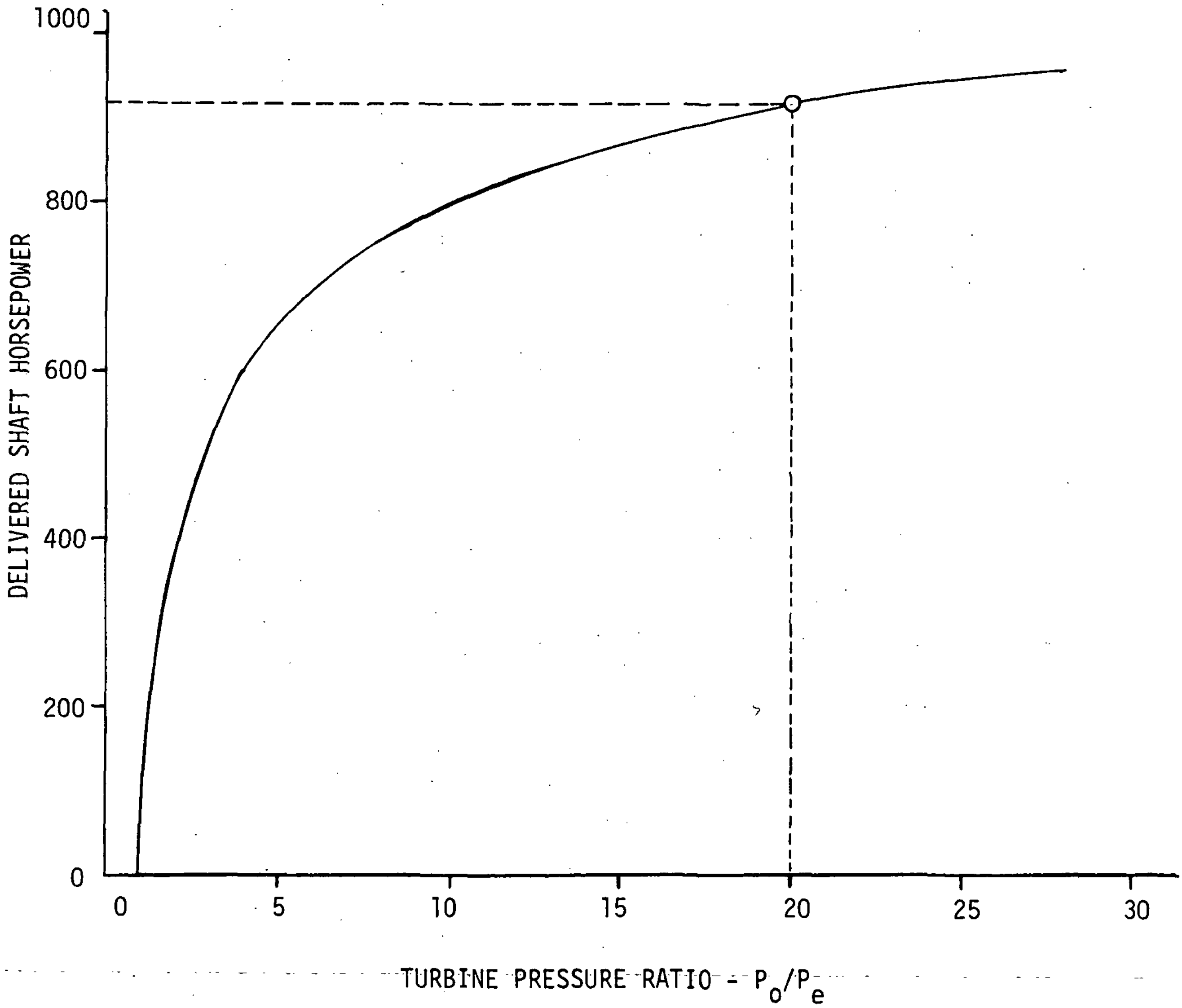


Figure C-54

TURBINE POWER - PRESSURE RATIO CURVE

- PARALLEL RCS
- ROTOR BLADE PERIPHERAL SPEED = 1000 FT/SEC
- HYDROGEN CONDITIONER



MULTIPLE CONTROL CONCEPTS

- PARALLEL RCS
- WITH MASS FLOW CONTROL

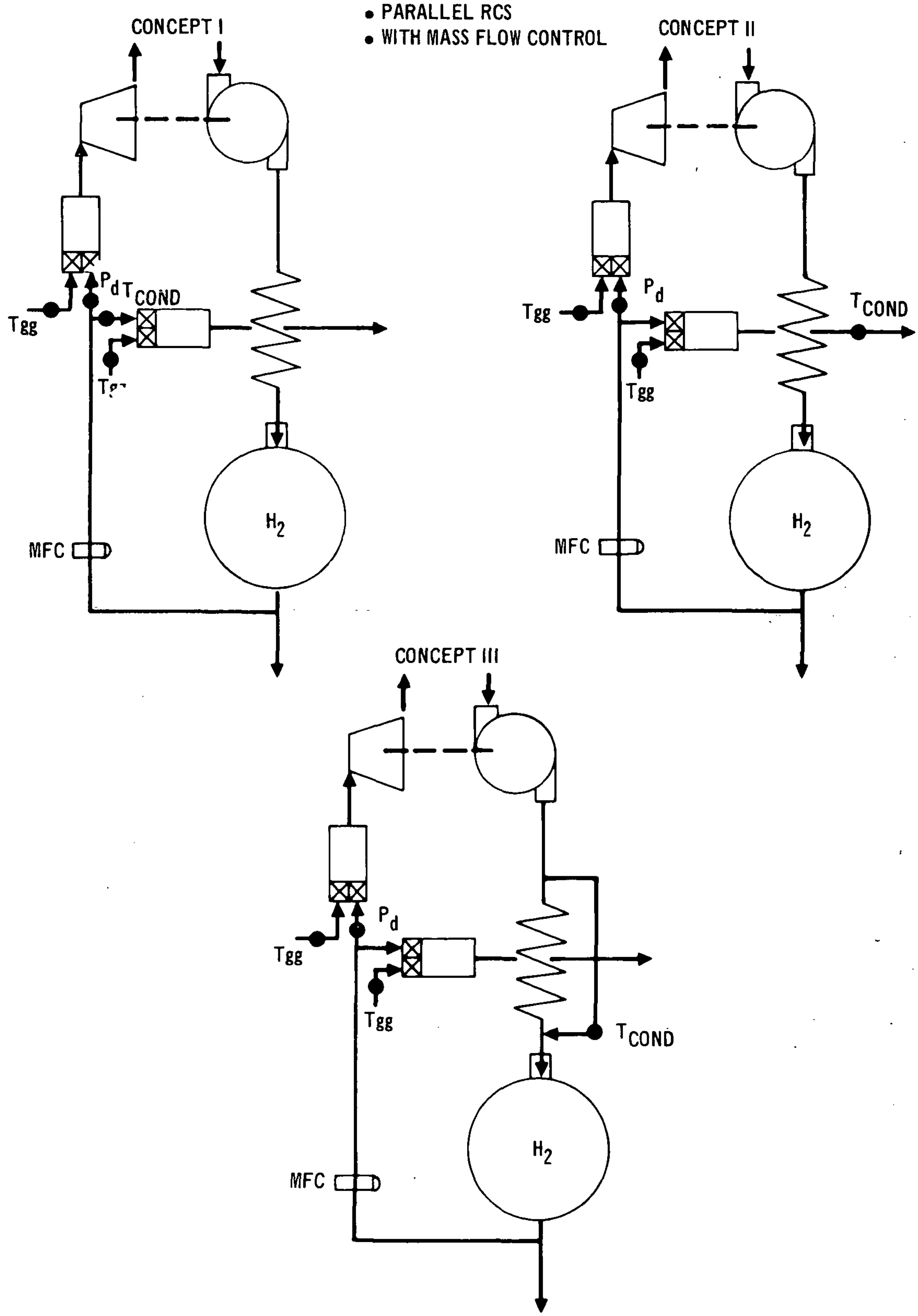


Figure C-56

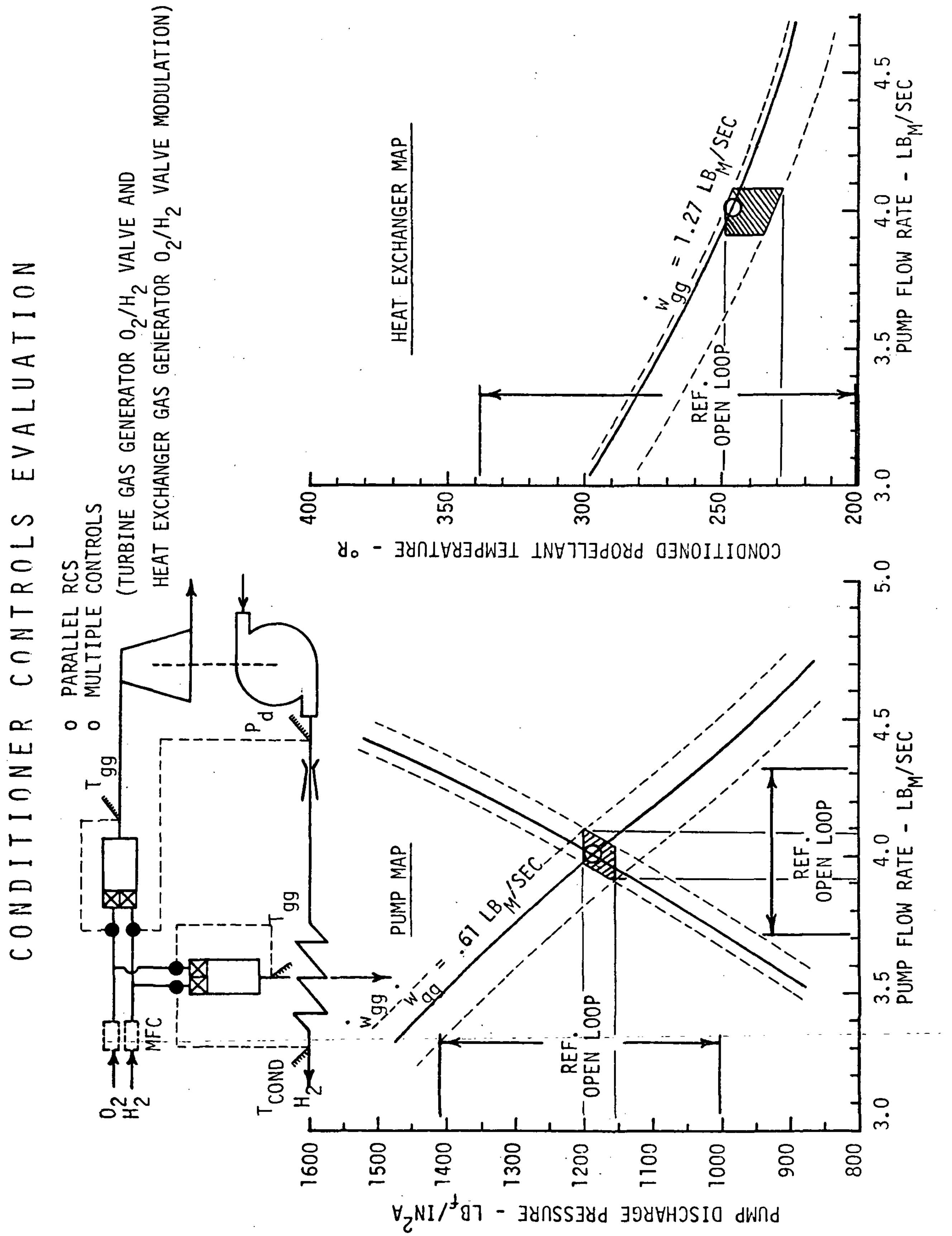

CONDITIONER CONTROLS EVALUATION
o PARALLEL RCS
o MULTIPLE CONTROLS
(TURBINE GAS GENERATOR O_2/H_2 VALVE AND
HEAT EXCHANGER GAS GENERATOR O_2/H_2 VALVE MODULATION)
O_2
H_2
MFC
T_{gg}
T_{gg}
P_d
T_{COND}
H_2
$\dot{w}_{gg}$
HEAT EXCHANGER MAP
$\dot{w}_{gg}$ = 1.27 LB_M/SEC
REF. OPEN LOOP
CONDITIONED PROPELLANT TEMPERATURE - °R
400
350
300
250
200
3.0
3.5
4.0
4.5
PUMP FLOW RATE - LB_M/SEC
PUMP MAP
$\dot{w}_{gg}$ = .61 LB_M/SEC
REF. OPEN LOOP
REF. OPEN LOOP
PUMP DISCHARGE PRESSURE - LB_f/IN^2
1600
1500
1400
1300
1200
1100
1000
900
800
3.0
3.5
4.0
4.5
5.0
PUMP FLOW RATE - LB_M/SEC

Figure C-57

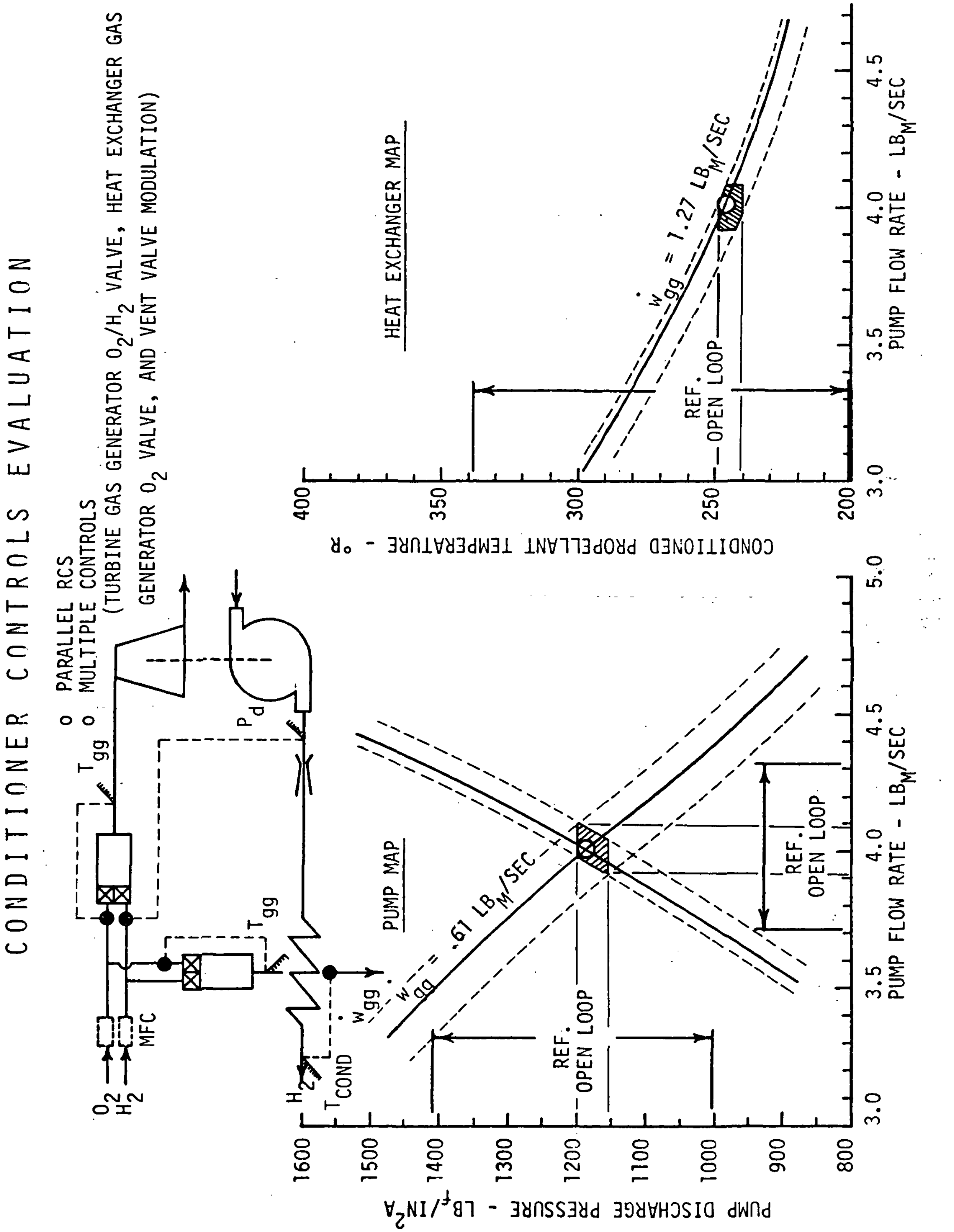

CONDITIONER CONTROLS EVALUATION
o PARALLEL RCS
o MULTIPLE CONTROLS
(TURBINE GAS GENERATOR O_2/H_2 VALVE, HEAT EXCHANGER GAS GENERATOR O_2 VALVE, AND VENT VALVE MODULATION)
O_2
H_2
MFC
T_{gg}
T_{gg}
P_d
$\dot{w}_{gg}$
H_2
T_{COND}
HEAT EXCHANGER MAP
$\dot{w}_{gg}$ = 1.27 LB_M/SEC
REF. OPEN LOOP
CONDITIONED PROPELLANT TEMPERATURE - °R
400
350
300
250
200
3.0
3.5
4.0
4.5
PUMP FLOW RATE - LB_M/SEC
PUMP MAP
$\dot{w}_{gg}$ = .61 LB_M/SEC
REF. OPEN LOOP
REF. OPEN LOOP
PUMP DISCHARGE PRESSURE - LB_f/IN^2A
1600
1500
1400
1300
1200
1100
1000
900
800
3.0
3.5
4.0
4.5
5.0
PUMP FLOW RATE - LB_M/SEC

