DTIC FILE COPY

(4)

AD-A198 524

NADC-89006-60

FABRICATION AND TESTING OF LIGHTWEIGHT HYDRAULIC SYSTEM SIMULATOR HARDWARE—PHASE II

Rockwell International
North American Aircraft
4300 East Fifth Avenue
P.O. Box 1259
Columbus, Ohio 43216

DTIC
S D
E

15 JULY 1988

Addendum Report for Period, 2 January 1986–15 July 1988

Prepared For

NAVAL AIR DEVELOPMENT CENTER
Aircraft and Crew Systems Technology Directorate
Warminster, PA 18974

89 8 09 008

UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

1a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED	1b. RESTRICTIVE MARKINGS
2a. SECURITY CLASSIFICATION AUTHORITY 2b. DECLASSIFICATION / DOWNGRADING SCHEDULE	3. DISTRIBUTION / AVAILABILITY OF REPORT Approved for Public Release Distribution Unlimited
4. PERFORMING ORGANIZATION REPORT NUMBER(S) NA-88-0011	5. MONITORING ORGANIZATION REPORT NUMBER(S) NADC 89006-60
6a. NAME OF PERFORMING ORGANIZATION Rockwell International Corp. North American Aircraft Operations 6b. OFFICE SYMBOL (If applicable)	7a. NAME OF MONITORING ORGANIZATION Naval Air Development Center (60613)
6c. ADDRESS (City, State, and ZIP Code) 4300 East Fifth Avenue Columbus, OH 43216	7b. ADDRESS (City, State, and ZIP Code) Warminster, PA 18974
8a. NAME OF FUNDING / SPONSORING ORGANIZATION Naval Air Systems Command 8b. OFFICE SYMBOL (If applicable) 930	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER N62269-80-C-0261
8c. ADDRESS (City, State, and ZIP Code) Department of the Navy Washington, DC 20361	10. SOURCE OF FUNDING NUMBERS PROGRAM ELEMENT NO / PROJECT NO / TASK NO / WORK UNIT ACCESSION NO

11. TITLE (Include Security Classification)
FABRICATION AND TESTING OF LIGHTWEIGHT SYSTEM SIMULATOR HARDWARE - PHASE II

12. PERSONAL AUTHOR(S)
Bickel, William N. and Haning, Robert K.

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Addendum	FROM ~~85Nov15~~ TO 88Jul15	1988 July 15	181

16. SUPPLEMENTARY NOTATION 86ja-2

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Aircraft Hydraulic Systems Lightweight Hydraulic Systems LHS

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

Follow-on work was conducted in a development program to design, fabricate, and test a full scale 8000 psi Lightweight Hydraulic System simulator. The principal task was the performance of an additional 600 hours of mission/profile cycling (1200 hours total). Three flight control actuators completed approximately 5 million cycles under simulated aircraft loading. All minor hydraulic components performed satisfactorily. The results demonstrated that 8000 psi hydraulic systems pose no technological problems for aircraft applications. Other tasks completed were ground support equipment re-work, piston seal evaluation test, coil tube design guidelines, black residue investigation, hybrid pump development, and tube fitting re-design.

20. DISTRIBUTION / AVAILABILITY OF ABSTRACT ☒ UNCLASSIFIED/UNLIMITED ☐ SAME AS RPT ☐ DTIC USERS	21. ABSTRACT SECURITY CLASSIFICATION UNCLASSIFIED
22a. NAME OF RESPONSIBLE INDIVIDUAL	22b. TELEPHONE (Include Area Code) / 22c. OFFICE SYMBOL

DD FORM 1473, 84 MAR — 83 APR edition may be used until exhausted. All other editions are obsolete

SECURITY CLASSIFICATION OF THIS PAGE
UNCLASSIFIED

PREFACE

This document is an Addendum to Report NADC-79024-60 and covers follow-on effort in a development program conducted by Rockwell International Corporation, North American Aircraft Operations, Columbus, Ohio, under Contract N62269-80-C-0261 with the Naval Air Development Center, Warminster, Pennsylvania. Technical direction was administered by Mr. J. Ohlson, Head, Materials Application Branch, Aircraft and Crew Systems Technology Directorate, Naval Air Development Center (60613).

This report presents the results of follow-on work in Phase II of a program to design, fabricate, and test a full scale 8000 psi Lightweight Hydraulic System in a ground simulator. This work is related to tasks performed under Contracts NOw-65-0567-d, N0019-68-C-0352, N00156-70-C-1152, N62269-71-C-0147, N62269-72-C-0381, N62269-73-C-0700, N62269-74-C-0511, N62269-75-C-0422, N62269-76-C-0254, N62269-78-C-0005, and N62269-78-C-0363.

The project engineer for Phase II of the LHS Advanced Development Program was Mr. W. Bickel. Acknowledgement is given to the following individuals for their contributions to this report.

Mr. D. Todd	Rockwell design engineer
Mr. B. Sapp	Rockwell metallurgist
Mr. E. Kauffman	Rockwell R&M engineer
Mr. R. Olsen	LTV project manager
Dr. D. Uhr	Pall scientist
Mr. R. Tarbell	Resistoflex application engineer

Appreciation is extended to those who provided helpful support and constructive criticism during the program; in particular, Mr. J. Ohlson and Mr. J. Dever of the Naval Air Development Center, Mr. G. Dow and Dr. D. Uhr of Pall Corporation, and Mr. L. Biafore and Mr. A. Eckles of Rockwell International.

Distribution/
Availability Codes
Avail and/or
Dist Special
A-1

QUALITY INSPECTED 4

TABLE OF CONTENTS

Section	Title	Page
	PREFACE	i
	TABLE OF CONTENTS	ii
	LIST OF FIGURES	iv
	LIST OF TABLES	v
1.0	INTRODUCTION	
	1.1 BACKGROUND INFORMATION	1
	1.2 PHASE I REVIEW	1
	1.3 PHASE II REVIEW	2
	1.4 PHASE II FOLLOW-ON WORK	3
2.0	SUMMARY	
	2.1 LHS SIMULATOR ENDURANCE TEST	5
	2.2 GROUND SUPPORT EQUIPMENT RE-WORK	6
	2.3 PISTON SEAL EVALUATION TEST	6
	2.4 COIL TUBE DESIGN GUIDELINES	6
	2.5 BLACK RESIDUE INVESTIGATION	7
	2.6 HYBRID PUMP DEVELOPMENT	7
	2.7 TUBE FITTING RE-DESIGN	7
3.0	LHS SIMULATOR ENDURANCE TEST	
	3.1 INTRODUCTION	8
	3.2 COMPONENT CHANGES	8
	3.3 TEST CONDITIONS	
	3.3.1 Cycling Program	9
	3.3.2 Temperature	10
	3.3.3 Performance Checks	12
	3.4 TEST RESULTS	
	3.4.1 Pumps	16
	3.4.2 Actuators	16
	3.4.3 Solenoid Valves	29
	3.4.4 Relief Valves	29
	3.4.5 Restrictors	29
	3.4.6 Filters	29
	3.4.7 Fluid	32
	3.4.8 Fittings	32
	3.4.9 Hoses	32
	3.4.10 Swivels	34
	3.4.11 Seal Test Fixture	37
	3.5 RELIABILITY UPDATE	39
4.0	GROUND SUPPORT EQUIPMENT RE-WORK	
	4.1 INTRODUCTION	42
	4.2 PUMP INFORMATION	42
	4.3 TESTS CONDUCTED	
	4.3.1 Pump Tests	45
	4.3.2 Temperature Stabilization Test	47
	4.3.3 Noise Level Test	52

TABLE OF CONTENTS (Cont'd)

Section	Title	Page
5.0	PISTON SEAL EVALUATION TEST	
	5.1 INTRODUCTION	54
	5.2 CANDIDATE PISTON SEALS	54
	5.3 TEST CONDITIONS	55
	5.4 TEST RESULTS	57
6.0	COIL TUBE DESIGN GUIDELINES	
	6.1 INTRODUCTION	59
	6.2 A-7 SPOILER/DEFLECTOR INSTALLATION	59
	6.3 DESIGN GUIDELINES	
	6.3.1 Introduction	60
	6.3.2 Basic Configurations	61
	6.3.3 Bending Configuration	61
	6.3.4 Torsional Configuration	64
	6.3.5 Coil Tube Design Examples	64
7.0	BLACK RESIDUE INVESTIGATION	
	7.1 INTRODUCTION	66
	7.2 SUMMARY OF RESULTS	66
	7.3 ROCKWELL COMMENTS	67
8.0	HYBRID PUMP DEVELOPMENT	
	8.1 INTRODUCTION	68
	8.2 DISCUSSION	69
9.0	TUBE FITTING RE-DESIGN	
	9.1 INTRODUCTION	70
	9.2 SUPPLIER EFFORTS	70
	9.3 ROCKWELL COMMENTS	71
	REFERENCES	72
	LIST OF ABBREVIATIONS	73
APPENDICES		
A	MISSION/PROFILE TEST LOG	75
B	ACTUATOR PISTON AND ROD SEALS	83
C	EVALUATION OF PISTON SEALS FOR LIGHTWEIGHT HYDRAULIC SYSTEMS (LTV REPORT 2-51700/5R-61)	85
D	COIL TUBE DESIGN EXAMPLES	125
E	BLACK RESIDUE INVESTIGATION (PALL SLS REPORT #1676)	133
F	RELIABILITY AND MAINTAINABILITY ANALYSIS UPDATE OF HYDRAULIC SYSTEM DATA	173

LIST OF FIGURES

Figure No.	Title	Page
1	Lightweight Hydraulic System Simulator	4
2	Crack in Barrel of Speed Brake Actuator	23
3	Fatigue Patterns	24
4	Piston Failure in L/H UHT Actuator	26
5	End Cap Failure in L.E. Flap Actuator	28
6	Hydraulic Swivel Joint Cycling Setup	35
7	LHS Simulator Reliability Growth, 0 to 1200 Hours of Operation	41
8	Modified AHT-63 Portable Test Stand	43
9	Test Stand Pump Installation	44
10	Test Stand Pressure/Flow Characteristics	46
11	Test Stand Pressure Ripple	48
12	Test Stand/LHS Simulator Pressure Ripple	49
13	Test Stand Temperature Stabilization	50
14	Test Stand/LHS Simulator Temperature Stabilization	51
15	Test Stand Noise Levels	53
16	Candidate Piston Seals	56
17	Bending Configurations	62
18	Torsional Configurations	62
19	Helical Coils	63
20	Bending Configuration Layout	63
21	Coil Tube Configuration Examples	65

LIST OF TABLES

Table No.	Title	Page
1	Mission/Profile Cycling Program	11
2	Total Time Minor Components Subjected to 8000 psi	13
3	Maintenance Action Totals	15
4	Pump Operating Time	17
5	Pump Performance Checks	17
6	Actuator Cycle Totals	18
7	Actuator Seal Cycling Time Totals	19
8	Dynamic Seal Leakage Summary	20
9	Control Valve Null Leakage Summary	21
10	Spoiler/Deflector and RFI Actuator Internal Leakage	21
11	Solenoid Valve Internal Leakage	30
12	Relief Valve Performance Summary	30
13	Restrictor Performance Summary	30
14	Filter Element Replacement	31
15	Fluid Contamination	33
16	Swivel Leakage Summary	36
17	Seal Test Fixture Seals	36
18	Seal Test Result	38
19	Summary of LHS Simulator Failures	40

THIS PAGE LEFT BLANK INTENTIONALLY

1.0 INTRODUCTION

1.1 BACKGROUND INFORMATION

This addendum report documents follow-on work completed in Phase II of the Lightweight Hydraulic System (LHS) Advanced Development Program. Phase II tasks originally planned and completed are discussed in the basic report, reference 1.

The objective of the LHS program is to employ 8000 psi technology to reduce the weight and volume of aircraft hydraulic systems, and improve the reliability and maintainability of hydraulic components. A series of exploratory development programs were conducted from 1965 to 1978. The current program, in progress since 1978, was initiated to develop a full scale lightweight hydraulic system, and was to be accomplished in three phases:

Phase I Design, fabricate and test 8000 psi components.

Phase II Fabricate full scale hydraulic simulator. Conduct performance and endurance tests.

Phase III Install 8000 psi hydraulic system in an A-7E aircraft. Conduct flight test program. (Due to funding limitations, this phase was not accomplished.)

1.2 PHASE I REVIEW

Hydraulic circuitry in the A-7E aircraft was re-configured from three independent power control systems operating at 3000 psi to two independent 8000 psi flight control systems and one 3000 psi utility system. Detail installation drawings were prepared for a flight test aircraft.

A full scale simulator structure was designed with aircraft component installations configured to represent an A-7E 8000 psi hydraulic flight control system. A modular approach was employed to facilitate fabrication and provide test flexibility. Two types of modules were used: power modules and load modules. The power modules each contained a reservoir, filters, relief valves, and other components typically used in aircraft hydraulic supply systems. The load modules provided a means to physically load the flight control actuators based on A-7E load/stroke design curves. Two power modules and four load modules were fabricated.

Major 8000 psi components designed include a pump, reservoir, and eight flight control actuators. Three pumps, two reservoirs, and five actuators were fabricated. Numerous minor 8000 psi components, such as relief valves, filters, and check valves were designed and built.

Tests conducted in Phase I were:

Piston rod seal development (400 hours)
System/component compatibility (150 hours)
Pressure impulse (40,000 cycles)
Component endurance (10,000 cycles)

Other work completed in Phase I included:

LHS specification preparation (34)
Math model development (preliminary)
Weight and space analysis (preliminary)
R&M assessment (preliminary)

1.3 PHASE II REVIEW

The LHS simulator structure and hydraulic systems were fabricated, Figure 1. Seven flight control actuators and four load modules were built. The power modules, load modules, and actuators (including those used in Phase I) were then installed in the simulator. Instrumentation and automatic controls were assembled.

Tests conducted on the LHS simulator were:

- Proof Pressure
- System Integration
- Baseline Data
- Dynamic Performance
- Endurance (600 hours)

Other work completed in Phase II included:

- Math model development
- System weight and space analysis (update)
- Reliability and Maintainability Assessment
- LHS Specifications (update)

1.4 PHASE II FOLLOW-ON WORK

Successful completion of the tasks planned for Phase II resulted in a decision to 1) continue LHS simulator endurance cycling, 2) examine several areas needing investigation, and 3) present information that was not available for inclusion in the basic report, reference 1. Phase II follow-on work reported herein is as follows:

- LHS simulator endurance test
  (additional 600 hours, 1200 hours total)
- Ground support equipment re-work
- Piston seal evaluation test
- Coil tube design guidelines
- Black residue investigation
- Hybrid pump development
- Tube fitting re-design

Figure 1. Lightweight hydraulic system simulator

2.0 SUMMARY

2.1 LHS SIMULATOR ENDURANCE TEST

The planned 1200 hours of mission/profile cycling were completed successfully. Endurance characteristics of the re-designed pumps were improved significantly over prior models tested; heat rejection and overall efficiency met design goals. Three flight control actuators satisfactorily withstood approximately 5 million cycles under simulated aircraft loading conditions. Actuator wear and control valve internal leakage were acceptable. Piston and rod seal performance were good. Three actuators failed to complete the 1200 hours of testing. In each case, the failure was attributed to a design deficiency or improper fabrication. The performance of solenoid valves, relief valves, check valves, restrictors, filters, fittings, hoses, swivels and disconnects were all satisfactory.

The 1200 hour test provided valuable learning experiences. Important lessons learned include:

1. Aluminum is not recommended for use in components operating at 8000 psi.

2. Use of reduced diametral clearances for actuator dynamic seals caused actuator wear due to slight mis-alignments resulting from tolerance build-ups. Since dynamic seal performance was satisfactory, standard diametral clearances should be considered.

3. Surface finishes of 32 rms or better are recommended for highly stressed areas, particularly in fillets.

4. The use of pump case drain filters the same size as return filters is recommended.

5. Maintenance of strict quality control during component fabrication is essential.

Phase II results demonstrated conclusively that 8000 psi hydraulic systems pose no technological problems. The design and fabrication of 8000 psi components require the same attention and control as 3000 psi components. Hazards to personnel operating 8000 psi systems are no different than those presented by 3000 psi systems.

2.2 GROUND SUPPORT EQUIPMENT RE-WORK

A 3000 psi portable test stand modified for 8000 psi service had excessive fluid temperature build-up due to high heat rejection in the 8000 psi pump. The pump was replaced with a new design unit. Operational checks on the re-worked test stand showed satisfactory temperature stabilization characteristics and excellent pump performance.

2.3 PISTON SEAL EVALUATION TEST

A 400 hour test was conducted to evaluate nine candidate piston seals for use in 8000 psi flight control actuators. Two candidates completed the test satisfactorily. Three replacement seals satisfactorily completed less than 400 hours.

2.4 COIL TUBE DESIGN GUIDELINES

Difficulties experienced with coil tubes fabricated for the LHS simulator spoiler/deflector actuator installation were examined. The use of hydraulic extension units with re-designed end swivels is recommended for this application. Design guidelines for coil tubing in less severe applications are presented.

2.5 BLACK RESIDUE INVESTIGATION

A comprehensive study of the black residue observed in filters used on the LHS simulator was completed with the following results:

- The residue is the result of normal wear processes.
- The residue material contains primarily four particle types -- all black in color: 1) organic (the major constituent), 2) aluminum, 3) iron, and 4) chromium.
- The black particles had no effect on system fluid cleanliness.

2.6 HYBRID PUMP DEVELOPMENT

A new pump concept was proposed that attempted to eliminate inherent shortcomings in axial piston pump designs and capitalize on certain design advantages of check valve pumps. Problems were encountered during fabrication and testing of the new pump and the project was terminated.

2.7 TUBE FITTING RE-DESIGN

Expander tool life for internally swaged 3/16-size fittings was unsatisfactory during fabrication of LHS simulator plumbing. The basic problem was due to the necessary small size of expander components and the high yield strength of the tubing material. Several approaches to solve the problem were tried by the supplier. The most successful solution involved re-design of the fitting tube receptacle.

3.0 LHS SIMULATOR ENDURANCE TEST

3.1 INTRODUCTION

This test was a continuation of LHS simulator mission/profile endurance cycling conducted in the first part of Phase II when 600 hours were completed, reference 1. The success of the initial program indicated that additional endurance testing would be desirable to further demonstrate the reliability of 8000 psi hardware. Testing reported herein covers the period of 600 to 1200 hours. Several changes in the LHS simulator and in cycling rates were made during the additional test time. These are discussed in following sections.

3.2 COMPONENT CHANGES

Changes made prior to and during the course of the additional 600 hours of endurance cycling were:

Simulator Cycling Hours Completed	Change
600	o Install pumps with larger pintle bearings
	o Install larger case drain filters (identical to return filters)
900	o Install aileron actuator with new body (actuator now fully operational). New piston and piston rod seals installed.

Simulator Cycling Hours Completed	Change
900	o Install new one-piece piston/piston rod in seal test fixtures. New piston and piston rod seals installed.
946	o Install fixture to cycle the speed brake swivels.
968	o Install small actuator to permit cycling the speed brake actuator control valve.

3.3 TEST CONDITIONS

3.3.1 Cycling Program

Simulator cycling was essentially the same as outlined in reference 1 with minor changes to accelerate some cycling. Primary flight control actuator load/stroke and cycle distributions were unchanged and based on the schedule given in Specification MIL-C-5503 as shown below:

	No. of Cycles	% of Total
100% stroke & load	50,000	1%
50% stroke & load	250,000	5%
10% stroke & load	700,000	14%
2% stroke & load	4,000,000	80%
	5,000,000	100%

The automatic cycling program was designed to profile a typical two-hour mission with time broken down into 24 five-minute steps, Table 1. All primary flight control actuators, except the spoiler/deflector, accumulated cycles as follows:

Load/ Stroke	Cycling Rate	Total Cycles
2%	3 Hz	5400
10%	1 Hz	900
50%	0.25 Hz	375
100%	0.12 Hz	72
	Total	6747/2 Hr

The cycling rate of 1 cpm for the L.E. flaps and speed brake actuator was not changed, but the number of 5 minute steps during which cycling occurred was increased from 2 to 10 to accelerate accumulation of cycles. Additional cycling periods were also added for the seal test fixture as shown on Table 1.

3.3.2 Temperature

Operating temperature conditions were unchanged. Cycling was conducted at room temperature with fan air circulation to simulate compartment air movement. Pump inlet fluid was maintained in the range of +190 to +210°F. Pump case drain fluid was not allowed to exceed +275°F. Actuator return fluid ranged from +100 to +240°F depending on location and cycling mode.

Table 1. Mission/Profile Cycling Program

Step No.	Roll Aileron/ Spoiler	Pitch UHT	Yaw, Rudder	Seal Fixture	AFCS Actuators	Speed Brake L.E. Flap	Actuator Loading	Pump RPM
1	0	0	0	0	Off	Off	Off	3400
2	0	0	0	0	Off	Off	Off	3400
3	2%	10%	50%	S	On	Off	Off	3400
4	10%	50%	2%	S	On	Off	Off	3400
5	50%	2%	10%	S	On	Off	Off	3400
6	0	0	0	0	Off	Off	Off	5900
7	50%	2%	10%	L	On	On	On	5900
8	10%	50%	2%	S	On	Off	On	5900
9	0	2%	100%	L*	On	On*	On	5900
10	2%	50%	0	S	On	On*	On	4900
11	100%	0	2%	L	On	On*	On	4900
12	2%	100%	50%	L*	On	Off	On	4900
13	2%	2%	2%	S	On	Off	On	5900
14	2%	2%	2%	S	On	Off	On	5900
15	2%	2%	2%	S	On	Off	On	5900
16	0	10%	50%	S	On	On*	On	5400
17	0	50%	100%	L*	On	Off	On	5400
18	50%	0	10%	L	On	On*	On	5400
19	100%	0	50%	L	On	On*	On	4400
20	0	50%	0	L*	On	Off	On	4400
21	50%	10%	0	L	On	On*	On	4400
22	0	0	50%	S	On	On	On	4400
23	50%	0	0	L	On	On*	On	4400
24	10%	100%	0	S	On	Off	On	4400

*Denotes Change

NOTE: S = short stroke (±0.10 inch) at 2 Hz

L = long stroke (±1.0 inch) at 0.07 Hz

3.3.3 Performance Checks

Component performance checks conducted at 750, 900, 1050, and 1200 hours of simulator operation were as follows:

Component	Test
Pump	Overall efficiency Heat rejection
Flight Control Actuators	Null leakage Piston seal leakage Rod seal leakage (accumulation)
Solenoid Valves	Internal leakage
Restrictors	Flow rate
Relief Valves	Internal leakage Cracking & re-seat pressure

Disassembly of simulator actuators for wear inspection was performed at 900 and 1200 hours.

3.4 TEST RESULTS

Pertinent events that occurred during the 600 to 1200 hour period of mission/profile cycling are listed in the Test Log presented in Appendix A. The total time that minor components were subjected to 8000 psi is given on Table 2. Maintenance actions are shown on Table 3. Summaries of component performance are presented in the following sections.

Table 2. Total Time Minor Components Subjected to 8000 psi

Description	Part No.	System/Location	Time, Hrs.
Accumulator	3321471	FC-2 Power Module	1050
Check Valve	P4-858	FC-1 Pump	261
Check Valve	95201-5	FC-2 Pump	750
Check Valve	95200-5	FC-1 Power Module	1366
Check Valve	95202-5	FC-2 Power Module	1050
Check Valve	P2-858	FC-2 Press. Regulator	1050+ Pressure Surge Test
Check Valve	P9-858	FC-1 Run-around	1200
Check Valve	P9-858	FC-2 Run-around	900
Check Valve	P11-858	FC-1 Speed Brake	1366
Check Valve	P8-858	LH&RH UHT Actuator	900
Check Valve	P1-858	Rudder Actuator	900
Check Valve	P1-858	FC-1 Speed Brake	1200
Check Valve	P10-858	FC-1 Speed Brake	1200
Filter	AD-A640-83Y1	FC-1 Power Module	1366
Filter	AD-A640-83Y1	FC-2 Power Module	1050
Hose	F37404008-0300	FC-1 Pump	766
Hose	F37404008-0300	FC-2 Pump	450
Hose	DE6964-3-0282	FC-2, Aileron Actuator	600
Hose	28404003-0214	FC-1&2, Spoiler & RFI	107
Manifold	8696-581002	FC-1 Power Module	1366
Manifold	8696-581201	FC-1 Power Module	1200
Pressure Gage	1218-63-1	FC-2 Power Module	1050
Pressure Snubber	95239	FC-1 Power Modeul	1366
Pressure Snubber	95239	FC-2 Power Module	1050
Pressure Trans-mitter	18-2143	FC-1 Power Module	1366
Pressure Trans-Mitter	18-2143	FC-2 Power Module	1050

Table 2. (Cont'd)

Description	Part No.	System/Location	Time, Hrs.
Quick Disconnect	AE80943H	FC-1 Pump Hose	1366
Quick Disconnect	AE81214H	FC-1 Pump Port	1366
Quick Disconnect	AE81215H	FC-1 Ground Service	1366
Quick Disconnect	AE80943H	FC-2 Pump Hose	1050
Quick Disconnect	AE81214H	FC-2 Pump Port	1050
Quick Disconnect	AE81215H	FC-2 Ground Service	1050
Relief Valve	1257A	FC-1 Power Module	1366
Relief Valve	1258	FC-2 Power Module	1050
Restrictor	REFX0380250AB	FC-1 Speed Brake	58 + 52 + small actuator @ 6 sec/stroke
Restrictor	95461-2	FC-2 L.E. Flap	11
Restrictor	95462	FC-2 L.E. Flap	11
Restrictor	95461-1	FC-2 L.E. Flap	11
Solenoid Valve	3221472	FC-1 Speed Brake	1366
Solenoid Valve	3321473	AFCS Pitch Actuator	450
Solenoid Valve	305100	FC-2 Power Module	900
Solenoid Valve	306700	FC-1 AFCS Yaw Actuator	150
Solenoid Valve	306750	FC-2 L.E. Flap	900
Swivel	L38910	FC-1 Speed Brake	58+
Swivel	L39010	FC-1 Speed Brake	58+

NOTE: Total test time includes Phase I.

Table 3. Maintenance Action Totals

	Low Pressure Component	8000 psi Component
Actuators		
Fixes (rework out-of-tolerance part)		1
Rod seal replaced		7
Piston seal replaced		5
Piston rod repaired		2
Fatigue failure		2
"Murphy" failure		1
Fatigue failure due to loose lock nut		1
Pumps		
Removal (excessive case drain flow)		3
Removal (excessive wear)		1
Minor Components		
Leaking	1	1
Malfunction		1
Fatigue failure		1
Erroneous labeling		1
Fitting Leaks	1	
Coil Tubing/Fitting Leaks/Failures		3
Hose Leaks		1
Filter Elements Replaced	5	
Load System		
Fatigue failure (load module)	6	
Hydraulic power supply	2	
TOTAL	15	31

3.4.1 Pumps

Pump operating time totals are listed in Table 4. Performance summaries for FC-1 and FC-2 -3 model pumps are given in Table 5. Overall efficiency and heat rejection were considered to be satisfactory. Transient response, stability, pressure ripple, pressure droop, and compensator drift of both pumps were excellent. Endurance characteristics of the -3 pumps were significantly better than the -2 models. This was attributed to the larger size pintle bearings used in the -3 model and less allowable transfer tube motion.

3.4.2 Actuators

Actuator cycle totals are presented in Table 6. The rudder actuator, yaw AFCS actuator, and control valve on the aileron actuator completed approximately 5,000,000 cycles and are still functioning satisfactorily. The R/H UHT, RFI, spoiler/deflector, and three L.E. flap actuators have accumulated significant numbers of cycles and are still operating satisfactorily. Three fatigue failures occurred: 1) the speed brake actuator cylinder barrel failed at 13,237 cycles; 2) the L/H UHT actuator FC-2 piston failed at 4,607,483 cycles; and 3) the end cap on one of the four L.E. flap actuators failed at 15,322 cycles. Discussions of these failures are given in the following paragraphs.

Actuator seal cycling time totals are presented on Table 7. Piston and rod seal leakage summaries are shown on Table 8; all leakage rates were considered to be satisfactory. Actuator control valve null leakage summaries are given in Table 9. Leakage rates in the rudder, RFI, spoiler, and aileron actuator control valves were all satisfactory; leakage increase during the 600 to 1200 hour test period varied from near zero to moderate.

Table 4. Pump Operating Time

Serial No.	Phase I	Phase II	Vickers Tests	Total
VICKERS MODEL PV3-047-1 & 2				
FC-1 346581	158.5	792.4	200	1150.9
FC-2 348168	130.2	148.6	-	278.8
Spare 346580	0.3	226.6	-	226.9
VICKERS MODEL PV3-047-3				
FC-1 420001	-	545.7	4	549.7
FC-2 422717	-	517.9	4	521.9
Spare 421568	-	84.4	4	88.4

NOTE: Hours are accumulated time and include:

- Simulator mission/profile cycling
- Simulator checkouts
- Simulator tests
- Pump tests
- Simulator demonstrations

Table 5. Pump Performance Checks, Vickers M/N PV3-047-3

	Simulator Running Time, Hours				
	600	750	900	1050	1200
FC-1 Pump, S/N 420001					
Pump operating time, hrs.	1	24.8	189.5	486.4	549.7
Heat rejection, BTU/min	315	300	340	318	295
Overall efficiency, %	85.9	88.5	88.8	88.8	87.9
Case flow, gpm	1.19	1.33	1.33	1.28	1.25
FC-2 Pump, S/N 422717					
Pump operating time, hrs.	1	96.9	193.2	359.2	521.9
Heat rejection, BTU/min	343	-	271	282	262
Overall efficiency, %	84.1	-	89.4	86.9	90.8
Case flow, gpm	1.35	-	0.86	0.87	0.84

NOTES: 1. FC-1 pump returned to supplier for rework at 620 hours on simulator.
2. FC-2 pump returned to supplier for rework at 614 hours on simulator.

Table 6. Actuator Cycle Totals

Actuator	SIMULATOR RUNNING TIME, HR								
	0	150	300	450	600	750	900	1050	1200
L/H UHT	1,054,800[a]	1,560,800	2,066,800	2,525,600	3,018,200	3,524,225	3,993,053	4,506,483[f]	
R/H UHT	-	-	0	357,100	667,400	1,048,268	1,163,042	1,595,303	2,101,328
Rudder	1,054,800[a]	1,554,100	2,060,100	2,532,400	3,038,400	3,544,425	4,050,450	4,563,675	5,045,187
Yaw AFCS	1,054,800[a]	1,560,800	2,066,800	2,572,800	3,078,900	3,584,925	4,090,950	4,604,175	5,110,200
RFI	27,400[b]	533,400	1,039,400	1,545,400	2,051,500	2,530,125	3,036,150	3,576,775	4,082,800
Spoiler	50,000[b]	151,000	252,000	353,000	454,100	555,125	632,723	673,895	774,920
Aileron	1,054,800[a]	1,466,400	1,972,400	2,477,900	2,984,400	3,490,425	3,979,620	4,492,845[c]	4,998,870
L.E. Flap	0	750	1,500	2,250	3,000	6,750	10,350	12,950	16,700[g]
Speed Brake	4,000	4,750	5,500	6,250	7,000	10,750	13,237[d]		
Seal Test Fixtures									
FC-1	-	-	0	510,750	1,021,500	1,532,250	2,043,000	[h]359,520	870,270
FC-2	-	-	-	0	510,750	1,020,825	1,532,250	[h]510,750[i]	
Hydraulic Swivels								323,856[e]	829,881

[a]Cycles accumulated in Phase I.

[b]Tests conducted by LTV in Phase II.

[c]New actuator body installed at 900 hours. See page 96 in Reference 1.

[d]Actuator cylinder failed.

[e]Swivel cycling begun at 954 hours.

[f]Actuator piston failed.

[g]Inboard actuator failed at 15,322 cycles

[h]New piston/piston rods and new seals installed at 900 hours.

[i]Cylinder barrel failed at 1050 hours.

Table 7. Actuator Seal Cycling Time Totals

Actuator	Rod	FC-1 Piston	C/D	C/D	FC-2 Piston	Rod
L/H UHT	N.A.	1078	1246 930	1246 930	1078	1246 1078
R/H UHT*	N.A.	900	900 900	900 900	900	900 900
Rudder	1366 300	1198	1366 1050	1366 1050	1198	1366 300
AFCS	1366	1050	1366			
RFI**	300 300	300	300 300	300 300	300	300 300
Spoiler	N.A.	1050	1200 1050	1200 1050	1050	1200 300
Aileron***	N.A.	300	300	300 300	300	300 300

NOTES: 1. Time values are hours.

2. Times include Phase I hours.

3. Center dam (C/D) and rod seals are 2-stage (except AFCS). Upper value is first stage; lower value is second stage O-ring.

*Piston/piston rod re-chromeplated and ground at 900 hours. No seal changes.

**Piston rod re-chromeplated and ground at 900 hours. All new dynamic seals installed by Shamban.

***New actuator body installed at 900 hours. Control valve assembly and piston rod assembly not changed. All new dynamic seals. Seals installed by Greene, Tweed.

Table 8. Dynamic Seal Leakage Summary

Actuator	No. Cycles/ 150 Hrs	Max. Allowable Leakage/150 Hrs.	System/Seal	SIMULATOR RUNNING TIME, HR 150	300	450	600	750	900	1050	1200
L/H UHT	506,000	20,000	FC-2 Rod	220	221	132	115	223	378	332	
			FC-1 Piston	0	T	T	T	-	T	T	
			FC-2 Piston	T	T	T	T	-	T	0	
R/H UHT	506,000	20,000	FC-2 Rod	-	-	82	111	151	50	403	734
			FC-1 Piston	-	T	1D	1D	-	30D	T	0
			FC-2 Piston	-	T	T	T	-	T	T	T
Rudder	506,000	20,000	FC-2 Rod	41	3	0	7	0	0	3	4
			FC-1 Piston	T	T	T	T	-	T	T	T
			FC-2 Piston	T	T	T	T	-	T	T	T
Yaw AFCS	506,000	20,000	FC-1 Rod	28	(1)	(1)	(1)	(1)	(1)	(1)	(1)
			FC-1 Piston	(2)	(2)	(2)	(2)	(2)	(2)	(2)	(2)
RFI	506,000	20,000	FC-2 Rod	45	0	0	0	0	0	0	0
			FC-1 Piston	(2)	-	-	-	-	-	-	-
			FC-2 Piston	(2)	-	-	-	-	-	-	-
Spoiler	101,000	4,000	FC-2 Rod	(3)	10	10	27	19	166	51	158
			FC-1 Piston	T	T	T	T	-	16cc/ min	0	T
			FC-2 Piston	T	T	T	T	-	0	T	T
Aileron	506,000	20,000	FC-2 Rod	272	434	242	367	161	216	20	545
			FC-1 Piston							T	T
			FC-2 Piston	T	1D	T	T	-	T	2D	2D
L.E. Flap	3,750	150	FC-2 Rod	75	12	34	(4)	10	78	114	
			FC-2 Piston	T	T	T	T				
Speed Brake	3,750	150	FC-1 Rod	1	0	0	0	17			
			FC-1 Piston	T	T	T	T				
Swivel Joint	506,000	5,000	Inboard						(5)	1904	2002
			Outboard							1544	0 (6)

NOTES: 1. Rod seal leakage values are number of drops accumulated in 150 hours.
2. Maximum allowable rod seal leakage is 1 drop/25 cycles (Ref MIL-C-5503).
3. Piston seal leakage values are: T = Trace (less than one drop/min); D = Drops/min

(1) Not meaningful because of servo valve face seal seepage.
(2) Not measured because all cylinder porting is internal.
(3) Not meaningful because of coil tube failure.
(4) Not meaningful because of coil tube fitting leakage.
(5) Swivels installed at 954 hours. Begin leakage measurements at 978 hours.

Table 9. Control Valve Leakage Summary

Actuator	System	SIMULATOR RUNNING TIME, HR.								
		0	150	300	450	600	750	900	1050	1200
L/H UHT	FC-1	49	21	28	63	66	67	120	120	-
	FC-2	22	28	27	41	53	74	112	107	-
R/H UHT	FC-1	-	-	39	55	75		108	104	160
	FC-2	-	-	25	41	58		76	61	66
Rudder	FC-1	4	5	21	32	35	30	32	41	30
	FC-2	6	8	17	35	53	30	45	31	20
Yaw AFCS	FC-1	110	112	132	148	161	174	214	144**	144
	-									
RFI	*FC-1	48	21	80	50	70		57	64	80
	*FC-2	68	65	116	126	149		83	82	88
Spoiler	*FC-1	25	24	65	124	97	91	76	82	104
	*FC-2	22	18	41	52	47	64	48	56	65
Aileron	-	-	-					***	8	21
	FC-2	-	26	23	41	24		29	6	22

NOTES: 1. Leakage values are in cc/min.
2. Inlet fluid temperature approximately +130°F.
3. Maximum allowable leakage: 125 cc/min (goal).
* Some of this leakage may be around shrink-fit control valve (sleeve O.D. has no seals).
** Moog valve M/N 56E-201, S/N 2.
***New actuator body installed at 900 hours.

Table 10. Spoiler/Deflector and RFI Actuator Internal Leakage

SIMULATOR RUNNING TIME, HOURS →			300	450	600	750	900	1050	1200
Actuator	System	Direction							
Spoiler/Deflector	FC-1	extend	38	42	57	46	48	58	57
		retract	46	100	79	106	92	76	65
	FC-2	extend	29	120	49	36	40	32	35
		retract	44	86	92	92	80	96	92
RFI	FC-1	extend	144	95	61		45	68	79
		retract	184	159	198		112	156	148
	FC-2	extend	100	122	111		97	76	108
		retract	76	124	113		107	96	152

NOTES: 1. Leakage values are cc/min.
2. Valve inputs are hard-over.