Figure C-58

CONDITIONER CONTROLS EVALUATION

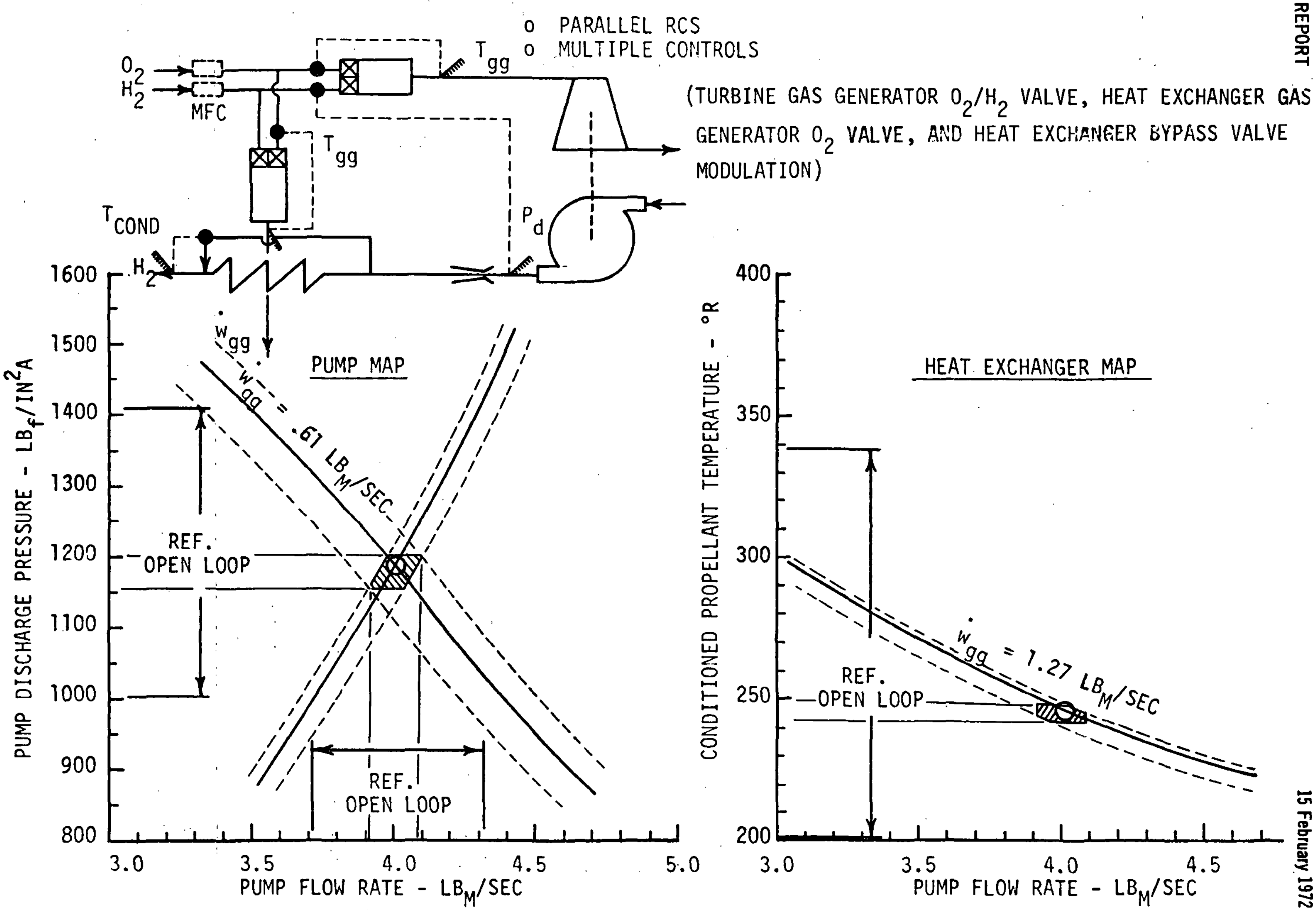


Figure C-59

WEIGHT SUMMARY CHART FOR MULTIPLE CONTROLS

- o Parallel RCS
- o With Mass Flow Controller

Controls	Prop	Nominal Control Value			Operating Band				ΔSystem Weight – Lbm*				
					$\dot{w}_p$ lbm/sec	P_d lbf/in.^{2}a	T_{cond} °R	T_{gg} °R	$\dot{w}_p$-Tol	P_d-Tol	T_{cond}-Tol	Total	
None	H_2				3.82-4.19	1074-1302	213-303	1692-2714	74	46	257	377	590
	O_2				11.48-12.58	1423-1705	393-627	1700-2698	55	23	135	213	
Concept I		T_{gg}	T_{cond}	P_d									
GGA O_2 Valves + TPA GGA H_2 Valve + Hex H_2 Valve	H_2	2000	246	1187	3.91-4.08	1155-1199	228-248	1808-2024	38	6	135	179	298
	O_2	2000	473	1565	11.79-12.30	1513-1582	422-477	1812-2030	26	3	90	119	
Concept II		T_{gg}	T_{cond}	P_d									
GGA O_2 Valves + TPA GGA H_2 Valve + Hex Vent Valve	H_2	2000	246	1187	3.91-4.08	1155-1199	242-248	1949-2031	38	6	26	70	112
	O_2	2000	473	1565	11.79-12.30	1513-1582	466-478	1885-2035	26	3	13	42	
Concept III		T_{gg}	T_{cond}	P_d									
GGA O_2 Valves + TPA GGA H_2 Valve + Hex By-pass Valve	H_2	2000	246	1187	3.91-4.08	1155-1199	243-248	1929-2016	38	6	17	61	100
	O_2	2000	473	1565	11.79-12.30	1513-1582	468-478	1923-2017	26	3	10	39	

*Referenced to system with perfect control.

active controls by themselves should be evaluated.

(2) The best controls for the three candidate RCS concepts are: gas generator 0_2 valve modulation for combustion temperature control; heat exchanger cold side bypass flow modulation for conditioned propellant temperature control; and gas generator H_2 valve modulation for pump discharge pressure (flow) control.

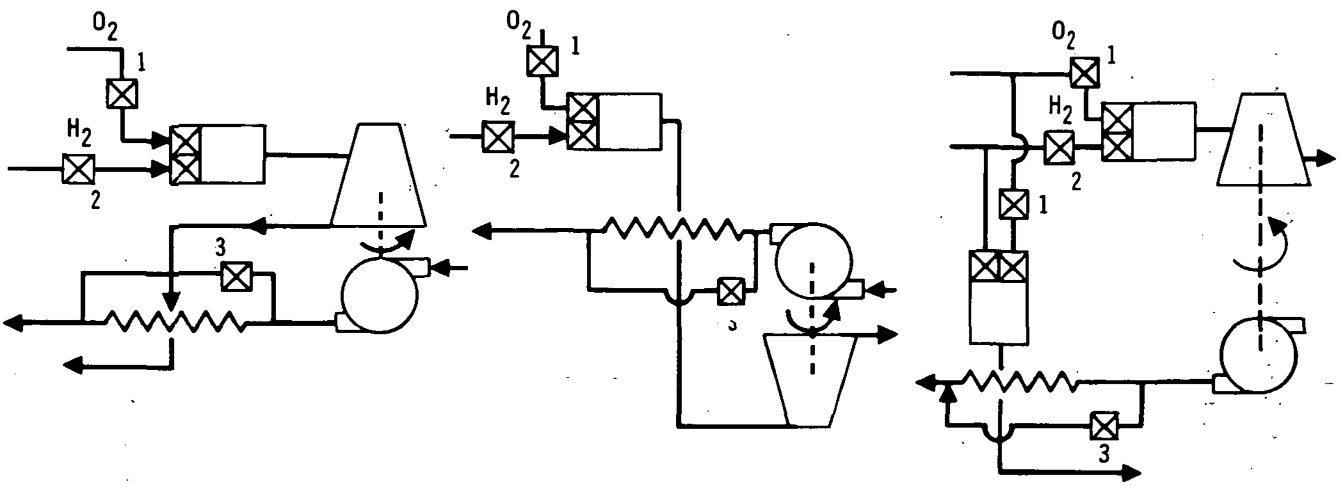
Simultaneous with the above analyses, detailed investigations were made of the heat exchanger design and of system performance during simulated missions. The heat exchanger analyses (Appendix D) showed that cold side bypass circuits on the hydrogen heat exchanger in the series-upstream turbine system and on both the hydrogen and oxygen heat exchangers in the parallel system were mandatory to preclude water condensation and freezing on the hot side tube walls. The mission operating analyses described in Section 4, paragraph 4.3, showed that conditioned propellant temperature and pump flow rate operating bands had a significant effect on system specific impulse and mixture ratio (thus propellant weight).

Hence, the control points selected for final evaluation (Figure C-61) reflected the mandatory controls required to limit gas generator combustion temperature and to preclude freezing in the heat exchanger. In addition, system weight evaluations included propellant weights based on mission operating analyses rather than the steady-state techniques previously employed. Other guidelines and constraints imposed for these evaluations were the same as shown in Figure C-23 with two exceptions: (1) no valve area constraints were imposed; and (2) identical control approaches were not required for both the hydrogen and oxygen conditioners.

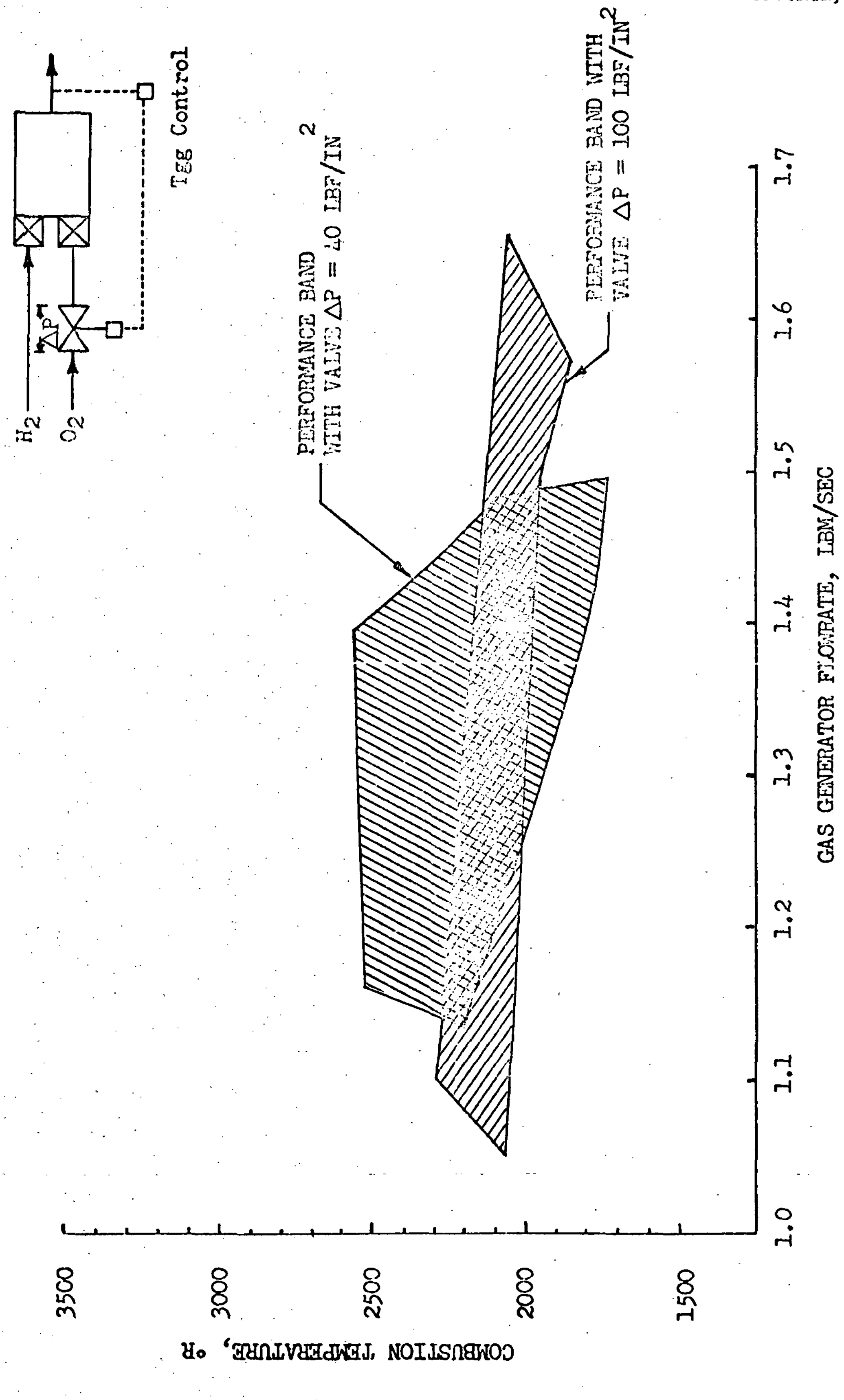
Initial investigations without mass flow control showed that the pressure drop across the gas generator 0_2 valve must be increased from 40 to approximately 100 lbf/in^2d to provide effective combustion temperature control (Figure C-62). Therefore, holding system regulated pressure and thruster chamber pressure constant at 400 and 300 lbf/in^2a, respectively, gas generator chamber pressure was reduced from 300 to 250 lbf/in^2a. This provided injector and valve pressure drops of 50 and 100 lbf/in^2d, respectively. Consistent with this change, all pertinent valve and component flow areas were resized to provide the required flow rate and temperature balances of Appendix B. The revised flow areas are tabulated in Figure C-63. Applying these areas, the gas generator inlet temperature bands of Figure C-4, and the component tolerances of Figure C-8, the effect of successively adding the controls of Figure C-61 was investigated. This was accomplished in three steps:

CANDIDATE CONTROL POINTS
(NO MASS FLOW CONTROL)

Series RCS (Turbine Upstream) | Series RCS (Turbine Downstream) | Parallel RCS



ACTIVE CONTROL POINT	INTERMEDIATE CONTROL FUNCTIONS	CONTROLLED PARAMETER
1. GAS GENERATOR OXYGEN VALVE 2. GAS GENERATOR HYDROGEN VALVE	TURBINE INLET TEMPERATURE, FLOW RATE, PRESSURE RATIO AND POWER	GAS GENERATOR COMBUSTION TEMPERATURE PUMP DISCHARGE PRESSURE PUMP FLOW RATE
3. HEAT EXCHANGER COLD SIDE BYPASS VALVE	HEAT EXCHANGER COLD SIDE FLOW RATE	CONDITIONED PROPELLANT TEMPERATURE

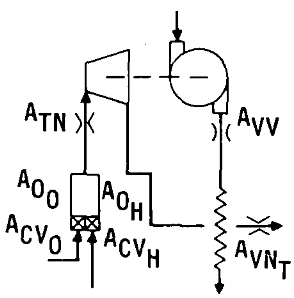
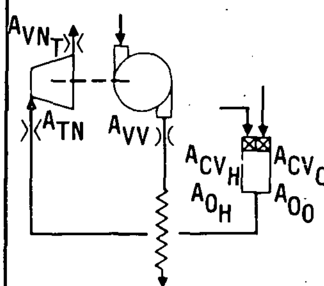
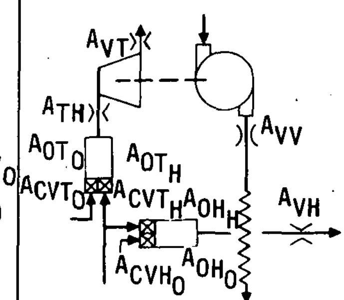


GAS GENERATOR PERFORMANCE MAP WITH COMBUSTION TEMPERATURE CONTROL

Figure C-62

SYSTEM VALVE/COMPONENT DESIGN AREAS

• GAS GENERATOR CHAMBER PRESSURE = 250 $LB_f/IN.^2A$

	SERIES TURBINE UPSTREAM	SERIES TURBINE DOWNSTREAM	PARALLEL
			
•GAS GENERATOR CONTROL VALVE AREA, $IN.^2$	H_2 , O_2 / H_2 , O_2	H_2 , O_2 / H_2 , O_2	H_2 , O_2 / H_2 ; O_2
A_{CV_O}/A_{CV_H}	0.0800, 0.0454/0.2430, 0.1382	0.0751, 0.0464/0.2287, 0.1414	–
A_{CVT_O}/A_{CVT_H}	–	–	0.0340, 0.0128/0.1021, 0.0380
A_{CVH_O}/A_{CVH_H}	–	–	0.0700, 0.0415/0.2060, 0.1227
INJECTOR ORIFICE AREA, $IN.^2$			
A_{O_O}/A_{O_H}	0.1221, 0.0695/0.373, 0.211	0.1109, 0.0685/0.338 , 0.209	–
A_{OT_O}/A_{OT_H}	–	–	0.0532, 0.0199/0.1692, 0.0632
A_{OH_O}/A_{OH_H}	–	–	0.0922, 0.0560/0.2935, 0.1779
•TURBINE ADMISSION AREA – ATN, $IN.^2$	1.291 , 0.7325	0.9635, 0.4700	0.5800, 0.2200
•PUMP CAVITATING VENTURI AREA – A_{VV}, $IN.^2$	0.0835, 0.0496	0.0864, 0.0501	0.0950, 0.0555
•VENT AREA, $IN.^2$			
A_{VN_T}	6.210 , 1.038	10.20 , 3.600	–
A_{VT}	–	–	8.800 , 3.450
A_{VH}	–	–	0.8250, 0.4400

(1) determining the operating bands on conditioned temperature, pump discharge pressure, and flow rate for each option using the mass flow and energy balances of Figure C-1; (2) determining total system weight with the design and sizing techniques of Reference N, and applying the conditioner operating bands found in Step 1; and (3) adjusting the system weights found in Step 2 applying the system mixture ratio/ specific impulse variances determined from mission duty cycle simulations described in Section 4, Paragraph 4.3.

Open-loop (no controls) conditioner performance for all three RCS concepts is summarized in Figure C-64, and the impact of successive control additions on total system weight is presented in Figures C-65 through C-67 for each RCS concept. It is noteworthy that systems without conditioner controls (open-loop) are more than 1,000 lb heavier than systems with perfect control.

For the series RCS concepts of Figures C-65 and C-66, the largest system weight benefit (approximately 600-700 lbm) is achieved with combustion temperature control. This is because gas generator O_2 valve modulation also reduces the operating bands on conditioned temperature, pump discharge pressure, and flow rate. In addition to the substantial weight savings effected, this control is considered mandatory to preclude potential turbine blade/heat exchanger tubing failures resulting from excessive gas inlet temperatures. Another large system weight reduction (approximately 200-300 lbm) is achieved through modulation of the hydrogen heat exchanger cold side bypass valve for conditioned temperature control. Although this control has no effect on the other conditioner interface parameters, it provides extremely tight control of hydrogen conditioned temperature. Oxygen conditioned temperature control is not employed for the two series concepts since it was felt that the additicnal weight savings (less than 100 lbm) would not justify the added complexity. Pump discharge pressure (flow) control through gas generator H_2 valve modulation provides a similar small weight reduction (approximately 100-150 lbm), however, more importantly it reduces the potential for heat exchanger development difficulty. A primary problem encountered in previous heat exchanger development programs (Saturn IV engine heat exchanger and Titan II autogeneous superheater) has been flow instability resulting from large fluid density variations and phase changes. This problem is minimized by tight control of cold side inlet conditions. Hence, the three control points shown in Figures C-65 and C-66 were selected for the two series RCS concepts. Typical performance achieved with these controls is illustrated in the hydrogen conditioner operating maps of Figures C-68 and C-69.