The UHT and yaw AFCS actuator valves experienced larger increases in null leakage over time and exceeded the design goal of 125 cc/min. The shrink fit valve sleeves in the RFI and spoiler/deflector actuators were believed to be leaking, see reference 1. Data taken during the last 600 hours of cycling disclosed no significant change in this leakage, Table 10.

Speed Brake Actuator. The speed brake actuator experienced a catastrophic failure when the cylinder barrel split after completion of 13,237 full load cycles, Figure 2. The piston rod was near the full extend position and under 40,000 pounds of compressive load at the time of failure. The actuator barrel was made of 7075-T651 aluminum alloy; the inner bore surface was hard anodized for wear protection.

The barrel was separated to permit a metallurgical examination of the fracture surfaces. A large number of crack initiation sites were observed throughout the entire inside surface of the barrel. Numerous fatigue patterns were seen on the fracture surfaces, Figure 3. The area that produced the final rupture is the large semi-circular fatigue pattern.

Scanning electron microscope analysis confirmed the primary cause of failure was fatigue, although examination of the crack initiation sites showed evidence of cleavage patterns that are indicative of brittle, plastic fracture. The cracks were believed to have been initiated by the anodic wear surface which exhibited brittle behavior under high stress, cyclic loading.

Anodic coatings can cause a significant reduction in the fatigue strength of wrought aluminum alloys. The extent to which fatigue strength is reduced depends upon the coating thickness and the composition and temper of the base alloy.

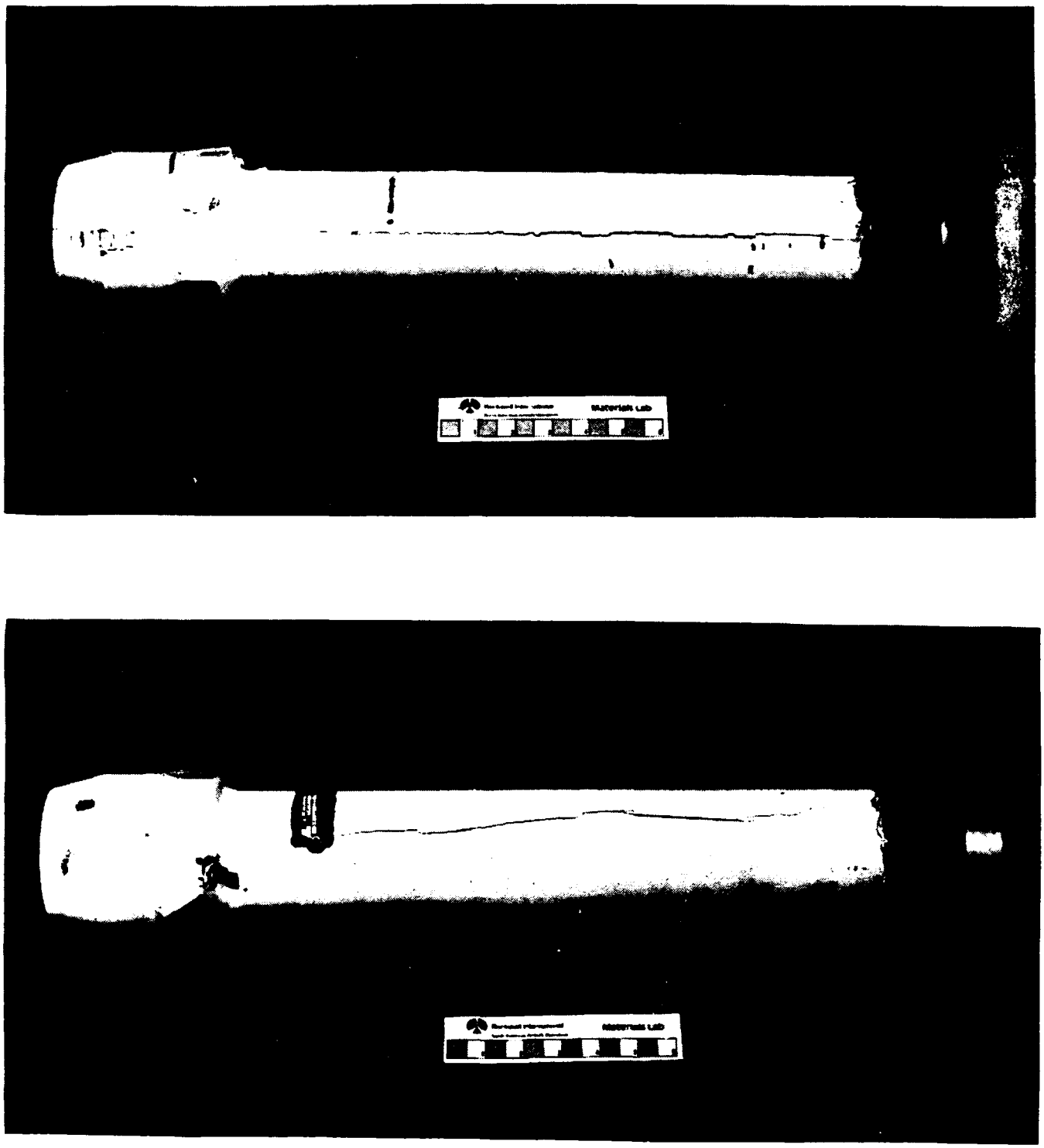
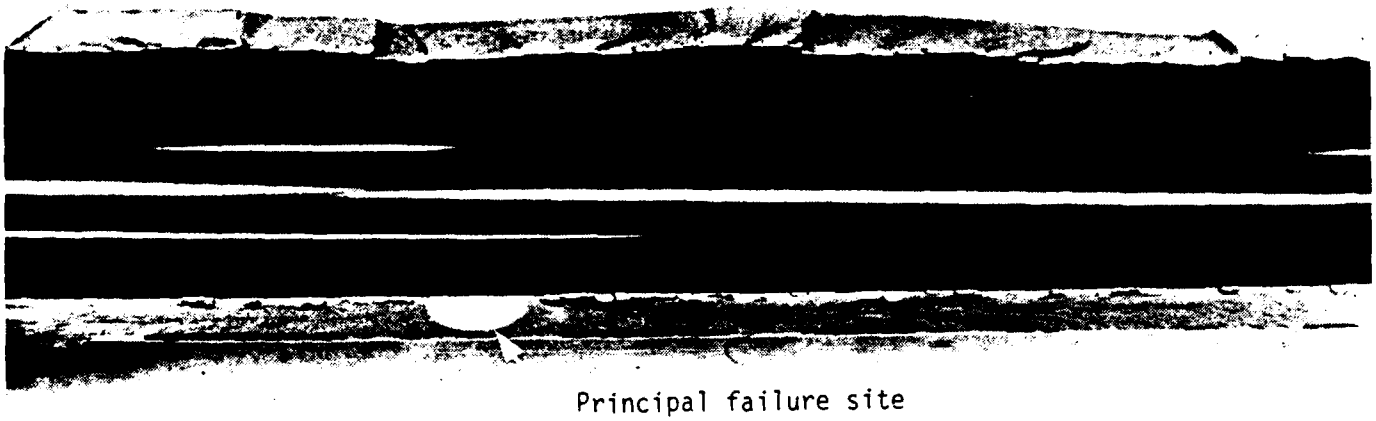


Figure 2. Crack in barrel of speed brake actuator



Principal failure site

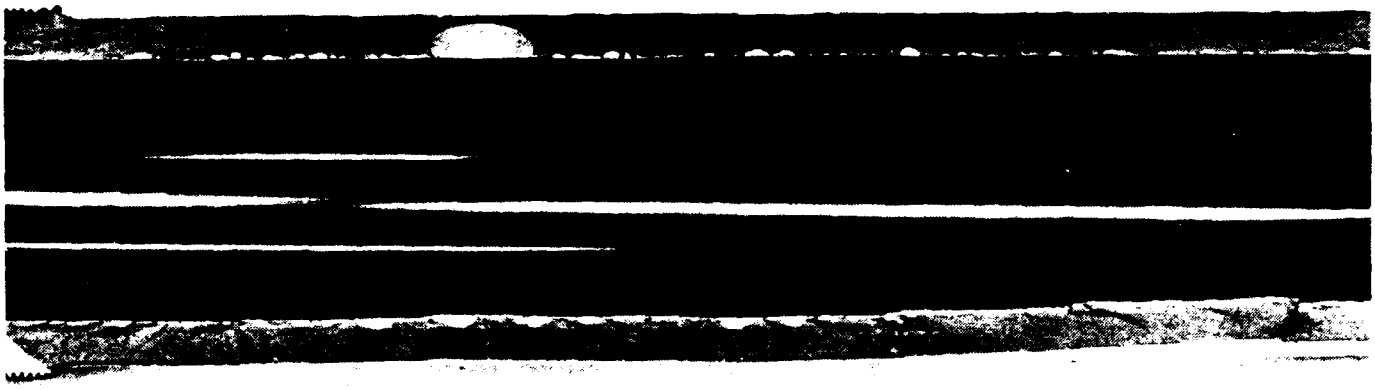


Figure 3. Fatigue patterns

The speed brake actuator was re-designed using a 4340 alloy steel barrel and a thinner wall end cap. The weight of the modified actuator was estimated to be slightly less than the original actuator. Fabrication of the re-designed actuator is planned if a second 1200 hours of endurance cycling is run on the LHS simulator.

Unit Horizontal Tail Actuator. The L/H UHT actuator malfunctioned when the FC-2 piston failed after completing 4,607,483 cycles. Although the actuator was still cycling, high internal leakage prevented operation at 100% stroke and load. Disassembly disclosed the forward edge of the piston was cracked causing the piston seal to fail, Figure 4. The piston was made of 4340 alloy steel heat treated to 180,000 psi (minimum). The fracture started in the fillet at the bottom of the piston seal groove.

Metallurgical examination of the fracture surface revealed fatigue striations which correlate to the latter stages of crack propagation. Macroscopic examination of the piston showed grinding marks in the fillet where the fracture started. A photomicrograph of the fillet cross-section disclosed the inside radius was not properly machined to insure a smooth, rounded profile. Measurements indicated the fillet radius was approximately 0.015 in.; the drawing call-out was $0.035^{+.000}_{-.015}$ in. The drawing requirement for surface finish in the fillet region was 125 RMS. Since the piston fillet is a high stress area, a 32 RMS finish would be more appropriate. In summation, the failure was caused by a stress concentration created by poor machining and inadequate surface finish requirements.

The 0.015 in. fillet radius and rough surface finish resulted in a notch sensitivity factor (K_T) of 5. Based on a calculated stress level of 45,000 psi in the fillet area and 4340 alloy steel fatigue data, reference 7, the expected life is 70,000 full load cycles. Actual cycle totals were as follows:

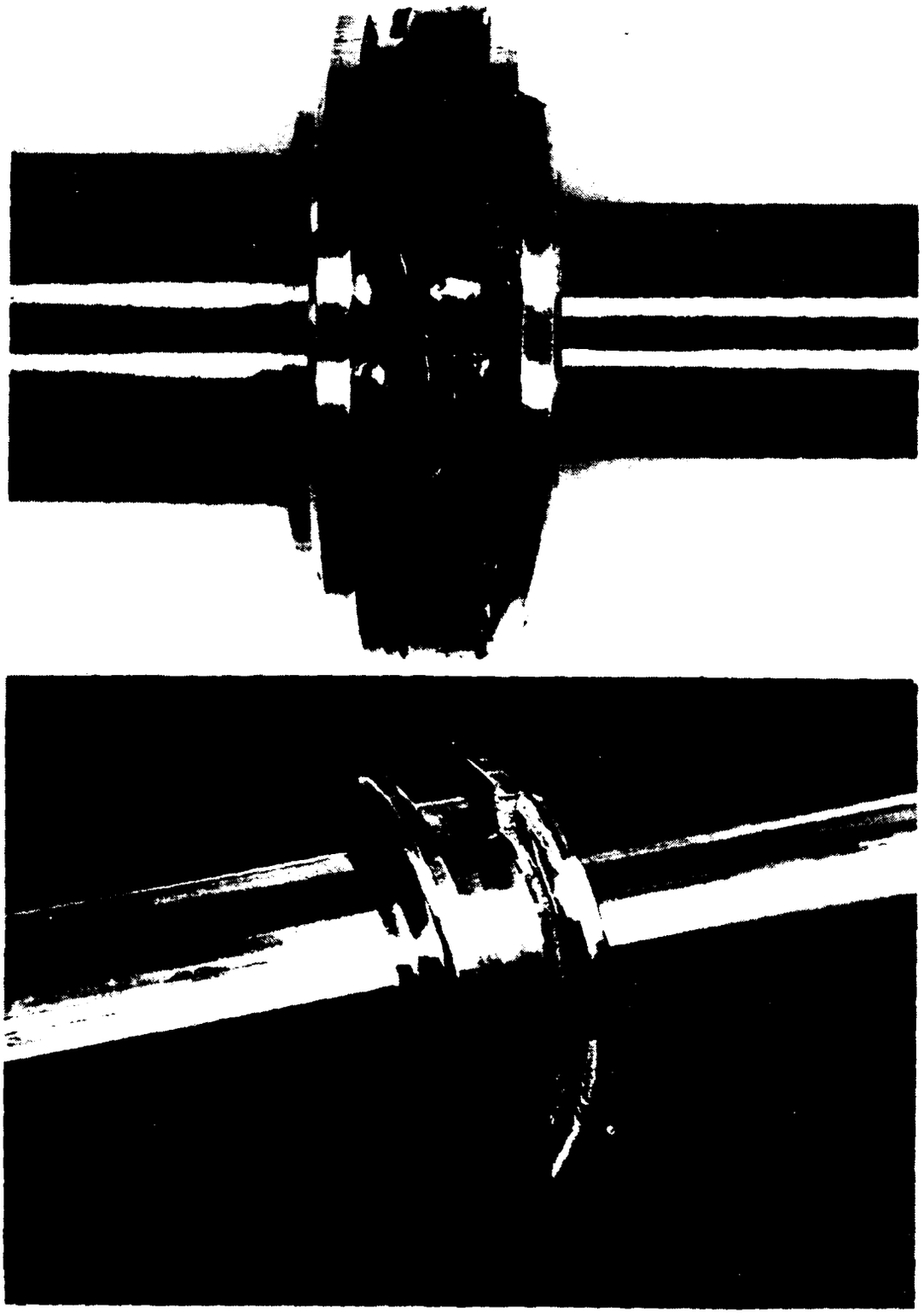


Figure 4. Piston failure in LH UHT actuator

Load/Stroke	Cycles
100%	46,000
50%	230,000
10%	645,000
2%	3,686,000

The piston thus appears to have performed satisfactorily based on its physical characteristics. Recommended changes to improve piston life are: 1) a thicker edge to reduce stress levels; 2) a larger radius fillet; and 3) a smoother fillet surface.

L.E. Flap Actuator. Actuator No. 2 (outboard actuator on inboard load module) failed after completing 15,322 cycles. The failure occurred when end cap P/N 83-00262-129 fractured in the threaded area, Figure 5. The end cap is made of 4340 steel heat treated to 200,000 psi (min.). Metallurgical examination disclosed the failure originated in a thread root. The primary crack causing the fracture traversed approximately 95% through the end cap wall thickness.

Actuator assembly instructions require that a 23,000 lb compressive preload be applied to the actuator prior to tightening the end cap lock nut, P/N 83-00262-119. (See drawing in reference 1, Appendix D.) The lock nut is then secured with lockwire. Examination of L.E. flap actuators No. 1, No. 3, and No. 4 revealed this lock nut was loose on all three actuators. The lockwire was intact, but had stretched sufficiently during cycling to permit the lock nut to turn. This condition was believed to have produced the actuator No. 2 failure when loss of the 23,000 lb preload allowed relative motion between the cylinder barrel and end cap. Impact loads thus occurred as the actuator cycled causing stress levels to exceed design values. Lockwire should not be employed to prevent relaxation of torque preloads. A more positive means such as using steel pins to prevent relative motion between the end cap and lock nut should be considered.

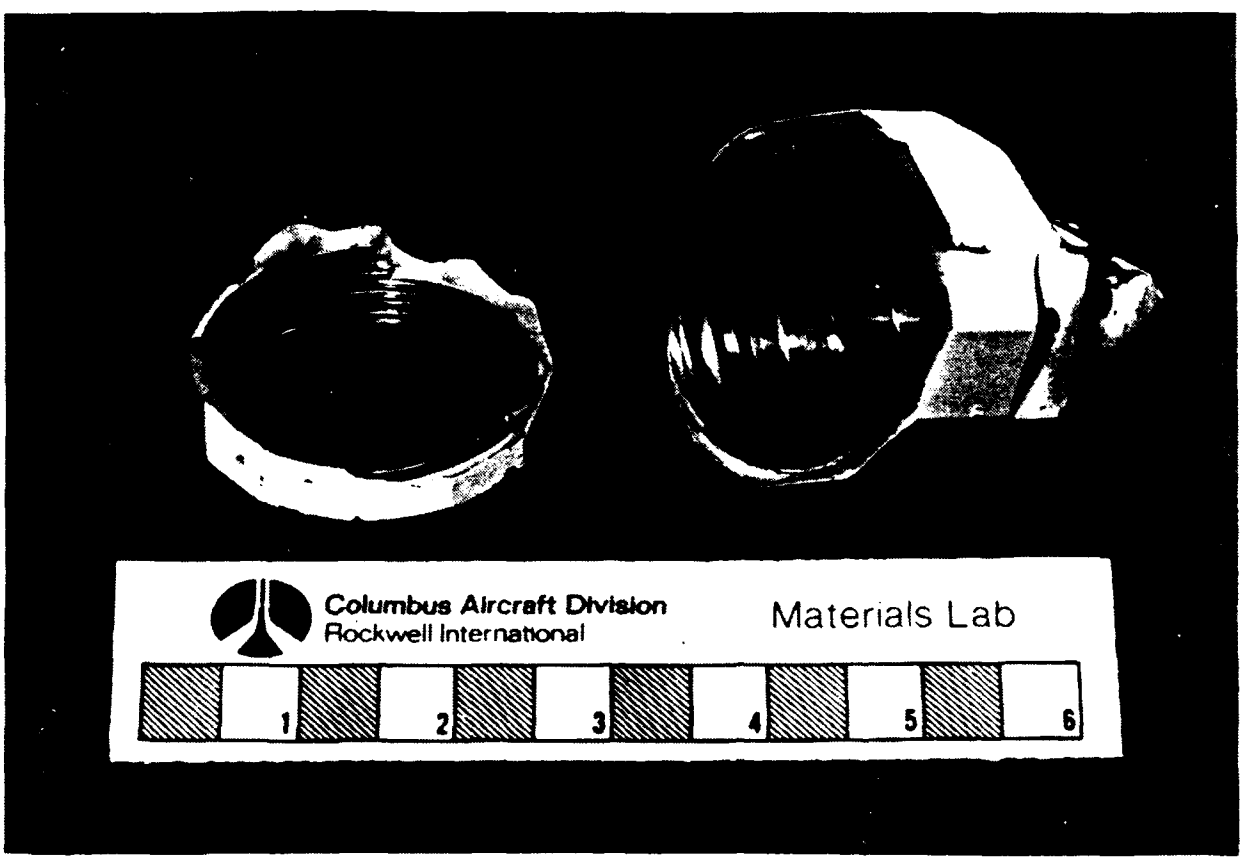


Figure 5. End cap failure in L.E. flap actuator

3.4.3 Solenoid Valves

A summary of internal leakage measured at 150 hour intervals is given on Table 11. The maximum allowable internal leakage goal was 10 cc/min. Four valves met this goal; the 4-way speed brake valve did not.

A seal failure occurred in the L.E. flap valve at 838 hours, see Appendix A. The cause was attributed to a mis-drilled internal port that allowed the seal to extrude. Other than the noted discrepancies, the performance of all valves was considered satisfactory.

3.4.4 Relief Valves

Relief valve performance is summarized on Table 12. All performance characteristics were satisfactory. No malfunctions or failures occurred.

3.4.5 Restrictors

Restrictor performance is shown on Table 13 A slight upward trending of flow occurred as simulator time accumulated. A poppet failure was discovered at the 900 hour performance check point, see Appendix A. The cause was due to insufficient webb area between the poppet holes. All other restrictors performed satisfactorily.

3.4.6 System Filters

New elements were installed in all filters at the beginning of the second 600 hours of mission/profile cycling. The pump case drain filter size was increased to provide more dirt holding capacity. The case drain and return line filters now have identical housings. All filters had 5 micron absolute ratings. Four return line element changes were made over the course of the test, Table 14. Element replacement intervals varied from 200 to 300 hours. Replacement intervals of 300 to 400 hours were considered as a goal.

Table 11. Solenoid Valve Internal Leakage Summary

Valve	Mode	SIMULATOR RUNNING TIME, HR.								
		0	150	300	450	600	750	900	1050	1200
Yaw AFCS (3-way) (P/N 3321473)	on	4	4	4	1500(6)*	4**		7	5	5
	off	8	7	7	8(9)*	8**		15	12	12
Yaw AFCS (3-way) (P/N 306750-1001)	on	--	---	---	T***	0		0	T	T
	off	--	---	---	T	3D		1D	T	1D
Speed Brake (4-way) (P/N 3321472)	on (ext.)	28	30	43	1600(52)*	32	42	68	44	60
	on (hold)	13	10	11	18(17)*	12	18	27	24	32
	off (ret.)	34	34	50	44(60)*	36	45	62	44	60
L.E. Flap (4-way) (P/N 306700)	on (ext.)	0	T	T	T	T		0*	T	T
	on (ret.)	T	T	T	T	T		0*	T	T
	off--	T	T	T	T	T		0*	T	T
FC-2 Reservoir (2-way) (P/N 305100)	off	--	---	12D	4D	9D		T	6D	

NOTES: 1. Leakage values are cc/min at 8000 psi except: T = Trace (less than one drop/min); D = Drops/min.
2. Inlet fluid temperature approximately +100°F.
3. Maximum allowable leakage goal: 10 cc/min.
* Leakage after installation of new seal.
** Used at 8000 psi for 450 hours; used at 2300 psi thereafter.
***Valve P/N 306750-1001 installed in FC-1 system. Valve P/N 3321473 installed in load system.

Table 12. Relief Valve Performance Summary

Valve	Simulator Hours	Cracking Pressure, psi	Reseat Pressure, psi	Internal Leakage
*FC-1 (M/N 1257A)	150	8900	8400	1 drop/min
	600	8900	8400	zero
*FC-2 (M/N 1258)	600	8750	8450	zero
**FC-1	1200	8625	8400	2 drops/min
**FC-2	1200	8450	8350	18 drops/min

*Test conducted using pump test stand.

**Test conducted on 8000 psi test bench.

Table 13. Restrictor Performance Summary

Restrictor	SIMULATOR RUNNING TIME, HR.				
	0	150	600	900	1200
Speed Brake (ext.) (P/N REFX0380250AB) (ret.)	4.0	4.02	4.02	4.04	4.04
	4.17	4.18	4.21	4.16	4.25
L.E. Flap, Inbrd., 1-way (P/N 95461-1)	1.18	1.18	1.23	1.14*	1.18
L.E. Flap, Inbrd., 2-way (P/N 95462)	1.27	1.24	1.33	1.33	1.35
L.E. Flap, Outbrd., 1-way (P/N 95461-2)	2.22	2.21	2.28	2.31	2.23
L.E. Flap, Outbrd., 2-way (P/N 95462)	1.18	1.18	1.23	1.24	1.25

NOTE: Flow in gpm at return pressure.

*Restrictor poppet failed. Flow value shown is for repaired restrictor.

Table 14. Filter Element Replacement

Running Time, Hrs.	Pump Case Drain		Pressure		Return	
	FC-1	FC-2	FC-1	FC-2	FC-1	FC-2
0	5u[1]		5u[1]		5u[1]	
78	5u[2]					
100	5u[3]					
150	15u[4]					
200					5u[4]	
250	5u[3]		5u[3]			
300		5u[1]		5u[1]		5u[1]
342		5u[2]				
460	15u[5]	15u[5]				
560						5u[2]
600*	5u[1]	5u[1]	5u[1]	5u[1]	5u[1]	5u[1]
742	5u[3]					
802					5u[5]	
920						5u[5]
1050					5u[5]	
1142						5u[5]

*Pump case drain filter size made the same as return filter size.

[1]New test
[2]Filter ΔP button operated
[3]New start with reworked pump
[4]Plumbing or setup change
[5]Element dirty

3.4.7 Fluid

Fluid contamination checks were made frequently during the first 600 hours of mission/profile cycling, reference 1. These checks, and the need to make filter patches of pump case drain debris, depleted system fluid. A persistent concern during this period was the black residue observed on the filter patches. The residue was subsequently identified. A discussion of the black residue is presented in Section 7.0.

Experience gained during the first 600 hours indicated that less frequent checks would be adequate and would preserve system fluid. Particulate contamination samples were therefore taken at the 900 and 1200 hour check points, Table 15. The fluid contamination level was better than NAS 1638 class 2; class 8 is the maximum acceptable.

3.4.8 Fittings

Fitting performance was excellent during the second 600 hours of endurance cycling. Only one leak developed and this was in a return line. The cause was attributed to a loosening of the tube/fitting swage.

Aircraft vibration levels were not simulated during mission/profile cycling. Such vibration would increase the failure rate of fittings. For this reason, it is recommended that flex stress/pressure impulse tests be conducted on various tube/fitting combinations to more fully demonstrate 8000 psi tube/fitting integrity.

3.4.9 Hoses

Test hose locations were 1) pump discharge lines, and 2) aileron actuator installation. Hoses were also used at the spoiler/deflector and RFI actuators, but these hoses were considered as interim fixes to permit LHS simulator testing and would not be used on an A-7 aircraft.

Table 15. Fluid Contamination

Hours	System	PARTICLE SIZE RANGE, MICRONS					
		1-5	5-15	15-25	25-50	50-100	100+
0	FC-1		1207	39	7	0	0
50	FC-1		16434	1298	182	20	2
64	FC-1		23944	1309	375	82	2
100	FC-1		9416	752	160	20	0
150	FC-1		1059	44	183	25	0
200	FC-1		12463	89	31	0	0
250	FC-1		3825	716	414	127	11
300	FC-1		394	69	19	2	1
490	FC-1		75810	206	92	20	0
490	FC-2		33788	24	10	4	0
542	FC-1		113532	17185	3493	257	2
542	FC-2		33983	1703	466	62	7
*600	FC-1		1984	353	109	32	2
*600	FC-2		1298	597	143	16	6
**600	FC-1	45	20	7	3	2	1
**600	FC-2	45	26	13	3	1	0.4
1200	FC-1		1614	61	18	4	0
***1200	FC-2		73601	112	51	9	0
Reference Standard							
NAS 1638, Class 8			64000	11400	2025	360	64

*Manual count made by Rockwell-Columbus. Data received 1-9-86.

**Manual count made by Pall Corporation. See Appendix H.

***Count for 5-15 micron range is suspect.

Titeflex hoses were used at the pumps in Phase I and Phase II up to the 900 hour point. Hose performance was excellent; no problems occurred. At 900 hours, Titeflex provided lighter weight hoses (1.5 lb versus 2.2 lb). These were installed and performance has been excellent. The hoses were special made and no part number was assigned.

Titeflex also provided a new design hose (no part number) for the aileron actuator installation. FC-1 pressure hose developed a leak in a metal joint after only 3.7 hours of use. The cause was attributed to faulty fabrication. An identical hose (except for length) in FC-2 system completed 300 hours satisfactorily. The hose originally installed in FC-1, Aeroquip P/N DE6964-3-0282, was used to replace the failed Titeflex hose.

3.4.10 Swivels

Two hydraulic swivel joints were mounted on the LHS speed brake actuator. The joints did not swivel as the actuator cycled because of the load module kinematics, reference 1. The speed brake actuator failed after completing 13,237 cycles, see Section 3.4.2. A setup was subsequently fabricated to oscillate the swivels and permit a better assessment of their endurance capabilities, Figure 6. LHS simulator roll axis controls were used to cycle the swivels. The cycling schedule was therefore based on primary flight control requirements; the swivels were designed for use on a secondary flight control actuator (20,000 cycle life). The accelerated cycling rate provided a means to quickly evaluate the potential life of the swivels. Swivel cycling began at the 954 hour point. Maximum oscillation amplitude was $\pm 35^{\circ}$. Test results are shown on Table 16. The swivel seals are single stage with a capped O-Ring. Swivel performance was considered satisfactory.

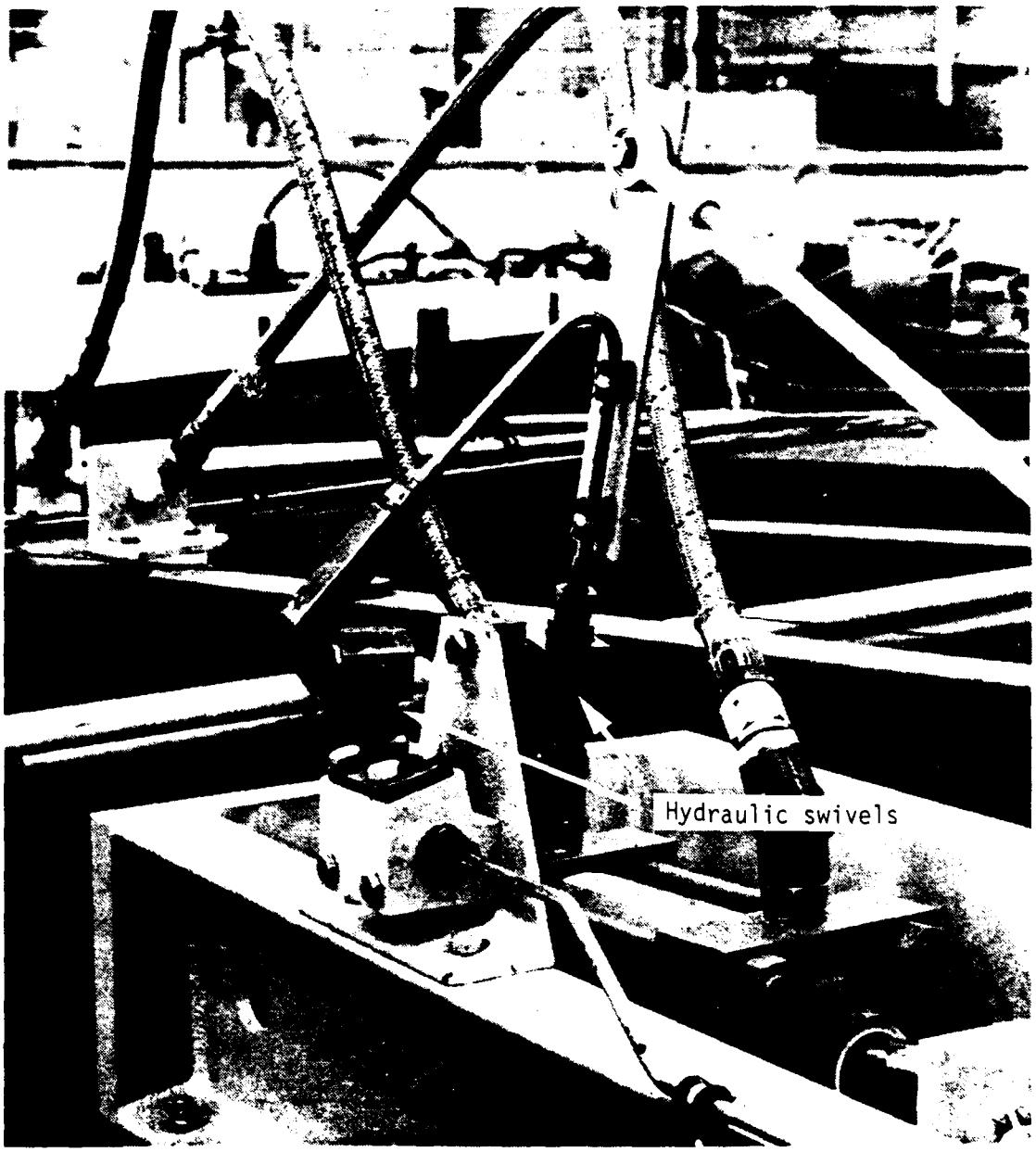


Figure 6. Hydraulic swivel joint cycling setup

Table 16. Swivel Leakage Summary

Simulator Running Time, Hr.	954	1,050	1,200
Swivel Cycles Completed	0	323,856	829,881
Leakage, Cycles/Drop			
Swivel P/N L38910 (outboard)		170	0 (See Note 2)
Swivel P/N L39010 (inboard)		210	2,002

NOTES: 1. Maximum allowable leakage is 1 drop/100 cycles.

2. Swivel P/N L38910 developed excessive leakage at 534,663 cycles. Single stage cap seal replaced with O-ring and cycling resumed.

Table 17. Seal Test Fixture Seals

Fixture	Type	Location	Stage	Part Number	Supplier
FC-1	Piston	-	-	S30772-3044 S36326-330-19-552	Shamban
	Rod	Inbd	1st	S36325-218-19-552	
			2nd	S30775-218P-19 S33157-218-19	
		Outbd	1st	S36325-218-19-552	
			2nd	S33353-218P-19	
FC-2	Piston	-	-	266-33000-964-1200	Greene, Tweed
	Rod	Inbd	1st	265-21800-964-1200	
			2nd	4635-21800H-964	
		Outbd	1st	265-21800-964-1200	
			2nd	4635-21800H-964	

3.4.11 Seal Test Fixture

A seal test fixture was plumbed into FC-1 and FC-2 systems at a location originally intended for the RH aileron module, see reference 1. Each fixture was a balanced actuator driven by an 8000 psi direct drive electro-hydraulic servo valve. A total of 4 piston rod and 2 piston head seal systems were evaluated. The cycling schedule is shown on Table 1.

The test fixtures were installed in the LHS simulator at the 300 hour point of mission/profile cycling. An evaluation of seal performance during the period of 300 to 900 hours was determined to be invalid due to misalignment in the 3-piece piston/piston rod assembly which caused undesirable side loading. New one-piece piston/piston rods were fabricated for use beginning at 900 hours. New piston head and piston rod seals were provided by Greene, Tweed and Shamban, Table 17.

Seal performance in FC-1 fixture was excellent after 300 hours and 870,270 cycles of operation. Seal evaluation in FC-2 fixture was cut short when the actuator barrel (4340 steel) developed a small crack at 1050 hours. The crack was attributed to a high stress level (84,000 hoop stress with 8,000 psi pressure applied), piston score marks on the barrel I.D. (due to the previously used 3-piece piston assembly), and use of the fixture in prior tests (3.25 million total accumulated cycles at the time of failure). Seal test results are summarized on Table 18. The condition of all seals at the conclusion of testing was satisfactory.

Table 18. Seal Test Results

Simulator Hours →	900	948	1,050	1,200
Cycles Completed				
FC-1 Fixture		0	359,570	870,270
FC-2 Fixture	0		510,750	(1)
Seal Leakage, drops/min (accumulated)				
FC-1 Fixture				
Inboard Rod Seal			114	121
Outboard Rod Seal			80	100
Piston Seal			T	T
FC-2 Fixture				
Inboard Rod Seal			(2)	(1)
Outboard Rod Seal			(2)	(1)
Piston Seal			T	(1)

(1) Fixture barrel failed at 1050 hours.
(2) Leakage not measured, but was satisfactory.

3.5 RELIABILITY UPDATE

A reliability trend analysis was developed in reference 1 covering simulator failures that occurred during the first 600 hours of mission/profile cycling. The reliability growth concept projects that Mean-Time-Between-Failure (MTBF) plotted versus time will approximate a straight line (using log-log scaling) if corrections are incorporated following each failure. Analysis details are discussed in Appendix F.

A summary of LHS simulator failures that occurred during the period of operation from 600 hours to 1200 hours is given on Table 19. This information was used to generate the updated reliability growth trend line shown on Figure 7. The MTBF for the first 600 hour period was 94 hours, reference 1. The MTBF calculated at the 1200 hour point was 123 hours, Figure 7. This is a 31% growth of MTBF over the first 600 hours of testing. The 123 hours MTBF is a 58% improvement over the A-7E 3000 psi hydraulic system (77.8 hours MTBF) as reported by 3M data and shown on Figure 7.

The test data confirms that 8000 psi hydraulic systems can be designed, fabricated, and operated successfully for extended periods of time without unusual problems occurring. The reliability growth as presented on Figure 7 demonstrates that design and quality factors that impact reliability can be identified through a test program, and that corrective actions can be taken to improve system reliability.

The test program did not simulate one important environmental influence on reliability -- airframe vibration. Although the hydraulic systems generated some vibration due to their own operation, an additional applied vibration could accelerate some failures and perhaps cause some that otherwise would not occur. Incorporation of this factor in the test program would probably reduce the observed reliability values somewhat.

Table 19. Summary of LHS Simulator Failures

Component	Simulator Hours	Failure	Location/Remarks
RH UHT Actuator	619	Malfunction	Control valve sticking; faulty fabrication.
O-ring	696	Leaking	Spoiler actuator C1 port Rosan fitting loose; faulty fabrication.
Channel Seal	838	Leaking	L.E. flap solenoid valve; faulty fabrication.
Speed Brake Act'r	850	Fatigue	Cylinder barrel; under-design.
'T' seal	853	Leaking	FC-1 reservoir piston low pressure seal; normal wear.
O-ring	898	Leaking	RFI actuator center dam bleed hole; normal wear.
Restrictor	900?	Fatigue	L.E. flap circuit; poppet failed; under-design.
Hose	904	Leaking	Aileron actuator FC-2 pressure; new design.
Fitting	998	Leaking	Rudder AFCS solenoid valve return port; loose swage.
LH UHT Actuator	1080	Fatigue	FC-2 piston edge cracked; under design and faulty fabrication.
L.E. Flap Act'r	1144	Fatigue	End cap fractured; lock nut worked loose permitting impact loading.

NOTE: Summary excludes coil tube failures, pump rework and non-hydraulic related mechanical failures.