CONDITIONER OPEN-LOOP PERFORMANCE

- GAS GENERATOR CHAMBER PRESSURE = 250 LB_f/IN^2A

Hydrogen

	SERIES RCS TURBINE UPSTREAM	SERIES RCS TURBINE DOWNSTREAM	PARALLEL RCS
GAS GENERATOR COMBUSTION TEMP, ^0R	1555–3290 (2000)*	1545–3272 (2000)	1562–3278 (2000)
PUMP FLOW RATE, LB_M/SEC	3.59 – 3.94 (3.77)	3.37 – 4.04 (3.72)	3.74 – 4.47 (4.11)
PUMP DISCHARGE PRESS., $LB/IN.^2A$	1368–1749 (1550)	1040–1677 (1330)	1022–1539 (1257)
CONDITIONED PROPELLANT TEMP, ^0R	219–320 (263)	205–323 (253)	205–332 (245)

Oxygen

	SERIES RCS TURBINE UPSTREAM	SERIES RCS TURBINE DOWNSTREAM	PARALLEL RCS
GAS GENERATOR COMBUSTION TEMP, ^0R	1555–3290 (2000)	1541–3277 (2000)	1567–3296 (2000)
PUMP FLOW RATE, LB_M/SEC	11.09–12.44 (11.77)	10.51–13.09 (11.71)	11.27–13.69 (12.43)
PUMP DISCHARGE PRESS., $LB/IN.^2A$	1665–2089 (1875)	1457–2286 (1817)	1356–2005 (1651)
CONDITIONED PROPELLANT TEMP, ^0R	396–656 (503)	388–710 (506)	374–724 (466)

*( ) DESIGN VALUE

Figure C-64

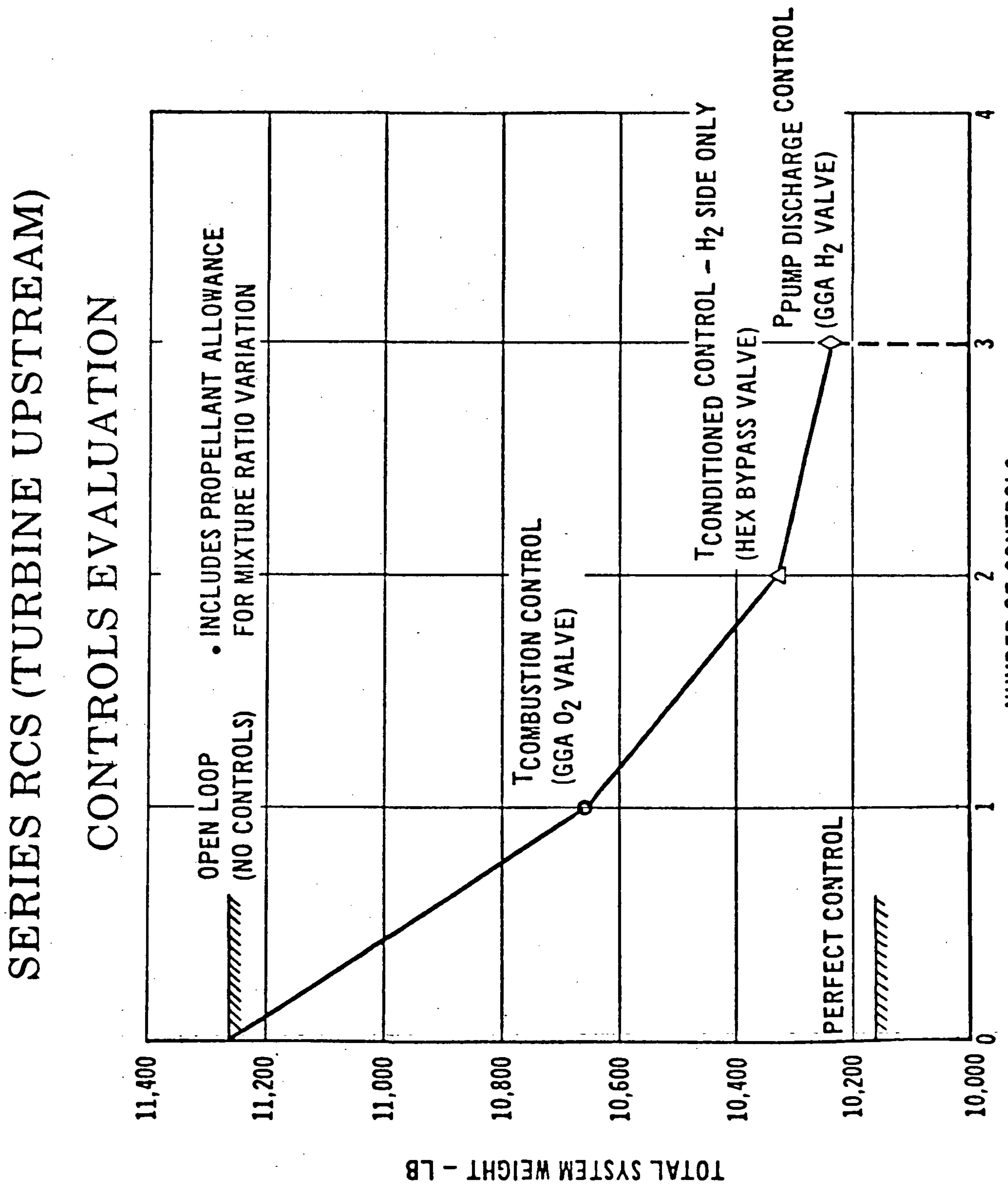

SERIES RCS (TURBINE UPSTREAM)
CONTROLS EVALUATION
• INCLUDES PROPELLANT ALLOWANCE
FOR MIXTURE RATIO VARIATION
OPEN LOOP
(NO CONTROLS)
$T_{COMBUSTION}$ CONTROL
(GGA O_2 VALVE)
$T_{CONDITIONED}$ CONTROL – H_2 SIDE ONLY
(HEX BYPASS VALVE)
P_{PUMP} DISCHARGE CONTROL
(GGA H_2 VALVE)
PERFECT CONTROL
TOTAL SYSTEM WEIGHT – LB
11,400
11,200
11,000
10,800
10,600
10,400
10,200
10,000
NUMBER OF CONTROLS
0
1
2
3
4

Figure C-65

SERIES RCS (TURBINE DOWNSTREAM) CONTROLS EVALUATION

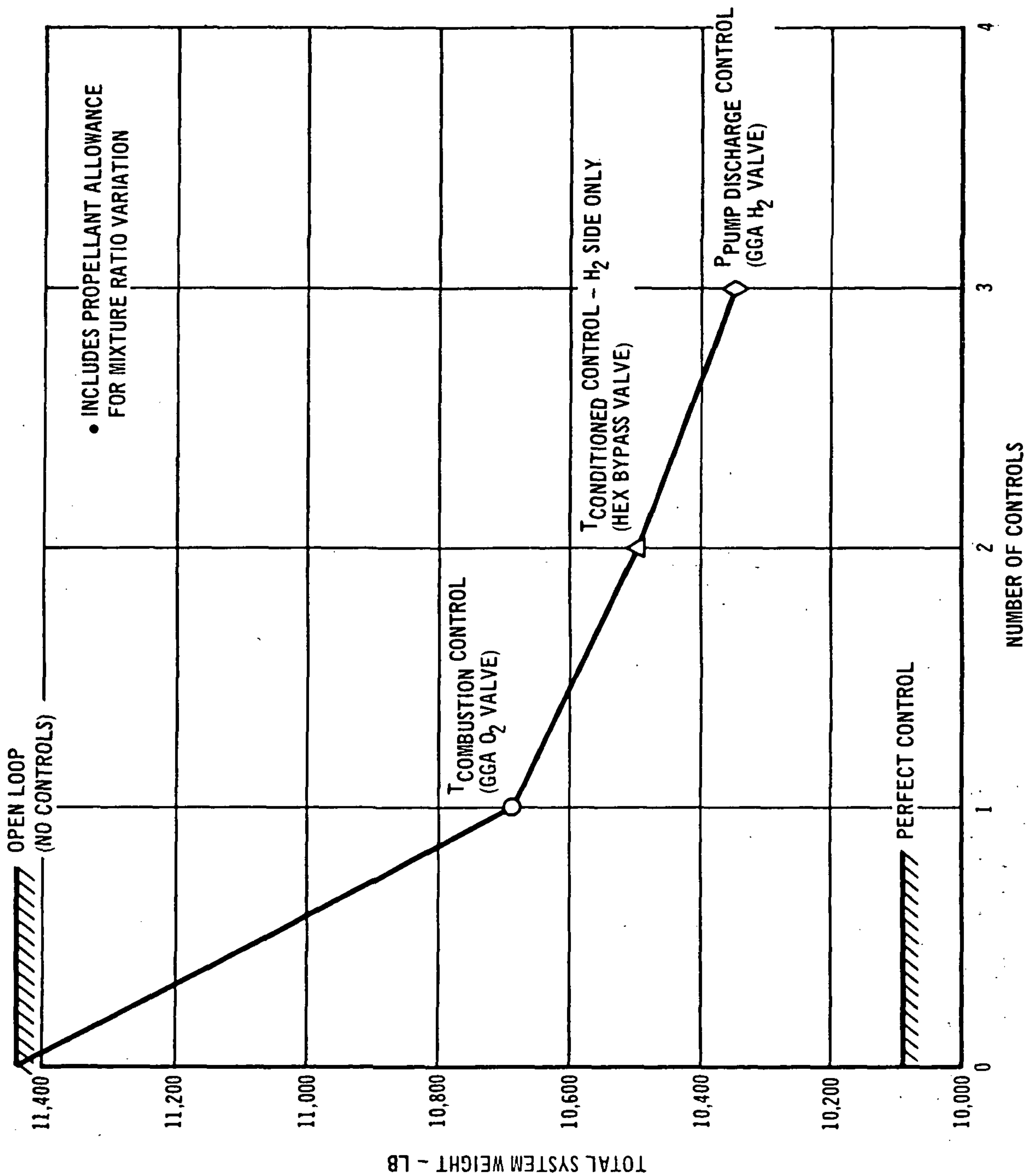


Figure C-66

PARALLEL RCS CONTROLS EVALUATION

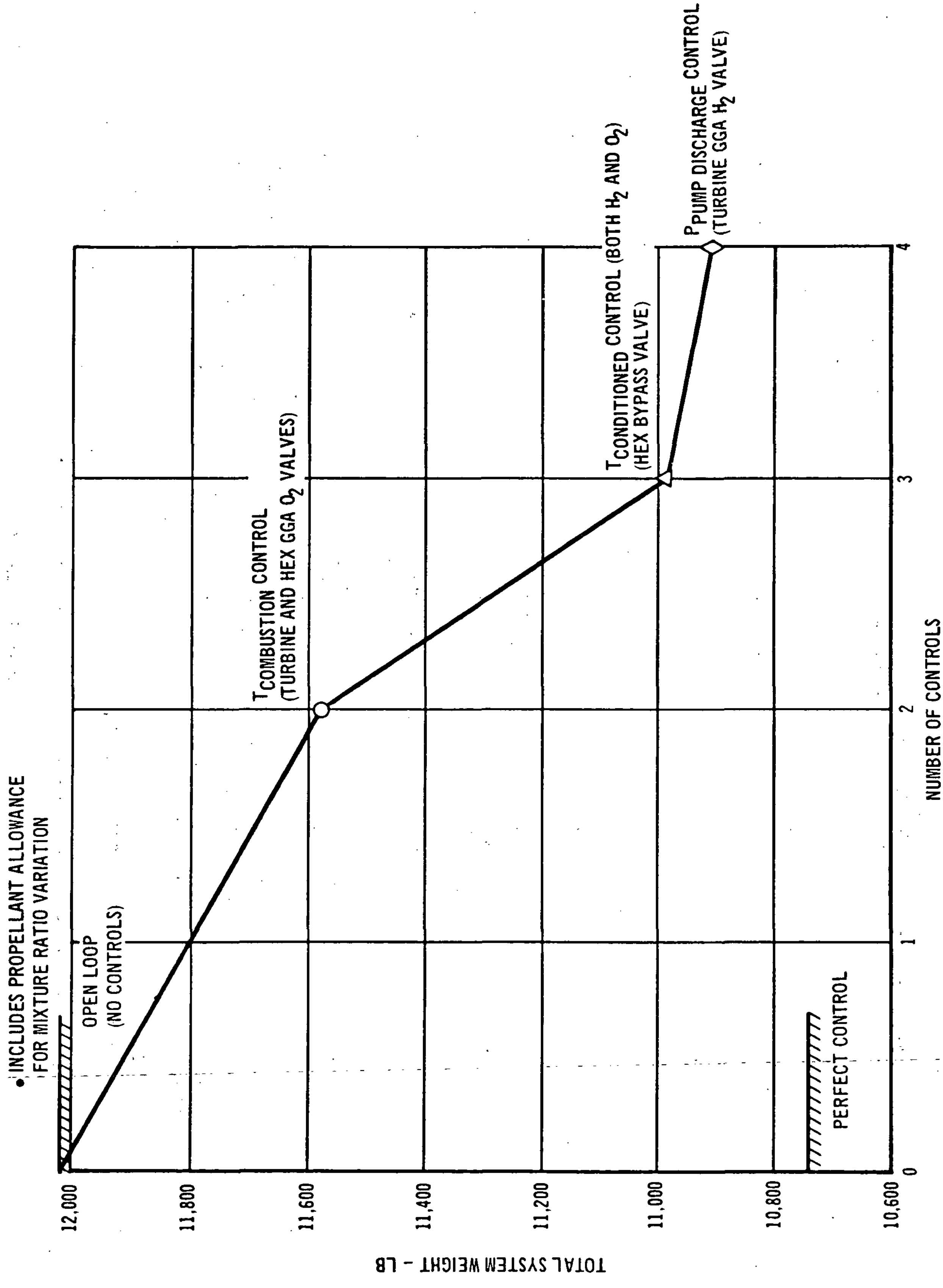


Figure C-67

SERIES RCS (TURBINE UPSTREAM) CONDITIONER PERFORMANCE MAP

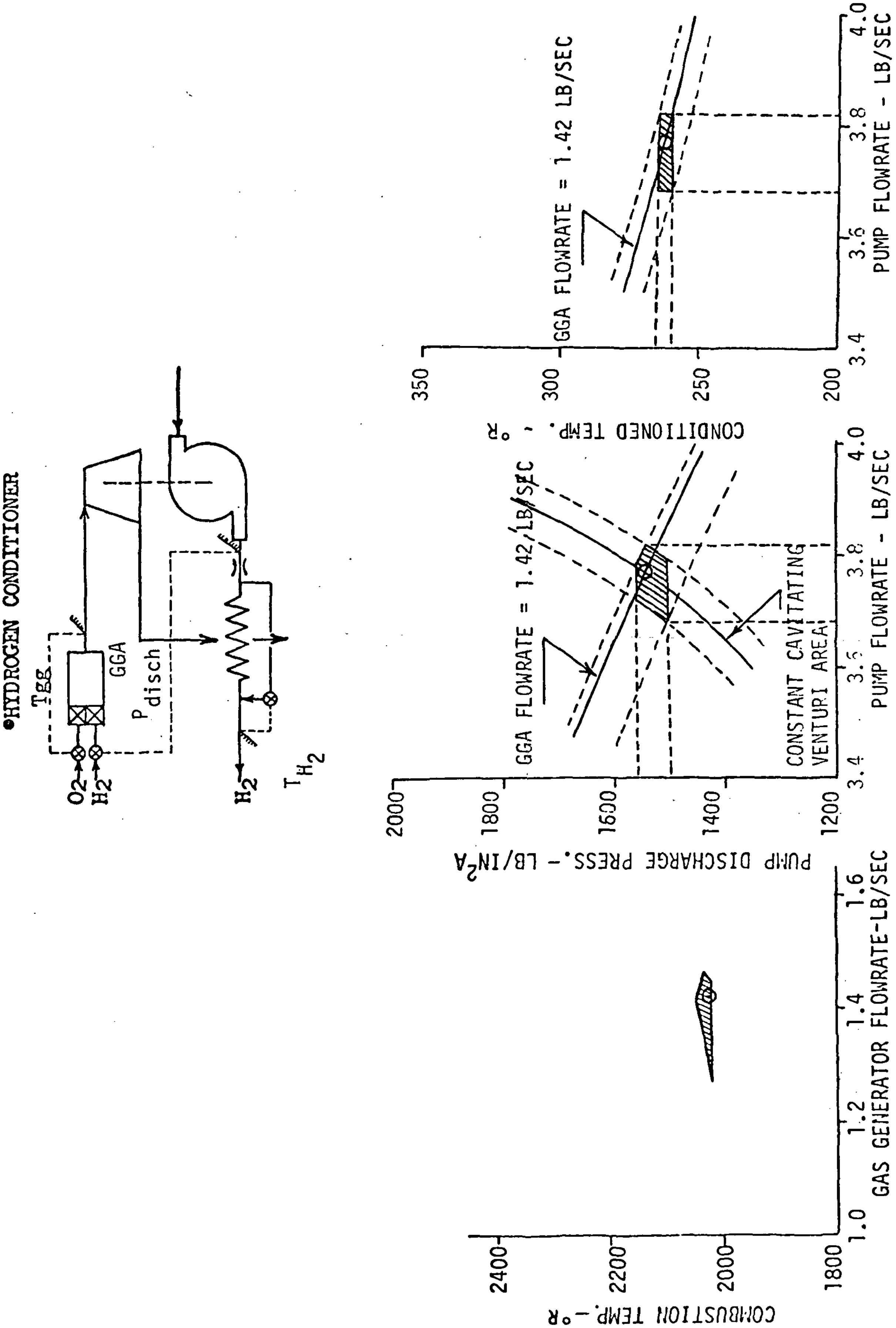


Figure C-68

SERIES RCS (TURBINE DOWNSTREAM) CONDITIONER PERFORMANCE MAP

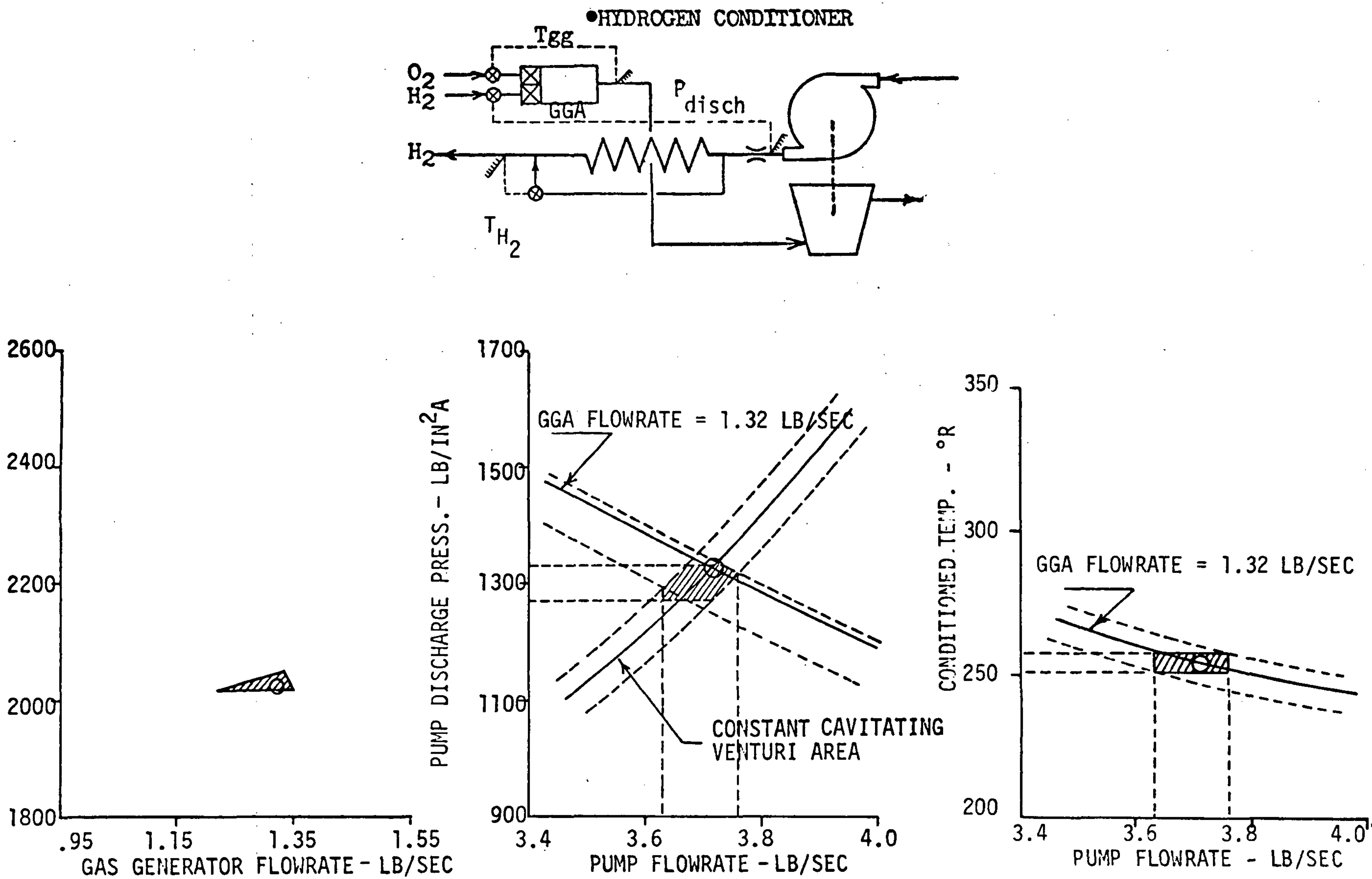


Figure C-69

The rationale for parallel RCS controls selection is similar to that for the two series concepts. However, unlike the series concepts in which oxygen conditioned temperature control provided only a modest decrease in system weight (< 100 lbm), this control provides a savings of approximately 200 lbm in the parallel RCS. This greater weights savings results from the fact that in the parallel RCS no control other than the gas generator O_2 valve is exercised over heat exchanger hot side flow. Hence, the operating band on conditioned temperature (without bypass flow modulation) approaches the open-loop limits. In the series RCS, modulation of the gas generator H_2 valve to provide constant pump discharge pressure also provides secondary control of heat exchanger hot side flow, narrowing the operational band on conditioned temperature. Therefore, because of the substantial weight savings, oxygen conditioned temperature control is desirable for the parallel RCS. Typical parallel RCS conditioner performance with selected controls is shown in the hydrogen operating maps of Figure C-70.