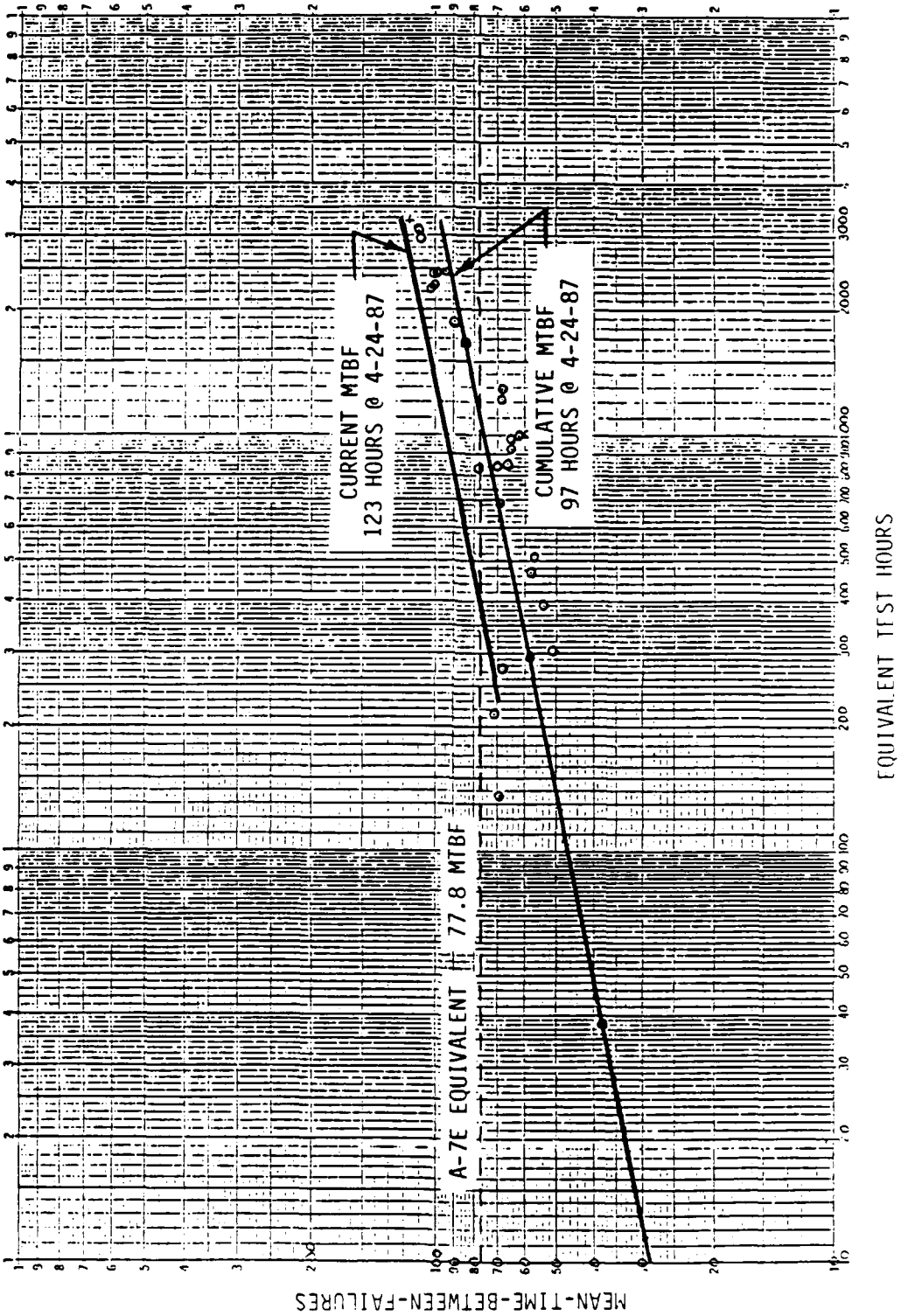


Figure 7. LHS simulator reliability growth, 0 to 1200 hours of operation

4.0 GROUND SUPPORT EQUIPMENT RE-WORK

4.1 INTRODUCTION

An existing 3000 psi AHT-63 portable test stand was modified to operate at 8000 psi, Figure 8. Descriptions of the test stand and demonstration tests are detailed in reference 1. Test stand performance was considered good except heat generation exceeded heat dissipation capabilities. The principal cause of the problem was heat rejection in the 8000 psi pump. This pump is a modified version of the Denison P64 Gold Cup series, and is an axial piston design with manual controls for displacement and pressure compensation.

The Denison pump was removed from the test stand and a Dynex/Rivett pump installed. The following sections describe the new pump and the tests conducted.

4.2 PUMP INFORMATION

The new GSE pump, Dynex/Rivett M/N PV4018-X6, is a modified version of a 20 year old 5000 psi design originally used on mining machinery and road building equipment. The unit is a check valve type pump with in-line pistons (10), a non-rotating block, and a fixed angle cam. Flow can be manually adjusted up to 10 gpm; pressure compensation is adjustable from 2000 to 9000 psi. The pump installation in the AHT-63 MOD test stand is shown in Figure 9.

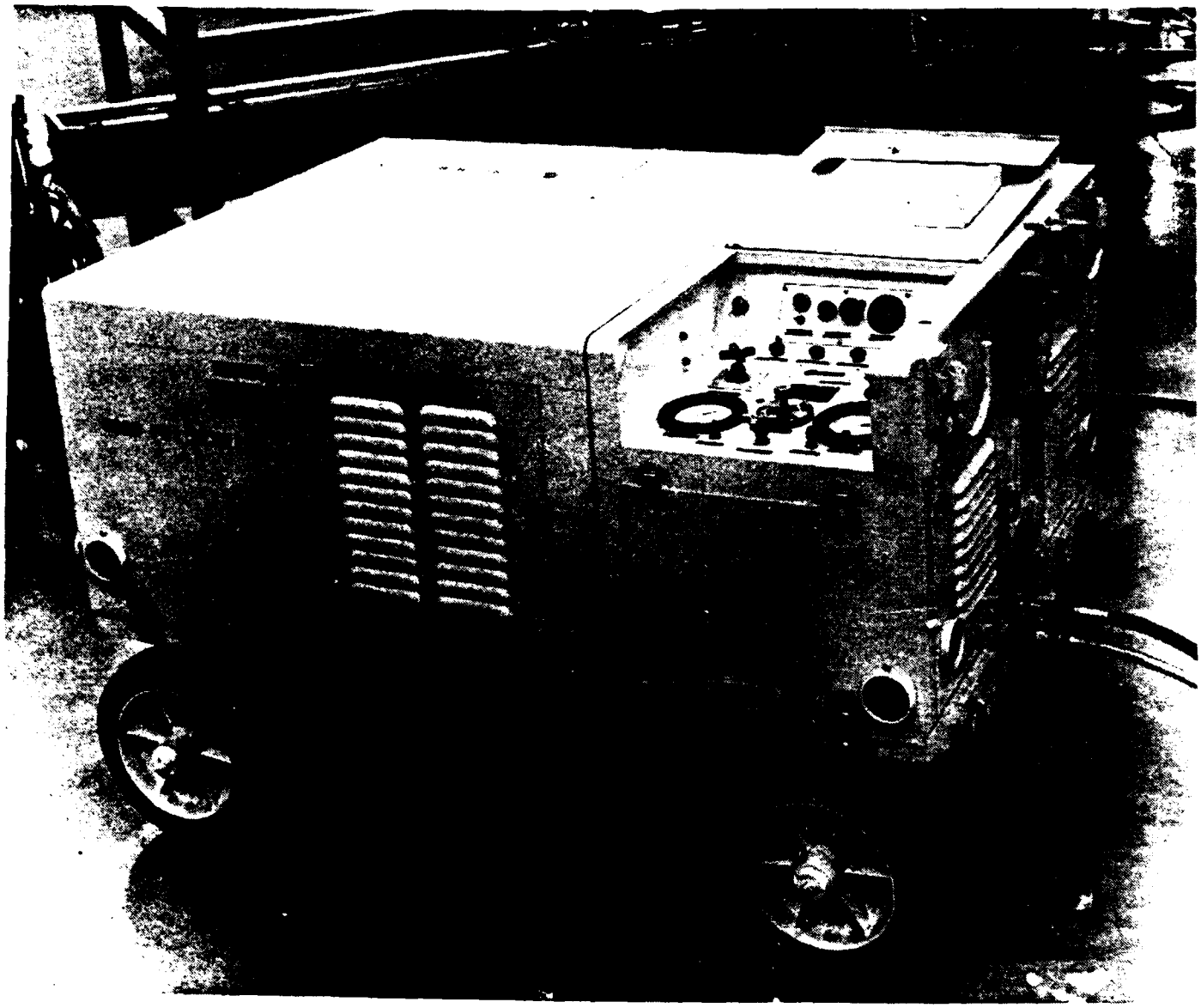


Figure 8. Modified AHT-63 portable test stand

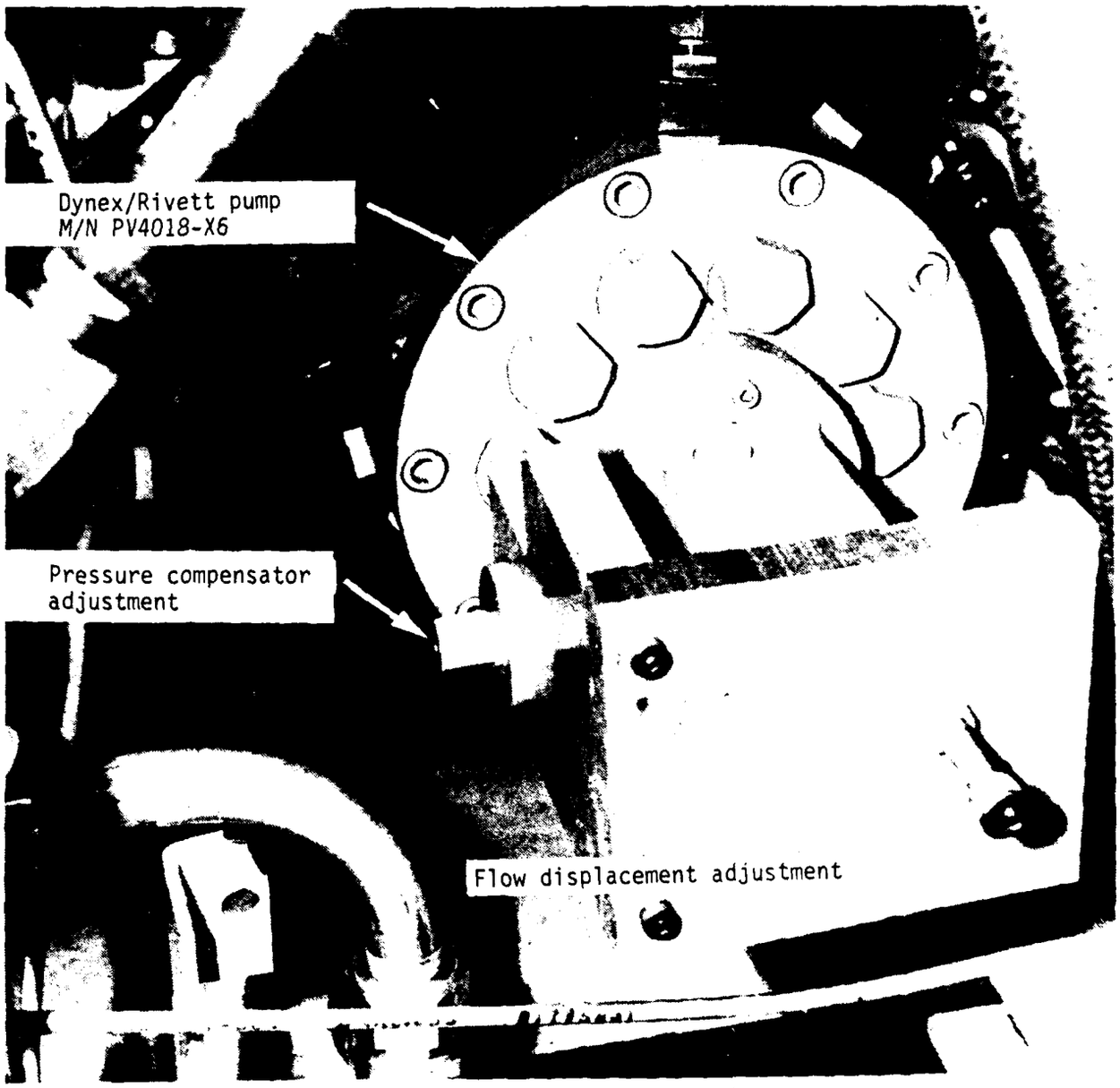


Figure 9. Test stand pump installation

4.3 TESTS CONDUCTED

Tnree types of tests were conducted on the re-worked test stand:

Pump performance
- Pressure/flow characteristics
- Pump ripple

Temperature stabilization
- Test stand alone
- Test stand and LHS simulator

Test stand noise level

4.3.1 Pump Performance Tests

Pump pressure/flow characteristics were determined using a needle (load) valve installed between the test stand pressure and return hoses. Pump displacement was set at 8 gpm and pressure was varied from full-flow pressure to zero flow pressure. Test data were obtained with the pump compensator set at 4000, 6000, and 8000 psi, Figure 10. Compensator operation was smooth and the pressure flow characteristics were considered excellent.

Pressure ripple in the pump discharge line was determined for the following configurations:

Hoses not connected to test stand

Hoses connected to test stand only

Hoses connected to LHS simulator

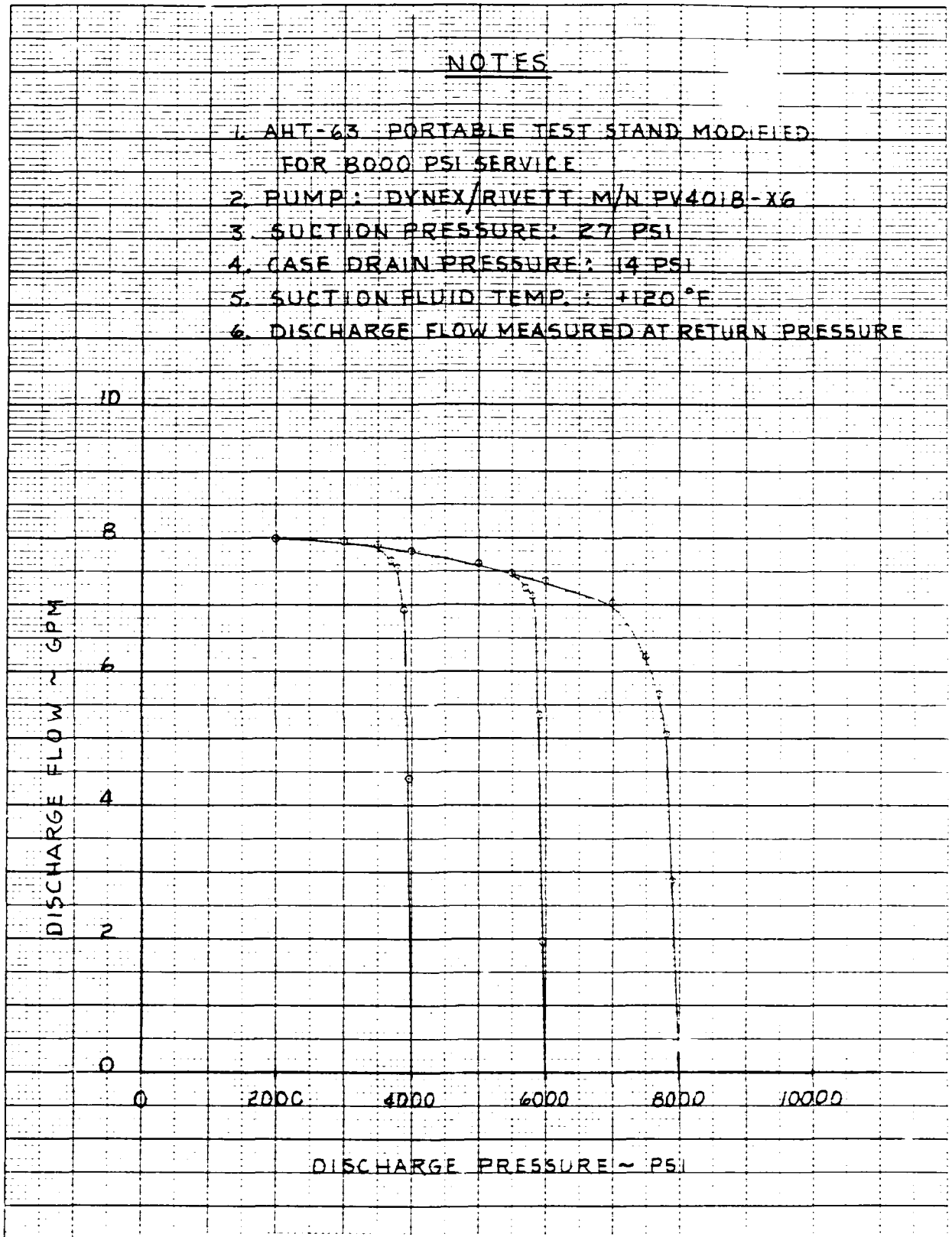


Figure 10. Test stand pressure/flow characteristics

A clamp-on type pressure transducer was used to measure pressure ripple, reference 1. Transducer locations and test data are shown on Figures 11 and 12. Ripple was less than 100 psi peak-to-peak with the test stand powering the LHS simulator. This was considered excellent.

4.3.2 Temperature Stabilization Test

The ability of the re-worked test stand to handle hydraulic throttling loads without excessive temperature build-up was demonstrated by conducting temperature stabilization tests. The test stand was operated (alone) with 8000 psi discharge pressure and continuous loads of 1 gpm, then 3 gpm. Temperature versus time data were recorded. Temperature stabilization was achieved at less than +150°F (pump inlet fluid), Figure 13. The goal was to achieve stabilization at temperatures no higher than +180°F.

Temperature data were taken with the test stand powering the LHS simulator. All flight control actuators were operated with 2% stroke at 3 Hz. Return flow was 2.1 gpm. Temperature stabilization occurred at +130°F (pump inlet fluid), Figure 14.

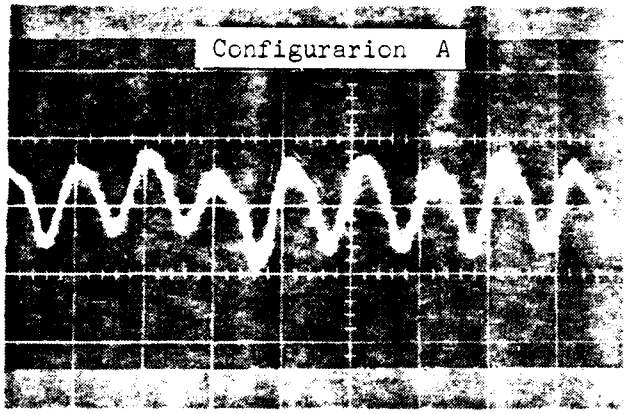
The excellent operating efficiency of the Dynex/Rivett pump was made evident by measuring test stand drive motor current. The motor is rated for 50 hp at 1200 rpm with 440 VAC 60 ampere power input. A comparison between the Denison and Dynex/Rivett pumps is given below:

	Motor current, amperes	
	No Flow, 8000 psi	8 gpm, 8000 psi
Denison pump	55	90
Dynex/Rivett pump	32	47

NOTES

1. Operating pressure level: 8000 psi
2. Discharge flow: 0.5 gpm
3. Pressure ripple measured in pump discharge line downstream of check valve
4. Configuration:
   - A Test stand hydraulic disconnects not used (dust covers in place)
   - B 15 ft. long pressure and return hose connected to test stand disconnects. Load valve installed between pressure and return hoses.
5. Pressure ripple trace scaling:

Ripple amplitude	100 psi/div
Time	0.005 sec/div



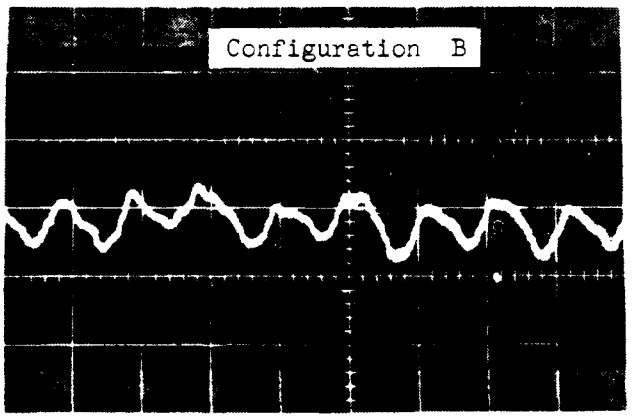
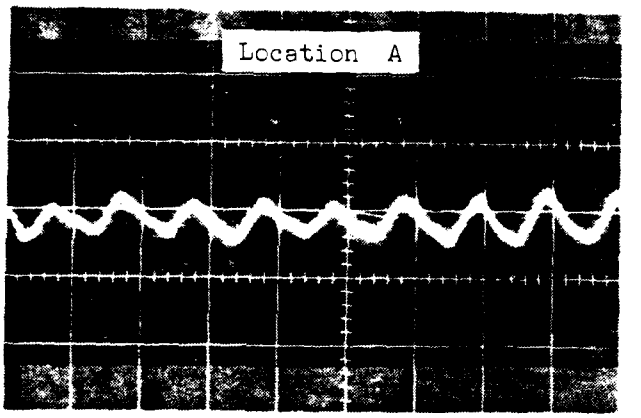


Figure 11. AHT-63 MOD. test stand pressure ripple

NOTES

1. Operating pressure level: 8000 psi
2. Discharge flow: 0.4 gpm
3. 15 ft. long pressure and return hoses connected to A-7E LHS simulator FC-1 system
4. Pressure ripple measurement locations:
   - A Test stand pump discharge line downstream of check valve
   - B LHS simulator pressure line 18 in. upstream of FC-1 pressure disconnect
5. Pressure ripple trace scaling:

Ripple amplitude	100 psi/div
Time	0.005 sec/div



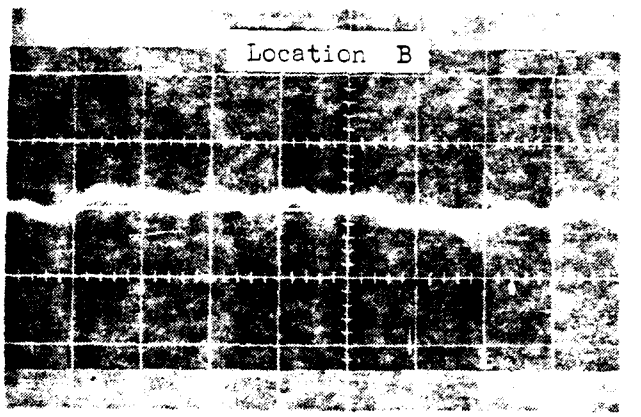


Figure 12. AHT-63 MOD. test stand/LHS simulator pressure ripple

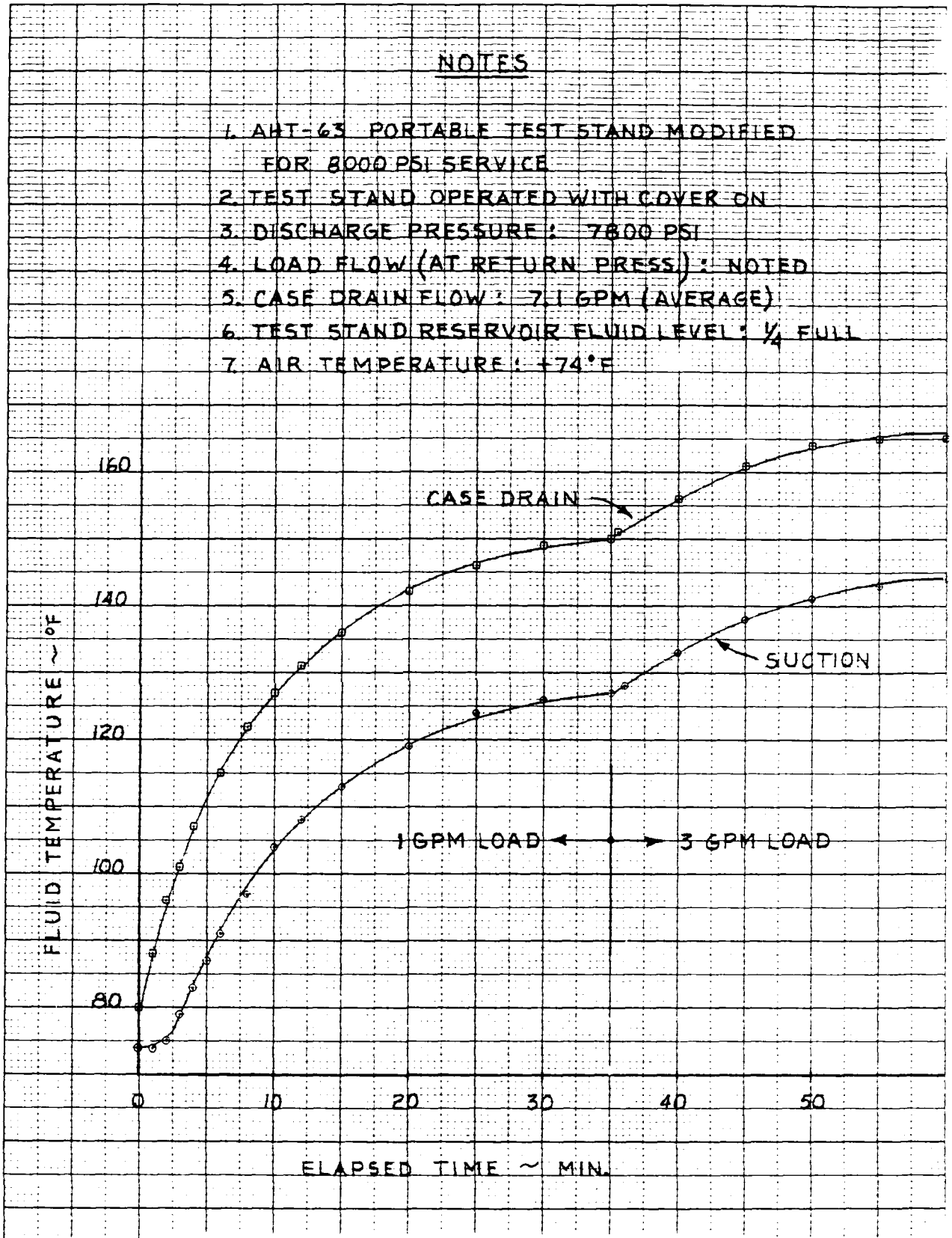


Figure 13. Test stand temperature stabilization

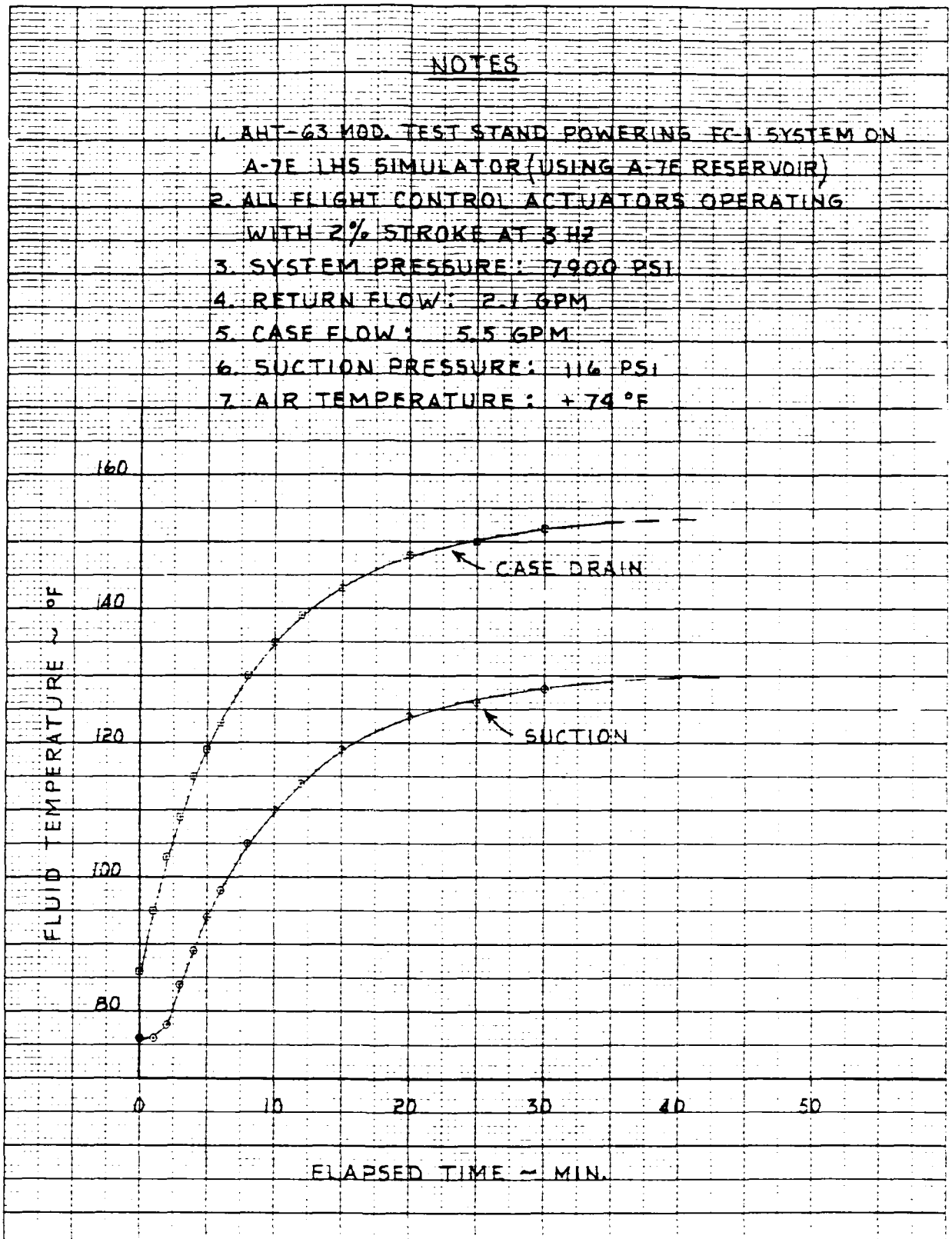


Figure 14. Test stand/LHS simulator temperature stabilization

4.3.3 Noise Level Test

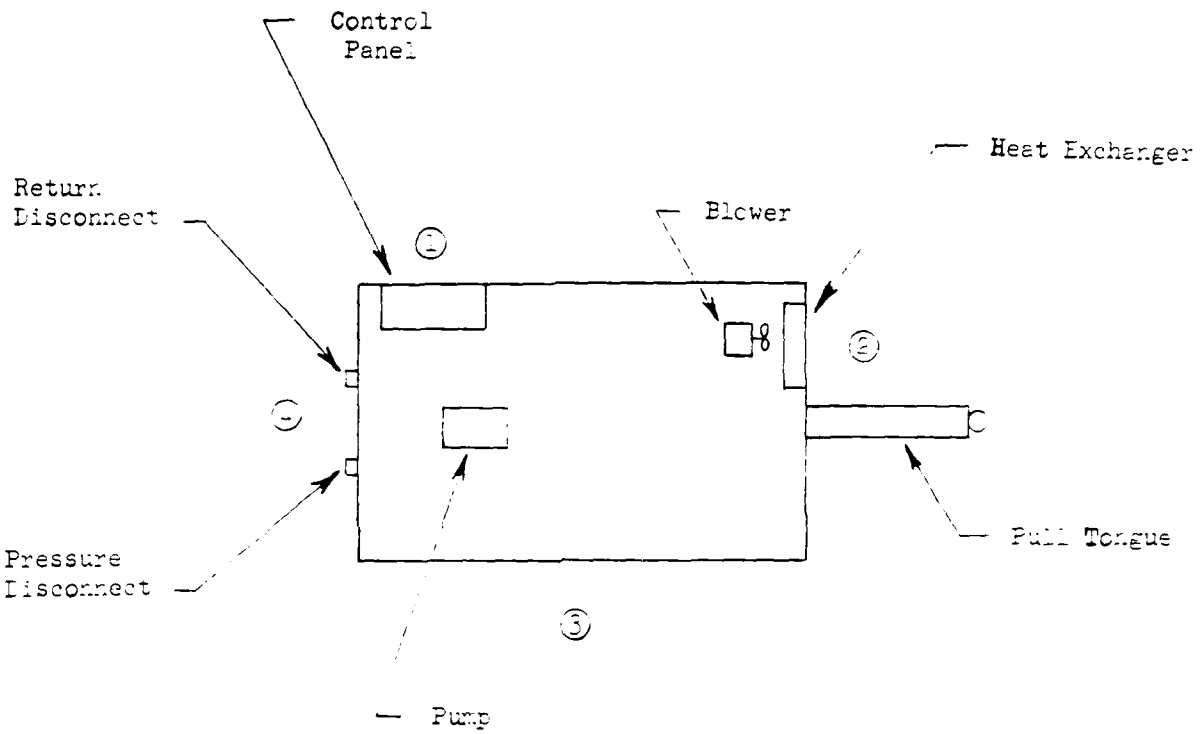
A survey was conducted to provide data to compare the noise level of the re-worked test stand with the original 3000 psi test stand before conversion to 8000 psi. The survey data are shown on Figure 15. Average noise levels of the two configurations are compared below.

	Noise Level, db (Average of 4 Locations)
3000 psi original test stand	90
8000 psi re-worked test stand	86

NOTES

1. Noise measurements taken with Metrosonic Metrologger II approximately 2 ft. from test stand at the indicated locations.

2. Data taken with test stand cover on.



Location	Noise Level, dBa
1	87
2	92
3	83
4	82

Figure 15. AHT-63 MOD. test stand noise levels

5.0 PISTON SEAL EVALUATION TEST

5.1 INTRODUCTION

Satisfactory dynamic seals are critical to the successful performance of lightweight hydraulic systems. Candidate rod seals were evaluated in a 400 hour test program conducted by LTV Aerospace and Defense Co. The results, reported in reference 1, were used to select rod seals for LHS simulator test actuators. Candidate piston seals were evaluated by LTV under the current contract and a summary of results is presented in this section. The complete LTV report is contained in Appendix C.

Major goals of the piston seal test program were:

- Evaluate several experimental and state-of-the-art piston seal systems.
- Make an assessment of the maximum diametral clearance tolerated.
- Evaluate, secondarily, two-stage rod seal systems with relatively high diametral clearance.

5.2 CANDIDATE PISTON SEALS

Four suppliers participated in the program by providing technical support and test seal samples.

C. E. Conover Co.
Greene, Tweed & Co.
W. S. Shamban Co.
Tetrafluor Co.

Fourteen seal configurations were studied for possible testing. A decision matrix was established and each of the 14 seal systems were rated against 15 weighted criteria. Nine candidates selected for endurance testing are shown on Figure 16.

5.3 TEST CONDITIONS

Six separate test actuators cycling simultaneously were used to screen the candidate piston seals. Endurance cycling was conducted until a candidate failed or until leakage became unacceptable at which time an alternate seal was installed. Dimensional measurements were taken to determine wear on the piston/piston rod, cylinder bore, end cap, and seals. Test conditions were:

Hydraulic fluid:	MIL-H-83282
System pressure:	8000 psi
Piston seal Δp:	6400 psi (80% of 8000 psi)
Fluid temperature:	
Endurance cycling	+250 to +275°F
Leakage checks	-40°F and ambient
Test duration:	400 hr (eight 50 hr blocks)
Endurance cycling:	
$\pm$1.75 in. stroke at 9 cpm	90% of each 50 hr block
$\pm$0.10 in. stroke at 30 cpm	10% of each 50 hr block

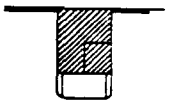
Candidate P1
266-21800-964-1200
"CGT" Seal
Greene, Tweed and Co.

Candidate P2
S30772-218P19, S35117
"Plus Seal II", interlocking backup ring set
W. S. Shamban and Co.

Candidate P3
S30772-218P19, S33157-218-19
"Plus Seal II", non cut backup rings
W. S. Shamban and Co.

Candidate P4
S34232-218P19
"Wedgpak" set
W. S. Shamban and Co.

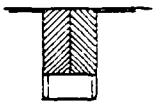
Candidate P6
CEC 5223-218
Dual Trapezoid Piston Seal
C. E. Conover and Co.



Candidate P8
TF 500A-7218(490)
Piston ring assembly
Tetrafluor Co.

Candidate P12
TF1146-218
Cap seal, rectangular shape or o-ring elastomer, heavy duty backup rings, anti-blowby notches in backup rings
Tetrafluor Co.

Candidate P13
S30772-218P19N, S33157-218-19
"Plus Seal II", Noncut backup rings
Anti-blowby notches in cap seal
W. S. Shamban and Co.



Candidate P14
S35119
Dual Piston Ring
W. S. Shamban and Co.

Figure 16. Candidate piston seals

5.4 TEST RESULTS

Results of the screening test were as follows:

Seal Candidate	Hours Completed	Seal Condition at Hours Completed
P1 (#2)	202.6	Good
P2	404.7	Excellent
P3	237.4	Seal Failed
P4	404.7	Excellent
P6	253.3	Seal Failed
P8	201.7	Seal failed. Cause not attributed to seal
P12	122.7	Excellent
P13	122.7	Fair
P14	202.6	Excellent

Based on the number of test hours completed, seal wear, seal leakage, and diametral clearance, the candidates were ranked as follows (1 = best):

Rank	1	2	3	4	5	6	7	8	9
Candidate	P4	P2	P14	P8	P12	P1	P6	P3	P13

Two circumstances prevented a valid assessment from being made of the effect of increased diametral clearance on piston seal performance: 1) the method of measuring diametral clearance proved to be unreliable; and 2) the actuator end caps influenced cylinder barrel stretch.

No rod seal failures occurred during the 404.7 hour test. The baseline system accumulated 2,714,918 cycles. Diametral clearance varied from 0.0028 to .0060 in. All rod seals were in good to excellent condition at the conclusion of testing.