The primary conclusions derived from the above controls evaluations are:

(1) Active controls are required to provide acceptable conditioner performance.

(2) With an all-active approach (no passive mass flow control), excellent control of combustion temperature and the conditioner interface parameters (conditioned temperature, pressure, and pump flow rate) can be achieved.

(3) Combustion temperature control is mandatory to prevent excessive turbine/heat exchanger gas inlet temperatures, and is best achieved using the gas generator O_2 valve.

(4) Conditioned temperature has the greatest impact on system weight and although some control is exercised through the gas generator O_2 valve, additional control is necessary to achieve substantial weight reductions. The best and most direct control approach is to modulate heat exchanger cold side bypass flow.

(5) Pump discharge pressure and flow rate control provides a further weight reduction and is desired to minimize potential heat exchanger cold side flow instability. It is best achieved by modulating the gas generator H_2 valve in response to pump discharge pressure.

PARALLEL RCS CONDITIONER OPERATING MAP

HYDROGEN CONDITIONER

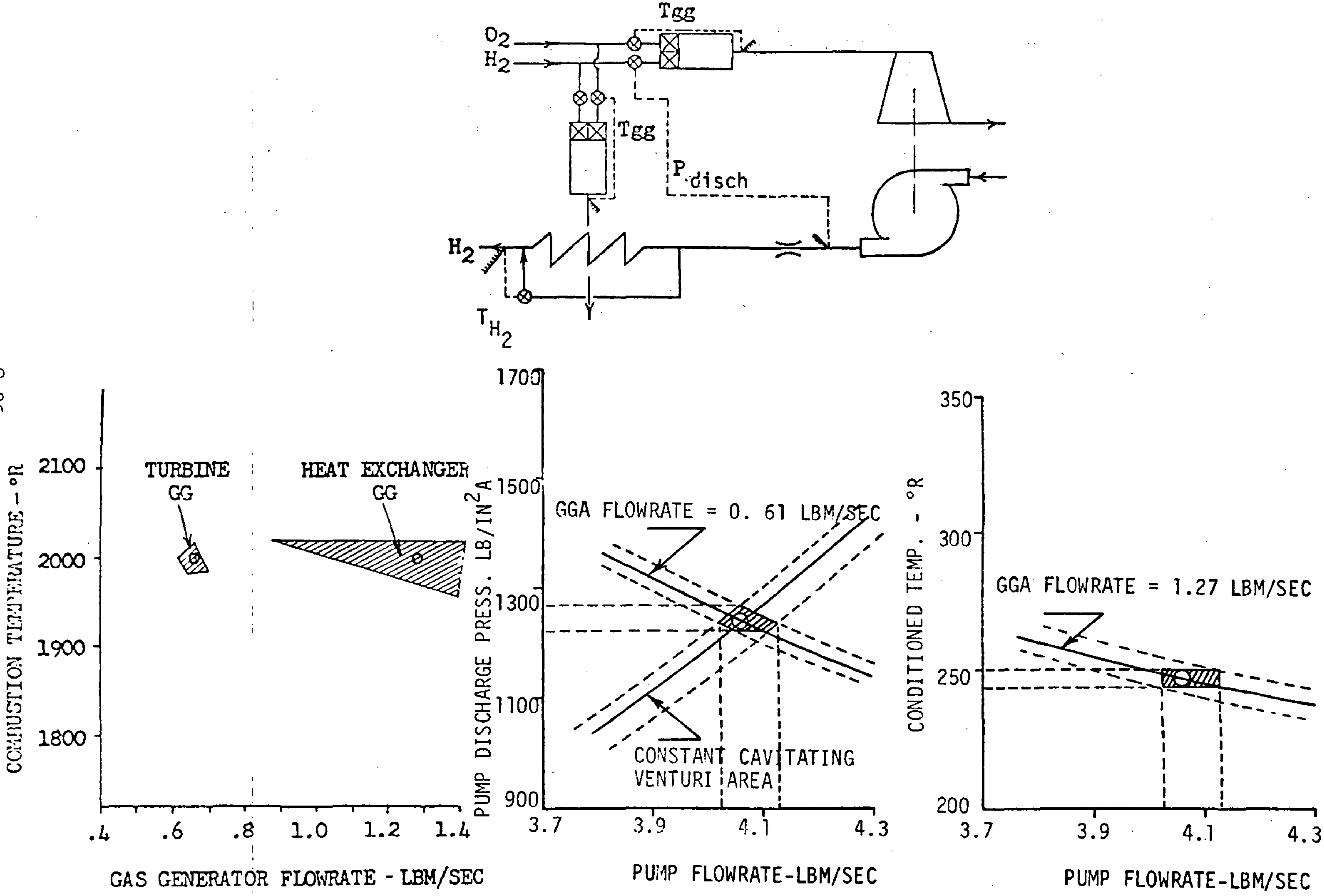


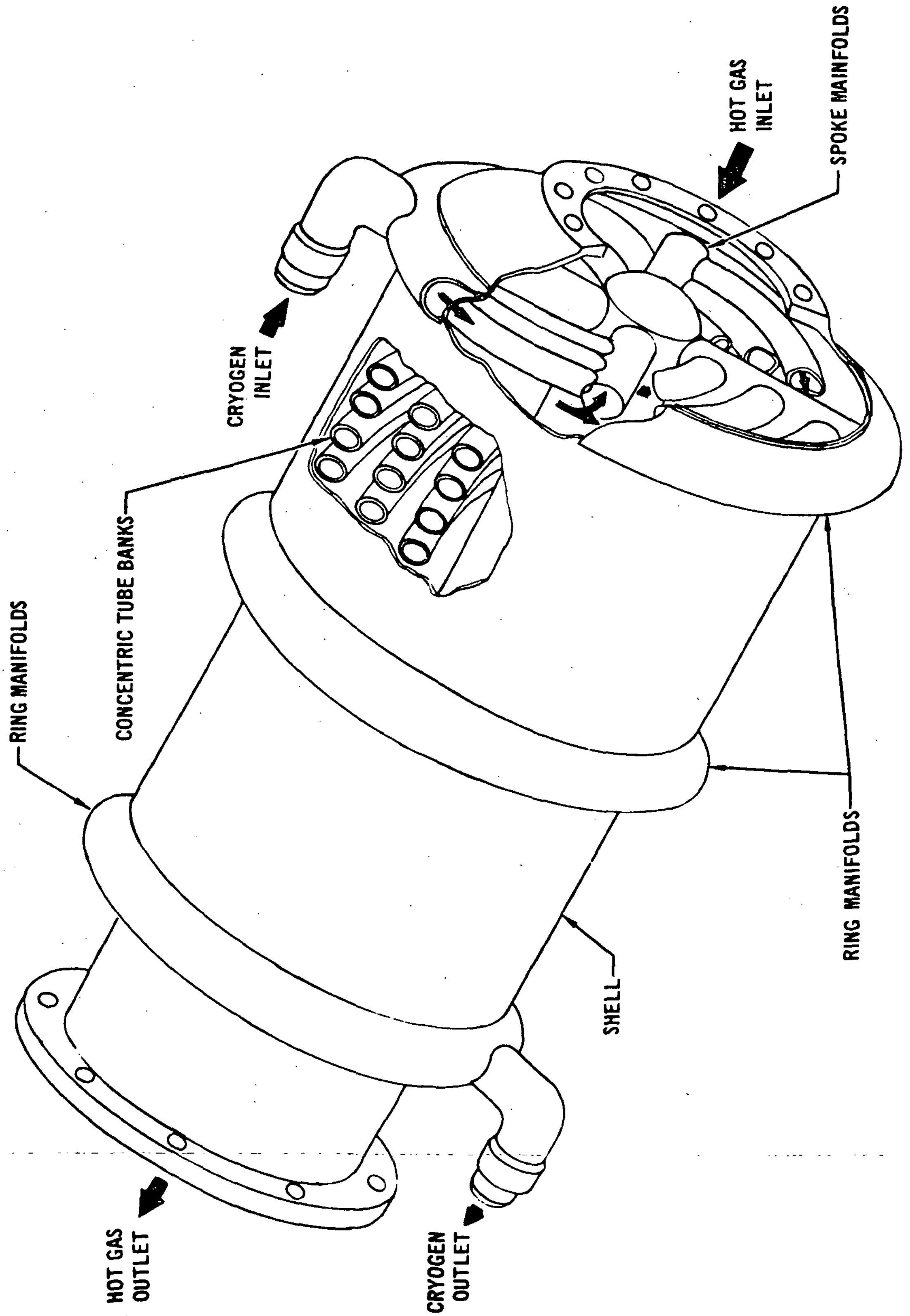
Figure C-70

APPENDIX D

FINAL HEAT EXCHANGER DESIGN

As discussed in Appendix A, preliminary heat exchanger designs (Figure D-1) for the series-upstream turbine RCS (hydrogen conditioner, only) and the parallel RCS (both oxygen and hydrogen conditioners) provided limited margin in operating conditions before freezing would occur on the hot side tube walls. Subsequent analyses of conditioner operation showed that for the worst case tolerances on gas generator inlet temperatures/pressures and component flow areas, water vapor in the gas generator exhaust flow would condense and freeze on the hot side tube walls. Various heat exchanger design alternatives were considered to prevent this icing condition. However, based on the rationale of Figure D-2, the only practical approach was to incorporate a cold side bypass circuit. This reduced the cold side heat exchanger flow rate and increased both the cold side exit temperature and bulk tube wall temperature, thus preventing ice formation. However, since cold side exit temperatures were higher than required for accumulator resupply, final thermal conditioning was achieved by mixing the heat exchanger cold side exit and bypass flows. The impact of this cold side bypass circuit on H_2O icing probability is shown in the example of Figure D-3 for the series-upstream turbine RCS, hydrogen conditioner. As shown by the imposed conditioner operating envelopes, icing can occur without bypass while it is precluded entirely with a 50% bypass loop. As a result, the affected heat exchangers were reconfigured to provide bypass circuits. Whereas icing was not probable on the series-downstream turbine RCS, a hydrogen heat exchanger bypass circuit was incorporated for the purpose of conditioned temperature control (Appendix C). The revised heat exchanger configurations for all three RCS concepts are described in Figure D-4, and operating performance maps are presented in Figures D-5 through D-10.

CONCENTRIC TUBE AND SHELL HEAT EXCHANGER MODEL



RATIONALE FOR USE OF HEAT EXCHANGER COLD SIDE BYPASS

PROBLEM - FOR WORST CASE OPERATING CONDITIONS, H_2O IN EXHAUST WILL CONDENSE AND FREEZE ON HOT SIDE TUBE WALLS (H_2 SIDE OF PARALLEL AND SERIES/UPSTREAM TURBINE RCS, AND O_2 SIDE OF PARALLEL RCS).

DISCUSSION-

DEPENDENT VARIABLES	CONSIDERATIONS
o HOT SIDE INLET TEMPERATURE	o CONSTRAINED TO 2000°R TO ALLOW CONVENTIONAL HEAT EXCHANGER/TURBINE DESIGN
o COLD SIDE ΔT	o ΔT'S OF APPROXIMATELY 200 AND 300°R ARE REQUIRED FOR H_2 AND O_2, RESPECTIVELY.
o HEAT EXCHANGER CONFIGURATION	o TUBE IN SHELL CONCEPT SELECTED AS A PRACTICAL APPROACH FOR LOW IMPULSE (1.0 - 3.0$\overline{M}$ LB-SEC) MISSIONS. SIZING IS CONSTRAINED BY PRACTICAL WEIGHT AND FABRICATION CONSIDERATIONS.
o RATIO OF HOT TO COLD SIDE FLOW	o MAY BE INCREASED THROUGH USE OF COLD SIDE BYPASS WITH NEGLIGIBLE EFFECT ON SYSTEM WEIGHT

SOLUTION - CAREFULLY CONFIGURE HEAT EXCHANGER AND INCORPORATE COLD SIDE BYPASS CIRCUIT TO PRECLUDE ICING. COLD SIDE BYPASS WILL ADD FLEXIBILITY TO HEAT EXCHANGER PERFORMANCE CAPABILITY.

Figure D-2

EXAMPLE HEAT EXCHANGER OPERATING CHARACTERISTICS

- *SERIES RCS (TURBINE UPSTREAM)*
- HYDROGEN CONDITIONER
- GGA COMBUSTION TEMP. CONTROL (GGA O_2 THROTTLE VALVE)
- TURBINE POWER CONTROL (GGA H_2 THROTTLE VALVE)

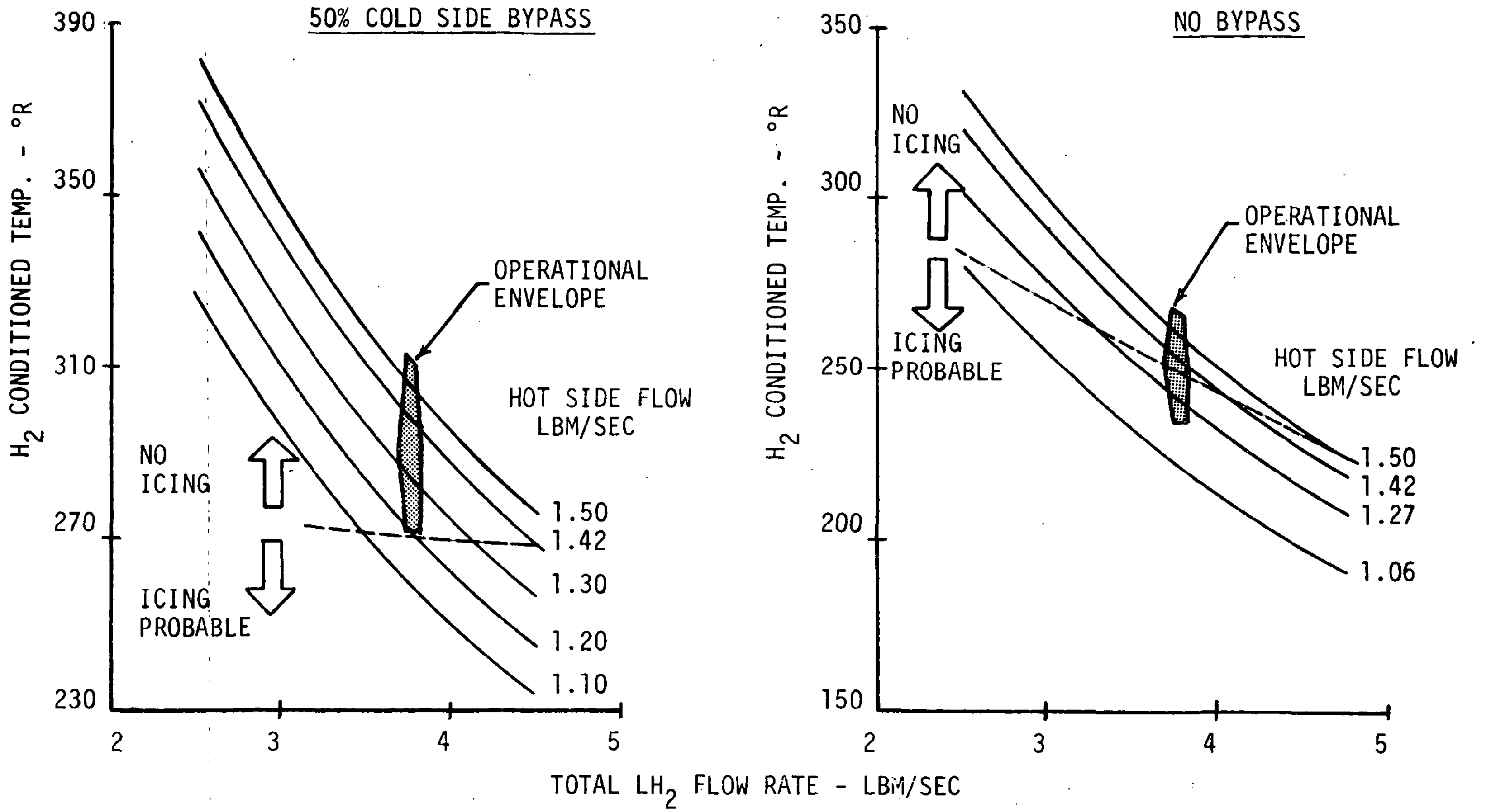


Figure D-3

HEAT EXCHANGER CONFIGURATIONS
(TUBE IN SHELL CONCEPT)

• HYDROGEN	SERIES TURBINE UPSTREAM	SERIES TURBINE DOWNSTREAM	PARALLEL
BYPASS RATIO	0.50	0.60	0.65
NO. CONCENTRIC RINGS	6/5*	5	5
NO. SPOKES	12	12	12
TUBE O.D. (IN.)	0.313/0.433*	0.313/0.433*	0.313/0.433*
TUBE WALL THICKNESS (IN.)	0.016	0.016	0.016
RADIAL GAP (IN.)	0.15/0.125*	0.10	0.10
EFFECTIVE SURFACE AREA (FT^2)	39.8	38.75	42.5
LENGTH (IN.)	39.9	29.1	31.5
WEIGHT (LBM)	63.7	42.7	48.4

• OXYGEN	SERIES TURBINE UPSTREAM	SERIES TURBINE DOWNSTREAM	PARALLEL
BYPASS RATIO	0.	0.	0.25
NO. CONCENTRIC RINGS	5	5	5
NO. SPOKES	12	12	12
TUBE O.D. (IN.)	0.25	0.34	0.34
TUBE WALL THICKNESS (IN.)	0.016	0.016	0.016/0.016*
RADIAL GAP (IN.)	0.150	0.150	0.150/0.125*
EFFECTIVE SURFACE AREA (FT^2)	23.0	28.1	48.3
LENGTH (IN.)	23.1	19.7	36.2
WEIGHT (LBM)	26.9	41.4	50.2

* UPSTREAM/DOWNSTREAM (OF CENTER RING MANIFOLD)

Figure D-4

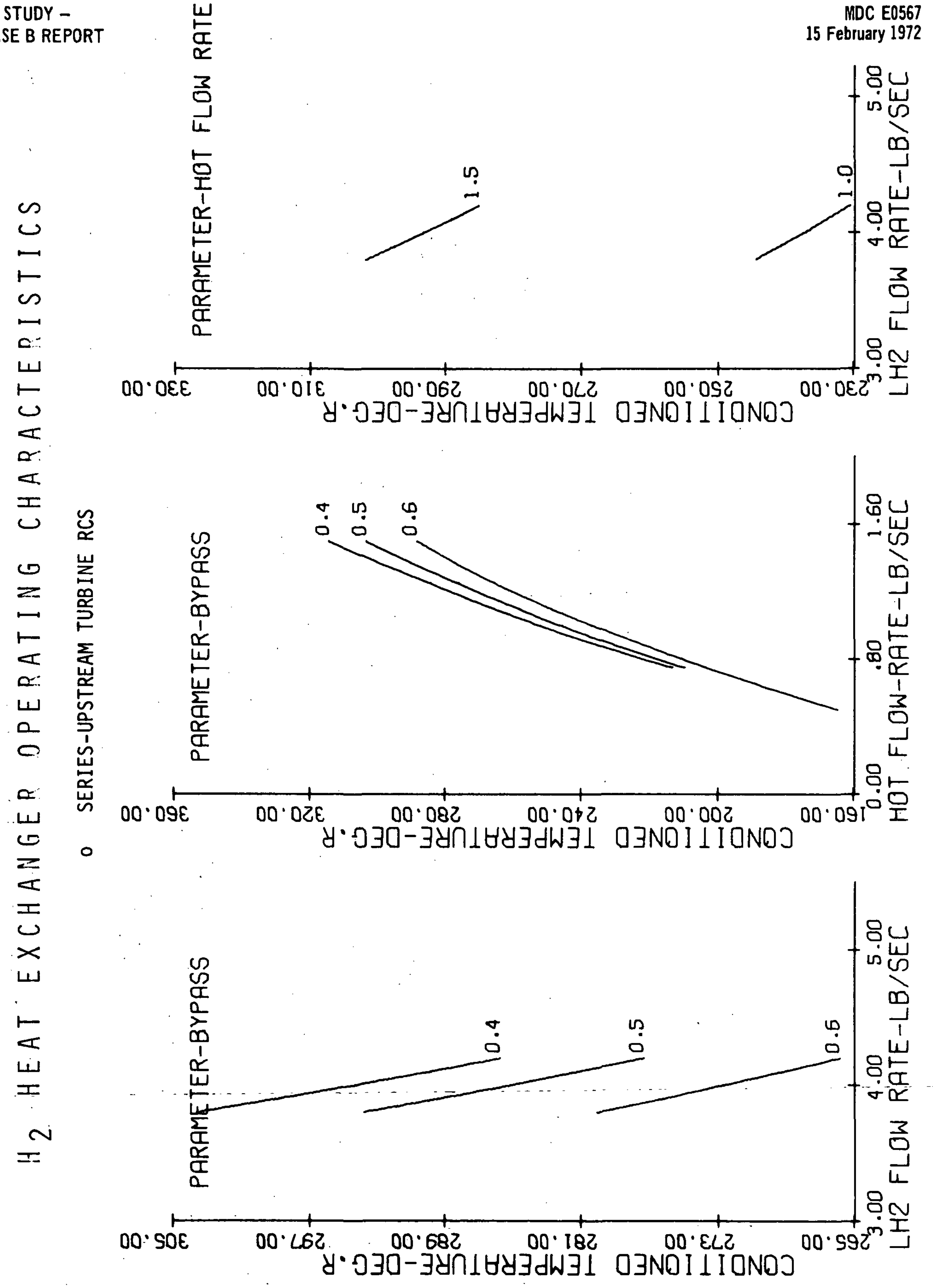

H2 HEAT EXCHANGER OPERATING CHARACTERISTICS
o SERIES-UPSTREAM TURBINE RCS
PARAMETER-HOT FLOW RATE
1.5
1.0
CONDITIONED TEMPERATURE-DEG.R
330.00
310.00
290.00
270.00
250.00
230.00
LH2 FLOW RATE-LB/SEC
3.00
4.00
5.00
PARAMETER-BYPASS
0.4
0.5
0.6
CONDITIONED TEMPERATURE-DEG.R
360.00
320.00
280.00
240.00
200.00
160.00
HOT FLOW-RATE-LB/SEC
0.00
.80
1.60
PARAMETER-BYPASS
0.4
0.5
0.6
CONDITIONED TEMPERATURE-DEG.R
305.00
297.00
289.00
281.00
273.00
265.00
LH2 FLOW RATE-LB/SEC
3.00
4.00
5.00

Figure D-5

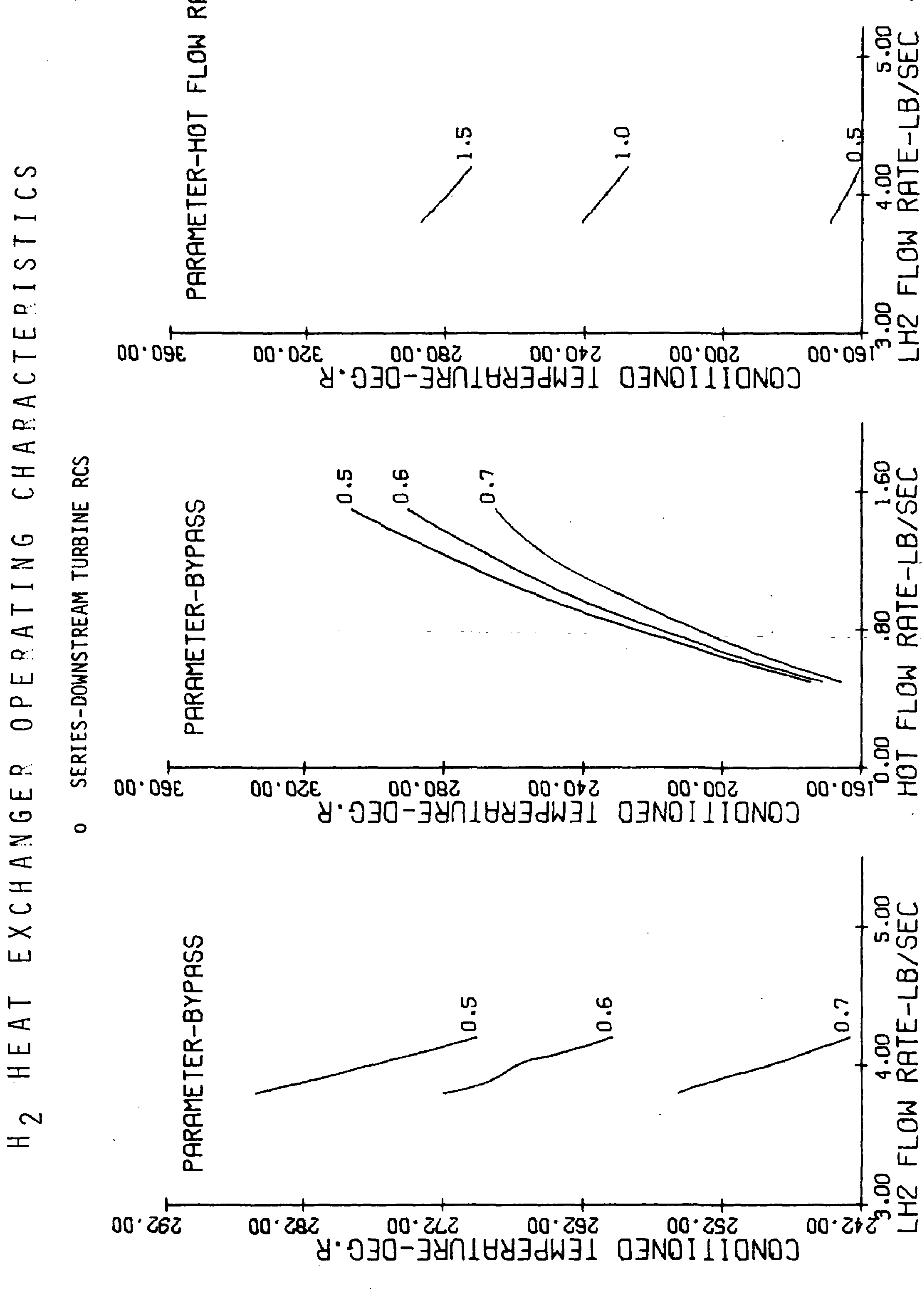

H2 HEAT EXCHANGER OPERATING CHARACTERISTICS
o SERIES-DOWNSTREAM TURBINE RCS
PARAMETER-HOT FLOW RATE
1.5
1.0
0.5
CONDITIONED TEMPERATURE-DEG.R
160.00
200.00
240.00
280.00
320.00
360.00
LH2 FLOW RATE-LB/SEC
3.00
4.00
5.00
PARAMETER-BYPASS
0.5
0.6
0.7
HOT FLOW RATE-LB/SEC
0.00
.80
1.60
PARAMETER-BYPASS
0.5
0.6
0.7
242.00
252.00
262.00
272.00
282.00
292.00

Figure D-6

H_2 HEAT EXCHANGER OPERATING CHARACTERISTICS

o PARALLEL RCS

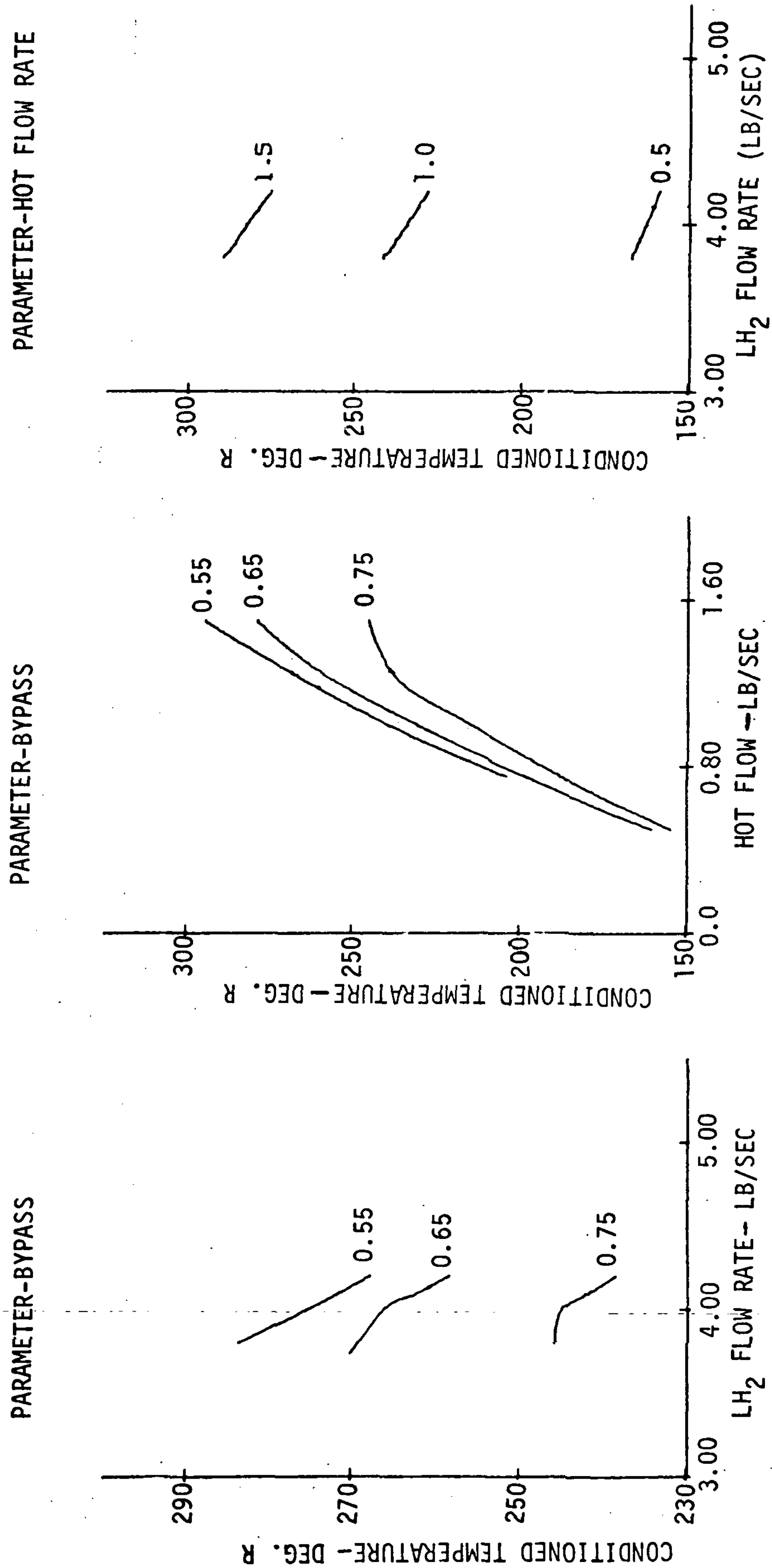


Figure D-7

O_2 HEAT EXCHANGER OPERATING CHARACTERISTICS

o SERIES-UPSTREAM TURBINE RCS

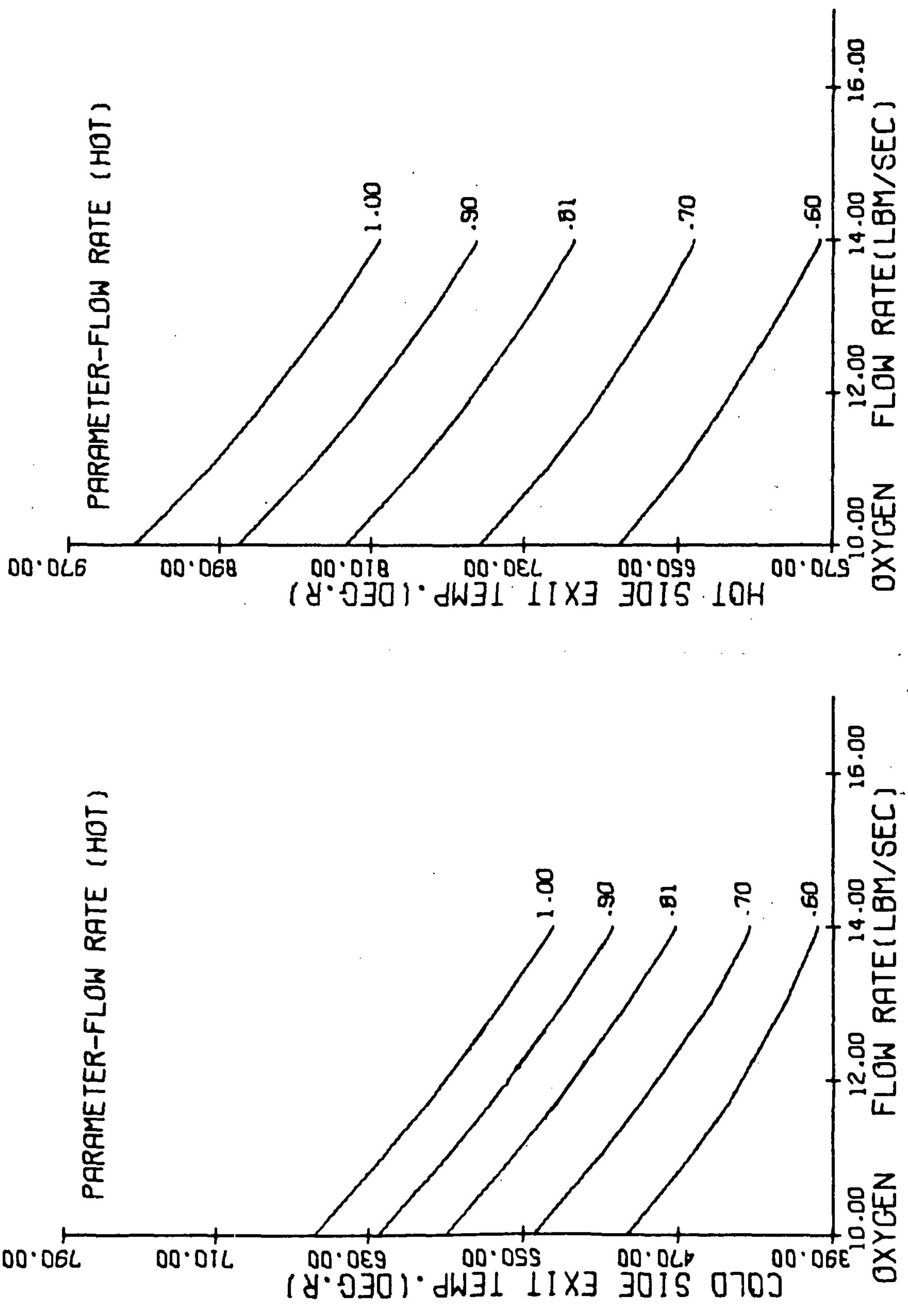


Figure D-8

O_2 HEAT EXCHANGER OPERATING CHARACTERISTICS

o SERIES-DOWNSTREAM TURBINE RCS

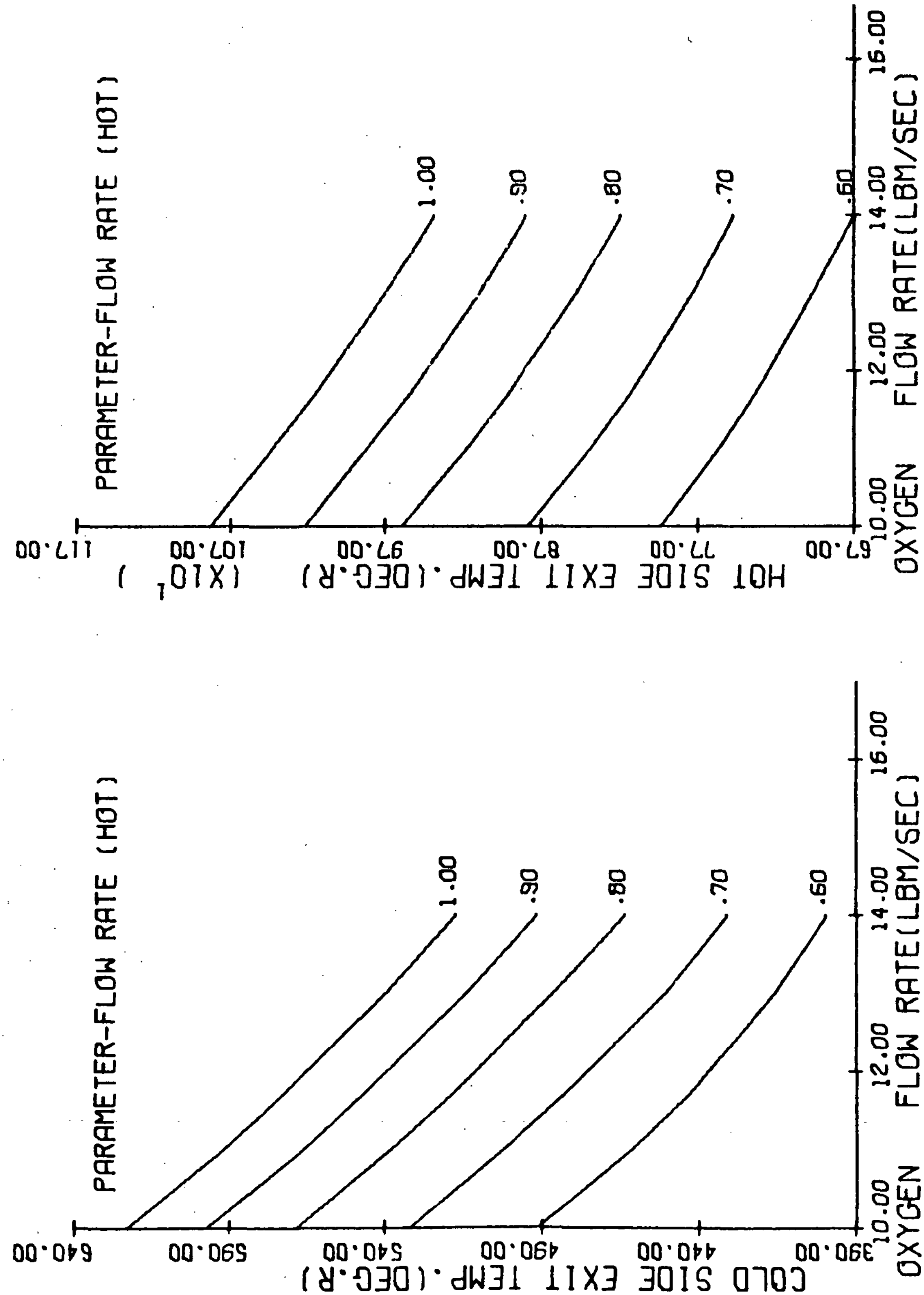
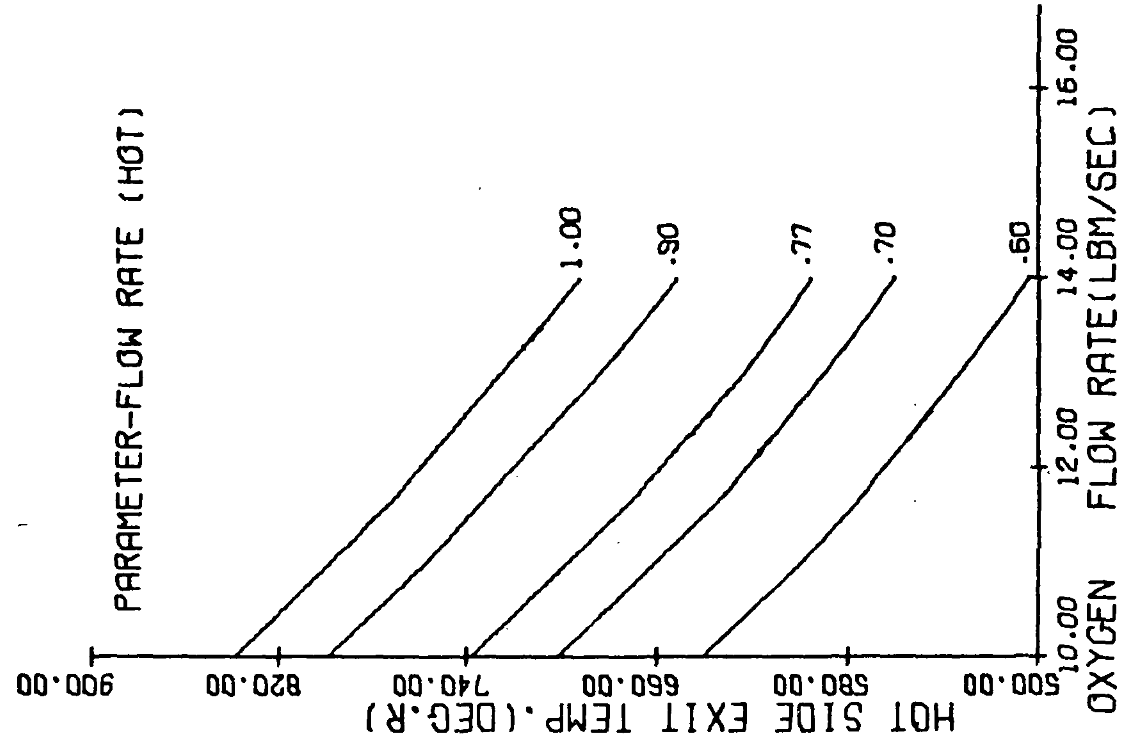