6.0 COIL TUBE DESIGN GUIDELINES

6.1 INTRODUCTION

Hydraulic extension/swivel units are employed to transmit power to the spoiler/deflector and RFI actuators on the A-7 aircraft. Coil tubing is used for these applications on the LHS simulator. Due to severe space constraints and complex motion of the spoiler/deflector actuator, coil tube stress levels were higher than desired. As a result, failures occurred, reference 1. Rockwell addressed this problem area in an attempt to find a solution.

6.2 A-7 SPOILER/DEFLECTOR INSTALLATION

When the A-7 spoiler/deflectors operate, high velocity air flow is directed through the spoiler/deflector compartment and around the spoiler/deflector actuator. This condition precludes the use of coil tubing due to vibration that would be induced in the tubing by the impinging air stream. Space constraints prevent the use of hoses -- even -3 size highly flexible Kevlar hoses. The best solution is the original configuration -- hydraulic extension units with end swivels. These units function well on the A-7 but have a history of leaking. (This was the reason they were not considered for use on the 8000 psi LHS simulator.) High friction in the end swivels may be a factor in the leakage problem. Pressurization at 3000 psi tightens the swivel due to the squeezing action of the hydraulic seal on the knuckle joint. Significant torques are required to move the swivel with pressure applied. This high friction and frequency of actuator operation accelerates seal wear and is believed to contribute to the leakage problem.

A number of swivel manufacturers were contacted concerning the possibility of making 8000 psi swivels for the spoiler/deflector application. Friction in the swivel joint was discussed, and it appears that reducing friction in a knuckle type swivel would be difficult. A different type of swivel, for example, two or three single-plane swivels in series, might be feasible. If a solution were found for the spoiler/deflector installation using hydraulic extension units with re-designed end swivels, this, of course would eliminate the coil tube problem. Coil tubing has important potential use in 8000 psi systems and the inability to use coil tubing in one installation should not detract from their use in other, more suitable applications.

The A-7 RFI actuator installation is more amenable to the use of coil tubing because space constraints are reasonable and actuator motion is simple. Although leakage did eventually occur in both the nested coil and flat tri-coil tubing used on the LHS simulator (see reference 1), these problems can be resolved by modifying the original coil tube assembly, such as using different wall thickness tubing, a different method of fitting attachment, or a different coil design. Factors affecting coil tube design are discussed in the next section.

6.3 DESIGN GUIDELINES

6.3.1 Introduction

Hydraulic tubing can safely withstand simultaneous pressurization, bending, and torsion stresses. Bending of coils or torsion of straight tubes are the principal modes of flexure and various proportions or combinations can be used. The following guidelines will assist the design of hard tube connections between two points which move relative to each other. Additional design information can be found in references 2, 3, 4, and 5.

6.3.2 Basic Configurations

If the fixed end of the section of tubing to be designed is near or can be brought near the same plane as the moving end, a coiled arrangement is indicated, Figures 17a and 17b. If the fixed end is located some distance from, and in a plane normal to the plane of the moving end, a torsional arrangement can be used, Figure 18a. The location of the fixed point relative to the moving point can be determined by using the shortest route of tubing and accepting whichever bending or torsional mode occurs. In most cases, either mode or a combination may be possible and should be considered before an optimized configuration is selected, Figure 18b.

Helical coiled tubing with angular motion, Figure 19a, has bending (tension/compression) stresses. Linear motion helical coil tubing as shown in Figure 19b has bending and torsional (shear) stresses. Simple torsion tube configurations principally operate with shear stresses, Figure 19a.

The point of maximum stress in all coil tube configurations is at the tube fitting. Bending stresses at the fitting should be kept below 2/3 of the normal tube bending allowable. Bending stresses in other portions of the coil may approach the allowable tube bending stress.

6.3.3 Bending Configuration

To achieve minimum bending moments at the end fittings, connect the starting and ending points of the moving end with a layout line extended to the fixed end point, Figure 20. Assume no fixity at the ends and calculate the resulting bending moments and stresses. Each element of the coiled tube configuration is then analyzed for deflections. The sum of the deflections relative to the reference axis will give the total motion of the moving end relative to the fixed end. Revisions to the trial configuration will be

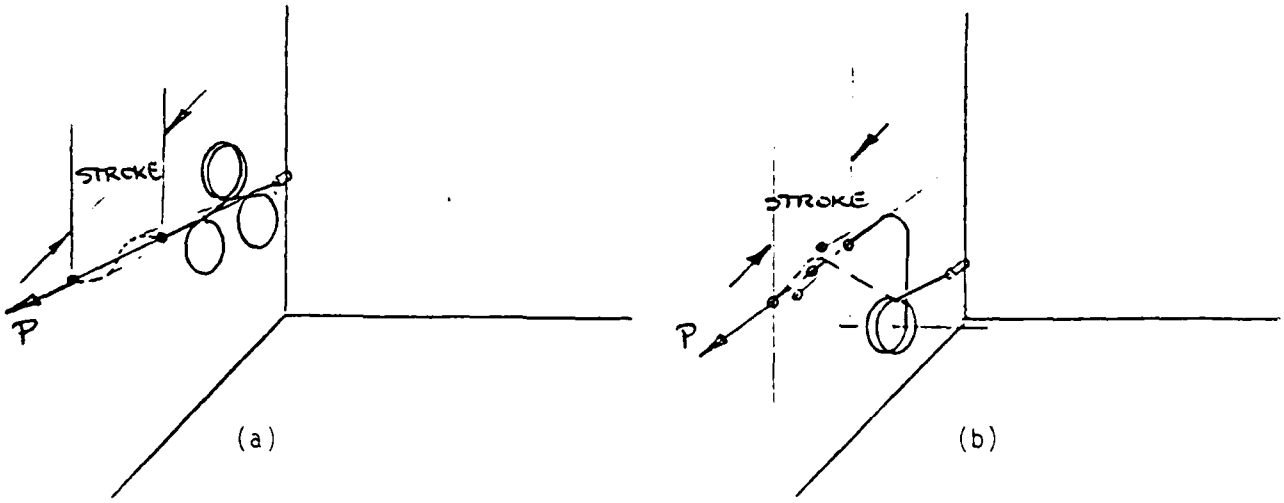


Figure 17. Bending configurations

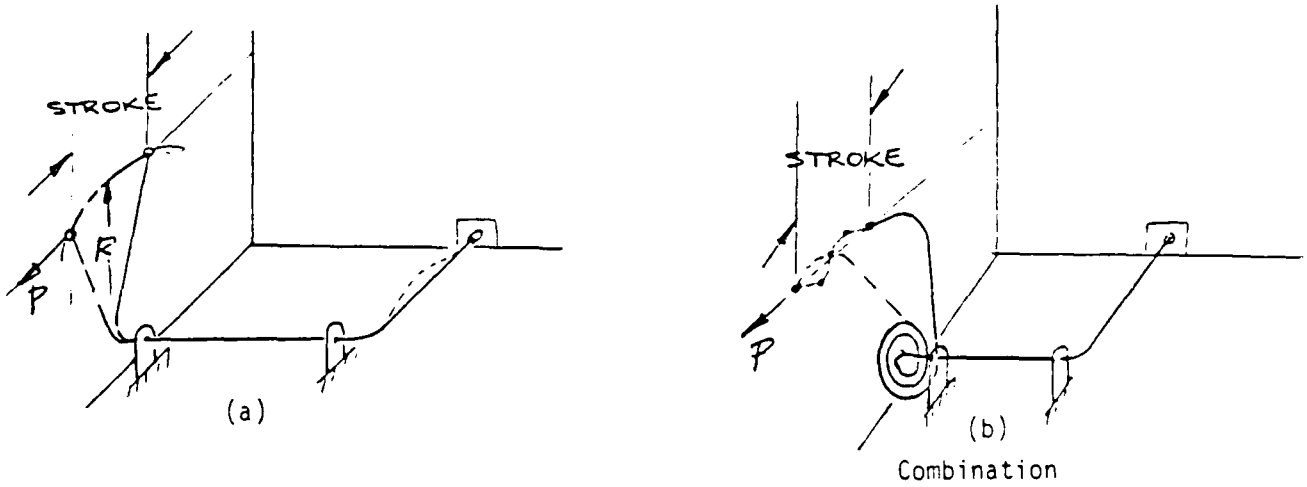


Figure 18. Torsional configurations

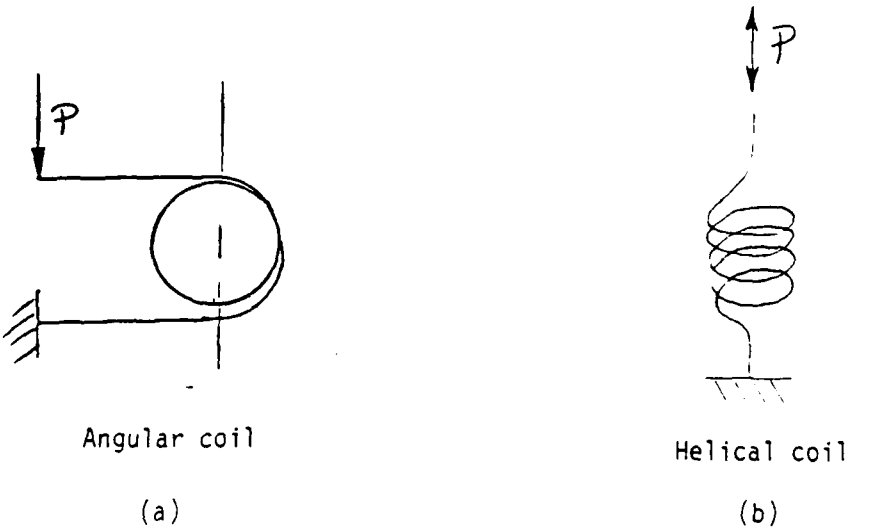


Figure 19. Helical coils

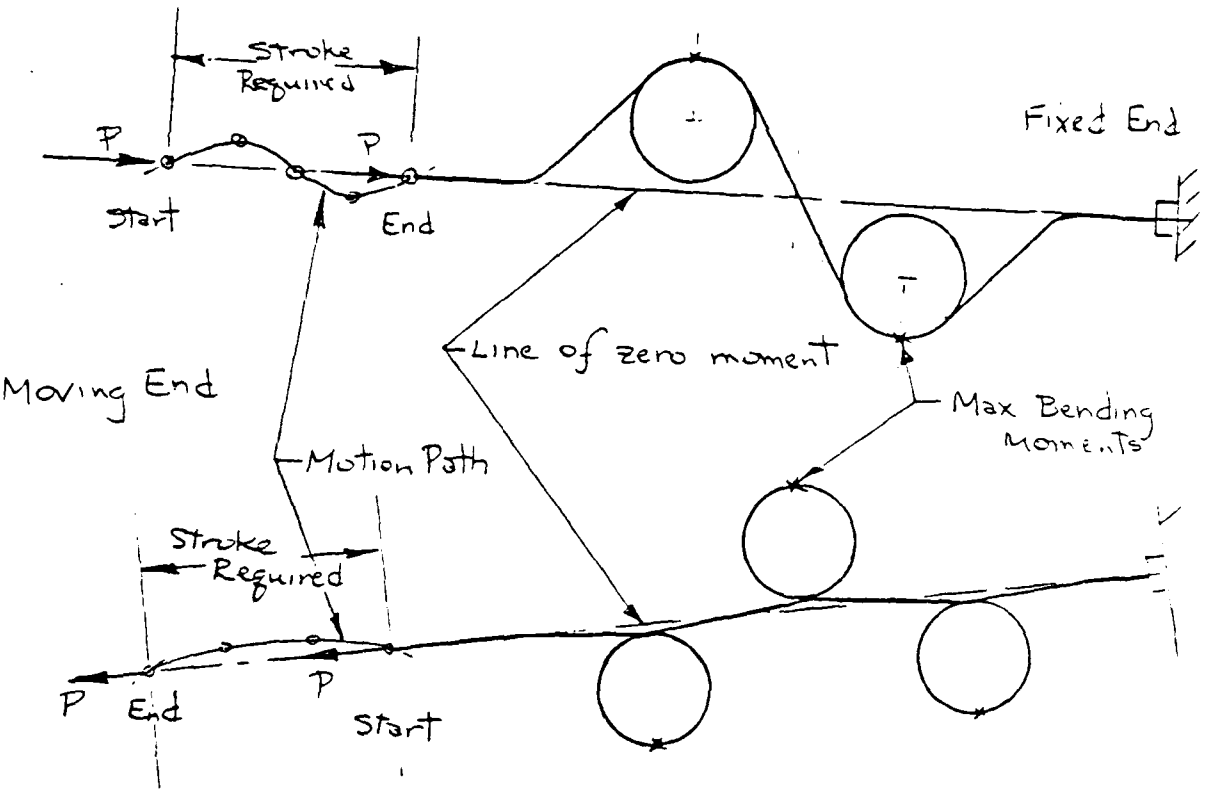


Figure 20. Bending configuration layout

necessary to achieve reasonable bending stresses with the required mechanical motion. Design to wind up coils (not unwind). To decrease the spring rate and bending stresses, increase the number of tube coils and/or coil diameter. Theoretical bending stresses are compared to allowable bending stresses to determine a satisfactory fatigue life. Additional revisions in the configuration may be required to produce a satisfactory fatigue life when internal pressure stresses, ovality stresses, and vibration stresses are considered.

6.3.4 Torsional Configuration

A tube shape operating principally in torsion has a long tube length (center section between supports) subjected to a torsional moment, Figure 18a. The straight section ends have bending stresses, and the curved sections contain stresses that vary from pure bending to pure torsion. Assume no fixity at the moving end and design a torsional configuration with a deflection of the moving end equal to the required mechanical motion while maintaining reasonable torsion stresses in the straight center tube and reasonable bending stresses in the end straights. Several trials may be required to keep stresses within allowable limits.

6.3.5 Coil Tube Design Examples

Eleven different coil tube configurations are depicted on Figure 21. Although other configurations are possible, the examples given cover most of the practical variations. Appendix D contains stress analyses of configurations 1, 2, and 3; a computer example is run for configuration 4.

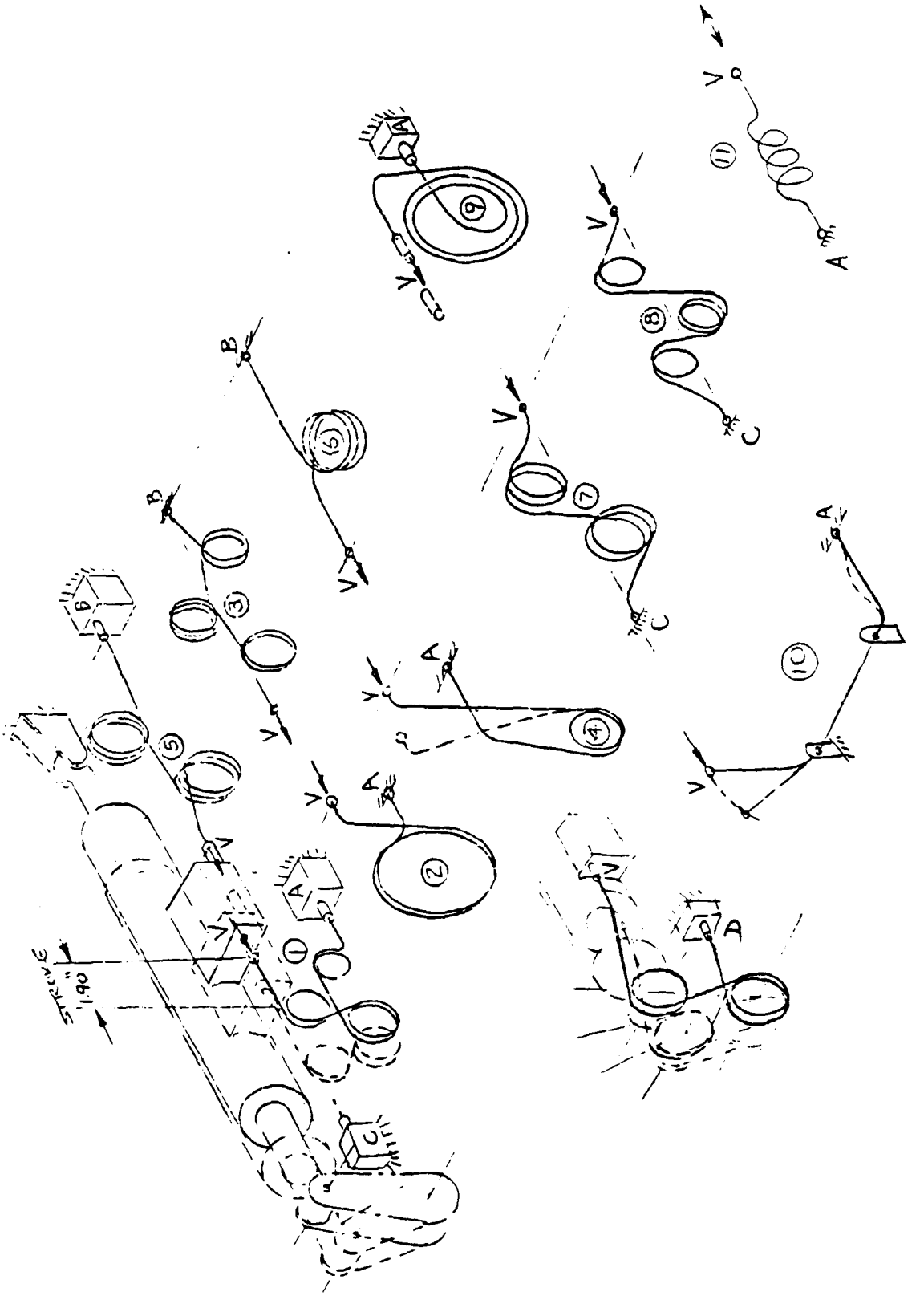


Figure 21. Coil tube configuration examples

7.0 BLACK RESIDUE INVESTIGATION

7.1 INTRODUCTION

A black residue was found on contamination patches made of return line filter debris collected during an exploratory LHS development program conducted in 1974, reference 6. The cause, source, or composition of the residue material were not established. The only immediate observable effect of the black particles was accelerated loading of filter elements. Effects of the residue on component performance and wear appeared to be negligible.

The residue continued to be a concern in subsequent LHS development programs. The residue was also found to occur in 3000 psi systems. Several attempts were made to identify the composition of the residue particles with inconclusive results.

The residue was discussed with personnel at Pall Corporation, Glen Cove, New York. Pall agreed to conduct an in-depth analysis of the particles using sophisticated test equipment. A two year study was conducted and documented in a comprehensive report. The complete APM report is contained in Appendix E. The results are summarized in the following section.

7.2 SUMMARY OF RESULTS

The "black residue" was determined to be primarily a composite of four particle types: A) organic black particles; B) aluminum black particles; c) iron black particles; and d) chromium black particles. Due to the difficulty in distinguishing one black particle from another, it was not practical to determine a percent distribution among the four particle types.

Organic particles were most numerous and made up roughly 50 to 75% of all particle types. Small inorganic particles were frequently imbedded in relatively larger organic particles forming an agglomerate. Organic and aluminum containing particles were found in greatest number on the pressure line filter. Pump case drain filters had a predominance of iron containing particles. Return line filters held a large number of chromium containing particles. Results of fluid contamination analyses showed that system cleanliness was NAS 1638 class 1 or better.

7.3 ROCKWELL COMMENTS

The black residue was the result of wear processes. Such wear typically occurs in all hydraulic systems that operate for extended periods of time. Operating pressure level could affect the rate of generation of wear particles. For example, higher wear rates probably occur in rod seals operating at higher pressure levels due to increased friction loads. The residue particles had no effect on fluid contamination; system filters maintained a fluid cleanliness level better than required (class 1 maintained, class 8 allowed).

8.0 HYBRID PUMP DEVELOPMENT

8.1 INTRODUCTION

Conventional aircraft pumps have good performance records, but are a major source of heat generation. Application of the same pressure compensated, axial piston, rotating barrel design to 8000 psi systems was anticipated to require heat exchangers to remove the additional heat generated. Further, the high internal loading in these designs operating at 8000 psi could cause extra wear and shorten pump life.

A new pump concept developed by MV Systems, Inc. addressed the needs of the LHS program, the shortcomings of the axial piston pump, and the advantages of check valve pumps. Pertinent features of the proposed 8000 psi hybrid pump were:

Design Feature	Advantages
Non-rotating cylinder block	No centrifugal force on pistons. Lower internal leakage, since rotating barrel lift-off cannot occur.
Low compensator actuation control force	Eliminates pintle bearing wear.
Piston shoe/rotating cam interface	Low hydrostatic leakage at no flow since pistons see no pressure.
Pump cooling	Low pressure inlet fluid used.

An unsolicited proposal for development of the hybrid pump was submitted to NAAO-Columbus by MV Systems, Inc. in August 1983. NADC agreed to participate in funding development of the pump and added this task to the LHS program. The pump concept and overall design were the responsibility of MV Systems, Inc. Pump detail design, fabrication, assembly, and development testing were to be performed by Hydrodyne (a division of F.P.I., Inc.). Two pumps were to be delivered. The program was scheduled for completion in July of 1985.

8.2 DISCUSSION

Fabrication and assembly of the hybrid pump was slowed by the need for Hydrodyne to sub-contract the manufacture of several components to outside shops. Since Hydrodyne had no test facilities, pump performance data were acquired in a Lockheed test laboratory approximately 40 miles from Hydrodyne. As a result, fabrication and testing of the pumps required more time than was anticipated. Several pump failures occurred during development testing that further delayed the program.

Rockwell and NADC personnel visited Hydrodyne in October of 1986 to review the status of pump development and observe pump operation at the Lockheed facility. The pump was then brought to NAAO-Columbus for evaluation testing. Although some aspects of pump performance were found to be satisfactory, it was apparent that a great deal of further development effort was required to achieve an acceptable unit.

MV Systems, Inc. severed their relationship with Hydrodyne in July of 1986. The current status of hybrid pump development is such that, in the best interests of all concerned, the development program was terminated.

9.0 TUBE FITTING RE-DESIGN

9.1 INTRODUCTION

An expander tool assembly consisting of a cage, three rollers, and a tapered mandrel is used to internally swage and attach Dynatube fittings to tubing. Each tube size has an individual swaging tool size. The device performs well for tube O.D.'s larger than 3/16 in. The expander had a very short life when used to swage 3/16 x .020 in. 3Al-2.5V titanium tubing. This made fabrication of -3 size tube assemblies for the LHS simulator troublesome.

The 0.147 in. tube I.D. limited the O.D. of the expander and prevented the use of adequate size rollers and mandrels. As a result, swaging stresses were excessive. Rollers usually failed after one or two swage operations; mandrels lasted for approximately 10 swages. Care was taken to assure that particles worn from the tubing, rollers, and mandrels were removed between swaging operations. All swaging was done by hand at a moderate rate; motor driven swaging was not used. Recommended lubricants were employed. The time required to continually refurbish the expanders and the cost of spare parts was a concern of both Rockwell and Crane Resistoflex. Fittings swaged by the -3 tooling performed well, however. Crane Resistoflex studied the problem and tried several approaches to find an answer. The following section discusses their efforts.

9.2 SUPPLIER EFFORTS

The basic problem was due to the necessary small physical size of expander components and the high yield strength of the tubing being swaged. Several different tool steels were evaluated in an attempt to improve the mandrel and roller materials; only minor benefits were attained. Various coatings and surface treatments were investigated; none offered any significant

improvement. A change in dimensional balance between the mandrel and rollers to achieve a better load distribution was tried. This approach increased expander life to approximately 20 swages, and caused (intentionally) the rollers to fail first. Visual inspection rather than disassembly could therefore be used to determine tool condition, and refurbishment costs would be lowered since rollers are less expensive than mandrels.

The most successful attempt involved re-design of the fitting tube receptable to reduce swaging forces and critical component loading. This approach, together with incorporation of minor improvements obtained in the preceding studies, produced a significant increase in tool life. Seven fitting receptacle designs were studied. The design selected for testing reduced tool loading by 47%. Swaging tests indicated tool life was now approximately 80 swages. The integrity of the swage joint was investigated by means of rotary flexure and presssure impulse tests. Results have been promising thus far, and further endurance cycling is planned in the near future.

9.3 ROCKWELL COMMENTS

The endurance characteristics of internally swaged fittings on the LHS simulator have been excellent. No catastrophic failures occurred, and only one swage area leak was observed during 1200 hours of cycling the LHS simulator. The need for improved -3 size tooling was evident, however, during fabrication of simulator plumbing. The need to solve the tooling problem was further encouraged by wider use of -3 size tubing in recently designed aircraft. The tooling/fitting modification proposed by Resistoflex appears to be satisfactory, but the integrity of the re-designed fitting in actual applications remains to be established.

REFERENCES

REFERENCE NO.

1 W. N. Bickel and R. K. Haning, Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware - Phase II, NA-85-0134, Rockwell International Corp., North American Aircraft Operations, Contract N62269-80-C-0261, January 1986. Unclassified. AD A-169 884

2 C. H. Cooke and R. D. Stouffer, Study of Coiled Tubing for Aircraft Hydraulic Systems, WADC-TR-55-121, Martin Marietta Corp., Contract AF 33/616/-2510, February 1955. Unclassified. AD 68 348

3 R. D. Stouffer, J. T. Hudson, and L. E. Freitag, Coiled Stainless Steel Tubing Design Manual, WADC TR-57-507, Martin Marietta Corp., Contract AF 33/616/-3464, December 1957. Unclassified. AD-142253

4 ARP 584, Coiled Tubing, Society of Automotive Engineers, March 1960.

5 K. E. Whitfill and R. B. Olsen, Development of Small Diameter Hydraulic Coiled Tubing Technology, NADC-79217-60, Vought Corporation, Contract N62269-80-C-0205, June 1981. Unclassified.

6 J. N. Demarchi and R. K. Haning, Lightweight Hydraulic System Hardware Endurance Test, NR75H-22, Columbus Aircraft Division, Rockwell International Corporation, Contract N62269-74-C-0511, March 1975, Unclassified. AD A-013 244

7 J. H. Stolpestad, Fatigue Design Properties Manual, NA72-1088, Rockwell International, June 1979.

LIST OF ABBREVIATIONS

AFCS	Automatic Flight Control System
BTU	British thermal unit
B-U	Back-up ring
cc	cubic centimeter
C/D	Center dam area
cpm	cycles per minute
db	decibel
°F	degrees Fahrenheit
FC-1	Flight control system No. 1
FC-2	Flight control system No. 2
ft	feet
gpm	gallons per minute
Hr	hour
Hz	cycles per second
I.D.	inside diameter
in.	inch
lb	pound
L.E.	Leading edge
LH	Left hand
LHS	Lightweight Hydraulic System
min.	minute (time) or minimum
MTBF	Mean-Time-Between-Failure
NAAO	North American Aircraft Operations
NADC	Naval Air Development Center

LIST OF ABBREVIATIONS (Cont'd)

No.	Number
O-R	O-ring
O/S	Outside
PEEK	Polyether etherketone
Δ P	Differential pressure
P/N	Part number
psi	pounds per square inch
RFI	Roll feel isolation
RH	Right hand
R&M	Reliability and Maintainability
RMS	Root-mean-square
sec	second (time)
S/N	Serial number
UHT	Unit horizontal tail
μ	micron

APPENDIX A

MISSION/PROFILE TEST LOG

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
9-27-85	600					Resume mission/profile cycling. Principal changes are: 1. Pumps have larger pintle bearings (Vickers M/N PV3-047-3) 2. Case drain filters now same as return filters 3. L.F. flap/speed brake actuator cycling rate increased
10-7-85	614	14 Hr.	FC-2	Pump	PV3-047-3	Pump developed high case flow and vibration. Pump sent to supplier for investigation. Install spare pump.
10-7-85	619	19 Hr.	FC-1	Pump R/H UHT Actuator	PV3-047-3 83-00211-102	Pump developed high case flow and vibration. Pump sent to supplier for investigation. R/H UHT actuator not operating with 2% inputs. Remove actuator. Cycling halted temporarily.
10-22-85	619					Resume mission/profile cycling using M/N PV3-047-2 pumps (FC-1 and FC-2 used during first 600 hours) and no R/H UHT actuator.
10-23-85	624					Install R/H UHT actuator. Control valve spool center land diameter reduced to eliminate sticking.
10-30-85	648	1,113,000	FC-1	Coil Tube (on RFI Actuator)	None	Tube leaking. Replace with Rockwell design coil tube.
11-6-85	664	352 Hr.	FC-1 & FC-2	Shear Pin (R/H UHT Module)	None	Pin failed. Continue cycling without R/H UHT actuator.
11-12-85	682		FC-2			Remove FC-2 pump M/N PV3-047-2. Install FC-2 pump M/N PV3-047-3
11-18-85	696	518,800	FC-1	Spoiler Actuator	83-00271	FC-1 cylinder port rosan fitting in control valve housing leaking. Replace O-Ring.

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
11-19-85	698					Install new shear pin in R/H UHT module. Resume cycling R/H UHT actuator.
12-2-85	726		FC-1			Remove FC-1 Pump M/N PV3-047-2. Install NADC pump M/N PV3-047-3 in FC-1.
12-10-85	742	21 Hr	FC-1	Pump	PV3-047-3	Pump has excessive wear (patch test). Remove NADC pump and send to supplier for investigation. Install spare M/N PV3-047-3 pump.
12-13-85	750					Shut down for component performance tests.
12-16-85	750	327,000	FC-1 & FC-2	Shear Pin (R/H UHT Module)	None	Shear pin failed. Failure discovered during component performance tests.
12-19-85	750		FC-1			Resume mission/profile cycling. R/H UHT actuator not being cycled. Spare pump M/N PV3-047-3 replaced with FC-1 pump M/N PV3-047-3.
1-8-86	774	1,519,650	FC-1	Control Valve (Seal Test Fixture)	50-489	Control valve malfunctioning. Find broken centering spring. Use valve without centering springs.
1-21-86	788					Install new shear pin in R/H UHT module. Resume cycling R/H UHT actuator.
1-31-86	818	818 Hr	Load	Reducer Bushing (Hyd. Power Supply)		Bushing failed. Replace with new bushing.
2-3-86	822	1,163,000	Load	Piston Rod (R/H UHT Module)		Load cylinder piston rod broke. Continue cycling without R/H UHT actuator.
2-6-86	830	12 Hr	Load	Reducer Bushing (Hyd. Power Supply)		Bushing cracked. Replace with new bushing. Re-plumb pressure line to reduce vibration level.

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
2-11-86	838	8950	FC-2	4-way solenoid valve (LE Flap System)	306700	Seal Failure. Send valve to supplier for investigation. Continue cycling without L.E. flaps.
2-11-86	840	193 Hr	FC-2	Pump	PV3-047-3	Excessive case drain leakage. Remove FC-2 pump M/N PV3-047-3 and send to supplier for investigation. Install spare pump M/N PV3-047-3
2-13-86	844					Cycling halted temporarily while rotary actuator load module installed on R/H wing leading edge.
3-3-86	844					Complete rotary actuator check-out tests. Install repaired 4-way solenoid valve in L.E. flap system. Resume mission/profile cycling.
3-4-86	845	1 Hr	FC-2	Control Valve (Rotary Actuator)	1056-C-001	Valve malfunctioning due to electronic problem. Continue cycling without actuator.
3-6-86	850	13,237	FC-1	Speed Brake Actuator	83-00201	Catastrophic failure. Actuator barrel split end-to-end. Continue cycling without speed brake actuator.
3-10-86	853	1019 Hr	FC-1	Reservoir	83-00241	Nuisance low pressure leak. Replace large seal on reservoir piston. Resume cycling.
3-11-86	858	632,723	FC-1 & FC-2	Spoiler Hinge (Load Module)		Outboard hinge failed. Continue cycling without spoiler actuator.
3-24-86	898	3,993,000	FC-1 & FC-2	Bearing (L/H UHT Module)	YD-129	Bearing in load module bellcrank failed. Continue cycling without L/H UHT actuator.
3-24-86	898	3,029,000	FC-1 & FC-2	RFI Actuator	83-00251	Leak out hole between rod seals in center dam area. Continue cycling.

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
3-24-86	900					Shut down for component performance tests and actuator seal inspections.
5-27-86	900	?	FC-2	1-way restrictor (L.E. Flap System)	95461-1	Failure discovered during performance checks. Time of failure not known. L.E. flap system appeared to function normally.
10-27-86	900					Install re-worked actuators: Aileron — New body, new piston and rod seals R/H UHT — Replate/grind piston rod, new piston seal RFI — Replate/grind piston rod, new piston and rod seals FC-2 Seal Test Fixture: New piston/piston rod and seals Spoiler Load Module: Replace failed hinge and bearings L/H UHT Load Module: Replace failed bearing
11-6-86	900	4,090,950	FC-1	Servo Valve	56E-201 S/N 1	Valve installed incorrectly on rudder AFCS actuator during actuator reassembly. 8000 psi applied to return port. Flexure tube failed. Install spare valve, S/N 2.
11-7-86	900					Resume mission/profile endurance cycling. Actuators not cycling are: L.E. Flap — GAR-Kenyon restrictors not yet received L.E. Rotary — Awaiting arrival of Fairey personnel for operational check-out. FC-1 Seal Test F. — Shamban seals not yet received. Speed Brake — Awaiting fabrication of new actuator
11-10-86	904	13,069 3.7 Hrs	FC-2 (Pressure)	Hose (@ Aileron Actuator)	None	New design Titeflex hose leaking. Install hose used previously at this location.
11-11-86	906	870,550 258 Hrs	FC-1 (Pressure)	Coil Tube (@ RFI Actuator)	None	Rockwell design coil tube leaking at tube/fitting interface. Replace tube with hose (interim fix).

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
11-13-86	918	911,400 270 Hrs	FC-1 (Return)	Coil Tube (@ RFI Actuator)	None	Rockwell design coil tube leaking in coil area. Replace with Rockwell made tri-coil tube (interim fix).
12-1-86	946					Install restrictors. L.E. flap actuators now operational. Install seals. FC-1 seal test fixture now operational.
12-8-86	954					Personnel from Fairey Hydraulics (England) check out rotary actuator drive electronics. Circuit problems encountered.
12-18-86	954					Resume mission/profile cycling using Rockwell built electronics to drive rotary actuator and swivel joints.
12-22-86	966					Receive new Dynex/Rivett pump for LHS ground cart. Begin pump installation in ground cart.
1-14-87	966	966 Hrs	FC-1 (SP.BR. Valve)	Check Valve	P1-858	Complete ground cart performance tests. Discover LHS check valve installed backwards due to incorrect marking.
1-15-87	968	0	FC-1 (SP.BR. Valve)	Actuator	Unknown	Install small actuator downstream of speed brake 4-way valve. Begin cycling 4-way valve.
1-22-87	984	405,348	-	Pin (R/H UHT Act'r)	None	Actuator base end support pin failed. Continue cycling without actuator while repairs are in work.
1-27-87	998	2000 cycles (Est.)	FC-2	Fitting	54040T-3	Fitting leaking on rudder AFCS 3-way valve return port. Replace fitting.
2-3-87	1010		FC-1 & FC-2	R/H UHT Act'r		Support pin and clevis repaired. Resume cycling actuator.

MISSION/PROFILE TEST LOG

DATE	SIMULATOR HOURS	COMPONENT CYCLES/HOURS	SYSTEM	COMPONENT	PART NO.	REMARKS
2-11-87	1036	12,600	FC-2	Fitting	MS21921-3	3000 psi MS fitting leaking (8000 psi line). Replace fitting.
2-17-87	1050					Shut down for component performance tests
2-25-87	1050	3,253,000	FC-2	Seal Test Fixture	4252-29	Crack in cylinder barrel discovered during piston seal leakage check. Remove fixture from simulator.
2-27-87	1050					Resume mission/profile endurance cycling.
3-11-87	1080	4,607,483	FC-2	L/H UHT Act'r	83-00211-101	Actuator piston lip cracked and piston seal failed. Resume cycling without L/H UHT actuator.
3-18-87	1096	4,721,331	FC-2	Rudder Act'r Servo Valve	215-62100-3	Pin hole leak near FC-2 cyl. #1 port. Housing is aluminum and designed for 3000 psi A-7 system. Install new housing.
3-24-87	1114	534,663	FC-2	Swivel Joint (Outboard)	L38910	Joint leaking 3 cc/min. Single stage seal designed for 20,000 cycle life. Replace cap seal with O-Ring. Resume cycling.
4-3-87	1144	15,322	FC-2	L.E. Flap Act'r	83-00261-101	End cap fractured in threaded section. Failure caused by loose lock nut P/N 83-00262-119. Need better method of securing nut. Resume cycling without inboard load module.
4-24-87	1200					Shut down for component performance tests and actuator seal inspections.

THIS PAGE LEFT BLANK INTENTIONALLY

APPENDIX B

ACTUATOR PISTON AND ROD SEALS

(NEW SEALS INSTALLED AT 900 HRS)

ACTUATOR	TYPE	SYSTEM	LOCATION	STAGE	RING	PART NUMBER	SUPPLIER
RFI	Piston	FC-1 & 2	--		T-S	7116MT-160-4750	Greene, Tweed
	Rod	FC-1 & 2	C/D	1st	+S B-U	S30775-116P-19 S35393-116	Shamban Shamban
				2nd	H-S	S34832-116P-19	Shamban
L/H Aileron	Piston	FC-1 FC-2			CGT CGT	266-11600-964-1180 266-21000-964-1180	Greene, Tweed Greene, Tweed
	Rod	FC-2	O/S	1st	CGT	265-21000-964-1180	Greene, Tweed
				2nd	RSR/W	235-21050-964-0300	Greene, Tweed
			C/D	1st & 2nd	EHP	5979A11300R002	Greene, Tweed
Seal Test Fixture	Piston	FC-1	---	---	+S PEEK B-U	S30772-3044 S36326-330-19-552	Shamban
	Rod	FC-1	Inbd	1st	+S PEEK B-U	S30775-218P-19 S36325-218-19-552	Shamban
				2nd	+S B-U	S30775-218P-19 S33157-218-19	Shamban Shamban
			Outbd	1st	+S PEEK B-U	S30775-218P-19 S36325-218-19-552	Shamban
				2nd	H-S	S33353-218P-19	Shamban
	Piston	FC-2	---	---	CGT	226-33000-964-1200	Greene, Tweed
	Rod	FC-2	Inbd & Outbd	1st	CGTL	265-21800-964-1200	Greene, Tweed
				2nd	TRAP	4635-21800H-964	Greene, Tweed

APPENDIX C

EVALUATION OF PISTON SEALS

FOR

LIGHTWEIGHT HYDRAULIC SYSTEMS

(LTV REPORT 2-51700/5R-61)

NADC-89006-60

Report 2-51700/5R-61
Final Report
27 November 1985

EVALUATION OF PISTON SEALS FOR LIGHTWEIGHT HYDRAULIC SYSTEMS

Submitted to:
Rockwell International
North American Aircraft Operations
4300 E. Fifth Avenue
Columbus, Ohio 43216

As Required by:

P. O. Number H4FC-089014-475

Prepared by:

Robert B. Olsen
Technical Project Manager

LTV Aerospace and Defense
Vought Aero Products Division

TABLE OF CONTENTS

EVALUATION OF PISTON SEALS
FOR
LIGHTWEIGHT HYDRAULIC SYSTEMS

SECTION		PAGE
1.	INTRODUCTION	1
2.	DISCUSSION OF PROBLEM	2
3.	SELECTION OF SEAL SYSTEMS	4
	1. METHOD OF DETERMINING CANDIDATES . . .	4
	2. DESIGN AND TEST CONDITIONS	4
	3. DESCRIPTION OF CANDIDATE SEAL SYSTEMS	4
	4. SEAL SYSTEM SELECTION METHOD	13
	5. CANDIDATES SELECTED FOR TEST	16
	6. ROD SEAL EVALUATIONS	17
4.	TEST PROCEDURE	20
5.	ANALYSIS OF TEST RESULTS	24
	1. PISTON SEAL TEST RESULTS	24
	2. ROD SEAL TEST RESULTS	~~31~~ 32
6.	CONCLUSIONS	~~36~~ 37

LIST OF ILLUSTRATIONS

FIGURE	TITLE	PAGE
1	PISTON SEAL CANDIDATES P1, P2, P3 and P4 . . .	6
2	PISTON SEAL CANDIDATES P5, P6, P7, and P8 . .	8
3	PISTON SEAL CANDIDATES P9, P10, P11 and P12 .	10
4	PISTON SEAL CANDIDATES P13, P14	12
5	BASELINE ROD SEAL SYSTEM TESTED	18
6	ALTERNATE ROD SEAL SYSTEM TESTED	19
7	TEST SYSTEM HYDRAULIC SCHEMATIC	22

LIST OF TABLES

TABLE	TITLE	PAGE
1	SEAL SYSTEM RATING DECISION MATRIX	15
2	SUMMARY OF PISTON SEAL TEST RESULTS	25
3	RECORD OF PISTON SEAL LEAKAGE (AMBIENT) . . .	26
4	RECORD OF PISTON SEAL LEAKAGE (-40°F)	27
5	BREAK-OUT PRESSURE TEST RESULTS	~~28~~ 29
6	SUMMARY OF ROD SEAL TEST RESULTS	~~32~~ 33

1.0 INTRODUCTION

Selection of seal systems for Lightweight Hydraulic Systems (LHS) is a critical task since complaints of system leakage constitute a very high proportion of reports of failures of 3000 and 4000 PSI systems in service. As LHS technology progresses into production designs, systems users will expect life and reliability of LHS components to equal or exceed that of comparable 3000 PSI components. In order to meet these expectations and to continue the trend of orderly development of LHS technology, this program of test and evaluation of piston seal systems was conducted.

The major goals of this program were:

(1) Evaluate wear and leakage of several experimental and state of the art piston seal systems.

(2) Make some assessment on total diametral clearance allowable before seal performance is affected.

(3) As a secondary evaluation, 19 sets of two stage rod seal systems were installed for monitoring of leakage and wear at relatively high diametral clearances.

The primary test conditions for this program were:

Pressure:	8000 PSIG
Fluid:	MIL-H-83282
Temperature:	-40 to +275 DEG F

The author wishes to express appreciation to the following firms who supplied the test candidates and were very instrumental in supporting the program with samples and technical support.

W. S. Shamban Co.
C. E. Conover Co.
Greene, Tweed and Co.
Tetrafluor Co.

2.0 DISCUSSION OF PROBLEM

As operating pressure increases, seals are greatly affected by diametral clearance or any characteristic of the seal gland that makes the diametral clearance look greater than it really is. Gland characteristics which make the diametral clearance seem larger are slope of gland wall and the radius or edge break on groove edges. Reduction of actual diametral clearance improves life of seals at 8000 PSI. Assuming these gland modifications are made, the seal materials are critical in achieving a long seal life. First, the elastomer must be compatible with the fluid and the temperature range predicted for the application. The elastomer must have some small percent swell to compensate for contraction at low temperatures. Second, plastic materials used for anti-extrusion devices must be a compromise between hardness to give long wear and some degree of cold flow to conform to cylinder wall and gland dimensions. Under pressure, backup rings used with o-rings on rod seals are continually cold flowing to compensate for material wear at the rod. As this occurs, the backup ring gets thinner and is reduced in cross-section. A very soft material will cold flow in this manner until the backup is completely worn through. With high radial loading even metal backups will wear through.

One of the most difficult problems to be overcome in piston seal applications is expansion of the cylinder wall when pressurized with a resultant increase in diametral clearance. The following example is provided to illustrate the problem. Assume that an actuator has an aluminum housing and a steel housing each with 1.493 inch bore and the piston is steel. MIL-G-5514F gland dimensions which give maximum diametral clearance are used in the example. A comparison of the clearances obtained with the two materials at 8000 psi hydraulic pressure and 275 degrees F is given below.

The 70 DEG F dimensions for the example are:

Cylinder bore.....1.493

Piston diameter.....1.488

Cylinder wall thickness..... .60 aluminum

..... .17 steel

Temperature -- deg F	Cylinder Material	Piston Material	Pressure -- psi	Max Diametral Clearance
70	Aluminum	Steel	0	0.0050
70	Aluminum	Steel	8000	0.0074
275	Aluminum	Steel	8000	0.0095
70	Steel	Steel	0	0.0050
70	Steel	Steel	8000	0.0068
275	Steel	Steel	8000	0.0068

With the aluminum cylinder, diametral clearance increase is 48 percent due to pressure and 90 percent due to pressure and temperature. With the steel cylinder, diametral clearance increase is 36 percent due to pressure, no increase is experienced due to temperature. Even if dimensions of the steel cylinder were changed to reduce clearances to 0.003 at 70 deg F, the growth due to pressure would be about the same. Increasing the wall thickness of the steel cylinder from .17 to .41 to reduce the increase in diametral clearance still results in a growth of 0.001 at 8000 PSI.

The conclusion is that because of the weight penalty, it is impractical to design cylinders with extremely thick walls to limit growth in diametral clearance. The most probable solution will be a combination of wall thickness design to limit diametral clearance to a recommended value and the use of piston seal systems which have demonstrated satisfactory life with reasonably achievable diametral clearances.

3.0 SELECTION OF SEAL SYSTEMS

3.1 Method of Determining Candidates

Letters and telephone calls were used to contact the W. S. Shamban Co., Greene, Tweed and Co., and Tetrafluor, Inc. The requirements for this program were given, and each company was requested to submit recommendations and sample seals for evaluation. Also, a heavy duty trapezoid seal, designed by G. K. Fling of LTV Aerospace and Defense Co. and fabricated by C. E. Conover and Co. was included in the seal selection process. As a result of this procedure, fourteen piston seal candidates were identified.

3.2 Design and Test Conditions

The following design and test conditions were defined for seal systems considered for selection:

Pressure:	8000 psi
Fluid:	MIL-H-83282
Temperature:	-65 to +275°F, however lowest test temperature will be -40°F due to use of MIL-H-83282 fluid.
Test Duration:	400 hours at 250 to 275°F (360 hours at +/- 1.75 in. stroke and 40 hours at +/- 0.10 in. stroke)
Bore:	1.490 + .001/-.000 16 RMS finish
Piston OD:	1.489 + .000/-.001
Groove OD:	1.248 + .000/-.001 for standard gland 1.057 + .000/-.001 for piston ring gland
Groove Width:	.304 + .010/-.000 for standard gland .126 +/-.001 for piston ring gland

3.3 Description of Candidate Seal Systems

Candidate P1; 266-21800-964-1200; Greene, Tweed & Co.

Material: Cap Seal - Graphite filled TFE: Greene, Tweed material P4

Backup Rings - Nylatron

Elastomer - Greene, Tweed nitrile compound; material code 964.

Characteristics: Seal consists of a plastic cap seal energized by a "T" shaped elastomer. The cap seal has a scarf-cut backup ring installed on each side. The backup rings and cap seal are loaded into the sealing surface by deformation of the elastomer when pressurized.

Configuration: See Figure 1

Candidate P2; S30772-218P19, S35117; W. S. Shamban Co.

Material: Cap Seal - Modified TFE with proprietary filler; Material Code TURCON 19

Backup Rings:
Outer - unfilled polyimide; material code AEROLON 22.
Inner - TFE with inorganic filler, material code AEROLON 20

Elastomer - Nitrile per MIL-P-83461

Characteristics: Plastic cap seal energized by special shape elastomer. The cap seal has scarf-cut backup rings installed on each side. The backup rings are a "piggy back" design with the outer backup installed on the circumference of the inner backup ring. The outer backup has an axial cut, the inner backup, a radial cut.

Configuration: See Figure 1.

Candidate P3; S30772-218P19, S33157-218-19; W. S. Shamban Co.

Material: Cap Seal - Modified TFE with proprietary filler; Material Code TURCON 19

Backup Rings - TURCON 19.

Elastomer - Nitrile per MIL-P-83461

Characteristics: Plastic cap seal energized by special shape elastomer. The cap seal has non-cut nominal .05 width backup rings installed on each side.

Configuration: See Figure 1.

Candidate P4; S34232-218P19; W. S. Shamban and Co.

Material: Seal - Nitrile elastomer per MIL-P-83461

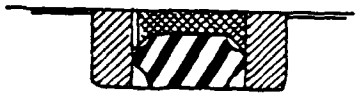
Backup Rings - modified TFE with proprietary filler; Material code TURCON 19.

Characteristics: Assembly consisting of special shape elastomer and backup rings. The seal has a backup ring on each side. The backup ring has an angled face toward the elastomer which reduces the radial loading of the elastomer and loads the backup into the cylinder wall when pressurized.

Configuration: See Figure 1.

Candidate P1
266-21800-964-1200
"CGT" Seal
Greene, Tweed and Co.

Candidate P2
S30772-218P19, S35117
"Plus Seal II", interlocking backup ring set
W. S. Shamban and Co.



Candidate P3
S30772-218P19, S33157-218-19
"Plus Seal II", non cut backup rings
W. S. Shamban and Co.

Candidate P4
S34232-218P19
"Wedgpak" set
W. S. Shamban and Co.

Figure 1. Piston Seal Candidates P1, P2, P3 and P4

Candidate P5; S33824-218P19; W. S. Shamban and Co.

Material: Seal - Nitrile elastomer per MIL-P-83461

Backup Rings - modified TFE with proprietary filler; Material Code TURCON 19.

Characteristics: Assembly consisting of o-ring and special shape backup rings. The seal has a backup ring on each side. The backup ring has an angled face toward the elastomer which reduces the radial loading of the elastomer and loads the backup into the cylinder wall when pressurized.

Configuration: See Figure 2.

Candidate P6; CEC5223-218; C. E. Conover and Co. Inc.

Material: Seal - Nitrile per MIL-P-83461

Backup Rings - Proprietary TFE blend; Material Code REVONOC 6200.

Characteristics; Elastomer seal of special shape with very thick cross-section special shape backup rings. The seal has a backup ring installed on each side. The backup ring has an angled face toward the seal which reduces the radial loading of the elastomer and loads the backup into the cylinder wall when pressurized. Requires wider gland than standard.

Configuration: See Figure 2.

Candidate P7; 7218MT-972-4780; Greene, Tweed and Co.

Material: Seal - Nitrile treated to reduce friction and wear; Material Code 972

Backup Rings - outer, nylon; inner TFE: material code 4780.

Characteristics: Special "T" shaped elastomer seal loads two stage backup rings into cylinder wall when pressurized. Seal has scarf-cut backup rings on each side. Backup rings are two stage with nylon outer backup and TFE inner backup.

Configuration: See Figure 2.

Candidate P8; TF500A-7218(490); Tetrafluor Co.

Material: Reinforced fluoropolymer

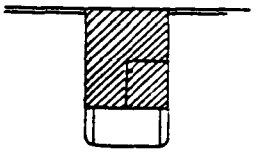
Characteristics: One piece piston ring with maraging steel expander.

Configuration: See Figure 2.

Candidate P5
S33824-218P19
"Delta" backup with M83461/1-218
o-ring
W. S. Shamban and Co.

Candidate P6
CEC 5223-218
Dual Trapezoid Piston Seal
C. E. Conover and Co.

Candidate P7
TF500A-7218(490)
"AGT" ring with two stage backup
rings
Greene, Tweed and Co.



Candidate P8
TF 500A-7218(490)
Piston ring assembly
Tetrafluor Co.

Figure 2. Piston Seal Candidates P5, P6, P7 and P8

Candidate P9; S35119, S30772-218P19, S33157-218-19; W. S. Shamban and Company

Material: Piston ring - Polyetheretherketone (PEEK) with CRES steel expander cap

Cap Seal - modified TFE with proprietary filler;
Material Code TURCON 19

Backup Ring - TURCON 19
Elastomer - Nitrile per MIL-P-83461

Characteristics: Seal system consists of dual piston rings installed on each side of a cap seal with heavy duty backup rings installed on each side.

Configuration: See Figure 3.

Candidate P10; S35119, S34232-218P19; W. S. Shamban and Co.

Material: Piston ring - Polyetheretherketone (PEEK) with CRES steel expander.

Backup Ring - Modified TFE with proprietary filler;
Material Code TURCON 19.

Elastomer - Nitrile per MIL-P-83461

Characteristics: Seal system consists of dual piston rings installed on each side of a special shape elastomer seal with special shape backup rings installed on each side.

Configuration: See Figure 3.

Candidate P11; M83461/1-218, MS27595-218; Military Standard

Material: Seal - Nitrile elastomer per MIL-P-83461
Backup Ring - unfilled TFE.

Characteristics: O-ring seal with noncut standard backup ring on each side.

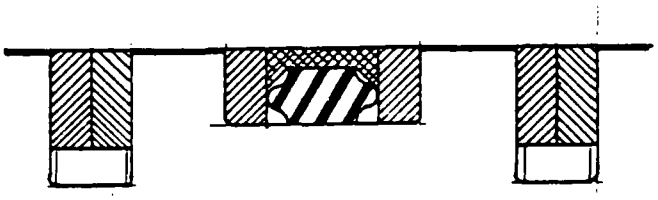
Configuration; See Figure 3.

Candidate P12; TF1146-218; Tetrafluor Co.

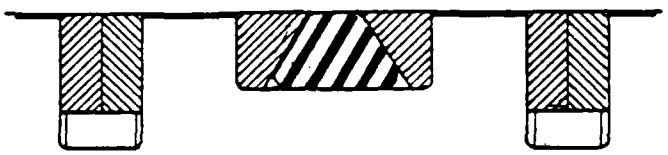
Material: Cap Seal - Reinforced fluoropolymer
Elastomer - Nitrile per MIL-P-83461
Backup Rings - Reinforced fluoropolymer

Characteristics: Thick cross-section cap seal energized by square or round cross-section elastomer. The seal has a heavy duty thickness scarf cut backup ring installed on each side. The backup rings have anti-blow by notches on both sides of the backup.

Configuration: See Figure 3.



Candidate P9
S35119, S30772-218P19
Three Stage Piston Seal
W. S. Shamban and Co.



Candidate P10
S35119, S34232-218P19
Three Stage Piston Seal
W. S. Shamban and Co.

Candidate P11
M83461/1-218, MS27595-218
O-ring and non cut backup rings
Military Standard

Candidate P12
TF1146-218
Cap seal, rectangular shape or
o-ring elastomer, heavy duty backup
rings, anti-blowby notches in backup
rings
Tetrafluor Co.

Figure 3. Piston Seal Candidates P9, P10, P11 and P12

Candidate P13, S30772-218P19N, 33157-218-19, W.S. Shamban

Material: Cap Seal - Modified TFE with proprietary filler;
Material Code TURCON 19
Backup Rings - TURCON 19
Elastomer - Nitrile per MIL-P-83461

Characteristics: Plastic cap seal with anti-blow by notches energized by special shape elastomer. The cap seal has a non-cut backup ring installed on each side.

Configuration: See Figure 4

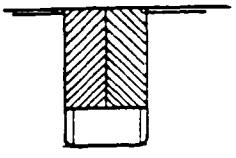
Candidate P14; S35119, W,S. Shamban

Material: Polyetheretherketone (PEEK) with 17-4PH stainless steel expander

Characteristics: Dual piston rings indexed by tang on expander to maintain 180 degree separation of cuts in rings.

Configuration: See Figure 4

Candidate P13
S30772-218P19N, S33157-218-19
"Plus Seal II", Noncut backup rings
Anti-blowby notches in cap seal
W. S. Shamban and Co.



Candidate P14
S35119
Dual Piston Ring
W. S. Shamban and Co.

Figure 4. Piston Seal Candidates P13, P14

3.4 Seal System Selection Method

A decision matrix was used to establish a relative rating of the 14 candidate systems considered. The decision matrix rated each seal system against 15 criteria. The criteria were weighted to place highest importance on sealing and suitability for pressure level. Next in importance were seal wear and complexity. The next lower importance was placed on friction, temperature compatibility, extrusion gap design, installation difficulty and seal fatigue. Finally, the lowest importance was placed upon abrasiveness of seal materials, possibility of a pressure trap within the seal system, space required, possibility of wrong installation, compatibility with fluid, and relative cost to produce. These 15 criteria and weighting factors are identified on the Seal Rating Decision Matrix as:

Criteria	Weighting Factor	Criteria	Weighting Factor
Sealing	0.10	Installation	0.08
Seal Wear	0.09	Space	0.04
Abrasion	0.05	Seal Deflection	0.08
Friction	0.08	Orientation	0.04
Pressure Trap	0.05	Complexity	0.09
Temperature	0.08	Compatibility	0.04
Extrusion Gap	0.08	Producibility	0.04
		Pressure Level	0.10

For the decision matrix, each candidate was evaluated against each criteria. If the candidate was excellent, above average, or good, a rating factor of 10 was given. If the candidate was average a rating factor of 6 was given. If the candidate was poor or fair a rating factor of 2 was given. The sum of the products of weighting factor and rating factor is the selection score for each seal system.

The seal system with the highest score was the top candidate. The rest of the candidates were ranked in descending order according to selection score. The Seal Rating Decision Matrix is shown on Table 1. The results of the seal Selection Matrix place the candidates in the following order:

RATING	CANDIDATE	DESCRIPTION
1	P8	One piece piston ring from Tetrafluor
2	P14	Dual piston rings from W.S. Shamban
3	P6	Heavy Duty Dual "Trapezoid" Piston Seal from C. E. Conover
4/5	P4	"Wedgpak" from W. S. Shamban
4/5	P2	"Plus" seal with piggy back backups from W. S. Shamban
6	P1	"CGT" seal from Greene, Tweed
7	P5	"Delta" backups and o-ring from W. S. Shamban
8/9/10	P10	Two stage seal of piston rings and o-ring/"Delta" backups from W. S. Shamban
8/9/10	P3	"Plus" seal with heavy duty backups from W. S. Shamban
8/9/10	P13	Same as P3 except with anti-blow by notches
11	P12	Capseal with "square" elastomer energizes and scarf cut heavy duty backups
12	P11	O-ring with noncut backups - MS standard
13	P9	Two stage of piston rings and "Plus" seal with noncut backups from W. S. Shamban
14	P7	"AGT" seal with two stage backups from Greene, Tweed

SEAL SYSTEM RATING DECISION MATRIX

CRITERIA	SEALING	SEAL WEAR	ABRASION	FRICTION	PRESSURE TRAP	TEMPERATURE	EXTRUSION	INSTALLATION	SPACE	SEAL DEFLECTION	ORIENTATION	COMPLEXITY	COMPATIBILITY	PRODUCIBILITY	PRESSURE LEVEL	SUM OF RF*WF	SEAL SYSTEM RATING
WF	0.10	0.09	0.05	0.06	0.05	0.06	0.06	0.06	0.04	0.06	0.04	0.09	0.04	0.04	0.10		
CANDIDATE																	
P1	6 0.60	10 0.00	6 0.30	10 0.75	10 0.50	6 0.45	6 0.45	6 0.45	6 0.23	6 0.45	2 0.00	2 0.18	6 0.23	6 0.23	10 1.00	6.75	6
P2	6 0.60	10 0.00	6 0.30	10 0.75	10 0.50	6 0.45	10 0.75	6 0.45	6 0.23	6 0.45	10 0.30	2 0.18	6 0.23	2 0.08	6 0.60	6.00	4/5
P3	6 0.60	10 0.00	6 0.30	10 0.75	2 0.10	6 0.45	6 0.45	2 0.15	6 0.23	6 0.45	10 0.30	2 0.18	6 0.23	6 0.23	10 1.00	6.35	8/9/10
P4	10 1.00	6 0.53	6 0.30	6 0.45	2 0.10	6 0.45	10 0.75	6 0.45	6 0.23	6 0.45	10 0.30	6 0.53	6 0.23	10 0.30	6 0.60	6.00	4/5
P5	10 1.00	6 0.53	6 0.30	6 0.45	2 0.10	6 0.45	10 0.75	6 0.45	6 0.23	2 0.15	10 0.30	6 0.53	6 0.23	10 0.30	6 0.60	6.50	7
P6	10 1.00	6 0.53	10 0.50	2 0.15	2 0.10	6 0.45	10 0.75	6 0.45	2 0.08	6 0.45	10 0.30	6 0.53	6 0.23	10 0.30	10 1.00	6.95	3
P7	10 1.00	2 0.18	6 0.30	6 0.45	10 0.50	6 0.45	10 0.75	10 0.75	6 0.23	2 0.15	2 0.08	2 0.18	6 0.23	2 0.08	6 0.60	5.90	14
P8	2 0.20	10 0.00	6 0.30	10 0.75	10 0.50	10 0.75	10 0.75	10 0.75	10 0.38	6 0.45	10 0.30	10 0.08	10 0.38	10 0.38	10 1.00	8.70	1
P9	6 0.60	10 0.00	2 0.10	10 0.75	2 0.10	6 0.45	10 0.75	2 0.15	2 0.08	6 0.45	10 0.30	2 0.18	6 0.23	2 0.08	10 1.00	6.15	13
P10	10 1.00	6 0.53	2 0.10	6 0.45	2 0.10	6 0.45	10 0.75	6 0.45	2 0.08	6 0.45	10 0.30	2 0.18	6 0.23	6 0.23	10 1.00	6.35	8/9/10
P11	10 1.00	2 0.18	10 0.50	6 0.45	2 0.10	6 0.45	6 0.45	6 0.45	6 0.23	10 0.75	10 0.30	6 0.53	6 0.23	10 0.30	2 0.20	6.25	12
P12	6 0.60	6 0.53	6 0.30	10 0.75	10 0.50	6 0.45	2 0.15	6 0.45	6 0.23	10 0.75	10 0.30	2 0.18	6 0.23	6 0.23	6 0.60	6.30	11
P13	6 0.60	10 0.00	6 0.30	10 0.75	2 0.10	6 0.45	6 0.45	2 0.15	6 0.23	6 0.45	10 0.30	2 0.18	6 0.23	6 0.23	10 1.00	6.35	8/9/10
P14	2 0.20	10 0.00	10 0.50	10 0.75	10 0.50	10 0.75	10 0.75	10 0.75	10 0.38	6 0.45	10 0.30	2 0.18	10 0.38	10 0.38	10 1.00	8.20	2

EXPLANATION OF AN ELEMENT: RF

RF*WF

TABLE 1. SEAL SYSTEM RATING DECISION MATRIX

3.5 Candidates Selected for Test

There were six test cylinders used to evaluate piston seals. Beginning with the first six candidates selected, the testing continued until a total of 400 hours of dynamic cycling was completed. If any candidate failed as indicated by leakage at a rate too high to measure, or clearly much greater than the nominal leakage measured at the beginning of the test, that candidate was replaced with one of the alternate seals recommended. Seals which began the test were selected based upon this rating in the selection procedure except as noted.

Seals tested first were:

CANDIDATE	COMMENT
P8	Rated No. 1 in selection procedure
P6	Rated No. 3 in selection procedure
P2	Rated No. 4/5 in selection procedure
P4	Rated No. 4/5 in selection procedure
P1	Rated No. 6 in selection procedure
P3	Rated No. 8/9/10 in selection procedure, but will be tested as a baseline since it is the current piston seal being used in the LHS program.

As failures occurred, the following alternate candidates were tested until the 400 hours of test time was accumulated.

CANDIDATE	COMMENT
P14	Rated No. 2 in selection procedure.
P12	Rated No. 11 in selection procedure. Has anti-blow by notches in backups
P13	Anti blow-by notches added to cap seal. Tied for 8/9/10 rating.

3.6 Rod Seal Evaluations

The rod seals were not candidates in this program, but leakage and wear were maintained to provide additional data.

The test cylinders used two different two stage unvented rod seals systems. One of the systems, identified as the baseline system, was the same design and configuration as currently being used in the power control actuators in the Rockwell LHS Endurance test. However, these seals were fabricated using materials for the backup rings and cap seals which may offer better performance than those previously tested for this rod seal system.

Inboard Rod Seal (Figure 6)

S30650-214-19 "Double Delta II" Cap Strip Seal

S33157-214-19 Heavy Duty Backup Ring

M83461/1-214 O-Ring

The material previously used for the cap string seal and the backup rings was Shamban TURCON 14, a modified TFE with fiberglass and molybdenum disulfide filler. The material tested was TURCON 19 which is a modified TFE with proprietary filler which is less abrasive than TURCON 14.

Outboard Rod Seal (Figure 6)

CEC5057C-214NC M83461/1-214 o-ring with two stage noncut backup rings of C. E. Conover material REVONOC 6200.

The material previously used for the backup rings was REVONOC 18158. The material tested was REVONOC 6200, a TFE blend which offers greater resistance to extrusion but is not abrasive.

An alternate rod seal system was tested for comparison with the baseline system. It consisted of identical "Trapezoid" rod seals using the Greene Tweed Code 964 nitrile elastomer, and Code P5 backup rings.

Inboard Rod Seal (Figure 7)

4635-21400D964 "Trapezoid" rod seal; Greene, Tweed & Co

Outboard Rod Seal (Figure 7)

4635-21400D964 "Trapezoid" rod seal; Greene, Tweed & Co.

The rod seals were monitored for leakage and wear throughout the test.

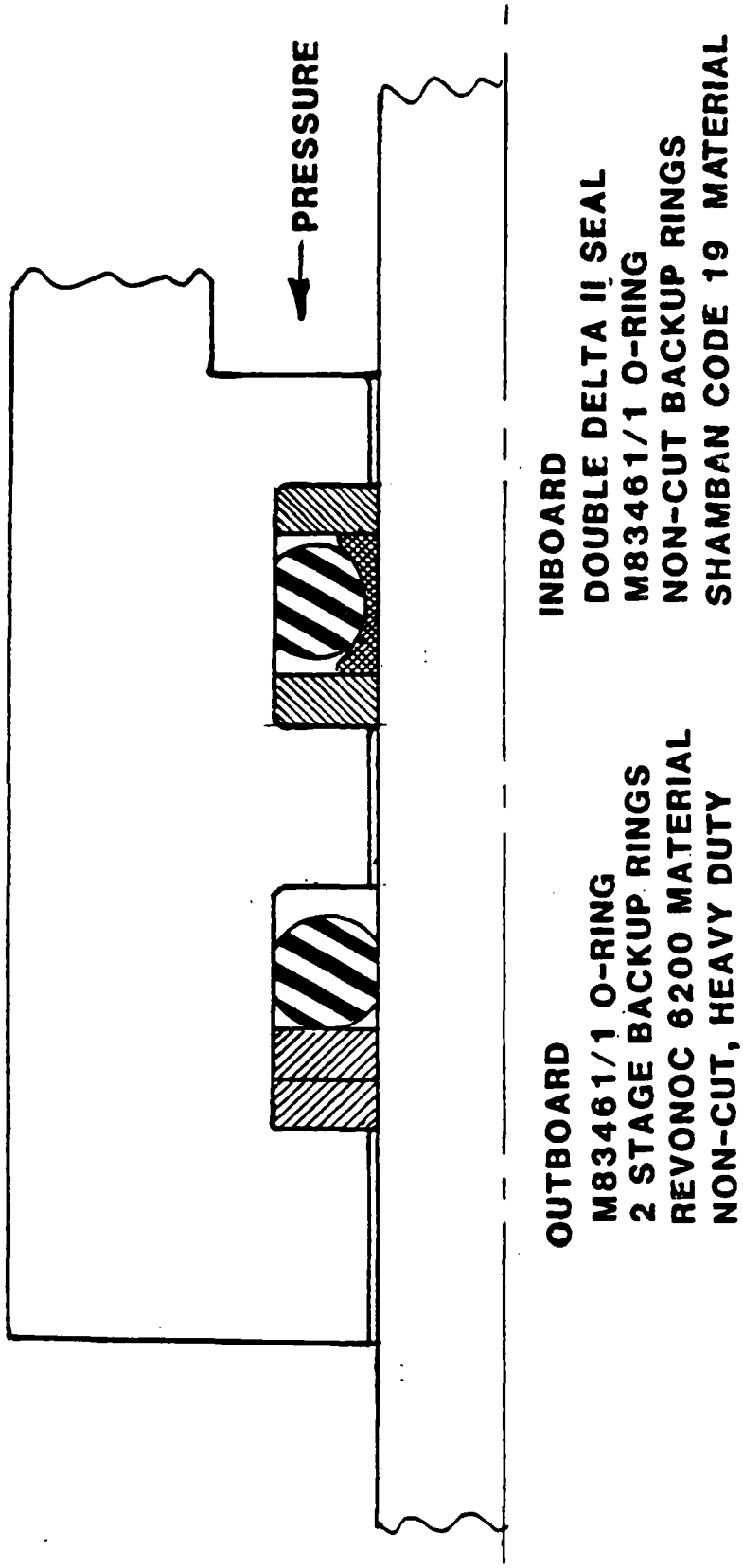


Figure 5. Baseline Rod Seal System Tested

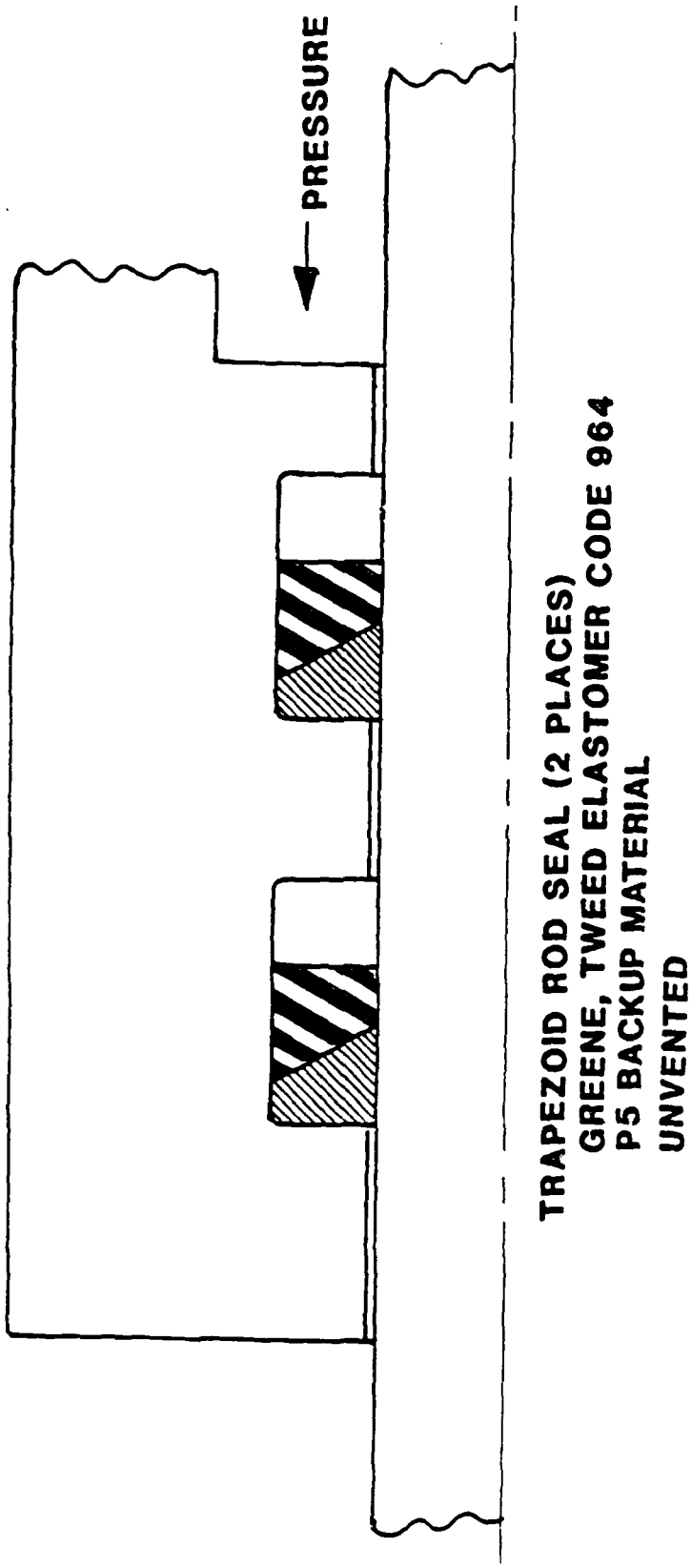


Figure 6. Alternate Rod Seal System Tested

4.0 TEST PROCEDURE

4.1 OBJECTIVE - The objective of this test is to determine long life piston seals for MIL-H-83282, -65°F to +275°F, 8000 psi hydraulic systems.

4.2 TEST MATERIAL AND EQUIPMENT

Six 83-00516 actuators

Piston Seals - Fourteen candidates from various suppliers

Rod Seals - Two configurations from three suppliers

TL25001 test fixture 8000 psi, MIL-H-83282 hydraulic power source.

4.3 TEST LAB TASKS

4.3.1 Test Fixture Rework - Strengthen loading cylinder clevis by welding on four stiffeners.

4.3.2 Dimensional Inspection of Test Articles - Measure and record part dimensions identified below before test. Parts with "*" to be measured after test also.

4.3.2.1 Seals and Backup Rings*

Cross-section diameter or thickness and width.

Dimensions of each seal and backup ring will be recorded and its identity will be maintained throughout the test and evaluation program

4.3.2.2 Piston and Rod

Rod diameter and surface finish
Piston O.D.
Piston seal gland.

4.3.2.3 End Cap

Rod bore diameter
Seal glands diameter

4.3.2.4 Barrel

I.D. and surface finish
O.D.

4.3.3 Test Setup (Figure 7) - Install six piston seals, twelve rod seals, and assemble six test actuators. The six 83-00516 test actuators will be installed in the TL25001 test fixture. A load cylinder will provide an opposing load in each direction corresponding to 80% of the total thrust of the test cylinders operating at 8000 psi.

4.3.4 Operational Test

3.4.1 Testing will be accomplished in eight 50 hour blocks. Ninety percent of each block will be accomplished at test specimen actuator piston amplitudes of +/- 1.75 inches and the remaining ten percent at amplitudes of +/-0.10 inches. Cycling will be accomplished at rates of nine and thirty cycles per minute for the piston amplitudes of +/-1.75 and +/-0.10 inches respectively. The test specimen actuators will not be permitted to bottom and will be limited in stroke by the installation of plugs in each end of the loading cylinder.

4.3.4.2 Fluid temperature at one port of two specimen actuators will be monitored and maintained between 250°F (1) and 275°F. Pressure available to drive the actuators will be 8000 psi and will be reacted by the load cylinder equivalent to eighty percent of the force output of the specimen actuators to produce a 6400 psi pressure differential across the piston seals.

4.3.4.3 During all operational tests, the following parameters will be periodically monitored and their results recorded at least once during each day of operation.

- Pump suction (pressure and temperature)
- Pump discharge (pressure and temperature)
- Pump case drain (pressure and flow)
- Load cylinder pressure
- Specimen actuator rod end leakage - provide a plastic tube and clean bottle at each end cap. Collect and record rod seal leakage periodically.
- Specimen actuator cycles
- Elapsed test time
- Specimen actuator pressures

Specimen actuator fluid temperatures will be recorded at least every 4 hours.