Figure D-9

O_2 HEAT EXCHANGER OPERATING CHARACTERISTICS

o PARALLEL RCS



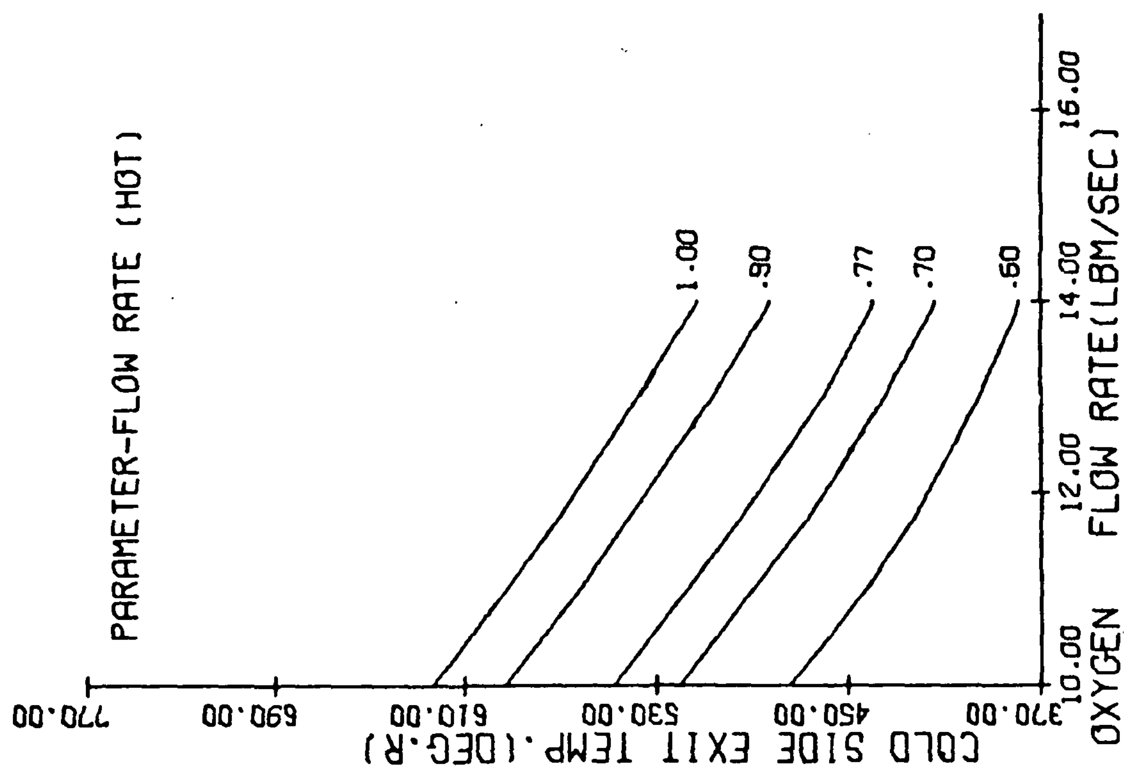


Figure D-10

APPENDIX E

CONDITIONER FAILURE MODE AND EFFECTS ANALYSIS

A failure mode and effect analysis (FMEA) was conducted for the candidate RCS concepts. The purpose of the FMEA was to examine each component failure mode and determine its effect on system operation. This analysis provided the basis for establishing the redundancy required to meet the fail-operational/fail-safe (FO/FS) failure criteria of Reference G, and for establishing preliminary instrumentation requirements for malfunction detection.

Figures E-1 and E-2 present schematics for the candidate RCS concepts. The component identification numbers shown on the schematics were assigned to provide a cross-reference between the schematics and the FMEA.

Figures E-3 and E-4 present a preliminary list of operating parameters monitored to provide control points and to detect malfunctions. The output of a single sensor at each control point is used to control each conditioner. Parallel redundant sensors are used for detection of critical malfunctions. A malfunction signal from either of the redundant sensors alerts the crew or shuts down the conditioner depending on the necessity for fast response.

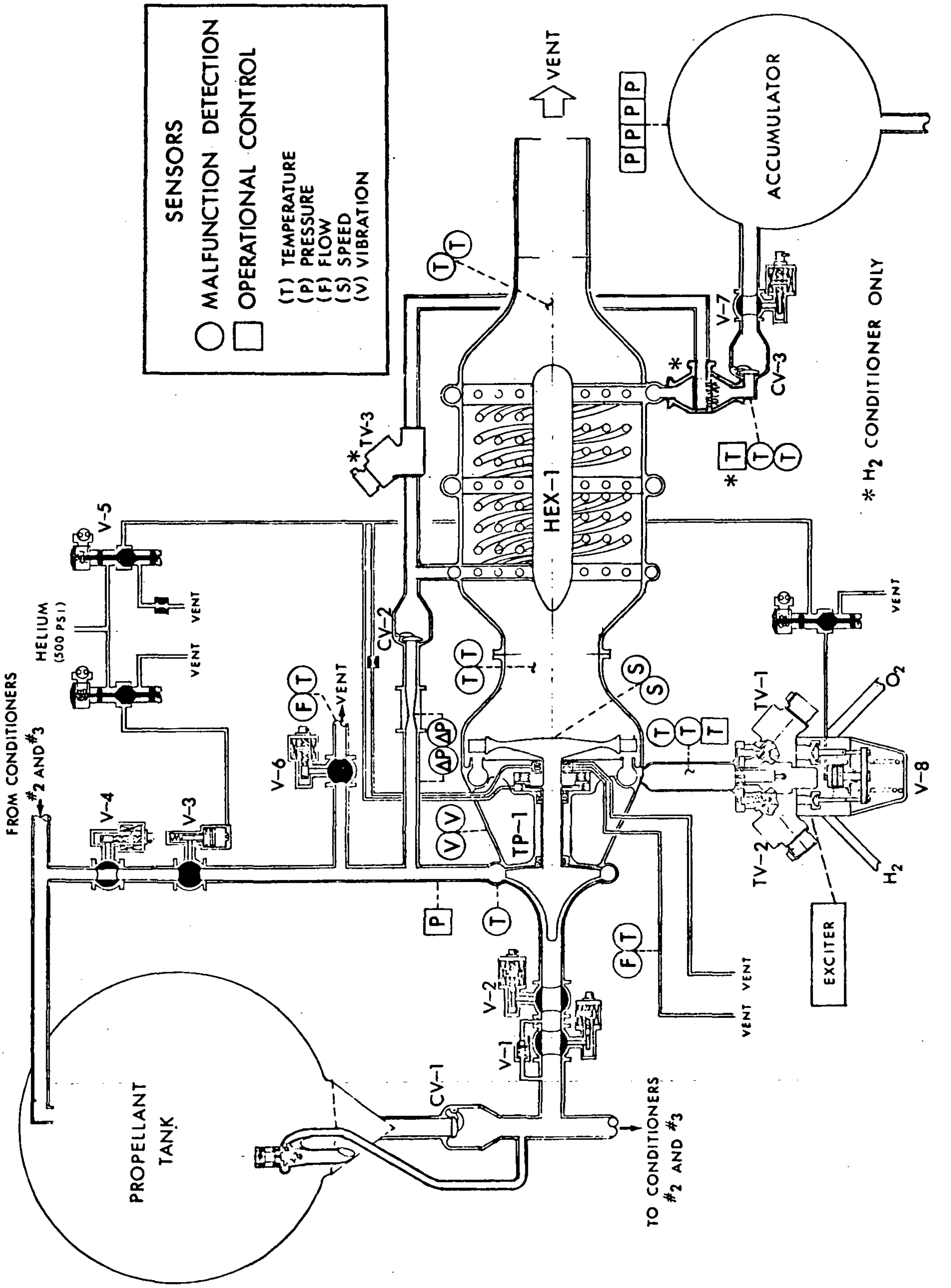
The FMEA discusses component failure modes for the following operating conditions or phases:

- Start
- Steady State Operation
- Shutdown and Idle
- Standby (Another Conditioner Operating)

Only those failure modes affecting system operation during a particular phase are discussed under that phase. A complete FMEA is presented in Figure E-5 for the "Series RCS" (series hot gas flow through the turbine and heat exchanger - one gas generator). The failure modes and effects are the same for the turbine downstream concept. The FMEA for the "Parallel RCS" (parallel hot gas flow through turbine and heat exchanger - separate gas generators) is limited to added components and those failure modes resulting in effects that differ from effects of the same failure mode in the "Series RCS". These additional items are presented in Figure E-6.

External leakage of propellants or hot gas are considered as subsystem failure modes. External leakage problems can be minimized by special attention to component design details and adequate structural design margins for the hot gas components.

SERIES
RCS CONDITIONER SCHEMATIC



PARALLEL RCS CONDITIONER SCHEMATIC

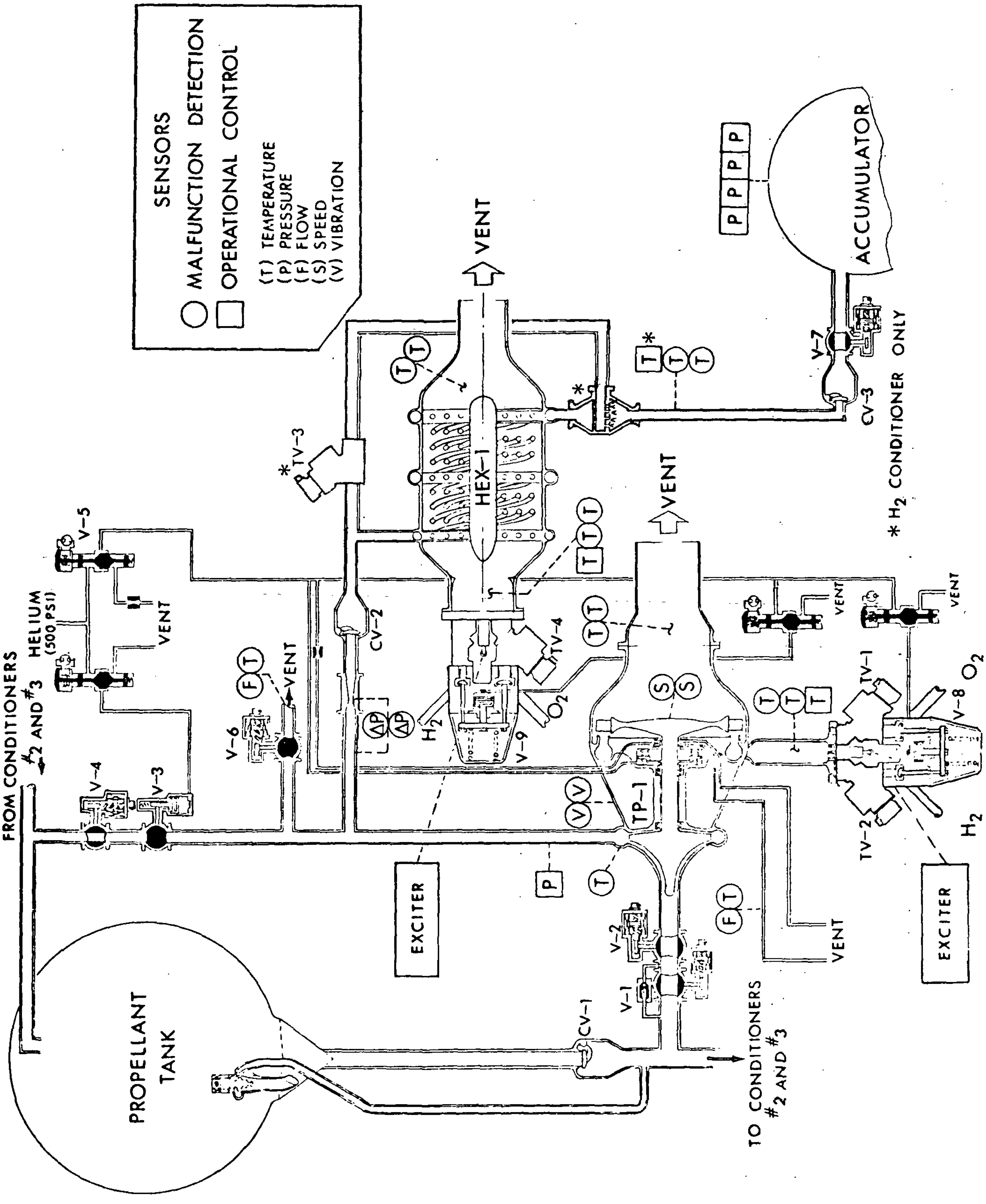


Figure E-2

CONDITIONER INSTRUMENTATION

SERIES RCS

PARAMETER	APPLICATION		
	OPERATIONAL CONTROL	MALFUNCTION DETECTION	
		AUTOMATIC SHUTDOWN	CREW DISPLAY
GGA COMBUSTION TEMPERATURE	√	√	
TURBOPUMP SPEED		√	
TURBOPUMP VIBRATION		√	
TURBOPUMP COLD FLOW DISCHARGE TEMPERATURE			√
TURBOPUMP COLD FLOW DISCHARGE PRESSURE	√		
VENTURI △ PRESSURE		√	
TURBINE DISCHARGE TEMPERATURE			√
O_2 TURBOPUMP SEAL VENT FLOW AND TEMPERATURE			√
TURBOPUMP VENT FLOW AND TEMPERATURE			√
HEAT EXCHANGER COLD SIDE DISCHARGE TEMPERATURE	√		√
HEAT EXCHANGER HOT SIDE TEMPERATURE		√	
ACCUMULATOR PRESSURE	√		√

Figure E-3

CONDITIONER INSTRUMENTATION

PARALLEL RCS

PARAMETER	APPLICATION		
	OPERATIONAL CONTROL	MALFUNCTION DETECTION	
		AUTOMATIC SHUTDOWN	CREW DISPLAY
GGA COMBUSTION TEMPERATURE	√	√	
TURBOPUMP SPEED		√	
TURBOPUMP VIBRATION		√	
TURBOPUMP COLD FLOW DISCHARGE TEMPERATURE			√
TURBOPUMP COLD FLOW DISCHARGE PRESSURE	√		
VENTURI Δ PRESSURE		√	
TURBINE DISCHARGE TEMPERATURE			√
O_2 TURBOPUMP SEAL VENT FLOW AND TEMPERATURE			√
TURBOPUMP VENT FLOW AND TEMPERATURE			√
HEAT EXCHANGER COLD SIDE DISCHARGE TEMPERATURE	√	√	
HEAT EXCHANGER HOT SIDE TEMPERATURE		√	
ACCUMULATOR PRESSURE	√		√

Figure E-4

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

START

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Sensor, Accumulator Pressure	Provides signal to the controller that accumulator pressure has dropped to the recharge level.	Senses higher than actual pressure.	In a nonredundant system this failure could allow accumulator pressure to fall below the critical recovery level.	This sensor is one of four redundant accumulator pressure sensors. The controller will incorporate voting or averaging logic to detect a malfunctioning sensor.
Valve, Helium, Pump Lift-off Seal and Seal Purge V-5 (Seal Purge applicable to O_2 only)	Opened at start command to supply high pressure helium to the lift-off seal bellows, pump seal purge line, and the GGA propellant valve pilot. Pump seal purge applicable to O_2 pump only.	Fails closed.	The conditioner will not start. This valve must open to supply high pressure helium to the GGA bipropellant valve for actuation.	The primary parameter for sensing failure of a conditioner to start and deliver propellant to the accumulator will be venturi Δpressure. When the start signal is generated a time delay is initiated. If a suitable flow rate has not been established through the venturi at the end of the time delay, the conditioner will be shut down and one of two standby conditioner assemblies will be activated. Failure of the GGA to provide power will be detected earlier by monitoring GGA combustion temperature, turbine speed, or pump discharge pressure.
Valve, GGA Bipropellant Shut-off V-8	Opened at conditioner start command to supply fuel and oxidizer to the GGA.	Fails closed.	The conditioner will not start.	Same as preceding entry.
		Slow open.	Conditioner may be shut down due to slow power buildup with the resulting slow buildup of pump discharge pressure.	Same as preceding entry.
Igniter, GGA (Exciter and Spark Plug)	Provides high voltage spark for propellant ignition.	Fails to provide spark.	Conditioner will not start.	Same as preceding entry - plus monitoring of exciter output to detect loss of high voltage power.
		Low power or intertent spark.	Possible delayed ignition or failure to ignite propellant. Could allow excessive accumulation of propellants in the GGA before ignition and result in a hard start with a potential for structural damage.	Same as preceding entry plus the effect of a delayed ignition should be investigated during system development. It may be necessary to verify that torch has ignited before initiating main propellant flow.
Injector, GGA Propellant	Provides optimum mixing of propellants in GGA combustion chamber.	Restricts fuel flow.	The oxidizer rich propellant mixture may result in GGA overtemperature condition which will cause shutdown of the conditioner	Overtemperature will be detected by monitoring GGA combustion temperature, and turbine exhaust temperature. Two standby conditioners provide FO/FS redundancy.
		Restricts oxidizer flow	The fuel rich propellant mixture results in combustion inefficiency and possible degraded performance. May cause shutdown if turbopump discharge pressure is inadequate. There is a potential for icing in the heat exchanger.	Low combustion temperature will be detected by monitoring GGA combustion temperature and turbine exhaust temperature. Degraded performance will be detected by monitoring venturi Δpressure and heat exchanger cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
Valve, Throttle, GGA Oxidizer TV-1	Opens gradually during start period to aid in controlling turbine power buildup.	Opens too fast.	Excessive GGA combustion temperature and turbine power buildup could result in structural failure of turbine blades or heat exchanger and/or degraded pump bearing life due to "sledding."	The effects of this failure will be detected by monitoring GGA combustion temperature, turbine exhaust temperature and turbopump vibration. Actual failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
		Fails to open or opens slowly.	Slow power buildup may result in shutdown if flow through the venturi does not reach the required level by the end of the time delay.	The effect of this failure will be detected by monitoring turbine speed and GGA temperature trends, and venturi Δ pressure. Two standby conditioners provide FO/FS redundancy.