NOTE: (1) This minimum temperature will be adjusted to limit maximum pump inlet oil temperature to 250°F

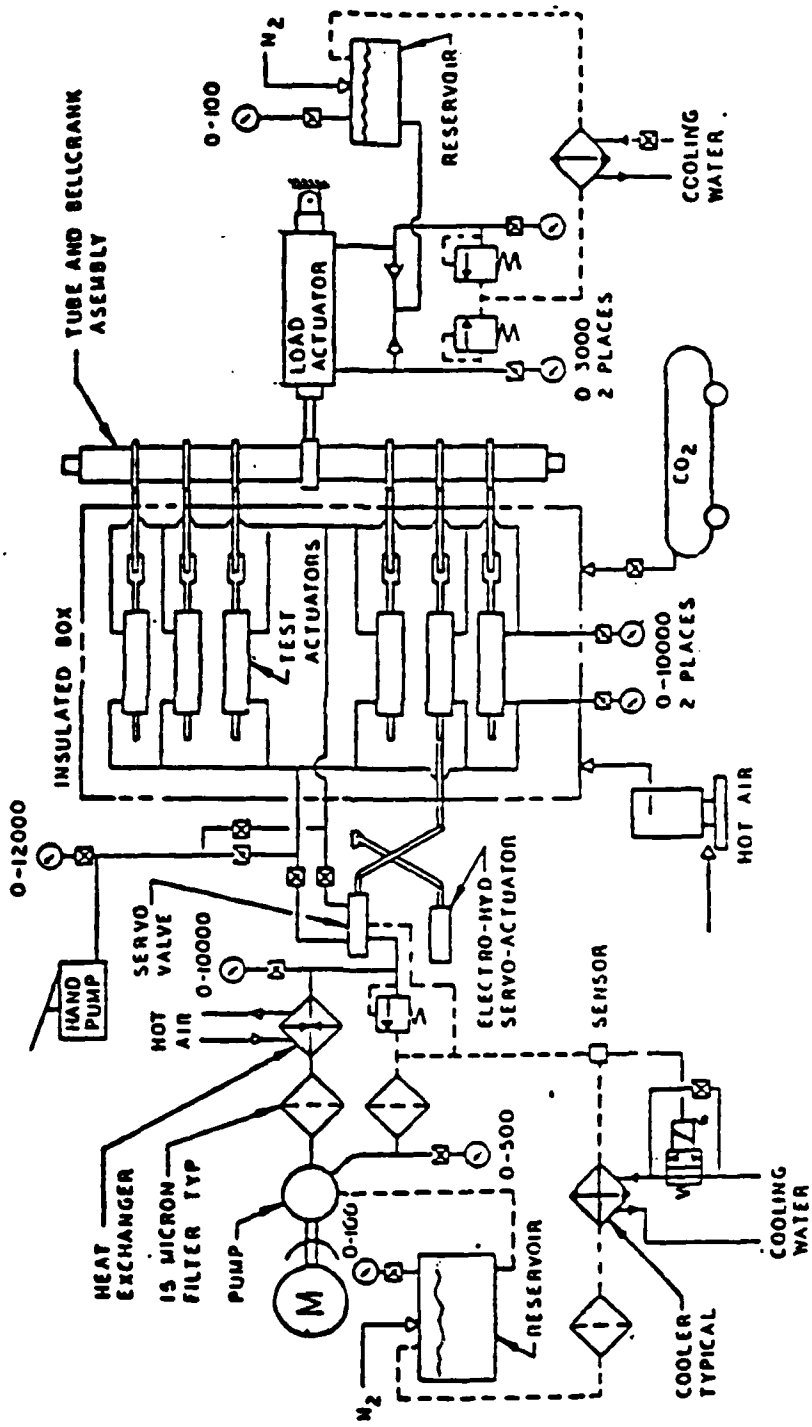


Figure 7. Test System Hydraulic Schematic

4.3.4.4 Leakage Tests - Static leakage tests on the piston seals will be performed prior to the start of cycling tests and at the completion of each block of fifty hours of operational testing.

These leakage tests will be accomplished at ambient temperature and at minus 40°F by pressurizing one end of the test actuator to values of 8,000 and 500 psi with the cylinder centered at mid stroke. Leakage quantities will be determined during the second and third minutes of a three minute test period and recorded. This leakage test will be accomplished by mechanically restraining cylinder and opening a needle valve connected to a drip tube (2).

The O.D. of the barrel at mid-point will be measured at 1000 psi pressure intervals prior to beginning test.

4.3.5 POST TEST - The test actuators will be disassembled and visually inspected.

NOTE: (1) A one minute period for tube drainage will be allowed prior to 2 minute leakage test.

5.0 ANALYSIS OF TEST RESULTS

5.1 Piston Seals Test Results

The test was concluded after 404.67 hours of cycling. Oil temperatures taken at one of the test cylinders on the average of every 19 minutes during the test indicate an average oil temperature of 240 degrees F. Applying the normal distribution to the data indicates 68 percent of the test was conducted at temperatures between 203 and 273 F degrees. Forty percent of the test time was at 250 F degrees or greater.

Two piston seal candidates completed more than 400 hours of testing. These were candidates P2 and P4. Two candidates failed during the test due to severe extrusion. These were candidates P6 and P3. Two candidates failed due to chipped chrome from the piston and was not the fault of the candidates. Candidate P1 had high leakage after 100 hours and was loaded with chrome chips from the piston. A second sample of P1 was placed back into the test. Candidate P8, a one piece piston ring, had high leakage after 200 hours. A chip from the groove edge created a large gap which allowed the piston ring to extrude and fail. A second sample of this seal was not immediately available and subsequently a piston ring from another source was substituted.

A program goal was to make some conclusions about upper limits on diametral clearance for piston seals. Accordingly, the pistons and cylinder barrels were dimensionally inspected prior to assembly of test cylinders. In order to measure cylinder barrel breathing, each assembly was mechanically restrained in the test fixture and the change in cylinder barrel radius was measured at pressure from 0 to 8000 psi using a dial indicator grounded on the cylinder end cap. This method was not reliable. It did not account for whether a cylinder barrel was centered or decentered in the end caps. As a result the data was irregular with one exception. Several of the barrels stopped expanding after 6000 psi was applied. It was determined by measurements after the test that the barrels expanded until the barrel outer diameter contacted the end cap bore into which the barrel was inserted. Therefore, all diametral clearances shown for piston seals is the sum of the measured unpressurized piston to bore clearance and the barrel to end cap bore clearance. A discussion of each candidate is presented in succeeding paragraphs. See Table 2 for a summary of piston seal test results. Table 3 and 4 give a detailed leakage record for each candidate. Table 5 provides average pressure to overcome seal breakout friction for each cylinder assembly in the test.

5.1.1 Candidate: P1 (SN's 001-004); 266-21800-964-1200, Greene, Tweed & Co.

Results: Two samples of this candidate were tested. The first sample was removed after 100 hours of testing. The reason for removal was excessive leakage. when the test actuator was disassembled, it was found that the chrome plating on one side of the piston had chipped. Subsequent cycling had scored the cylinder walls. The seal was in excellent condition except for an accumulation of metal chips around the sealing surface. When the seal was removed, it was observed that the elastomer had split through the mold parting

TABLE 2. SUMMARY OF PISTON SEAL TEST RESULTS

CYL ASSY NO. REF	CANDIDATE NO.	DESCRIPTION	DIA CLR	HOURS TESTED	LEAKAGE ml/min (3)	SEAL CONDITION	CYL BORE CONDITION
5	P4	Special Shape elastomer & backup	.0021	401.9	.02/.04	Exc	Fair
3	P2	Capseal "piggy back" backups	.0011	404.67	0.4/0.3	Exc	Good
6A	P14	Dual piston ring set with expander	.0050	202.9	33/44	Exc	Exc
6	P8	One piece piston ring with expander	.0021	201.75	48/*	Poor (2)	Good
1A	P12	Cap seal with very thick backups	.0025	122.7	0.5/0.5	Exc	Fair
2	P1	Cap seal with "T" Elastomer	.0020	100.0	.08/*	Exc (1)	Poor (1)
2A	P1	Cap seal with "T" Elastomer	.0044	202.6	5.9/4.5	Poor	Fair
4A	P13	Same as P3 except notches in cap seal	.0021	122.7	0.5/0.0	Fair	Poor
1	P6	Heavy duty special shape elastomer & backup	.0028	253.3	.08/*	Poor	Good
4	P3	Cap Seal with non-cut backups	.0021	237.4	0.6/*	Poor	Good

NOTES: (1) Chrome had chipped off one side of the piston and caused severe damage to the piston OD and cylinder bore. It is believed the chrome was chipped during the 2nd 50 hour block when the rod end jam nut backed off and allowed the piston to bottom internally against the end cap.

(2) Chrome had chipped off one edge of the seal gland. The piston ring extruded through the gap created.

(3) Average 8000 psi ambient temperature leakage for all blocks prior to removal/leakage at removal. "*" indicated leakage too great to measure in 2 minute period or could not achieve test pressure.

TABLE 3. RECORD OF PISTON SEAL LEAKAGE -- ML/MIN (500 psi/8000 psi)
AMBIENT TEMPERATURE

PISTON SEAL CANDIDATE	TIME - HRS 0	50	100	150	200	250	300	350	400
P1	0.1/0.08	0/0	38/*						
P2	0.05/0.1	0/0.3	0.05/0.03	0.22/0.48	0.2/0.6	0./0.08	0/0.35	0.25/1.05	0/0.3
P3	0.08/0.05	0/0	0/0.05	0.33/0.55	2.53/2.48	*/*			
P4	0.1/0.1	0/0	0/0	0.13/0.08	0.2/0.4	0.13/0.15	0/0.35	0.1/0.65	0/0.35
P6	0.1/0.08	0/0	0/0	0.2/0.22	1.0/1.5	*/*			
P8	17/44	9/16.6	7.5/90	4.8/42.0	22.8/*				
P1					(2)/0.01	0/0.65	0.4/0.98	7.4/16	9.0/4.5
P12						(2)/0.01	0.08/0.21	0.05/0.7	0/0.45
P13						(2)/0.08	0/0.35	0/0.6	0/0
P14					(2)/(4)	6/24	16/23	68/52	13/44

NOTES: (1) "*" Indicates leakage rate is too great to measure at test pressure required
(2) Leakage not measured
(3) 80 ML/MIN at 3000 psi

TABLE 4. RECORD OF PISTON SEAL LEAKAGE -- ML/MIN (500 psi/8000 psi)
-40°F TEMPERATURE

PISTON SEAL CANDIDATE	TIME - HRS 0	50	100	150	200	250	300	350	400
P1	4.55/3.25	0/0.25	*/*						
P2	1.6/1.88	0.38/1.0	0/0.22	0.32/0.28	0.2/0.55	0.2/0.45	0/0	0/0.05	0.4/1.3
P3	4.13/1.18	0/0	0.58/.03	0/0.10	0.58/0.25	*/*			
P4	1.38/1.13	0/0	0/0	0/0.03	2.9/0.18	0.28/0.90	1.25/2.50	0.15/2.3	3.2/0.3
P6	0.78/1.28	0/0	0/0	0.05/0.05	0/0	*/*			
P8	8.4/39.5	32/11	17.5/86	1.0/16	15/*				
P1					(2)/(2)	0.53/0.5	0/0.05	0.19/1.4	2.5/2.5
P12						(2) 275HR	0/0.08	0/1.3	0.05/0.95
P13						(2) 275HR	0/0	0.15/0.48	1.5/0.5
P14					(2)/(2)	4.95/48	3.4/1.35	14/5.0	1.4/8.5

NOTES: (1) "*" Indicates leakage rate is too great to measure in 2 minute period or could not achieve test pressure.
(2) Leakage not measured
(3) Oil temperature stabilized at -40°F at start of test. Test temperature allowed to increase during leakage test.

TABLE 5. BREAK-OUT PRESSURE TEST RESULTS

ASSY	AVG PRESSURE TO BREAKOUT -- psig	SEALS IN ASSY REF RE	PISTON	LE
1	155	BL	P6	BL
2	176	BL	P1	BL
3	116	BL	P2	BL
4	126	BL	P3	ALT
5	139	BL	P4	BL
6	127	BL	P8	BL
1A	176		P12	BL
2A	183	ALT	P1(1)	BL
4A	185	ALT	P13	BL
6A	126	BL	P14	ALT

BL = Baseline Rod Seal System
ALT = Alternate Rod Seal System
RE = Rod End of Cylinder
LE = Lug End of Cylinder

NOTES: (1) Retest

line. The elastomer was flexible and had not hardened. The chrome chipping may have occurred when the rod end jam nut backed off during the second block of cycling, which allowed the piston to severly bottom out internally.

Candidate P1, (SN's 152-155), 266-21800-964-1200; Greene, Tweed & Co.

This sample was installed at 201.75 hour to replace the sample above which was damaged by chrome chips. At the conclusion of the test, this sample had accumulated 202.9 hours at a diametral clearance of .0044. The appearance of the plastic portions of the seal was good. The backups had approximately 2 percent wear radially. The cap strip was worn approximately 1 percent of its thickness. The elastomer shows evidence of extrusion on both sides of the "T" portion of the seal. The plastic portions of the seal have some embedded chrome chips. One source of the chrome was a 1.0 mm chip missing from the side of the piston on the side toward the rod. The cylinder bore on this assembly was lightly scored along the length of the barrel. This assembly had a new piston and different cylinder barrel from the assembly which failed above in the first test of candidate P1.

5.1.2 Candidate P8; TF 500A-7218 (490), Tetrafluor Co

Results: After 201.75 hours of testing, it was not possible to pressurize the piston to 8000 psi with the hand pump, indicating high leakage. The test actuator was disassembled. The piston ring had extruded completely through. Approximately .19 inches of the seal was missing around the circumference. The cause of the failure is attributed to a chipped edge on the groove edge approximately the same width as the missing seal. It was concluded the failure was not the fault of the seal.

5.1.3 Candidate P3; S30772-218P19, S33157-218-19, W. J. Shamban Co.

Results: At the end of block 5 (253.3 hour), during the leakage test, it was not possible to pressurize the cylinder to 500 or to 8000 psi. Flow from the leakage drip tube was a continuous stream. The cylinder was removed and torn down for inspection of the piston seal. The seal was almost completely gone. between the 200 hour and the 250 hour leakage test, the seal had extruded severly. The backup rings were completely gone. Only a 1.0mm piece of the elastomer remained. Working clearance was .0021 on the diameter.

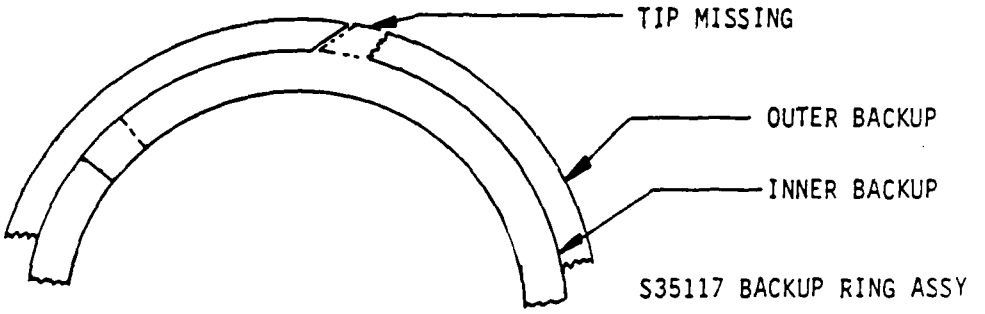
5.1.4 Candidate P6, CEC 5223-218, C. E. Conover & Co

Results: At the end of block 5, (253.3 hour) during the leakage test, it was not possible to pressurize the cylinder. Upon disassembly, the piston seal was severely extruded. Heat from fluid leakage had hardened the elastomer. The trapezoid shaped backup rings had extruded and worn completely through. The elastomer, was still flexible, but had been broken into many small pieces. The diametral clearance was .0028.

5.1.5 Candidate P2; S30772-218P19, S35117; W. S. Shamban Co.

Results: This seal survived the full 404.67 hour test without failure at a diametral clearance of .0011 inches. Overall appearance of the various parts of the seal is very good. The cap strip does not show any thin spots or evidence of extrusion. The elastomer is in excellent condition. The backup ring assemblies are in good condition with one interesting peculiarity. As

shown on the sketch below, the end of each outer backup is broken or worn at the cut. The inner backups have slightly cold-flowed into the irregular gap and, from the appearance of the inner backup, operated in this manner for some time. There is no way to be absolutely certain the tips were not sheared off at installation, however it has low probability because the piston, with seals installed, was placed in a chamfered cylindrical guide to install the piston and seals into the cylinder barrel. This was a hand fit operation with all pieces visible during installation, and witnessed by one person other than the assembler.



5.1.6 Candidate P4; S34232-218P19; W. S. Shamban & Co

Results: This seal survived the entire 400 hour test at a clearance of .0021 without failure. The elastomer is in excellent condition with no extrusion. The backup rings show approximately 32 percent wear in the radial direction and approximately 2 percent in width.

5.1.7 Candidate P12; TF 1146-218; Tetrafluor Co

Results: This seal was installed at 282 hour, therefore at the conclusion of the test, the seal had survived 122.7 hours at a diametral clearance of .0025 inches. All pieces of the seal are in excellent condition and exhibit very little wear or extrusion. The cap strip was evenly worn, had no thin spots, and appears to be of adequate thickness. The anti-blowby notches on each side of the thick backup rings were examined. The notches which were against the groove walls were closed and estimated to be non-functional. The grooves next to the cap strip were still open and appeared to be functional. No chrome chips were embedded in the plastic parts.

5.1.8 Candidate P13; S30772-218P19N, S33157-218-19; W. S. Shamban Co

Results: At the conclusion of the test, this candidate had accumulated 122.7 hour at a diametral clearance of .0021. Wear of the cap seal and backup was uniform around the circumference. The backup on the lug side of the groove had worn much more than the backup on the rod side. The radial wear of the rod side backup was approximately 2.7 percent compared to 19.9 percent for the lug side backup. Examination of cylinder parts and seal pieces does not indicate any reason for the difference in wear of the backup rings except that the cylinder barrel had numerous shallow axial score marks. Both backups had to operate over the score marks. There were no chrome chips embedded in the cap seal or backups. The cap seal did not have any thin spots. The anti-blow by notches on each side of the cap seal were closed up and estimated to be approaching a non-functional condition.

5.1.9 Candidate P14; S35119; W. S. Shamban Co.

Results: At the conclusion of the tests, this candidate had survived 202.9 hours at a diametral clearance of .0050 inches. Appearance of the seal was excellent. Wear of the seal was not measureable. Both halves of the seal had numerous chrome ships embedded in the sealing surface. The cylinder bore had a very good finish and had no score marks. The piston was examined and was not the source of the chips.

5.2 Rod Seal Test Results

There were no rod seal failures during the test. Both rod seal systems gave very low leakage and acceptable wear.

A total of 14 installations of the baseline rod seal system and 5 installations of the alternate rod seal systems were evaluated. The baseline rod seal system accumulated 2,714,918 cycles and the alternate system accumulated 821059 cycles. Diametral clearance varied from .0028 to .0060 for the first stage of the 19 installations. Regardless of time in the test or clearance all rod seals were in good to excellent condition upon removal with two exceptions. On cylinder assemblies 2A and 6A the first stage G-T trapezoid rod seal was installed with the backup towards cylinder pressure. The elastomer was badly extruded on assembly 2A and nibbled extensively on assembly 6A. Despite the reversed installation of the first stage, the external leakage of the two sets of seals was no greater than measured for correctly installed rod seals of the same design.

The baseline rod seal system and the alternate rod seal system both performed equally. Average leakage for the baseline rod seal system was .10 drop/25 cycles compared with .11 drop/25 cycles for the alternate system. See Table 6 for a summary of rod seal performance.

TABLE 6. SUMMARY OF ROD SEAL TEST RESULTS

ASSEMBLY NO	END CAP	ROD SEAL DESCRIP	DIA CLR	TOTAL HOURS	TOTAL CYCLES	TOTAL LEAKAGE(1)	DROPS PER 25 CYC	CONDITION
1	ROD END	BASELINE						
		1ST STAGE	.0060	253.33	216957	2	.02	EXCELLENT
		2ND STAGE	.0068					EXCELLENT
	LUG END	BASELINE						
		1ST STAGE	.0033	253.33	216957	9	.09	EXCELLENT
		2ND STAGE	.0037					EXCELLENT
2	ROD END	BASELINE						
		1ST STAGE	.0036	100.0	85500	NA	NA	EXCELLENT
		2ND STAGE	.0044					EXCELLENT
	LUG END	BASELINE						
		1ST STAGE	.0037	100.0	85500	NA	NA	EXCELLENT
		2ND STAGE	.0039					EXCELLENT
3	ROD END	BASELINE						
		1ST STAGE	.0037	404.67	345993	38	.05	EXCELLENT
		2ND STAGE	.0037					EXCELLENT
	LUG END	BASELINE						
		1ST STAGE	.0036	404.67	345993	26	.04	GOOD
		2ND STAGE	.0038					EXCELLENT

NOTE: (1) Milliliters

TABLE 6. SUMMARY OF ROD SEAL TEST RESULTS - CONTINUED

ASSEMBLY NO	END CAP	ROD SEAL DESCRIP	DIA CLR	TOTAL HOURS	TOTAL CYCLES	TOTAL LEAKAGE(1)	DROPS PER 25 CYC	CONDITION
4	ROD END	ALTERNATE		215.9	184594	15.5	.04	
		1ST STAGE	.0035					EXCELLENT
		2ND STAGE	.0037					EXCELLENT
	LUG END	ALTERNATE		215.9	184594	120.5	.33	
		1ST STAGE	.0038					EXCELLENT
		2ND STAGE	.0038					EXCELLENT
5	ROD END	BASELINE		401.9	343624	217.0	.32	
		1ST STAGE	.0035					EXCELLENT
		2ND STAGE	.0037					GOOD
	LUG END	BASELINE		401.9	343624	51.5	.07	
		1ST STAGE	.0098					GOOD
		2ND STAGE	.0054					EXCELLENT
6	ROD END	BASELINE		201.75	172496	NA	NA	
		1ST STAGE	.0039					EXCELLENT
		2ND STAGE	.0039					EXCELLENT
	LUG END	BASELINE		201.75	172496	NA	NA	
		1ST STAGE	.0037					EXCELLENT
		2ND STAGE	.0039					EXCELLENT

NOTES: (1) Milliliters

TABLE 6. SUMMARY OF ROD SEAL TEST RESULTS - CONTINUED

ASSEMBLY NO	END CAP	ROD SEAL DESCRIP	DIA CLR	TOTAL HOURS	TOTAL CYCLES	TOTAL LEAKAGE(1)	DROPS PER 25 CYC	CONDITION
1A	ROD END	BASELINE	(3)	(3)	(3)	(3)	(3)	(3)
		1ST STAGE						
		2ND STAGE						
	LUG END	BASELINE						
		1ST STAGE	.0020	122.7	104908	12.5	.06	EXCELLENT
		2ND STAGE	.0032					EXCELLENT
2A	ROD END	ALTERNATE						
		1ST STAGE	.0032	202.9	173479	9.5	.03	POOR (2)
		2ND STAGE	.0040					FAIR
	LUG END	BASELINE						
		1ST STAGE	.0036	202.9	173479	99.5	.29	EXCELLENT
		2ND STAGE	.0040					EXCELLENT

NOTES: (1) MILLILITERS
(2) BACKUP AND SEAL WERE INSTALLED REVERSED
(3) NOT RECORDED. THIS ROD HAD A TUNGSTEN CARBIDE SURFACE COATING WHICH WAS NOT PART OF THE TEST. A NON-OPTIMUM SEAL MATERIAL WAS USED.

TABLE 6. SUMMARY OF ROD SEAL TEST RESULTS - CONTINUED

ASSEMBLY NO	END CAP	ROD SEAL DESCRIP	DIA CLR	TOTAL HOURS	TOTAL CYCLES	TOTAL LEAKAGE(1)	DROPS PER 25 CYC	CONDITION
4A	ROD END	ALTERNATE						
		1ST STAGE	.0037	122.7	104908	18.5	.09	EXCELLENT
		2ND STAGE	.0037					EXCELLENT
	LUG END	BASELINE						
		1ST STAGE	.0037	122.7	104908	90.5	.43	EXCELLENT
		2ND STAGE	.0037					EXCELLENT
6A	ROD END	BASELINE						
		1ST STAGE	.0037	202.9	173479	11.5	.03	EXCELLENT
		2ND STAGE	.0037					EXCELLENT
	LUG END	ALTERNATE						
		1ST STAGE	.0039	202.9	173479	17.5	.05	POOR (2)
		2ND STAGE	.0041					FAIR

NOTES: (1) MILLILITERS
(2) BACKUP AND SEAL WERE INSTALLED REVERSED
(3) NOT RECORDED. THIS ROD HAD A TUNGSTEN CARBIDE SURFACE COATING WHICH WAS NOT PART OF THE TEST. A NON-OPTIMUM SEAL MATERIAL WAS USED.

6.0 CONCLUSIONS

6.1 PISTON SEALS

Based upon the diametral clearance in the installation, length of time in the test, seal wear, and leakage the candidates are rated in the following order with best performing listed first.

P4 - P2 - P14 - P8 - P12 - P1 - P6 - P3 - P13

Chipped chrome from the edge of the seal groove caused premature failure of candidate P8. It is estimated that P8 would have placed just below P14 if the gap created by the chip had not caused the seal to fail.

The only two failures in the test were P6 and P3. Both of these seals were in the process of extruding away when removed. More extrusion resistant materials are needed to extend the life of these two configurations.

It was a goal at the beginning of the program to evaluate the piston seals at diametral clearances in the range of .0025 to .0040. However, the cylinder barrel expansion with pressure was limited by the clearance between the end caps and the couter diameter of the cylinder barrel. Therefore the initial clearances in assemblies 1 thru 6 were in the range of .0011 to .0028. The replacement cylinder assemblies 1A, 2A, 4A, 6A with candidates P12, P1, P13 and P13 respectively had clearances ranging from .0021 to .0050. The only candidates apparently affected by the high diametral clearance were P1 with a clearance of .0044 and P6 with a clearance of .0028.

The anti-blowby notches in the backup ring adjacent to the capseal on Candidate P12 appeared to be functional after 122.7 hours of testing. The backup rings on this design were approximately .10 inch thick and could tolerate the reduction in cross-section at the notches. It is estimated that .05 nominal width backups would not tolerate notching. The anti-blowby notches on Candidate P13 which were in the edges of the capstrip appeared to be non-functional after 122.7 hours. Protection against blowby should be a matter of concern in future design.

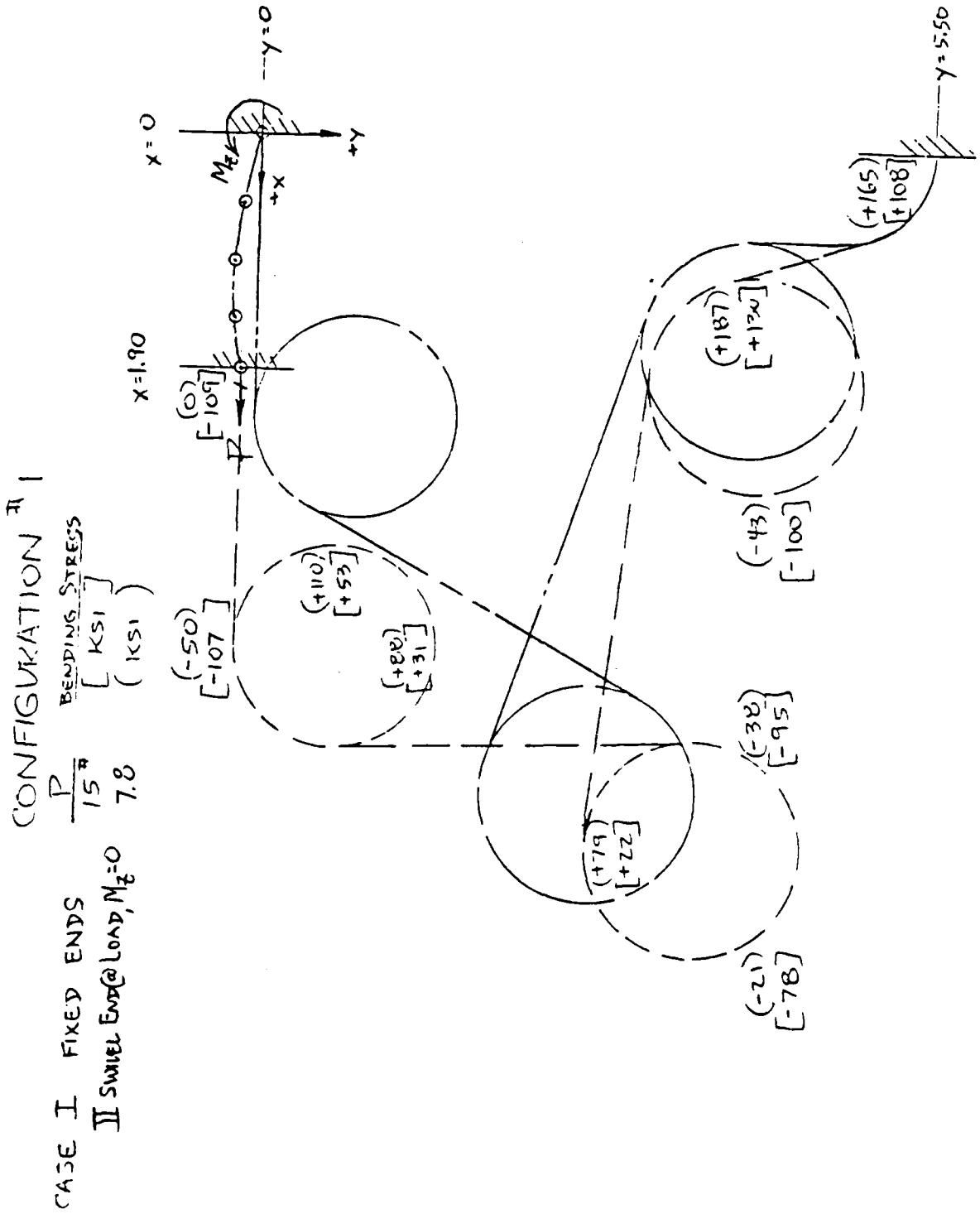
The breakout pressures do not represent any significant differences among the piston seal and rod seal combinations with an average for the 10 assemblies of 152 psi or 147 pounds thrust. One hundred forty seven pounds is 1.9 percent of the cylinder thrust at 8000 psi.

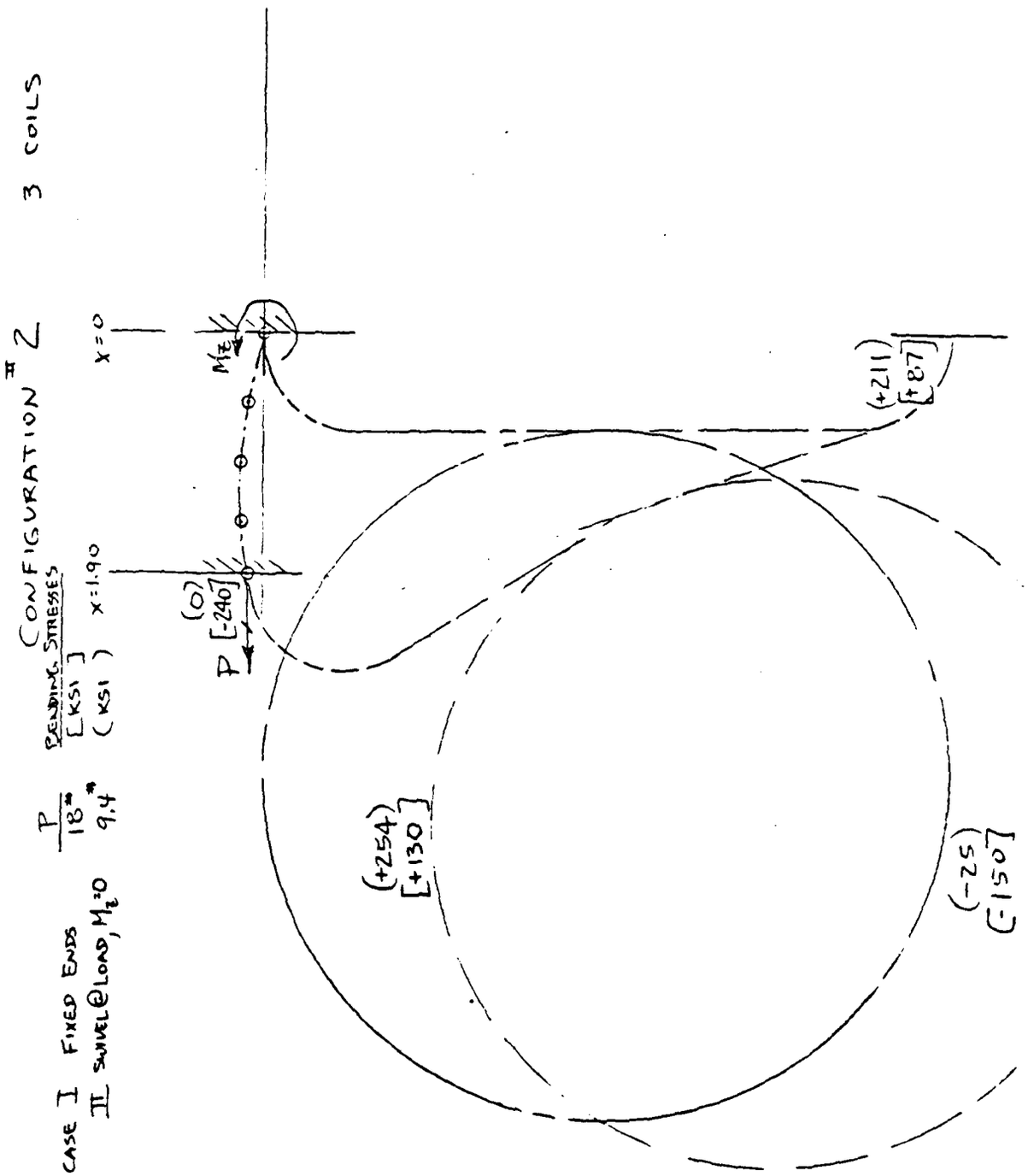
6.2 ROD SEALS

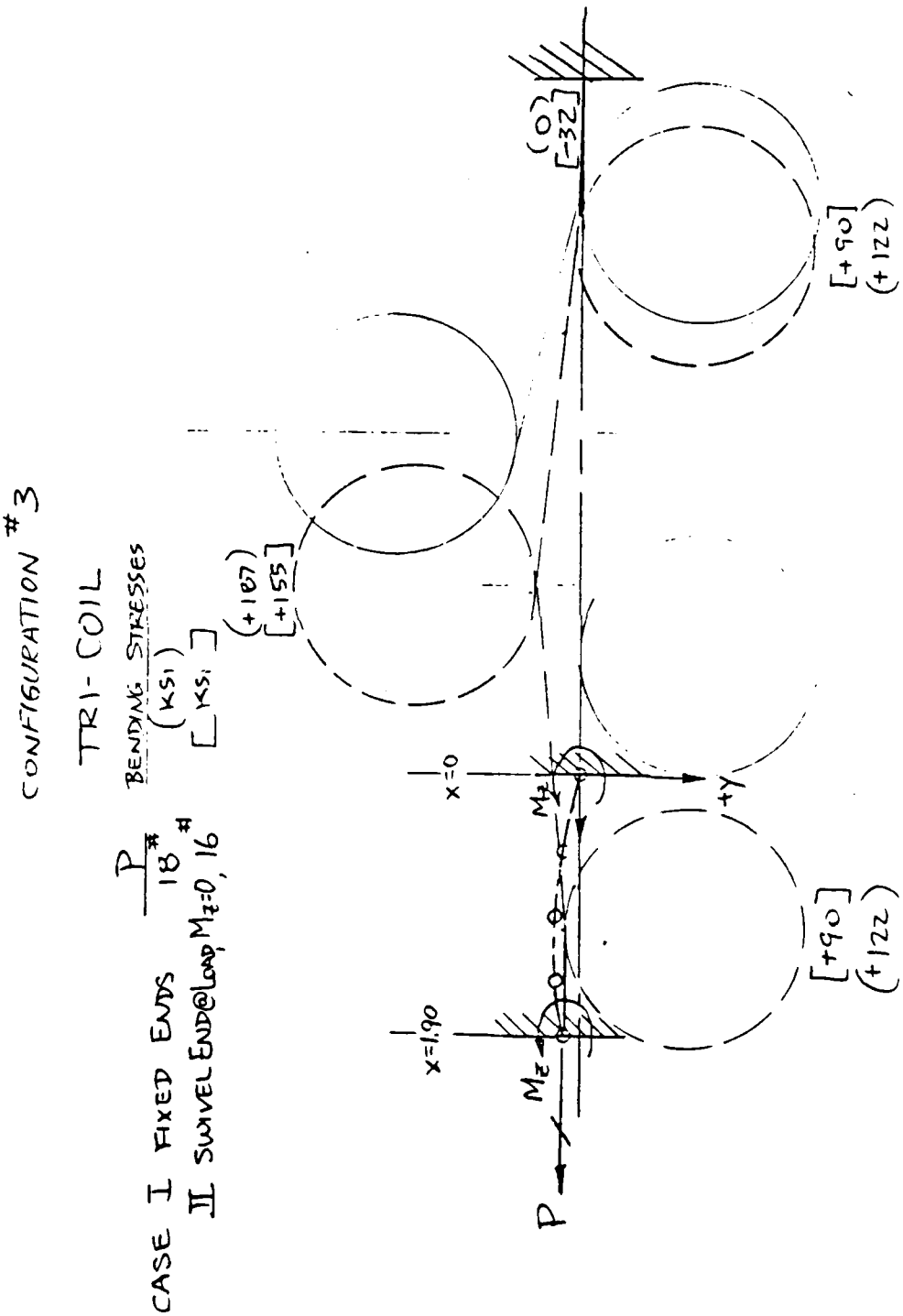
The two rod seal systems performed equally. Diametral clearance at the first stage seal ranged from .0028 to .0060 with an average of .0039 inches. It is concluded that these systems will operate satisfactorily at diametral clearances up to .004 inches.

APPENDIX D

COIL TUBE DESIGN EXAMPLES


CONFIGURATION #1
CASE I FIXED ENDS
II SWIVEL END @ LOAD, M_z=0
P/15# 7.8
BENDING STRESS
[KSI]
(KSI)
x=0
x=1.90
y=0
y=5.50
M_z
+x
+y
P
(0) [-109]
(-50) [-107]
(+110) [+53]
(+88) [+31]
(+187) [+124]
(-43) [-100]
(+165) [+108]
(+79) [+22]
(-38) [-95]
(-21) [-78]

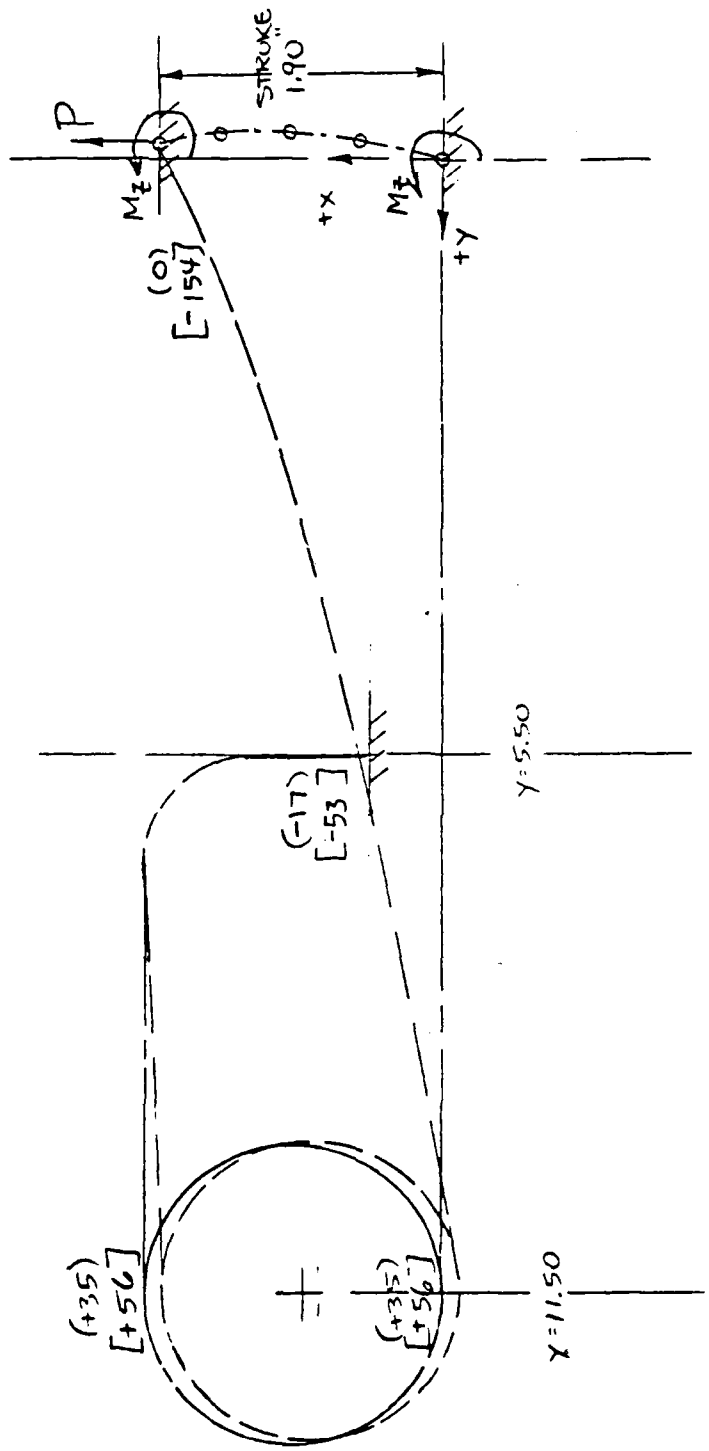

CONFIGURATION # 2
3 COILS
CASE I FIXED ENDS
II SWIVEL @ LOAD, $M_t = 0$
P
18#
9.4#
BENDING STRESSES
[KSI]
(KSI)
x=1.90
x=0
M_t
P [-240]
(0)
(+254)
[+130]
(-25)
[-150]
(+211)
[+87]


CONFIGURATION #3
TRI-COIL
BENDING STRESSES
(KSI)
[KSI]
CASE I FIXED ENDS
II SWIVEL END@LOAD, M_z=0, 16
P/18#
(+187)
[+155]
(0)
[-32]
[+90]
(+122)
[+90]
(+122)
x=0
x=1.90
M_z
M_z
P
+Y

CONFIGURATION 4

MODIFIED 540° HELICAL COIL TUBE

		P/6"	BENDING STRESSES
CASE I	FIXED ENDS		[KSI]
II	SWIVEL @ LOAD, $M_z = 0$	1	(KSI)



```
ST

 85/02/14. 11.13.30.
PROGRAM   DTCOIL

1000 ST_COIL540
1001 3,1
1002 15,2,1,1,14,0,0,0,1
1203 0.0,1.0,1,0,0,0,1,0
2001 1,0,0,0
2002 2.0,11.54,0
2003 3,.293,12.25,.06
2004 4,1.,12.54,.09
2005 5,1.707,12.25,.12
2006 6,2.0,11.54,.18
2007 7,1.707,10.83,.24
2008 8,1.0,10.54,.27
2009 9,.293,10.83,.3
2010 10.0,11.54,.36
2011 11,.293,12.25,.42
2012 12,1.,12.54,.45
2013 13,1.707,12.25,.48
2014 14,2.,11.54,.54
2015 15,2.,5.54,.54
3001 1,0,0,0,1,1,1
3002 15,1,1,1,1,1,1
4001 1,28.5E06,.285,11.2E06,8.65E-06,.283
5001 1,1,.0106,.0053,.0053,.00003765,.00003765,.0000753,.000402,.000402
6001 1,1,2,15,1
6002 2,2,3,6,1
6003 3,3,4,7,1
6004 4,4,5,8,1
6005 5,5,6,9,1
6006 6,6,7,10,1
6007 7,7,8,11,1
6008 8,8,9,12,1
6009 9,9,10,13,1
6010 10,10,11,14,1
6011 11,11,12,15,1
6012 12,12,13,15,1
6013 13,13,14,10,1
6014 14,14,15,1,1
9500 AXIALUNIT
9501 1,1
10000 1,0,0,0,0,0,0,0,0
11001 1,1,0,0,0,0,0
 READY.
```

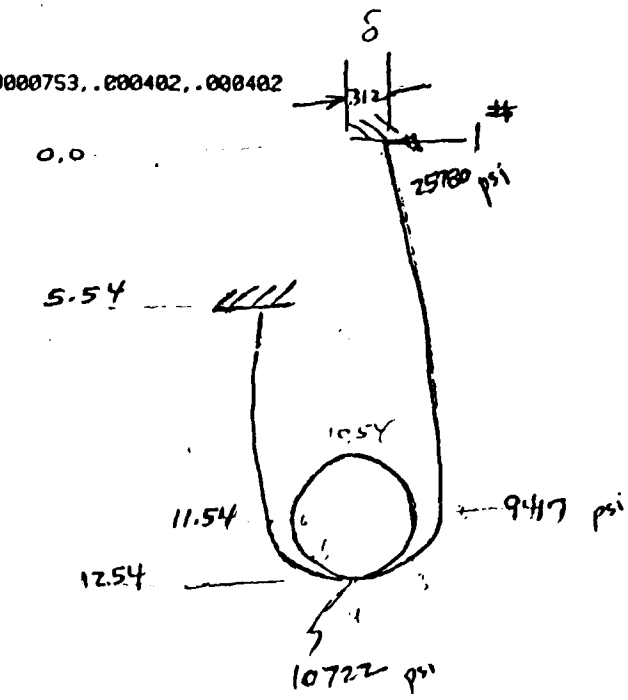
M_z Fixed End

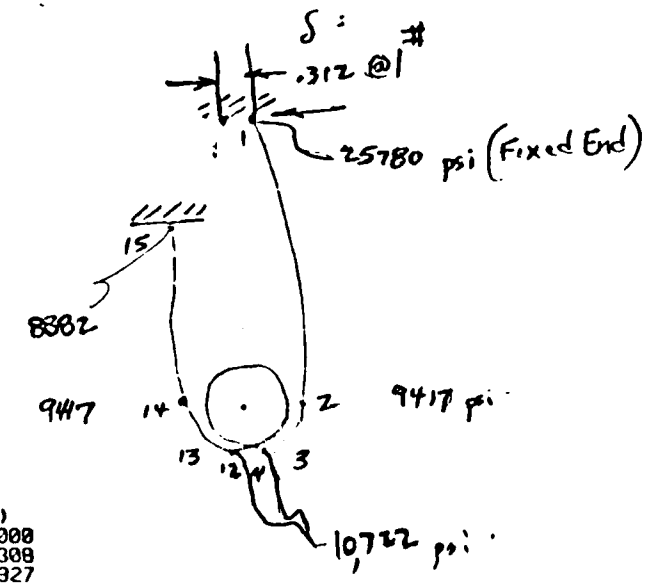
#

CONFIGURATION #4

COMPUTER EXAMPLE

FIXED END BENDING STRESSES (UNIT LOAD)





02/13/85 10.42.10. PAGE 2

RESULTS FOR LOADING CASE NO. 1 #

JOINT DISPLACEMENTS - (GLOBAL COORDINATE SYSTEM)

JOINT NO.	D-X (IN)	D-Y (IN)	D-Z (IN)	R-X (RAD)	R-Y (RAD)	R-Z (RAD)
1	.312108	-.029313	-.006314	0.000000	0.000000	0.000000
2	.026339	-.029313	-.005920	.000068	-.002028	.028008
3	.006542	-.021223	-.004992	.000039	-.003140	.026327
4	-.000790	-.003589	-.002724	.000006	-.003253	.023512
5	.005513	.012024	-.000397	-.000010	-.003321	.020699
6	.019088	.017710	.000589	-.000022	-.003326	.018221
7	.031086	.012676	-.000373	-.000000	-.003287	.016253
8	.035461	.001765	-.002608	.000025	-.003248	.014658
9	.031336	-.008052	-.004960	.000042	-.003201	.013064
10	.022527	-.011611	-.005856	.000054	-.003136	.011098
11	.015299	-.009709	-.004914	.000051	-.003027	.008622
12	.013109	-.003593	-.002813	.000044	-.002873	.005808
13	.014203	-.000498	-.000857	.000002	-.002674	.002094
14	.015345	.000000	-.000106	-.000035	-.002358	.000511
15	0.000000	0.000000	0.000000	0.000000	0.000000	0.000000

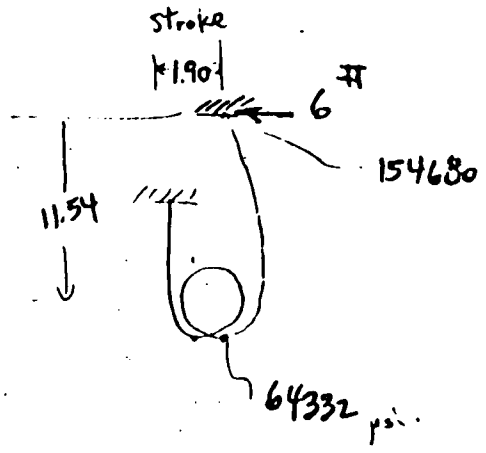
MEMBER STRESSES - (LOCAL COORDINATE SYSTEMS)

MEMBER NO.	JT NO	AXIAL (PSI)	BEND.-Y (PSI)	BEND.-Z (PSI)	COMBINED (PSI)
1	1	0.0000	-5493.5360	20285.8703	25779.4062
	2	0.0000	-1989.2599	7428.1885	9417.4484
2	2	-35.8784	279.1247	-7627.0361	-7042.0303
	3	-35.8784	-450.7219	9391.1746	-9077.7749
3	3	-87.2151	407.1175	-9441.6761	-9936.0087
	4	-87.2151	-534.1468	10155.7073	-10777.0693
4	4	-87.2151	549.0203	-10163.6178	-10799.8622
	5	-87.2151	-559.4809	9438.4504	-10085.1463
5	5	-35.8784	535.4212	-9429.3784	-10000.6781

$\times 6^{\#}$ for $\delta = 1.90''$

	6	-35.8784	-478.2205	7657.8372	-8171.0362
6	6	35.8784	541.7917	-7652.7773	8230.4474
	7	35.8784	-370.1045	6933.6338	6294.7117
7	7	87.2151	418.5342	-5933.0305	6420.7008
	8	87.2151	-291.5949	5189.0193	5567.7392
8	8	87.2151	281.0435	-5194.2168	5562.4724
	9	87.2151	-270.5889	5919.3842	6277.1892
9	9	35.8784	283.0534	-5915.1229	6234.0547
	10	35.8784	-340.2541	7686.6641	8052.7066
10	10	-35.8784	624.1989	-7669.2359	-8320.3132
	11	-35.8784	-795.7961	9433.3744	-10205.0489
11	11	-87.2151	600.6041	-9443.6989	-10221.5181
	12	-87.2151	-776.2475	10163.8672	-11027.3298
12	12	-87.2151	469.7719	-10165.6497	-10722.6367
	13	-87.2151	-544.8020	9444.2985	-10076.3155
13	13	-35.8784	-339.2528	-9377.5428	-9752.6740
	14	-35.8784	167.6556	7613.4043	-7816.9383
14	14	.0000	-1089.2509	-7428.1885	9417.4484
	15	.0000	-1081.2752	-6981.2008	8882.4760

READY.



APPENDIX E

BLACK RESIDUE INVESTIGATION

(PALL CORPORATION SLS REPORT # 1676)

Pall Corporation

30 Sea Cliff Avenue · Glen Cove, N.Y. 11542 · USA
Phone 516-671-4000 · TWX 510 223 0606
Telex 126329 · Cable Pallco Glen Cove

SCIENTIFIC & LABORATORY SERVICES DEPT.
Director: Erwin Kirnbauer
FILTER PERFORMANCE TESTS
PROCESS & SYSTEM CONTAMINATION CONTROL
PARTICULATE & MICROBIAL FLUID CONTAMINATION ANALYSIS
WEAR ANALYSIS
BIOLOGICAL CONTROLS

March 25, 1986

SLS Report #1676

Analysis Of "Black Residue" Filtered By APM 5 Micron Absolute Filters During Testing Of A Prototype 8,000 psi Lightweight Hydraulic System (LHS) For Rockwell International (North American Aircraft Division) Columbus, Ohio

Purpose

This report is a comprehensive summary of results and conclusions from laboratory analysis of LHS contaminants and fluid samples conducted over a two year period. The analyses aimed at identifying a black residue that has appeared persistently on system filters throughout the test program.

Summary

The results of analysis show that the "black residue" in the LHS is made-up of at least four distinctly different particles types. These particles are characterized below by elemental analysis and primary location filtered by APM 5 micron absolute filters.

a. iron containing (mainly case drain)
b. aluminum containing (mainly pressure-line)
c. organic (mainly pressure-line)
d. chromium containing (mainly return-line)

The results of analysis of fluid samples from the APM (5 micron absolute) filtered LHS show that cleanliness is being maintained at levels corresponding to NAS-1638 Class 1, or cleaner.

Scientific & Laboratory Services Dept.

Background

A "black residue" is being filtered by APM (5 micron absolute) filters operating in the Rockwell prototype 8,000 psi light weight hydraulic system (LHS) currently being developed for the U.S. Navy. The "black residue" becomes visible on the upstream surface of the APM filters after short periods of operation during each test and increases in amount with increasing hours of operation. Pressure-line, return-line and case drain APM (5 micron absolute) filters in the LHS are all filtering this black colored contaminant. Appendix C gives a chronological review (provided by Rockwell) of the "black residue" problem up to the time Pall Scientific and Laboratory Services Department began participation to help determine the materials makeup of the black particles. Collection of samples of "black residue" for analysis was first conducted by Pall SLS Department in January, 1984. At that time, samples of the filtered contaminant were collected from each APM filter location depicted in Figure I. It should be noted that the APM case-drain filter element used initially in the LHS (Figure I) was an APM (15 micron absolute) filter. Pall SLS Department recommended that the LHS configuration be changed to that depicted in Figure II. These changes were completed and a repeat sampling was conducted about three months later (March, 1984).

All samples were submitted to Pall Scientific and Laboratory Services Department for analysis of the residue and to determine if any change in contaminant types or particle size distribution resulted from the reconfiguration (Figure I → II) of the case-drain line to downstream of the return-line APM filter. It should also be noted that the APM case-drain filter installed in LHS configuration II (Figure II) was an APM 5 micron absolute filter.

Samples collected first quarter 1984 for analysis of the "black residue" will be related in this report to LHS configuration I or configuration II (Figure II) as follows.

Sample Set I

"Black Residue" Sample No	Description
1	pressure-line; configuration I
2	return-line; configuration I
3	case-drain; configuration I
4	pressure-line; configuration II
5	return line; configuration II
6	case-drain; configuration II

Samples 1 through 6 consisted of contaminants rinsed from the upstream surfaces of these APM filters. The filters could not be released by Rockwell for laboratory analysis at the time of sampling.

Note that only one loop of the LHS was in operation when these samples were collected. Consequently, no FC designation is applied to these samples as was applied to samples submitted later in the test program.

In March, 1985 a second set of (filter bowl) fluid samples containing "black residue" was collected from the APM 5 micron filters and submitted by Rockwell for evaluation versus particles in the supply of a original condition MIL-H-83282 fluid used in the prototype LHS. This set of samples is described as follows:

Sample Set II

"Black Residue" Sample No.	Description
7	FC-2; return-line; configuration II
8	FC-1; return-line; configuration II
9	FC-2; pressure-line; configuration II
10	FC-1; pressure-line; configuration II
11	FC-2; case-drain; configuration II
12	original condition MIL-H-83282

In June 1985, five (5) fluid samples were collected by Rockwell from the APM 5 micron absolute filtered LHS. Sampling valves on the APM filter housings or valves installed in the LHS downstream of APM filters were used to collect the fluid. These fluid samples were associated with a 600 hour test of the 8,000 psi system and are described below.

Sample Set III

Fluid Sample No.	Description
1	FC-1 (No. 1)
2	FC-1 (No. 2)
3	FC-2 (No. 1)
4	FC-2 (No. 2)
5	LHS Reservoir

The fluid samples submitted as Sample Set III were analyzed for the purpose of determining the level of fluid cleanliness being maintained by APM (5 micron absolute) filtration.

Included with Sample Set III was a small sample of seal debris from a LHS activator to be analyzed for elemental composition.

Analytical Approach - (see Appendix A for a description of analytical procedures)

I. Analysis of "Black Residue" and other particles

A. Sample Set I (Collected January and March 1984)

Representative collections of particles from each APM filter location were prepared from samples and then individual particles were isolated manually under a particle counting microscope according to color, shape and morphology. Isolated particles were transferred to clean secondary analysis membranes, where possible, or swept to cleaner areas of the primary membrane if transfer was impossible. Each type of particle identified and isolated in this fashion was photomicrographed and analyzed for elemental composition (XES).

In addition, 300 to 400 particles from each sample (i.e. a representative collection of particles) were sized and counted using particle counting techniques to determine the relative numbers of each type of particle (identified above) in a collection of particles from each sample.

B. Sample Set II (collected March, 1985)

Approximately 50 particles in a representative collection from each sample were individually analyzed for elemental composition using x-ray fluorescent emission spectrophotometry. The number of particles in each collection of 50 containing the same combination of elements was counted.

II. Fluid Analysis (collected in June, 1985)

Each fluid sample in Sample Set III was particle counted per SAE ARP 598A (modified). In addition, a gravimetric analysis (per SAE ARP-785) and a Karl Fischer total water content analysis (per DIN 51-777) was conducted on the reservoir oil sample (No. 5).

III. Elemental Analysis of the Seal Debris

The seal debris submitted with sample set III was analyzed for elemental composition using x-ray fluorescent emissions spectrophotometry (XES).

Results of Analysis

I. Analysis of Black Residue and other Particles

A. Particle Isolation and Photomicrographing

A total of seven particle types (I through VII) were isolated from thousands of particles in Sample Set I based on color, shape and morphology. Appendix B contains photomicrographs of each of the particle types found.

B. Elemental Analysis

Each of the seven particle types isolated from Sample Set I were analyzed for elemental composition. Based on this analysis the "black residue" was differentiated into four particle types that contribute to the black appearance of the contaminants collected on APM filter elements in the lightweight hydraulic system. Photomicrographs of the four "black" particles types were taken and are presented in Appendix B.

Photomicrographs include the black particle photos under perpendicular as well as oblique lighting because the appearance of iron and chromium containing particles changed versus the angle of the light source. The categorization of particle types based on elemental analysis is presented below and also in Appendix B.

Scientific & Laboratory Services Dept.

Elemental Compositions Of Type I "Black" Particles And Primary Particle Type At Each Filter Location In System Configuration II

Black Particle Type (I)	Elemental Analysis (XES)	Found Most In Filter Location
I (a)	major iron; trace chromium	case drain
I (b)	major aluminum; minor sulfur; traces chromium/zinc	pressure-line
I (c)	(organic particles suspected no significant XES response) limited to trace aluminum	pressure-line
I (d)	major chromium	return-line

Elemental Compositions Of Particles Types II Through VII

Particle Type	Elemental Analyses (XES)
II	minor Sulfur
III	minor aluminum, silicon and sulfur
IV	major silicon; minor magnesium and titanium; traces chlorine, copper and iron
V	major copper; minor tin; traces aluminum, silicon and sulfur
VI	minor sulfur and cadmium; traces aluminium, silicon and sodium
VII	major sulfur; minor aluminum, silicon an calcium; traces cadmium and potassium

C. Percent Distribution and Size Range of Significant Particle at Each Filter Location by Particle Type

Pressure-line

System Configuration I - (Sample Set I, No. 1)

Particle Type	I	II	III	IV	V	VI	VII
% by number	67	13	14	5	None Found	<1	<1
size range, microns	1-120	5-40	20-50	5-50	-	-	-

System Configuration II - (Sample Set I, No. 4)

Particle Type	I	II	III	IV	V	VI	VII
% by number	54	None Found	33	None Found	None Found	4	2
size range, microns	1-200	-	1-100	-	-	5-40	5-40

Return-Line

System Configuration I - (Sample Set I, No. 2)

Particle Type	I	II	III	IV	V	VI	VII
% by number	41	40	17	<2	<2	<2	<2
size range, microns	1-300	15-300	20-50	-	-	-	-

System Configuration II - (Sample Set I, No. 5)

Particle Type	I	II	III	IV	V	VI	VII
% by number	48	30	17	<3	<3	<3	<3
size range, microns	1-350	15-350	20-120	-	-	-	-

Case Drain

System Configuration I - (Sample Set I, No. 3)

Particle Type	I	II	III	IV	V	VI	VII
% by number	73	10	9	<3	<5	<3	<3
size range, microns	1-120	15-200	15-200	-	20-100	-	-

System Configuration II - (Sample Set I, No. 6

Particle Type	I	II	III	IV	V	VI	VII
% by number	60	12	19	<3	<3	<3	<3
size range, microns	1-350	15-350	15-120	-	-	-	-

It should be noted that only highly represented particles comprising 90 to 95 percent of the distribution were analyzed. The balance of particles (5 micron or less) in a collection were scanned quickly and size and percent by number in the collection estimated.

D. Elemental Analysis of Particles In Sample Set II

Approximately 50 particles selected at random from each sample in Set II (i.e. 7 through 12) were analyzed for elemental metals composition. Metals in each particle analyzed were used as a basis to sum up the number under each metals grouping. However, some trace metals were ignored in deciding which particles to group under a particular combination of metals. For example, many particles classified as being organic contained traces of one or more metallic elements that responded to XES analysis. In these cases, the judgement of the analyst was relied upon to distinguish between organic and inorganic particulate materials.

The results of this analysis are provided in the table below.

Scientific & Laboratory Services Dept.

Grouping of Representative Collection of Particles in Samples From Set II Based on Elemental Analysis*

Sample Set II	No. 7 LHS, FC-2 (Return)	No. 8 LHS, FC-1 (Return)	No. 9 LHS, FC-2 (Pressure)	No. 10 LHS, FC-1 (Pressure)	No. 11 LHS, FC-2 (Case Drain)	No. 12 Original Condition (MIL-H-83282)
Organic	37	34	27	27	27	34
AL		1	6	5	3	
Ca			1	3		
Cl	5	2		2		
Cr		14				
Cu					1	
Fe	1	1	1		24	1
S	1	1		3		1
Si	1		2	3		2
Ti			1	2		2
Al,Cr			1			
Al,Si			2	2	1	1
Al,Ti					1	
Al, Cl, Ca					1	
Al, Cl, Si	2					
Al, Ca, K, Si			1			
Al, Ca, Fe, Si, Ti			1			
Ca, Si	2					2
Cl, Na, Si			1			1
Cl, Na			1	1		1
Cr, Cd	1					
Cr, Fe		1				
Cr, K				1		
Fe, Cl, Si			1			
Fe, Si				1		1
S, Cd, Na				1		
S, Si	1					
S, Cd	1					
S, Sb			4		1	
S, Cl, Cd		1				
S, Ca, Cl, Si				1		
Si, Mn					1	
Si, Cl			1			
Particle Count Total	52	55	51	52	59	46
% Organic Particles	74	62	53	52	46	74

*Key To Elements

Al = aluminum	K = potassium	Si = Silicon
Ca = calcium	Mn = manganese	Ti = Titanium
Cl = chlorine	Na = sodium	Sb = Antimony
Cr = chromium	Fe = Iron	Cd = Cadmium
S = sulfur		

Organic = No element heavier than sodium

E. Elemental Analysis of Black Particulate Contaminants In Sample Set II (collected March, 1985).

Sample Description	Particle Type	Elements	Comments
		Elemental Analysis	
7. LHS, FC-2 (return)	1	traces of silicon, chlorine, potassium and calcium	Particle types 1 and 3 may be agglomerates of organic and inorganic particle
	2	traces of silicon, chlorine, barium, iron, nickel, copper and zinc	
	3	traces of silicon, chlorine, calcium and nickel	---------
	4	major aluminum, chlorine and calcium	----------
8. LHS, FC-2 (case drain)	1	major silicon, traces of chlorine, potassium and nickel	----------
	2	no elements with atomic number higher than sodium	likely to be organic
	3	major iron	---------
	4	traces of silicon, chlorine, potassium and nickel	Particles types 4 and 5 may be agglomerates of organic and inorganic particles
	5	traces of chlorine and potassium	
9. LHS, FC-1 (return)	1	traces of chlorine and sulfur	Particle types 1-2 may be agglomerates of organic and inorganic particles
	2	trace of chlorine	
	3	no element with atomic number higher than sodium	
	4	major sulfur; minor aluminum and chlorine; traces of silicon, barium, chromium, iron, nickel, copper and zinc	--------
	5	major silicon; traces of chlorine, chromium and iron	-
10. LHS, FC-1	1	major chromium and iron; traces of silicon and chlorine	---------
	2	minor chromium; traces of chlorine and nickel	----------
	3	minor silicon, potassium and calcium; trace of chlorine	----------
	4	minor chromium; trace of chlorine	may be the same as particle type 2
11. LHS, FC-1	1	major aluminum and silicon; traces of chlorine, titanium, copper, zinc and cadmium	----------
	2	major aluminum	----------
	3	major chlorine and potassium; minor sulfur; traces of sodium and silicon	----------
	4	No element with atomic number higher than sodium	likely to be organic
	5	major chlorine; traces of aluminum and silicon	---------

Scientific & Laboratory Services Dept.

II. Fluid Analysis of Sample Set III Associated with 600 Hour Period of Testing (collected June, 1985)

A. Optical Particle Counting

Particle Count Data On Fluid Samples Submitted in June 1985 re. 600 Hour Test, Rockwell LHS with APM (5 micron absolute) Filtration

Fluid Sample Designation (2)	∿NAS 1638 Class (1)	No. of Particles Per Milliliter (3)					
		1-15u	5-15u	15-25u	25-50u	50-100u	>100u
1. FC-1 (No. 1)	∿Class 3	618	252	71	68	15	3.5
2. FC-1 (No. 2)	∿Class 1	45	20	7	3	2	1
3. FC-2 (No. 1)	∿Class 1	26	42	23	7	1	0.3
4. FC-2 (No. 2)	∿Class 0	45	26	13	3	1	0.4
5. Reservoir, LHS	∿Class 0	436	51	8	2	0.7	0.3

Note: 1) ∿NAS 1638 Class is determined here based on particle size range larger than 5 microns and highest particle count for each sample. See Appendix D for a chart of NAS1638 Classes.

2) APM 5 micron absolute filtration is employed in this system in pressure-line, return-line and case drain locations.

3) Particle counting - Method - SAE ARP-598A.

Results of Miscellaneous Analyses on the LHS Reservoir Fluid Sample Submitted June, 1985

A. Gravimetric Analysis (Method ARP-785)

- Membrane - 0.2um
- Solvent - Freon

Results: 2.0 mg/liter of particles

B. Total Water (Method - Karl Fischer per DIN51-777)

Results: 134 ppm H_2O

III. Results of Elemental Analysis of Seal Debris from Rockwell LHS Actuator, Submitted June, 1985

The gelatinous material was analyzed by XES and found to contain silicon, chlorine and chromium.

Discussion

The results of analysis of Sample Set I show that the "black residue" consists of at least four distinctly different particle types. These are categorized as follows:

Black Particle Type	Elemental Composition	Found Most in Filter Location
I (a)	major iron, trace chromium	Case drain
I (b)	major aluminum, minor sulfur	pressure-line
I (c)	organic	pressure-line
I (d)	major chromium	return-line

Based on the results of analysis of Sample Set II, it is estimated that organic type particles make-up a large percentage of particulate contamination (i.e. ∿50-75 percent) in all samples.

Owing to the difficulty in distinquishing one black particle from another it was considered too laborious to analyze collections of black particles to determine a percent distribution among the four particle types identified.

The results of analysis of Sample Set II show that a significant number of particles contain chlorine. The sample of actuator seal wear debris submitted for analysis in June, 1985 was analyzed for elemental composition and found to contain silicon, chlorine and chromium. Consequently, at least one source of chlorine has been identified.

The results of analysis of fluid samples in Set II show that the APM (5 micron absolute) filters are performing as expected to maintain cleanliness of NAS 1638 Class 1, or cleaner.

Conclusions

1. The results of analysis shows that "black residue" is a composite of four particle types. Organic black particles (A) are most numerous (i.e. contain no element heavier than sodium). Aluminum containing black particles (B) are not distinquishable from organic particles when examined microscopically. Organic and aluminum containing particles were found in greatest number on the pressure-line filter. Iron containing particles (C) and Chromium containing particles (D) have a black appearance when viewed under an oblique (<90°) light source. Therefore, it is estimated that the later two particle types (C and D) contribute significantly to the black appearance of filtered particles. The case drain filters held a predominance of iron containing particles. The return-line filters held a large number of chromium containing particles.

Scientific & Laboratory Services Dept.

2. The results of particle counting fluid samples from the LHS FC-1 and FC-2 hydraulic loops show that APM, 5 micron absolute filtration is maintaining particulate contamination at levels represented by NAS 1638 Class 1 or cleaner. It is believed that sample 1 (i.e. FC-1, No. 1) is not representative of the rest of the samples collected. Excess contaminants may have been introduced into sample 1 during the sampling procedure.

Daniel R. Uhr, Jr.

Daniel R. Uhr, Jr., Ph.D.
Staff Scientist

DU/ds

SYSTEM CONFIGURATION I

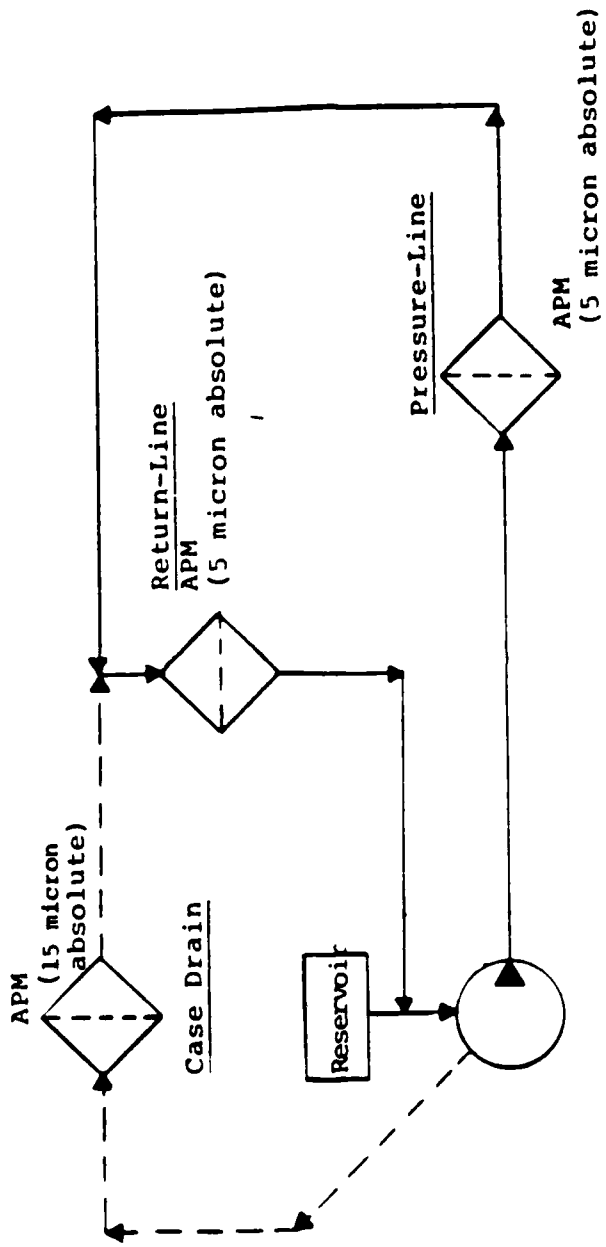


FIGURE 1

SYSTEM CONFIGURATION II

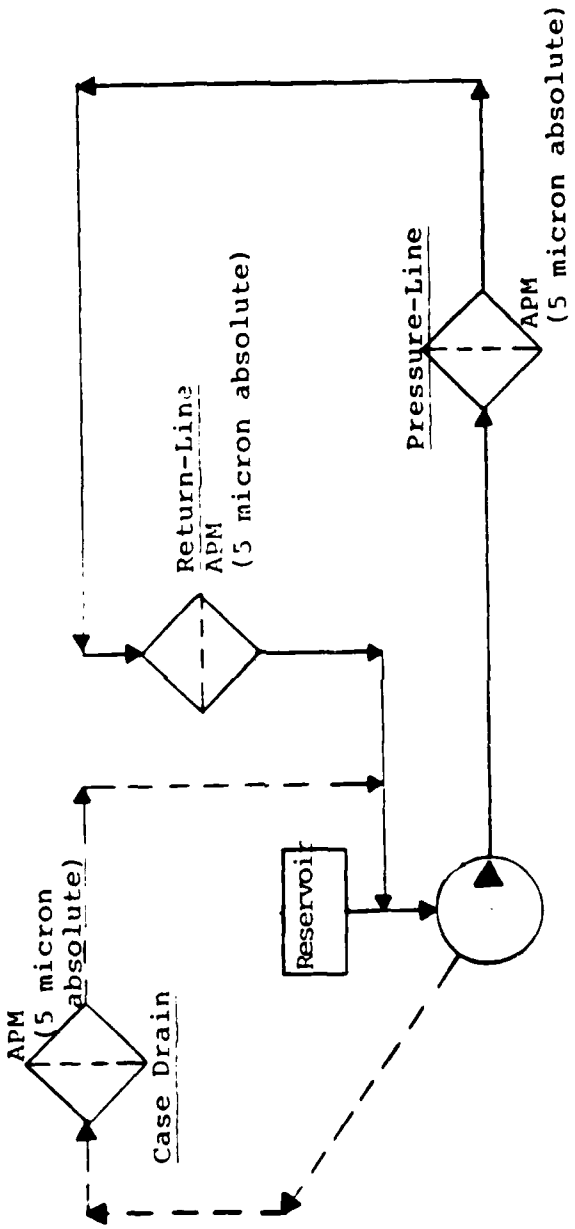


FIGURE 2

Appendix A

Analytical Procedures

All analytical procedures were performed in the facilities of the Scientific and Laboratory Services (SLS) Department of Pall Corporation in Glen Cove, New York.

1. Optical Particle Counting

The procedure employed for counting contaminant particles in fluid samples parallels the industry-wide standard of S.A.E. ARP 598A, "The Determination of Particulate Contamination in Liquids by the Particle Count Method". A volume of sample fluid is drawn through a gridded, 0.8 analysis membrane disc of 25mm diameter (2.8 cm^2 effective filtration area).

When a gel and/or a precipitate is present which interferes with particle counting the analysis membrane is made transparent utilizing type A immersion oil to enable visualizing the contaminant particles. Occasionally gels prevent counting in the 1-5 m range. In these cases the reported number of particles in this range are extrapolated using the SAE-ISO code purpose.

2. Photomicrographs

Color photomicrographs of the contaminant collected on a representative area of the analytical membrane surface are obtained at 100X magnification. A Vanox polarizing microscope with camera attachment is employed for this purpose.

3. Elemental Analysis of Contaminants

The contaminant particles are examined with a Vanox polarizing microscope. As appropriate, shape, coloration, crystallinity and refractive index are observed.

Semiquantitative analysis of the contaminant material is performed with an ISI Scanning Electron Microscope equipped with a Kevex 5100 X-ray Microprobe Analyzer. This spectrometer detects chemical elements in contaminant material with atomic numbers equal to or greater than eleven (i.e. sodium, silicon, iron). It does not permit detection of those lighter chemical elements commonly found in organic material, such as carbon, oxygen, and nitrogen.

Appendix B

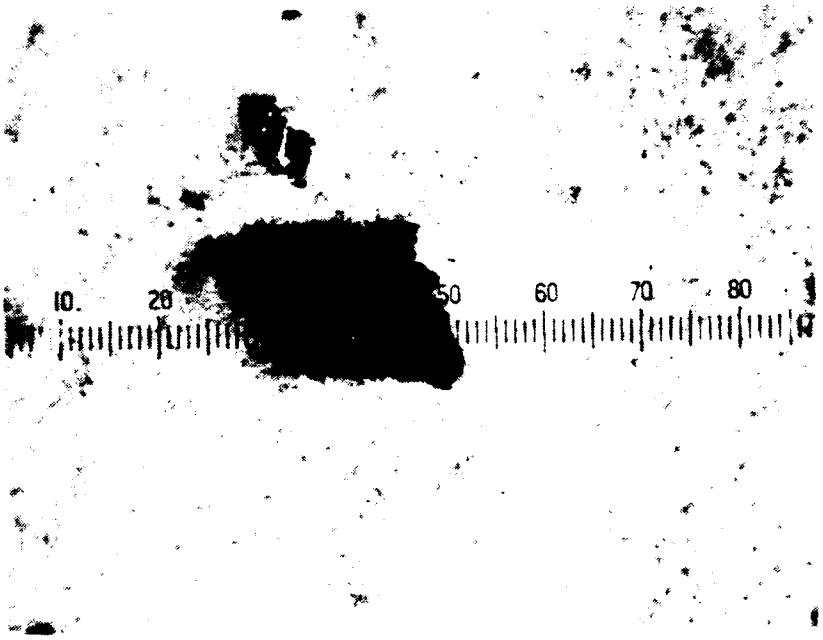
Photomicrographs of Contaminant Particle Types From Sample Set I (collected January and March, 1984).