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

START

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, Throttle, GGA Fuel TV-2	Opens gradually during start period to aid in controlling turbine power buildup.	Fails in low flow position.	Excessive GGA combustion temperature and turbine power buildup could result in structural failure of turbine blades or heat exchanger and/or degraded pump bearing life due to "sledding."	The effects of this failure will be detected by monitoring GGA combustion temperature, turbine exhaust temperature, and turbopump vibration. Actual failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
		Fails full open or opens too fast.	The effect on conditioner start up should be insignificant.	The actual failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
Valve, Check, Propellant Tank Outlet CV-1	Opens to allow propellant flow to the turbopump.	Fails closed (remote failure mode).	Propellant cannot be extracted from the liquid storage tank. This results in loss of auxiliary propulsion. This is catastrophic to crew and vehicle.	The effects of this failure will be detected by monitoring turbine speed. The turbopump will cavitate and turbine will overspeed. Due to the remote probability of this failure mode occurring, redundancy has not been incorporated in current schematics. However, due to the catastrophic consequences of this failure, the need for redundancy should be reevaluated during detail valve design.
Valve, Turbopump Inlet V-1 or V-2	No function during start phase except to remain in open position.	Inadvertent closure (remote failure mode).	Loss of liquid propellant flow to the conditioner will result in pump cavitation and turbine overspeed.	The effect of this failure will be detected by monitoring turbine speed. The actual failure will be detected by a valve position sensor. Two standby conditioners will provide FO/FS redundancy.
Valve, Bypass V-3	Remains open during start phase until turbopump outlet pressure reaches level required for flow through the conditioner and then closes to force full flow through the conditioner.	Fails open or partially open.	Adequate cold flow through the heat exchanger cannot be established. If flow is above the conditioner shutdown level and the conditioner continues to operate, both the cold side and hot side outlet temperatures will be above normal.	The effects of this failure will be detected by monitoring venturi Δpressure and by monitoring conditioner hot side and cold side outlet temperature. Two standby conditioners provide FO/FS redundancy. The actual valve failure will be detected by a valve position sensor.
		Fails closed or closes prematurely.	Effect of this failure is not fully defined. Probable effect is pump cavitation with resulting turbopump overspeed.	The effect of this failure will be detected by monitoring turbine speed. Actual valve failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
Valve, Bypass Isolation V-4	No function during the start phase except to remain in the open position	Closes inadvertently.	Same as preceding entry.	Same as preceding entry.
Sensor, Turbopump Discharge Pressure	Monitors turbopump outlet pressure. Output is used by controller to close the bypass valve when turbopump outlet pressure reaches level required to start flow through the conditioner.	Low or no output.	The bypass valve will not be closed and adequate cold flow through the conditioner cannot be established.	The effect of this failure will be detected by monitoring venturi Δ pressure. Actual failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy.
		High output.	The bypass valve will close prematurely. This may cause pump cavitation and resulting turbopump overspeed.	The effect of this failure will be detected by monitoring turbine speed. Actual failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

START

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, Turbopump Vent V-6	Remains closed unless recirculation pumps have failed. In this case the valve is used to condition the pump for start up.	Opens inadvertently or leaks.	Excessive propellant loss plus the possibility that adequate cold flow through the conditioner cannot be established. Fire hazards exist if leakage of hydrogen occurs in earths atmosphere.	The effect of this failure will be detected by monitoring venturi Δ pressure and by monitoring flow rate in the vent line. The inadvertent open will be detected by a valve position sensor. At least two other valves in each flow path provides FO/FS redundancy for this failure in conjunction with two standby conditioner assemblies.
Valve, Check, Heat Exchanger Inlet CV-2 or Valve, Check Heat Exchanger Outlet CV-3	Opens as pump bypass valve closes to allow cold flow through the heat exchanger.	Fails closed (Remote failure mode).	Pump will be deadheaded when bypass valve closes. Probable effect is pump cavitation with resulting overspeed unless absence of flow through the venturi causes shutdown before cavitation occurs.	The effect of this failure will be detected by monitoring venturi Δ pressure and/or turbine speed. Two standby conditioners provide FO/FS redundancy.
Valve, Conditioner Cold Flow Discharge Isolation V-7	No function except to remain open until conditioner isolation is required.	Inadvertent closure at or during start phase. (Remote failure mode)	Same as above.	Same as above - plus the detection of the actual valve failure by a valve position sensor.
Turbopump TP-1	Provides power to transfer low pressure cryogenic propellant from the cryogenic storage tank through a thermal conditioner to an accumulator.	Liftoff seal fails closed during start.	The seal may be damaged by high speed rubbing against seat surface. This will result in excess leakage during idle periods	Excess leakage from O_2 pump will be detected by monitoring flow in the O_2 seal dump line. The effects of leaking H_2 from the pump to the turbine will be detected by monitoring turbine exhaust temperature. Redundant isolation valves for the conditioner provide FO/FS redundancy in conjunction with two standby conditioners.
		Structural failure of pump or turbine blades or bearing failure.	Pump output degraded plus vibration due to imbalance in high speed rotating parts or to failed bearings. Potential for damage to surrounding equipment unless shutdown is effected immediately. For pump blade failure, turbine overspeed is a probable effect.	Effects of failure will be detected by monitoring turbopump vibration, turbine speed or venturi Δ pressure. Two standby conditioners provide FO/FS redundancy. Pump and turbine housings will be designed to contain thrown pump or turbine blades.
		Dynamic seal leakage (O_2 pump)	No effect unless there are multiple failures Redundant seals are provided for both O_2 and hot gas with intermediate seals purged with helium.	If leakage is significant, effects of failure will be detected by monitoring flow in the O_2 seal dump line. Redundant isolation valves in conjunction with two standby conditioners provide FO/FS redundancy.
		Dynamic seal leakage (H_2 Pump)	No effect unless both seals leak. If both seals leak, H_2 will flow to the turbine and lower the turbine exhaust temperature. This may result in icing in the heat exchanger and degraded conditioner performance.	If leakage is significant the effects of failure will be detected by monitoring turbine exhaust temperature and heat exchanger cold side outlet temperature. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Heat Exchanger HEX - 1	Provides heat transfer interface between hot turbine exhaust and cold propellant.	Internal leakage (O_2 heat exchanger)	O_2 introduced into the hot turbine exhaust will cause higher than normal hot gas temperature. Large leak may be catastrophic.	Effects of failure will be detected by monitoring heat exchanger hot gas temperature. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners if failure can be isolated before catastrophic structural failure or fire occurs.
		Internal leakage (H_2 heat exchanger).	Significant leak will cause degraded performance due to lowered hot gas temperature in heat exchanger and possible icing.	Effects of failure will be detected by monitoring heat exchanger hot side temperature and cold side discharge temperature. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.
		Coil Icing	Degraded performance due to poor heat transfer.	Effects of failure will be detected by monitoring heat exchanger cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
Sensor Malfunction Detection 1. GGA Combustion Temperature 2. Turbopump Speed 3. Turbopump Vibration 4. Turbopump Cold Flow Discharge Temperature 5. Turbopump Vent Flow and Temperature 6. Turbine Discharge Temperature 7. O_2 Turbopump Seal Vent Flow and Temperature 8. Venturi Δ Pressure 9. HEX Cold Side Discharge Temperature 10. HEX Hot Side Temperature 11. Accumulator Pressure	Monitors conditioner performance and provides output used to generate emergency shutdown command or crew alert.	Fails to detect malfunction.	No effect. Each critical parameter will be monitored by two redundant sensors. A malfunction signal from either sensor will initiate emergency shutdown or crew alert.	Sensor failure will be detected by ground checkout. Redundant sensors provide fail-safe detection of critical malfunctions.
		Erroneous malfunction signal.	Associated conditioner will be shut down for time critical failure indications or crew given false alert.	Sensor failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy. The desirability of providing a manual override of an emergency shutdown should be evaluated.
Conditioner Assembly Hot Gas Components and Lines	Generate and direct the flow of hot gas through the conditioner.	Leaks hot gas externally.	Potential damage to surrounding equipment or structure from the hot gas. This is a structural failure and is included to call attention to possible damage to other equipment.	A method of detecting hot gas leakage has not been selected. Adequate structural design of the hot gas components and thermal protection for surrounding equipment will minimize the probability of this failure occurring and the effect of this failure if it does occur. Two standby conditioners provide FO/FS operational redundancy.

START

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

STEADY STATE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Sensor, Accumulator Pressure. (1 of 4)	Continuously monitors accumulator pressure and provides signal to the controller.	High output.	In a nonredundant system, this failure would cause conditioner shutdown before the accumulator is fully charged. This would increase the number of conditioner cycles.	This sensor is one of four redundant accumulator pressure sensors. The controller will incorporate voting or averaging logic to detect a malfunctioning sensor.
Valve, Helium, Pump Lift-Off Seal and Seal Purge V-5 (Seal Purge Applicable to O_2 only)	Remains open during steady state operation to supply high pressure helium to the lift-off seal bellows O_2 pump seal purge line and the GGA propellant valve pilot.	Closes inadvertently.	Affected conditioner will be shut down. This nonsequenced shutdown will subject pump and upstream system to back pressure.	The primary parameter for sensing inadvertent shut down of a conditioner will be venturi Δ pressure. Loss of flow in the presence of an accumulator recharge command will be used to activate one of the two standby conditioner units. The propellant tank outlet check valve provides protection for positive expulsion screen against back pressure. Additional redundancy may be required in this area.
Valve, GGA Bipropellant Shutoff V-8	Remains open during steady state operation to supply fuel and oxidizer to the GGA.	Closes inadvertently.	Same as preceding entry.	Same as preceding entry.
Injector, GGA Propellant	Provides optimum mixing of propellants in the GGA combustion chamber.	Restricts fuel flow due to clogging.	Degraded turbine power. The O_2 throttle valve will throttle O_2 flow to keep GGA combustion temperature within limits.	Critical reduction in turbine power will be detected by monitoring venturi Δ pressure. Two standby conditioners provide FO/FS redundancy.
		Restricts oxidizer flow due to clogging.	Possible low GGA combustion temperature resulting in degraded turbine power and low conditioned propellant temperature with a potential for icing in the heat exchanger.	Critical reduction in turbine power will be detected by monitoring venturi Δ pressure. Low conditioned propellant temperature will be detected by monitoring heat exchanger cold side outlet temperature. Two standby conditioners provide FO/FS redundancy.
Valve, Throttle, GGA Oxidizer TV-1	Controls mass flow of oxidizer to the GGA to maintain GGA combustion temperature at the desired level.	Fails to provide adequate flow.	Same as preceding entry.	Same as preceding entry.
		Allows excessive flow.	Excessive GGA combustion temperature will result in increased turbine power and excessive hot gas temperature in the heat exchanger. In the worst case, the temperature will cause structural failure of turbine blades, combustion chamber and/or the heat exchanger with potentially catastrophic effects if failure is not detected and shut down effected quickly.	The effects of this failure will be detected by monitoring GGA combustion temperature, turbine speed, and turbine exhaust temperature. Two standby conditioners provide FO/FS operational redundancy if failure is detected and isolated before catastrophic structural failure results.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

STEADY STATE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Sensor, GGA Combustion Temperature.	Monitors GGA combustion temperature and provides output used to drive the GGA oxidizer throttle valve for control of combustion temperature.	Loss of output or low output.	Drives O_2 throttle valve to a "higher than required" flow position. Excessive GGA combustion temperature will result in increased turbine power and excessive hot gas temperature in the heat exchanger. In the worst case, the temperature will cause structural failure of the turbine blades, combustion chamber and/or the heat exchanger with potentially catastrophic effects if failure is not detected and shutdown effected.	The effects of this failure will be detected by monitoring GGA combustion temperature, turbine speed, and turbine exhaust temperature. Two standby conditioners provide FO/FS operational redundancy if failure is detected and isolated before catastrophic structural failure results. Redundant temperature sensors will be used to detect over-temperature conditions in the GGA in addition to the sensor used to drive the O_2 throttle valve. Preliminary analysis indicates that adequate time will be available to sense failure and shut down the conditioner to prevent catastrophic failure.
		High output.	Drives O_2 throttle valve to a "lower than required" flow position. Low GGA combustion temperature results in degraded turbine power and low conditioned propellant temperature with a potential for icing in the heat exchanger.	Critical reduction in turbine power will be detected by monitoring venturi Δpressure. Low conditioned propellant temperature will be detected by monitoring heat exchanger cold side outlet temperature. Two standby conditioners provide FO/FS redundancy.
Valve, Throttle, GGA Fuel TV-2	Controls mass flow of fuel to the GGA for controlling turbine power to provide a constant turbopump outlet pressure.	Fails full open or allows excessive flow.	Increased turbine power will result in above normal turbine speed and turbopump discharge pressure. This condition is not critical, however, shutdown and switching to a standby conditioner will be accomplished.	The effects of this failure will be detected by monitoring turbine speed, pump discharge pressure and venturi Δ pressure. Two standby conditioners provide FO/FS redundancy.
		Restricts fuel flow.	The O_2 valve will be driven to a low flow position to keep GGA combustion temperature within limits resulting in degraded turbine power.	Critical reduction in turbine power will be detected by monitoring venturi Δpressure, turbine speed, and pump discharge pressure. Two standby conditioners provide FO/FS redundancy.
Sensor, Turbopump Discharge Pressure.	Monitors turbopump discharge pressure and provides output used to drive the GGA fuel throttle valve for control of turbopump power and discharge pressure and flow rate.	High output.	The H_2 throttle valve is driven to a "lower than required" flow position. The O_2 throttle valve will also driven to a low flow position to control GGA combustion temperature. This decreases turbine power.	Same as preceding entry.
		Loss of output or low output.	The H_2 throttle valve will be driven to a "higher than required" flow position. This will increase turbine power and result in above normal turbine speed and turbopump discharge pressure. This condition is not critical, however, shutdown and switching to a standby conditioner will be accomplished.	The effects of this failure will be detected by monitoring turbine speed and venturi Δpressure. Two standby conditioners provide FO/FS redundancy.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS
SERIES RCS

STEADY STATE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, Turbopump Inlet V-1 or V-2	No function during steady state operation except to remain in the normally open position.	Closes inadvertently (Remote failure mode)	Loss of liquid propellant flow to the conditioner will result in pump cavitation and turbine overspeed.	The effect of this failure will be detected by monitoring turbine speed. The actual failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
Valve, Bypass V-3	No function during steady state operation except to remain in the closed position.	Opens inadvertently or leaks excessively.	Adequate cold flow through the heat exchanger cannot be maintained resulting in conditioner shutdown.	The effects of this failure will be detected by monitoring venturi Δ pressure and heat exchanger cold side and hot side discharge temperatures. Valve failure to open will be detected by a position sensor. Two standby conditioners provide FO/FS redundancy.
Valve, Turbopump Vent V-6	No function during steady state operation except to remain in the normally closed position.	Opens inadvertently or leaks excessively.	Excessive propellant loss plus the possibility that adequate cold flow through the heat exchanger cannot be maintained resulting in conditioner shutdown. Fire hazard results if leakage occurs in earth's atmosphere.	The effects of this failure will be detected by monitoring venturi Δ pressure, heat exchanger cold side and hot side temperatures and vent line flow rate. The valve "fail open" will be detected by a position sensor. Two standby conditioners provide FO/FS redundancy in conjunction with redundant isolation valves.
Valve, Conditioner Cold Flow Discharge V-7	No function during steady state operation except to remain in the normally open position.	Closes inadvertently.	Flow through the conditioner is stopped, deheading the turbopump. Probable effect is pump cavitation with resulting turbine overspeed unless absence of flow through the venturi causes shutdown before cavitation occurs.	The effect of this failure will be detected by monitoring venturi Δ pressure and/or turbine speed. The actual valve failure will be detected by a position sensor. Two standby conditioners provide FO/FS redundancy.
Turbopump TP-1	Provides power to transfer low pressure cryogenic propellant from the storage tank through a conditioner to an accumulator at high pressure.	Structural failure of the pump or turbine blades, or bearing failure.	Degraded performance plus vibration due to imbalance in high speed rotating parts, or due to failed bearings. Seized bearings could cause shaft failure. For pump blade failure, turbine overspeed is a probable effect. Potential exists for damage to surrounding equipment unless shutdown is effected immediately.	The effects of this failure(s) will be detected by monitoring turbopump vibration, turbine speed, or venturi Δ pressure. Pump and turbine housings will be designed to contain thrown turbopump or turbine blades. Two standby conditioners provide FO/FS operating redundancy.
		Dynamic seal leakage. (O_2 Pump)	No effect unless there are multiple failures. Redundant seals are provided for both O_2 and hot gas with intermediate seals purged with helium.	If leakage is significant the effects of failure will be detected by monitoring flow in the O_2 seal dump line. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.
		Dynamic seal leakage. (H_2 Pump)	No effects unless both seals leak. If both seals leak, H_2 will flow to the turbine and lower the temperature of hot gas flowing to the heat exchanger. Depending on the extent of leakage, conditioner performance may be degraded.	Degraded conditioner performance will be detected by monitoring heat exchanger cold side discharge temperature. Below normal hot gas temperature will be detected by monitoring turbine exhaust and heat exchanger hot gas temperatures. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

STEADY STATE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Heat Exchanger HEX-1	Provides heat transfer interface between hot turbine exhaust and cold propellant.	Internal leakage (O_2 heat exchanger)	O_2 introduced into the hot turbine exhaust will cause higher than normal gas temperature. Large leak may be catastrophic.	Effects of failure will be detected by monitoring heat exchanger hot gas temperature. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners if failure can be isolated before catastrophic structural failure or fire occurs.
		Internal leakage (H_2 heat exchanger)	Significant leak will cause degraded performance due to lowered hot gas temperature in heat exchanger and possible icing.	Effects of failure will be detected by monitoring heat exchanger hot side temperature and cold side discharge temperature. Redundant isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.
		Coil icing.	Degraded performance due to poor heat transfer.	Effects of failure will be detected by monitoring heat exchanger cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
Sensor, Malfunction Detection 1. GGA Combustion Temperature 2. Turbopump Speed 3. Turbopump Vibration 4. Turbopump Cold Flow Discharge Temperature 5. Turbopump Vent Flow and Temperature 6. Turbine Discharge Temperature 7. O_2 Turbopump Seal Vent Flow and Temperature 8. Venturi Δ Pressure 9. HEX Cold Side Discharge Temperature 10. HEX Hot Side Temperature 11. Accumulator Pressure	Monitors conditioner performance and provides output used to generate emergency shutdown command or crew alert.	Fails to detect malfunction.	No effect. Each critical parameter will be monitored by two redundant sensors. A malfunction signal from either sensor will initiate emergency shutdown or crew alert.	Sensor failure will be detected by ground checkout. Redundant sensors provide fail-safe detection of critical malfunctions.
		Erroneous malfunction signal.	Associated conditioner will be shut down for the critical failure indications or crew given false alert.	Sensor failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy. The desirability of providing a manual override of an emergency shutdown should be evaluated.
Conditioner Assembly Hot Gas Components and Lines	Generate and direct the flow of hot gas through the conditioner.	Leaks hot gas externally.	Potential damage to surrounding equipment or structure from the hot gas. This is a structural failure and is included to call attention to possible damage to other equipment.	A method of detecting hot gas leakage has not been selected. Adequate structural design of the hot gas components and thermal protection for surrounding equipment will minimize the probability of this failure occurring and the effect of this failure if it does occur. Two standby conditioners provide FO/FS operational redundancy.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

STEADY STATE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, Throttle, HEX Cold Side Bypass TV-5 (H_2 Conditioner only)	Throttles HEX cold side H_2 flow to control HEX cold side discharge temperature. Throttle range is limited.	Fails in high flow position.	HEX cold side discharge temperature will be less than nominal. However, the limited throttle range results in insignificant effects. May result in greater use of fuel.	The effect of this failure will be detected by monitoring HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
		Fails in low flow position.	HEX cold side discharge temperature will be above nominal. However, the limited throttle range results in insignificant effects.	Same as preceding entry.
Sensor, HEX Cold Side Discharge Temperature. (H_2 conditioner)	Continuously monitors HEX cold side discharge temperature. Output is used to drive the HEX cold side bypass throttle valve.	Low output.	HEX cold side bypass throttle valve is driven to a low flow position with same effects as preceding entry.	Same as preceding entry.
		High output.	HEX cold side bypass throttle valve is driven to a high flow position. The HEX cold side discharge temperature will be above nominal range the limited throttle range results in insignificant effects.	Same as preceding entry.
Liquid-Vapor Mixer (H_2 Conditioner Only)	Injects the HEX LH_2 bypass flow into the HEX cold side discharge to provide optimum mixing of liquid and gaseous H_2.	Clogging.	The HEX cold side discharge temperature will be above normal. Extensive clogging may require switching to a standby conditioner.	The effect of this failure will be detected by monitoring HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

SHUTDOWN & IDLE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Sensor, Accumulator Pressure (1 of 4)	Continuously monitors accumulator pressure and provides signal to the controller.	Low output.	In a nonredundant system this failure would result in accumulator overcharge and a premature conditioner start signal for the next cycle.	This sensor is one of four redundant accumulator pressure sensors. The controller will incorporate voting or averaging logic to detect a malfunctioning sensor.
Valve, Helium, Pump Lift-off Seal and Seal Purge V-5 (Seal purge applicable to O_2 only)	Closes to stop flow of helium to lift off seal and seal purge and the GGA propellant valve pilot.	Fails full open. (O_2 conditioner) Remote failure mode.	The pump lift-off seal will be kept open and the seal purge will continue resulting in loss of helium through the purge line and loss of O_2 through the dynamic seals and seal dump line.	The effects of this failure will be detected by monitoring flow in the pump O_2 seal dump line and by monitoring the flow of helium to the conditioner. The actual valve failure will be detected by a valve position sensor. Two standby conditioners in conjunction with either redundant helium systems or redundant helium isolation valves will provide FO/FS redundancy.
		Fails full open (H_2 conditioner) Remote failure mode.	The pump lift-off seal will be kept in the open position. This will allow some fuel leakage past the dynamic seals into the turbine and then into the heat exchanger. Fire hazard results if failure occurs in earth's atmosphere.	Significant leakage will be detected by monitoring the turbine exhaust temperature which will drop rapidly. The actual valve failure will be detected by a valve position sensor. Redundant conditioner isolation valves provide FO/FS redundancy in conjunction with two standby conditioners.
		Leaks.	Loss of helium.	A method for detecting helium leakage has not been selected. Potential methods include acoustic devices, and flow sensors in helium valve supply line or vent line. Redundant helium supplies will provide FO/FS redundancy. Loss of one helium supply will affect only one fuel and one oxidizer conditioner.
Valve, GGA Bipropellant Shut-off V-8	Closes during shutdown to stop the flow of propellants to the GGA and remains closed.	Fails to close at shutdown or leaks both fuel and oxidizer.	Combustion will continue in the GGA. This results in continued flow of hot gas through the heat exchanger with zero or very limited cold flow. This could result in structural damage to the heat exchanger due to overheating.	The effects of this failure will be detected by monitoring GGA combustion temperature, turbine speed, turbine exhaust temperature, heat exchanger hot side temperature, and heat exchanger cold side outlet temperature. Actual valve failure will be detected by a valve position sensor. Two series redundant isolation valve in each GGA propellant line provide FO/FS redundancy in conjunction with two standby conditioners.
		Leaks Oxygen during idle periods.	This will result in loss of oxygen and an O_2 rich mixture ratio at the next start. This will result in high GGA combustion temperature which may cause conditioner shutdown or could potentially cause structural failure of GGA, turbine or heat exchanger.	Significant loss of O_2 will be detected by monitoring O_2 accumulator pressure. High GGA combustion temperature will be detected by monitoring GGA temperature, and turbine exhaust temperature. A method of detecting valve leakage during flight has not been selected. Potential candidates include acoustic devices and flow sensors. Redundant GGA isolation valves and two standby conditioners provide FO/FS redundancy. To reduce safety hazard, it may be desirable to cycle at least one of the isolation valves with the GGA propellant valve for a more positive shutoff.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES PCS

SHUTDOWN & IDLE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
		Leaks H_2 during idle periods.	This will result in loss of H_2 and a fuel rich mixture ratio at the next start. A fire hazard is created by this failure while vehicle is in the earth's atmosphere.	Significant loss of H_2 will be detected by monitoring H_2 accumulator pressure. A method of detecting valve leakage during flight has not been selected. Potential candidates include acoustic devices, and flow sensors. Redundant GGA isolation valves and two standby conditioners provide FO/FS redundancy. To reduce fire hazard in the earth's atmosphere it may be desirable to cycle at least one of the isolation valves with the GGA propellant valve for a more positive shutoff.
Valve, Throttle GGA Oxidizer TV-1	Drives to fifty percent flow position in preparation for the next start cycle.	Stays in high flow position.	If not detected this failure will cause excessive GGA combustion temperature at next startup resulting in conditioner shutdown. Excessive pump acceleration may result in bearing "sledding."	This failure will be detected by monitoring valve position. If not detected, the effects of failure at next startup will be detected by monitoring GGA combustion temperature and turbine exhaust temperature. Two standby conditioners provide FO/FS redundancy.
Valve, Throttle GGA Fuel TV-2	Drives to fifty percent flow position in preparation for the next start cycle.	Stays in high flow position.	If not detected this failure will produce fuel rich GGA propellant mixture at the next startup. The effect on conditioner startup should be insignificant.	The actual failure will be detected by monitoring valve position. Two standby conditioners provide FO/FS redundancy.
Valve, Check, Propellant Tank Outlet CV-1	Must close to enable the recirculation pump to provide propellant flow through the turbopumps during the idle periods to maintain liquid in the pumps. This valve also protects the propellant tank positive expulsion screen against back pressure if the bypass valve (V-3) fails closed during conditioner shutdown.	Fails open.	The recirculation pump will be unable to force liquid propellant through the turbopumps during idle periods. This allows pump temperature to rise resulting in the formation of gas in the pump. This may cause pump cavitation at next startup if undetected. The effect of double failure of CV-1 and V-3 has not been fully determined with respect to positive expulsion screen damage.	The effect of this failure will be detected by montitoring turbopump outlet temperature during the idle periods. The turbopump vent valves (V-6) provide a backup method of cooling turbopumps by bleeding propellant overboard.
Valve, Turbopump Inlet V-1 or V-2	No function during these phases except to remain in the open position.	Closes inadvertently.	Prevents circulation of liquid propellants through the pump. This will allow pump temperature to rise and formation of gas bubble in the pump. If undetected this failure will cause pump cavitation at next startup.	The rise in pump temperature will be detected by monitoring pump outlet temperature. The valve failure will be detected by a valve position sensor. Two standby conditioners provide FO/FS redundancy.
Valve, Bypass V-3	Opens at shutdown signal to allow recirculation flow through the pump and also to prevent any shutdown back pressure transients.	Pilot Valve leaks.	Loss of helium.	A method of detecting helium leakage has not been selected. Potential methods include acoustic devices and flow sensors. Redundant helium supplies will provide FO/FS redundancy.
		Fails closed.	Recirculation flow through this pump is prevented and any shutdown back pressure must go through the pump. The propellant tank outlet check valve will protect the positive expulsion screen from back pressure	The effect of this failure will be detected by monitoring turbopump outlet temperature during the idle periods. The actual valve failure will be detected by a valve position sensor. Two standby turbopumps provide FO/FS redundancy. The turbopump vent valve (V-6) provides a backup method of cooling the turbopump if necessary.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

SHUTDOWN & IDLE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, Bypass Isolation V-4	No function during this phase except to remain in the open position.	Closes inadvertently.	Same as preceding entry.	Same as preceding entry.
Check Valve, Heat Exchanger Inlet CV-2 or Check Valve, Heat Exchanger Outlet CV-3	Closes during shutdown to prevent high pressure gaseous propellant from the accumulator leaking back to the cryogenic storage tank.	Fails open or leaks.	No effect unless both valves fail. Double failure allows gaseous propellant to flow from the accumulator to the cryogenic storage tank.	The effects of a double failure will be detected by monitoring accumulator pressure, propellant storage tank pressure, venturi pressure and turbopump outlet temperature. The conditioner isolation valve (V-7) provides fail safe redundancy. The fail open failure mode is considered remote due to the high pressure differential between the accumulator and the cryogenic storage tank.
Valve, Turbopump Vent. (V-6)	No function except to remain closed.	Opens inadvertently or leaks.	Loss of propellant. Results in fire hazard if leakage occurs in earth's atmosphere.	The failure will be detected by monitoring flow rate in the vent line. Redundant isolation valves and two standby conditioners provide FO/FS redundancy.
Turbopump TP-1	Pump stops and lift off seal seats to prevent leakage of propellant through the pump dynamic seals.	Lift off seal leaks. (O_2 Pump)	Loss of O_2 through the dynamic seals and seal dump line.	Significant loss of O_2 will be detected by monitoring flow in the O_2 seal dump line. Redundant isolation valves and two standby conditioners provide FO/FS redundancy.
		Lift off seal leaks. (H_2 Pump)	Loss of H_2 through the dynamic seals into the turbine. Fire hazard results from leakage in earth's atmosphere.	Significant loss of H_2 will be detected by monitoring the turbine exhaust temperature which will drop rapidly. Redundant conditioner isolation valves and two standby conditioners provide FO/FS redundancy.
Heat Exchanger HEX-1	No function during these phases.	Internal leakage (O_2 heat exchanger)	Leakage during shutdown will cause high temperatures in the heat exchanger. Large leak could be catastrophic. Leakage during the idle period may result in accumulation of O_2 in the heat exchanger which could produce high temperatures at next start up.	The effects of this failure will be detected by monitoring heat exchanger hot side temperature during shutdown. If leakage starts during the idle period heat exchanger hot side temperature will be below normal. Redundant conditioner isolation valves and two standby conditioners provide FO/FS redundancy if failure can be isolated before catastrophic structural failure of fire occurs.
		Internal leakage (H_2 heat exchanger)	Some loss of H_2. Fire hazard if leakage occurs in earth's atmosphere. Significant leak will cause rapid cool down of heat exchanger to below normal temperature.	The effects of this failure will be detected by monitoring heat exchanger hot side temperature. High propellant loss will be detected by monitoring propellant quantity in the cryogenic storage tank. Redundant conditioner isolation valves and two standby conditioners provide FO/FS redundancy.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

SHUTDOWN & IDLE

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Recirculation Pump. (One of two in each propellant tank)	Continuously circulates liquid propellant through the turbopumps in order to maintain liquid in the pumps during idle periods.	Fails to function.	No effect unless both pumps in fuel, or oxidizer tank fail. Double failure would stop recirculation flow and allow pumps to heat up resulting in formation of gas.	Effects of double failure will be detected by monitoring turbopump outlet temperature. The turbopump vent valves (V-6) provide fail safe method of conditioning pumps for start up.
Sensor, Malfunction Detection 1. GGA Combustion Temperature 2. Turbopump Speed 3. Turbopump Vibration 4. Turbopump Cold Flow Discharge Temperature 5. Turbopump Vent Flow and Temperature 6. Turbine Discharge Temperature 7. O_2 Turbopump Seal Vent Flow and Temperature 8. Venturi Δ Pressure 9. HEX Cold Side Discharge Temperature 10. HEX Hot Side Temperature 11. Accumulator Pressure	Monitors conditioner performance and provides output used to generate emergency shut down command or crew alert.	Fails to detect malfunction.	No effect. Each critical parameter will be monitored by two redundant sensors. A malfunction signal from either sensor will initiate emergency shutdown or crew alert.	Sensor failure will be detected by ground checkout. Redundant sensors provide fail safe detection of critical malfunctions.
		Erroneous malfunction signal.	Associated conditioner will be shut down for time critical failure indications, or crew given false alert.	Sensor failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy. The desirability of providing a manual over ride of an emergency shut down should be evaluated.
Conditioner Assembly Propellant Lines, Components, and Fittings.	Controls the flow of cryogenic and gaseous propellant through the conditioner.	Leaks H_2 externally.	Loss of H_2 and creation of fire hazard while in the earth's atmosphere. Leakage into a confined area while on orbit may result in fire or explosion during reentry into the earth's atmosphere.	Significant propellant loss will be detected by monitoring propellant quantity and accumulator pressure decay. A method of locating the leak has not been selected. Internal component redundancy will minimize this potential problem.
		Leaks O_2 externally.	Probably no effect other than loss of O_2.	Same as preceding entry.

FAILURE MODE AND EFFECTS ANALYSIS

SERIES RCS

STANDBY

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
The failure modes for this condition are the same as for "Idle" except for the following:				
Valve, Bypass V-3	Closes at start-up command in all conditioners except the one to be operated.	Fails open.	Allows an open path by which the operating turbopump could possibly ingest gas from the cryogenic storage tank. This could result in pump cavitation and cause emergency shut down.	The actual valve failure will be detected by a valve position sensor. Pump cavitation, if it occurs, will be detected by monitoring turbine speed. The pump inlet valves (V-1 and V-2) provide FO/FS isolation for this failure.
Check Valve, Heat Exchanger Inlet CV-2 or Check Valve, Heat Exchanger Outlet CV-3	No function during this phase except to remain closed.	Leaks.	No effect unless both valves leak. If both check valves leak, gaseous propellant from the accumulator may be ingested by the operating turbopump. This could result in pump cavitation and cause emergency shut down of the operating conditioner.	The method for detecting valve leakage has not been selected. Pump cavitation, if it occurs, will be detected by monitoring turbine speed. The conditioner isolation valve (V-7) provides failsafe redundancy for the double check valve failure when detected.

Figure E-5 Continued

FAILURE MODE AND EFFECTS ANALYSIS
PARALLEL RCS

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
The failure modes and effects for the "Parallel RCS" conditioner is essentially identical to the "Series RCS" with the following exceptions and additions				
Sensor, Turbopump Discharge Pressure.	Monitors turbopump outlet pressure. The output is used by the controller to close the bypass valve and start the HEX GGA during the start phase and to drive the turbine GGA fuel throttle valve during steady state operation.	High output during start phase.	The bypass valve will close and the HEX GGA will start prematurely. This may cause pump cavitation and resulting turbine overspeed and conditioner shut down. Premature HEX GGA start should not have a significant effect.	Pump cavitation and overspeed will be detected by monitoring turbine speed. Actual failure will be detected by ground checkout. Two standby conditioners provide FO/FS redundancy.
		Low or no output during start phase.	The bypass valve will not close and the HEX GGA will not start. Adequate flow through the venturi cannot be established and the conditioner will be shutdown.	The effects of this failure will be detected by monitoring venturi Δ pressure and HEX GGA combustion temperature. Two standby conditioners provide FO/FS redundancy.
Igniter, HEX GGA (Exciter and Spark Plug)	Provides high voltage spark for propellant ignition.	Low power or intermittent spark during start phase.	Possible delayed ignition could allow excessive accumulation of propellant in the GGA and heat exchanger before ignition. This may result in a hard start with a potential for structural damage.	The failure will be detected by monitoring the exciter output. The effect of a delayed ignition should be investigated during system development. It may be necessary to verify that torch ignition has occured before initiating main propellant flow.
		Fails to provide spark during start phase.	The HEX GGA will not start. This will result in extremely low propellant temperature at the HEX cold side discharge. If not detected this could drop accumulator temperatures to a critical level.	This failure will be detected by monitoring exciter output. HEX cold side discharge temperature and GGA combustion temperature. Two standby conditioners provide FO/FS redundancy.
Valve, HEX GGA Bipropellant Shut off V-?	Opens on command to supply fuel and oxidizer to the HEX GGA and remains open until conditioner shutdown.	Fails closed during start phase.	Same as preceding entry.	Same as preceding entry.
		Closes inadvertently during steady state.	HEX GGA will be shut off and HEX cold side propellant discharge temperature will drop rapidly. If not detected this could drop accumulator temperature to a critical level.	The effects of this failure will be detected by monitoring GGA combustion temperature and HEX cold side propellant discharge temperature. Two standby conditioners provide FO/FS redundancy.
		Fails open at shutdown.	HEX GGA will continue to supply hot gas to the heat exchanger after cold flow has stopped. This could result in structural damage to the heat exchanger if repeated for several cycles.	The effects of this failure will be detected by monitoring GGA combustion temperature, and HEX hot side and cold side discharge temperature. Two standby conditioners provide FO/FS redundancy in conjunction with redundant isolation valves in each GGA propellant supply line.
		Leaks O_2 during idle period.	This will result in loss of O_2 and and O_2 rich mixture ratio at the next start. This could result in high GGA combustion temperature which may cause conditioner shut down or could potentially cause structural failure of GGA or heat exchanger.	Significant loss of O_2 will be detected by monitoring O_2 accumulator pressure. High GGA combustion temperature will be detected by monitoring GGA combustion temperature and HEX hot side temperature. A method of detecting valve leakage during flight has not been selected. Potential candidates include acoustic devices and flow sensors. Redundant GGA isolation valves and two standby conditioners provide FO/FS redundancy. To reduce the safety hazard, it may be desirable to cycle at least one of the isolation valves with the GGA propellant valve to provide a more positive shutoff.

Figure E-6

FAILURE MODE AND EFFECTS ANALYSIS
PARALLEL RCS

COMPONENT	FUNCTION	FAILURE MODE	FAILURE EFFECT	DETECTION/REDUNDANCY
Valve, HEX GGA Bipropellant Shut-off V-9 (continued)		Leaks H_2 during idle mode.	This will result in loss of H_2 and a fuel rich mixture ratio at the next start. A fire hazard is created by this failure while vehicle is in the earth's atmosphere.	Significant loss of H_2 will be detected by monitoring H_2 accumulator pressure. A method of detecting valve leakage during flight has not been selected. Potential candidates include acoustic devices and flow sensors. Redundant GGA isolation valves and two standby conditioners provide FO/FS redundancy. To reduce the fire hazard in the earth's atmosphere, it may be desirable to cycle at least one of the isolation valves with the GGA propellant valve to provide a more positive shut off.
Injector, HEX GGA propellant	Provides optimum mixing of propellants in GGA combustion chamber.	Restricts fuel flow during start.	An O_2 rich mixture ratio may result in GGA overtemperature and cause shut down of the conditioner.	Overtemperature will be detected by monitoring GGA combustion temperature and HEX hot side temperature. Two standby conditioners provide FO/FS redundancy.
		Restricts fuel flow during steady state operation.	Degraded hot gas flow rate through the heat exchanger. The O_2 throttle valve will throttle O_2 flow to keep GGA combustion temperature within limits. This will result in low HEX cold side discharge temperature. There is a potential for icing in the heat exchanger.	The effects of this failure will be detected by monitoring HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
		Restricts oxidizer flow during start or steady state operation.	Possible low GGA combustion temperature due to fuel rich mixture ratio. This results in low HEX cold side discharge temperature. There is a potential for icing in the heat exchanger.	The effects of this failure will be detected by monitoring GGA combustion temperature and HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.
Valve, Throttle, HEX GGA Oxidizer TV-4	Throttles the flow of oxidizer to control GGA combustion temperature.	Fails to allow adequate flow.	Same as preceding entry.	Same as preceding entry.
		Allows excessive O_2 flow.	An O_2 rich mixture ratio may result in GGA overtemperature and cause shutdown of the conditioner. In the worst case the temperature may cause structural failure of the GGA or heat exchanger with potentially catastrophic effects if failure is not detected and isolated quickly.	Overtemperatures will be detected by monitoring GGA combustion temperature, HEX hot side temperature and HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy if failure is detected and isolated before catastrophic structural failure results.
Sensor, HEX GGA Combustion Temperature	Monitors GGA combustion temperature output is used to drive the GGA oxidizer throttle valve for control of combustion temperature.	Loss of output or low output.	Drives O_2 throttle valve to a "higher than required" flow position resulting in same effects as preceding intry.	Same as preceding entry.
		High output.	Drives O_2 throttle valve to a "lower than required" flow position. Resulting fuel rich mixture ratio may result in low GGA combustion temperature and cause low HEX cold side discharge temperature. There is a potential for icing in the heat exchanger.	The effects of this failure will be detected by monitoring GGA combustion temperature and HEX cold side discharge temperature. Two standby conditioners provide FO/FS redundancy.

Figure E-6 Continued

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