Note: The particles chosen to represent each of the seven particle types filtered by APM (5 micron absolute) filters in the LHS are the largest that could be found in representative collections of particles from respective samples. Consequently, no conclusions regarding actual size distribution can be drawn from data presented here in Appendix B.

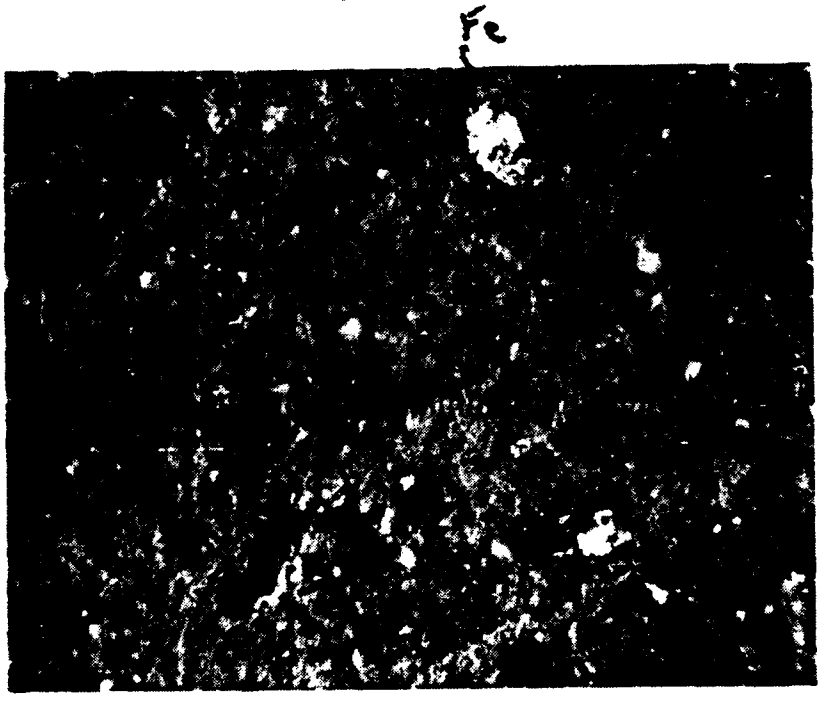
Black Particle Type - I(a) (Cont'd)

The "black residue" particles labeled I(a) contained major iron with a trace of chromium and was found in greatest amount on the APM (5 micron absolute) filter in the case-drain location.

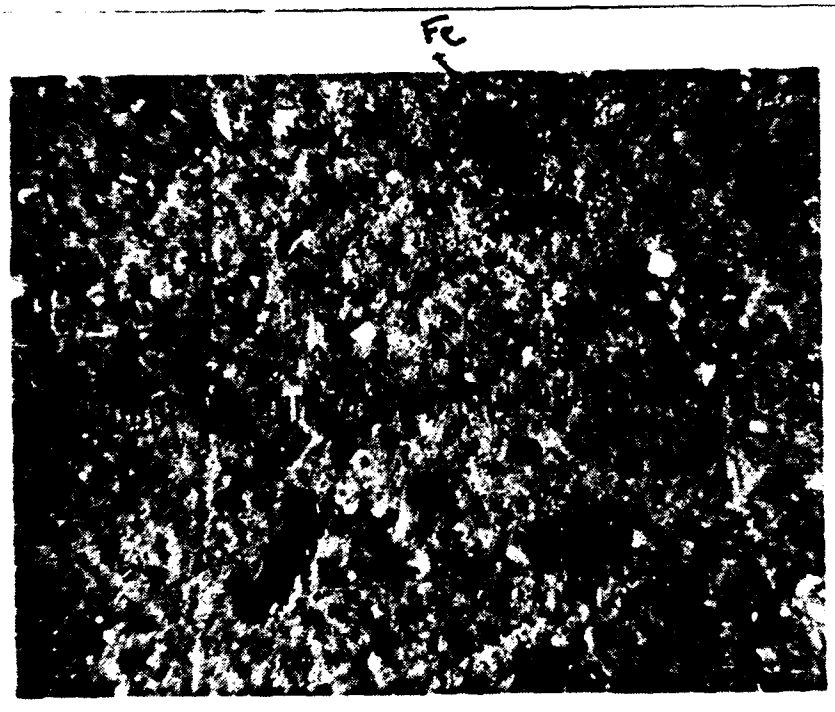
Oblique Lighting
(200X - Magnification
Reticle: 6.8 microns/div)



The Type I(a) particle depicted above looked black regardless of the angle of lighting. This particle is photomicrographed at 200X to show the surface roughness that may explain its dark appearance under all angles of incident of light.



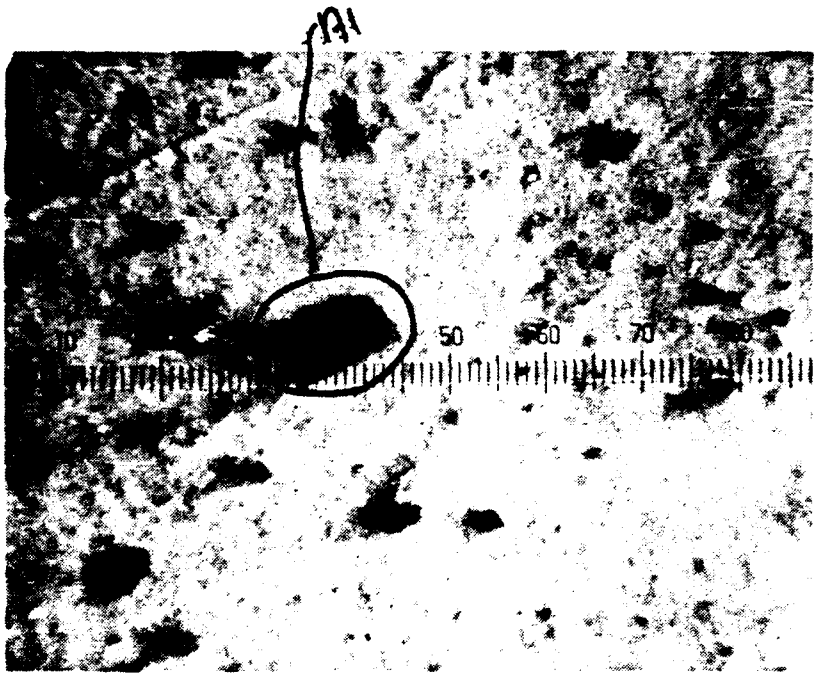
Perpendicular Lighting
(100X-Magnification
Reticle: 13.6 microns/div)



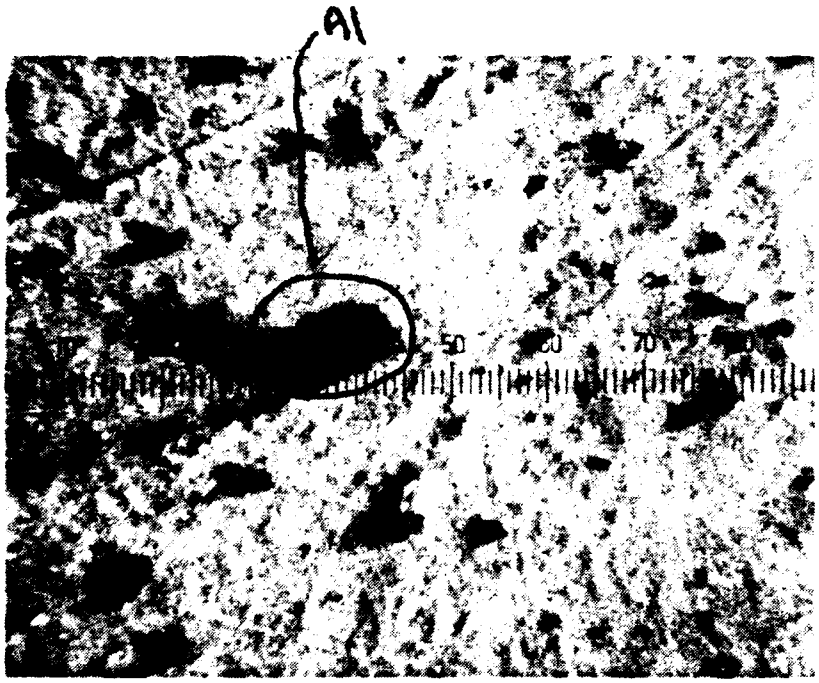
Oblique Lighting
(100X Magnification)
Reticle: 13.6 microns/div)

Black Particle Type - I (a)

The "black residue" particle labeled I(a) contained major iron with a trace of chromium and was filtered in greatest number by the APM (5 micron absolute) filter in the case drain location. Changing the angle of lighting made some of these particles appear reflective but did not change the appearance of others.



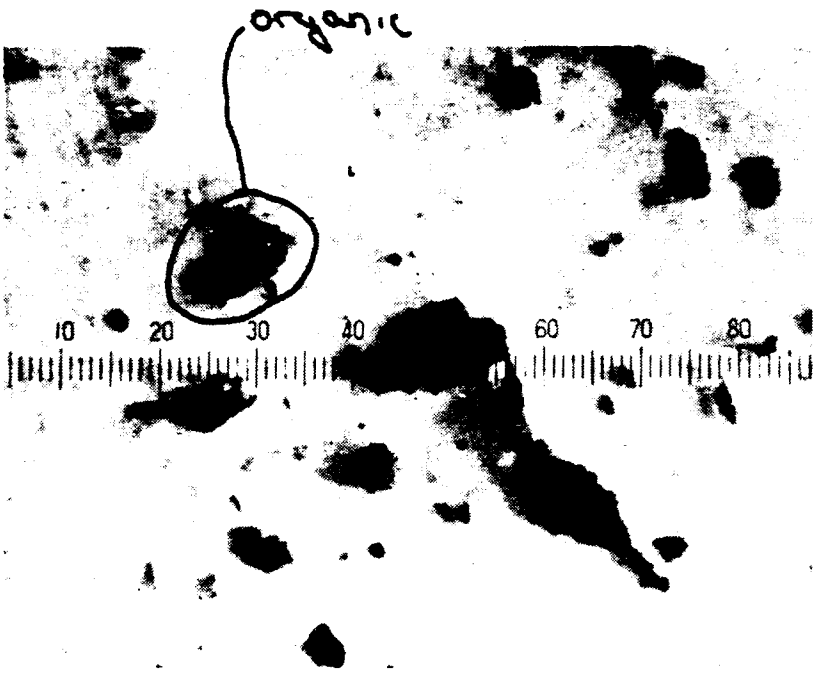
Perpendicular Lighting
(100X-Magnification
Reticle: 13.6 microns/div)



Oblique Lighting
(100X Magnification)
Reticle: 13.6 microns/div)

Black Particle Type - I (b)

The "black residue" particle labeled 1(b) contained major aluminum, minor sulfur with a trace of chromium and zinc. These particles were filtered mainly by APM (3 micron absolute) filters in the LHS filter pressure-line location. These particles appeared black under all angles of of the light source used to photomicrograph these particles.

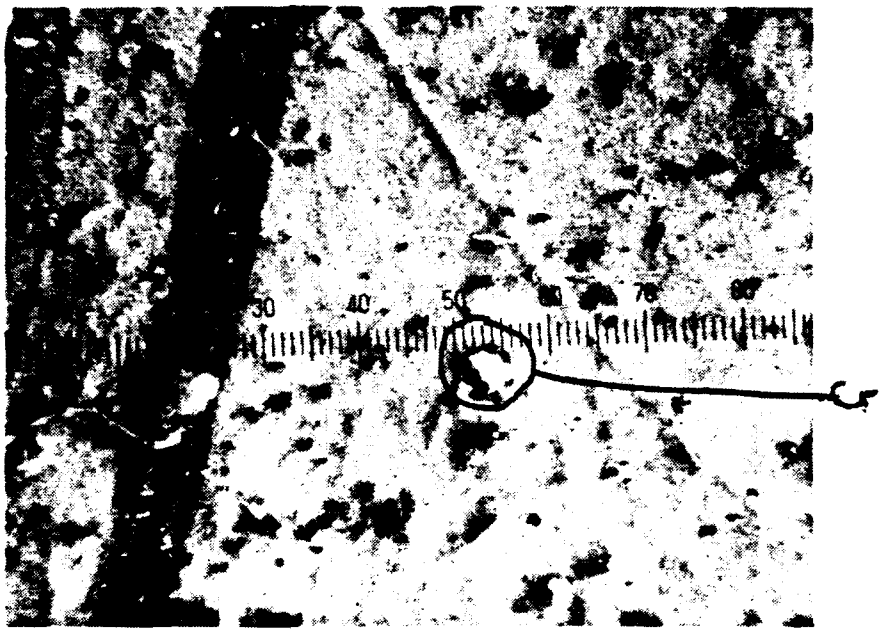


Perpendicular Lighting
(100X-Magnification
Reticle: 13.6 microns/div)

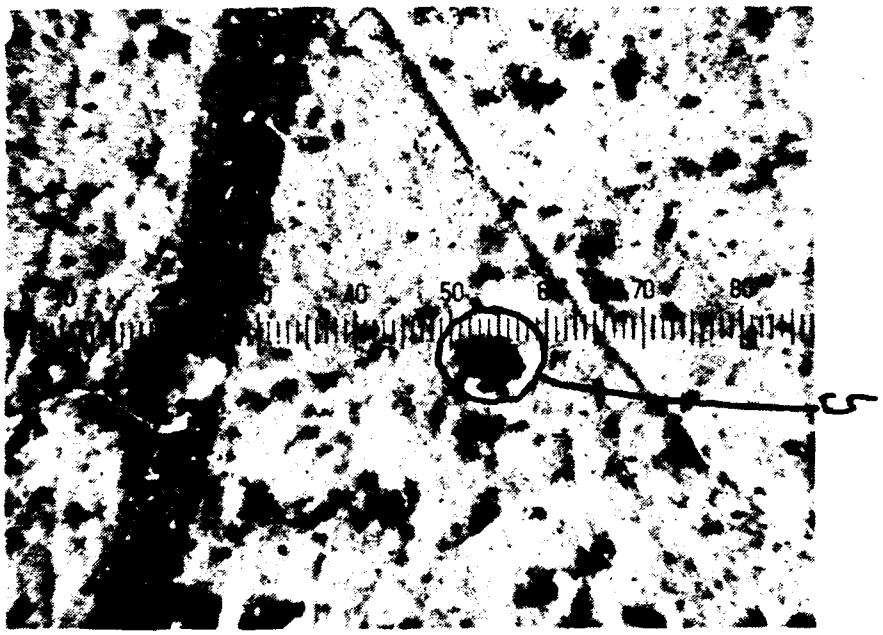
Oblique Lighting
(100X Magnification)
Reticle: 13.6 microns/div)

Black Particle Type - I (c)

The "black residue" particle labeled I(c) contained only a trace of aluminum. It is believed this particle may be composed largely of organic materials (i.e. carbon; oxygen; nitrogen). These particles were filtered mainly by AFM (5 micron absolute) filters in the pressure-line of the IHS. These particles appeared black under all angles of the light source.



Perpendicular Lighting
(100X-Magnification
Reticle: 13.6 microns/div)



Oblique Lighting
(100X Magnification)
Reticle:- 13.6 microns/div)

Black Particle Type - I (d)

The "black residue" particle labeled I(d) contained major chromium and was filtered in greatest number by the APM (5 micron absolute) filter in the return-line location. Changing the lighting angle showed a reflective surface that varied depending on the orientation of the particle on the analytical membrane.

Particle Type II

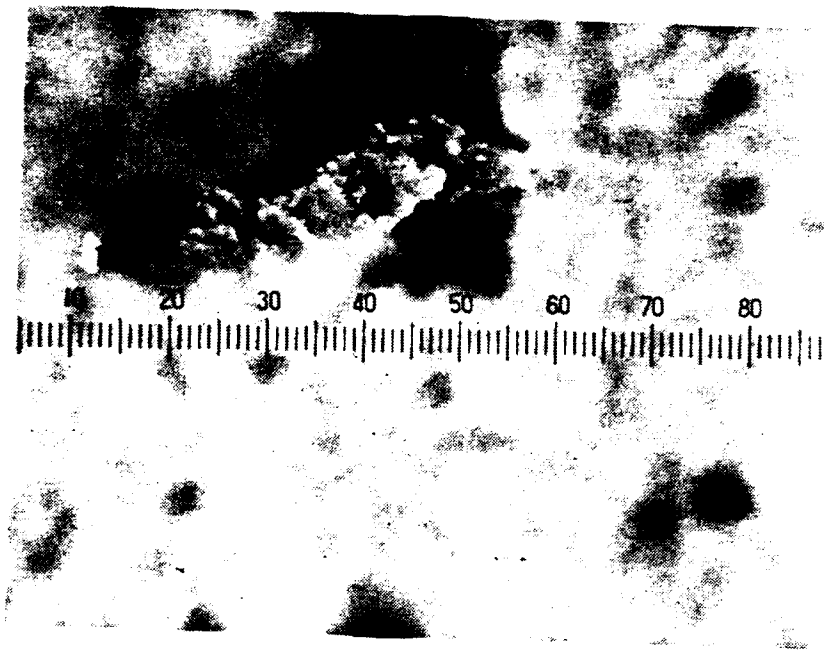
The translucent particles depicted in this photomicrograph were representative of Type II particles filtered mainly by AFM (5 micron absolute) filters in the pressure-line and return-line locations. The only element contained in this particle which had an atomic number higher than sodium was sulfur (in minor amounts).

The magnification is 100X. The Reticle measures 13.6 microns per division.

Particle Type III

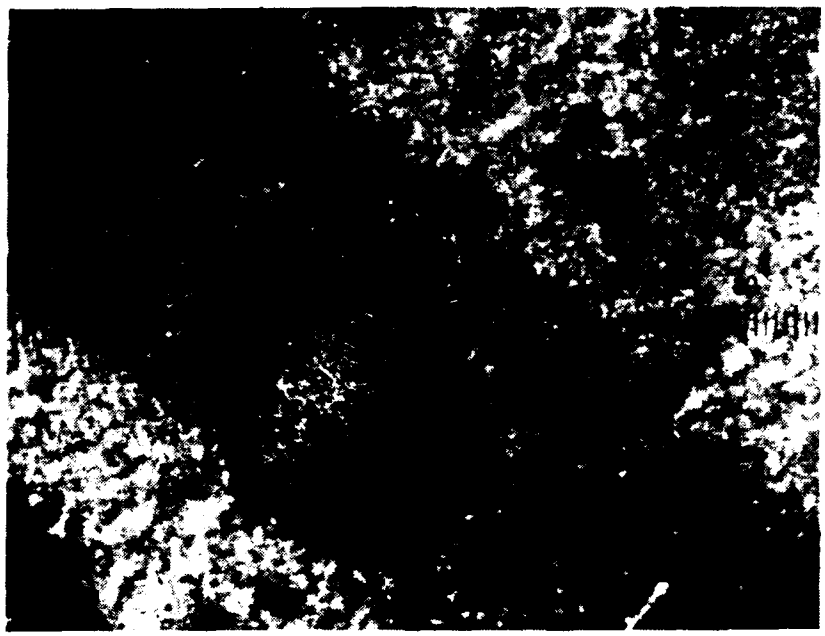
The white, glassy looking particle depicted in this photomicrograph is representative of Type II particles filtered by APM (5 microns absolute) filters in pressure-line, return-line and case-drain locations.

The results of elemental analysis of this particle type showed the presence of aluminum, silicon and sulfur in minor amount. The magnification is 200X. The reticle measures 6.5 microns per division.



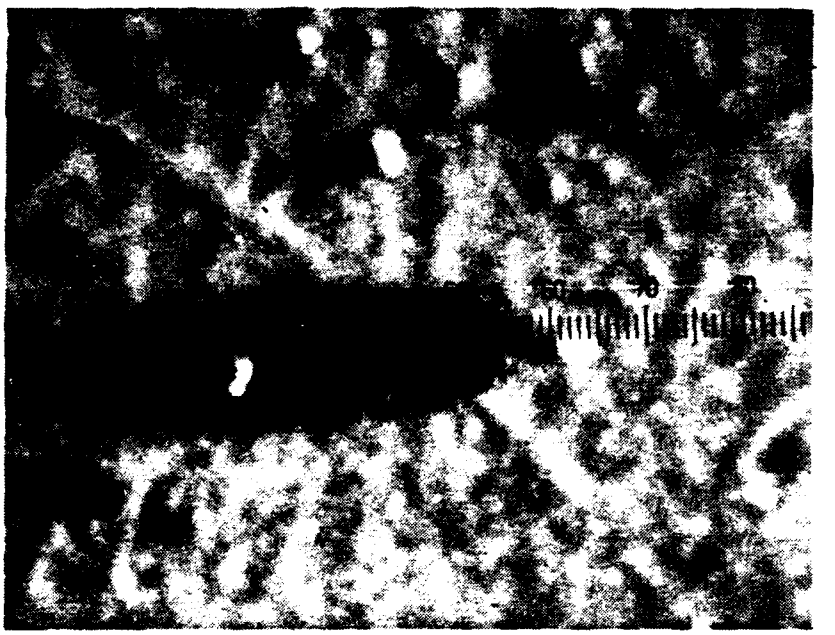
Particle Type IV

The light yellow/green particle depicted in this photomicrograph is representative of Type IV particles filtered by AFM (5 micron absolute) filters in pressure-line, return-line and case-drain locations. The elemental composition of this type of particle includes major silicon, minor magnesium and titanium and traces of chlorine, copper and iron. The Magnification is 200X. The reticle measures 0.5 microns per division.



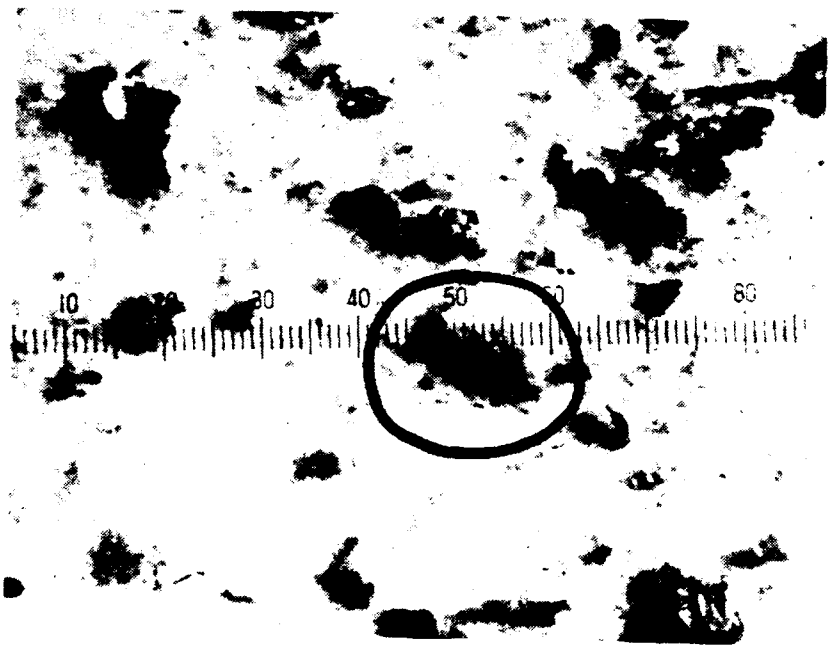
Particle Type V

The bronze looking particle depicted in this photomicrograph was representative of Type V particles filtered by APM (5 micron absolute) filters primarily in the case-drain location. Some were also found in the return-line location. The particle type contained major copper, minor tin and traces of aluminum, silicon and sulfur. The Magnification is 400X. The reticle measures 3.4 microns per division.



Particle Type VI

The orange colored particle depicted in this photomicrograph is representative of Type VI particles filtered by APM (5 micron absolute) filter in pressure-line, return-line and case-drain locations. The elemental composition of Type VI particles included minor sulfur and calcium; and traces of aluminium, silicon and sodium. The magnification is 200X. The reticle measures 6.8 microns per division.



Particle Type VII

The particle circled in this photomicrograph is representative of Type VII particles filtered by APM (5 micron absolute) filters is pressure-line, return-line and case-drain locations.

These particles contain major sulfur, minor aluminum, silicon and calcium and traces of cadmium and potassium. The magnification is 200X. The reticle measures 0.5 microns per division.

APPENDIX C

CHRONOLOGICAL INFORMATION COVERING

THE "BLACK RESIDUE"

1975

2.8 FLUID

LHS development tests, begun in 1968, were conducted initially with MIL-H-5606 hydraulic fluid, Reference 2. This fluid exhibited poor shear stability at high operating pressure levels due to polymeric additives used to improve its viscosity-temperature coefficient. MIL-H-27601 was employed for the tests reported in References 4 and 5 because of its excellent shear stability. MIL-H-27601 is a high temperature hydraulic fluid and is very viscous at low temperatures. MIL-H-83282 was evaluated in tests reported in References 6 and 12. This fluid is rated for use at temperatures from -40 to +400°F, is shear stable, and compatible with MIL-H-5606. The endurance test was conducted using MIL-H-83282 identified as "Experimental Less Flammable Synthetic Hydrocarbon Fluid" XRM-206-A (MLO 68-5), batch #186. The fluid was supplied by NADC and was manufactured by Mobil Oil Corporation, New York, N.Y.

Test Procedure

Two parameters were evaluated: viscosity and contamination. Fluid samples were taken after completion of 0, 16, 50, and 100 hours of endurance testing. Viscosity was measured using standard Canon-Fenske pipet equipment at +100°F. Fluid contamination was determined with a Hiac Automatic Particle Counter M/N PC204.

Test Results

Viscosity and contamination of fluid samples taken during the test are listed below:

Endurance Hours Completed	Fluid Viscosity, Centistokes	Fluid Contamination, Number of Particles per 100 ml			
		Micron Size Range			
		10-25	25-50	50-100	Over 100
0	17.48	6,665	0	0	0
16	17.59	22,960	45	0	0
50	17.70	11,985	10	0	0
100	17.71	49,265	20	0	0

System filter elements were visually examined at the 50 hour check point. All four elements were found to contain a black residue. Nominal quantities of larger size contaminates were also observed. The elements were cleaned ultrasonically and reinstalled in the system. The filters were inspected after completion of endurance cycling; the black residue was again observed. The return line filter element was washed with a solvent to remove the contaminates and the collected residue was examined under a microscope. The majority of the particles were less than one micron (0.00004 in.) in size; some aggregation was present. The composition and source of the particles was not established. The residue is discussed in Section 3.2.

The size of particulate matter is measured in terms of microns. One micron equals 0.0000394 inch. The largest dimension of a particle determines its size. Contamination level is based on particle size distribution and concentration in a 100 ml sample of fluid. Care must be exercised when taking a sample to prevent introduction of additional contaminants.

Six classes of particle contamination levels have been established by the Navy. Class 5 is the maximum acceptable level for hydraulic systems in Naval aircraft. The endurance test results are compared to Class 5 allowables below:

Endurance Hours Completed	Fluid Contamination Number of Particles per 100 ml — Micron Size Range			
	10-25	25-50	50-100	Over 100
Class 5 (Ref.)	21,400	3,130	430	41
0	6,665	0	0	0
16	22,960	45	0	0
50	11,985	10	0	0
100	49,265	20	0	0

The major contributor to the count observed in the 10-25 micron range was believed to be particles making up the black residue noted in Section 2.8. This level of contamination can accelerate loading of filter elements. The test system has four filters with stainless steel wire mesh elements rated at 10 microns nominal, 25 microns absolute. The contamination data above indicates the filters were performing satisfactorily.

Chemical qualitative analysis tests were performed to determine primary constituents of the black residue. The residue was found to be approximately 30% ferrous material (by weight). The remaining 70% was not identified. An infrared spectrograph was used to determine if organic compounds were present; none were indicated. Pure carbon, however, cannot be detected by the spectrograph. The source of the black particles was not established.

The endurance test was run with MIL-H-83282 fluid from batch #186 - - the same batch used for the evaluation tests reported in Reference 12. No black residue was observed in the Reference 12 tests. Effects of the residue on the endurance test were considered minimal in view of the excellent results, however further study in this area is warranted.

1978

During the first 100 hours of testing, Reference 7, a black residue was observed on filter elements in the system. Particles making up the residue were extremely small--less than one micron in size. The source and composition of the residue was not determined. Subsequently, information was obtained (Reference 12) which indicated the residue might be the result of fluid degradation caused by (1) oxidation, (2) thermal cracking, or (3) nitration, any of which could lead to the formation of insoluble deposits.

The reservoir was pressurized with nitrogen at 30 psig during the first 100 hours of endurance cycling. The second 100 hours of cycling were conducted with the reservoir pressurized by an inert gas--argon. This was done to minimize the possibility of oxidation/nitration reactions in the fluid.

High particle counts were observed in the 10-25 micron range based on fluid contamination checks made during the first 100 hours of cycling. All four filters in the system were rated for 10 microns (nominal). One filter element was changed for the second 100 hours of testing. The pump case drain element was replaced with a 5 micron absolute filter to lower the particle count in the 10-25 micron range. The three 10 micron (nominal) filter elements were cleaned ultrasonically prior to beginning the extended endurance test.

It should be noted that automatic particle counting equipment employed during the first 100 hours of testing was not capable of sensing contamination below 10-25 microns in size. The equipment has since been upgraded and currently counts particles in the 5 to 15 micron range.

Fluid viscosity and contamination were the parameters used to evaluate fluid performance. A decrease in viscosity would indicate poor shear stability; high particle counts in the 25 to 100 micron range would be evidence of poor lubricity and component wear. Viscosity and contamination of fluid samples taken during the test are listed below:

Endurance Hours Completed	Fluid Viscosity, Centistokes	Fluid Contamination Number of Particles per 100 ml Micron Size Range			
		10-25	25-50	50-100	100+
*0	17.48	6,665	0	0	0
*50	17.70	11,985	10	0	0
*100	17.71	49.265	20	0	0

*Data from Reference 7

		Micron Size Range				
		5-15	15-25	25-50	50-100	100+
100	16.85	95145	462	139	15	60
150	--	39318	315	18	1	0
200	16.88	3637	37	10	8	1
Class 5 System		87000	21400	3130	430	41
Class 1 System		4600	1340	210	28	3

3.0 DISCUSSION

The most critical components in an aircraft hydraulic system are pumps and seals--failure of either can result in serious circumstances. Testing conducted in the LHS program has demonstrated that current state-of-the-art hardware designed for use at 8000 psi performs well.

Pump performance in the extended endurance test was completely satisfactory. There were no malfunctions, discrepancies, or trends indicating possible problems. Pump wear was normal. Future LHS pumps will be new and original designs rather than modified existing designs. References 3 and 8. This will result in a more optimized unit.

Seal performance in the 200 hour endurance test was better than expected. Standard O-rings and TFE backups were found to be satisfactory for use in static applications (diametral, face, and boss type seals). This was a major determination that will minimize costs and greatly simplify conversion from 3000 to 8000 psi systems. Off-the-shelf components performed well in dynamic seal applications. This, again, was an important finding since special piston and rod seals will not have to be developed for 8000 psi actuators.

A black residue was observed on the filter elements at the conclusion of testing; the residue was also noted during the first 100 hours of cycling, Reference 7. The tiny black particles (<1 micron in size) were suspected of being caused by fluid oxidation, thermal cracking, or nitration, Reference 12. In an attempt to impede formation of the particles, the system reservoir was pressurized with argon during the extended endurance test. (Nitrogen was used for the first 100 hours of testing, Reference 7.) Argon did not eliminate the residue, but did appear to reduce the quantity of particles developed.

A sample of the black residue was examined by the Rockwell International Science Center in Thousand Oaks, California. An Electron Spectroscope for Chemical Analysis (ESCA) was employed to determine constituents in the residue. The results were:

Carbon (as graphite)	95%
Carbon bonded to oxygen (CO)	4%
Sulphur	less than 0.1% *
Silicon	less than 0.1% *
Iron	less than 0.1% *
Nitrogen	less than 0.1% *
Fluorocarbons	less than 0.1% *

* Amount below instrument sensitivity

(1981)

Naval Air Engineering Center
Advanced Technology Section
Lakehurst, NJ 08733
20 February 1981

North American Aircraft Division
Rockwell International
4300 E. Fifth Avenue
PO Box 1259 (Dept 871)
Columbus, OH 43216

Bill/Steve:

We completed the analysis of the two LHS fluid samples, the results of which are attached. It is my guess that there is a possible bearing problem, judging from the debris found in the case drain sample.

I have tried to describe the sources of the various debris so that you can draw your own opinions based on your knowledge of the system. The information about the debris comes from our "Wear Particle Atlas". Bill has a copy.

If we can be of any more help, just call.

Sincerely,

Pete

P. V. CIEKURS, P.E.
Project Engineer

PVC:flt
Encl: Summary of Analysis of Wear Debris from LHS Impulse Lab Tests

20 February 1981

SUMMARY OF ANALYSIS OF WEAR DEBRIS TAKEN FROM LHS IMPULSE LAB TESTS

Prepared by P. Ciekurs and T. Kelly

1. LHS RESEVOIR RETURN LINE FILTER SAMPLE

a. Spectrometric Analysis (ppm)

Fe - 2.2	Mg - 0.5	Na - 4.6	
Ni - 0.3	Si - 4.3	Sn - 4.5	Zn - 4.7
B - 0.2	Mo - 1.9	V - 0.6	

b. Optical Microscope Observations

Wear modes observed were: rubbing or normal wear, some scuffing, some cutting, and a substantial amount of fatigue chunks. Also in evidence were several spherical particles. Majority of the debris was steel or steel alloy. Typical spheres measured 7 - 10μm in diameter and fatigue particles measured 8μm x 4μm.

2. LHS PUMP CASE DRAIN FILTER SAMPLE

a. Spectrometric Analysis (ppm)

Fe - 1.7	Mg - 0.3	Na - 0.9
Ni - 0.6	Si - 1.0	Sn - 5.8
B - 0.3	Cd - 3.5	Mo - 2.4
V - 1.1	Zn - 4.1	

b. Optical Microscope Observations

Debris present consisted of rubbing wear with equal amounts of laminar and fatigue particles in majority. Typical dimensions of laminar were 20μm x 15μm and fatigue were 7μm x 8μm. One sphere measuring 4μm was observed.

3. DISCUSSION

Generally rubbing wear is associated with normal sliding wear, although substantial increases in quantity over a period of time are an indicator of possible abnormality. Since these are discreet samples, the rubbing debris is of little significance. Scuffing wear is typically generated by gears and is due to a combined sliding and rolling situation. Scuffing is caused by excessive speed and/or load which tends to generate heat and breaks down the lubricant film. This heat and lack of lubricant usually results in oxidized particles, none of which were found in these samples. The cutting wear is typically generated by a harder contaminent coming between two sliding surfaces. The presence of this could possibly be due to the silicon (usually sand) found in the first sample. The fatigue particles are definitely abnormal and the result of microspalling of the wear surface. This type of debris is typically associated with rolling contact fatigue found in bearings. Spherical particles are also associated with fatigue but the quantities found in these samples were small compared to the large quantities typically found in fatigue situations. Another source of spheres is cavitation or from outside sources such as grinding or welding. Typically rolling fatigue generates few spheres over 3μm while grinding and erosion is frequently over 10um. Laminar particles are also associated with rolling contacts and are formed by other debris passing through the contact. The presence of both laminar and fatigue particles in the case drain samples could indicate a bearing problem in the pump.

12450

1983

DEPARTMENT OF THE AIR FORCE
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES (AFSC)
WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

6 JUL 1983

REPLY TO ATTN OF: MLBT (Ms. L. Gschwender)

SUBJECT: Analysis of Fifty Hour Case Drain Precipitate from 8000psi MIL-H-83282 Tests

TO: Rockwell International
Attn: Mr. R.K. Haning
Columbus Aircraft Division
4300 E. Fifth Avenue
P.O. Box 1259
Columbus, OH 43216

1. Subject filtered material from a Rockwell 8000psi test was analyzed. The precipitate was found to be only 1.5% inorganic. This inorganic part consisted of typical hydraulic system component materials (Fe, Zn, Cr, Sn, Al etc.). Shiny metallic particles were visible to the unaided eye.

2. The organic part of the precipitate was found to contain an aliphatic ester (the rubber swell additive probably) likely with some of the hydrocarbon base oil. The largely organic nature of the precipitate indicates a fluid decomposition and/or interaction of some kind. Further conclusions would be speculative at this time.

3. If there are questions or comments, please call.

Carl E. Snyder Jr

CARL E. SNYDER, JR.
Fluids, Lubricants and Elastomers Branch
Nonmetallic Materials Division

204500

NADC-89006-60

1984

NAVAL AIR DEVELOPMENT CENTER
AIRCRAFT AND CREW SYSTEMS TECHNOLOGY DIRECTORATE
WARMINSTER, PA 18974

6061
8 MAR 1984

MEMORANDUM

From: 60613, J. Dever
To: 6061, J. Ohlson

Subj: Analysis of Black Contaminants on Hydraulic Filter Element (M8815/6-6) from LHS Simulator at Rockwell

Ref: (a) NADC 77218-30, LHS Extended Endurance Test of 30 Sep 1978

1. Objective

A 5 micron absolute filter element (M8815/6-6) removed from the pump case drain line of the LHS simulator at Rockwell after 100 hours was analyzed to determine the nature of the black contaminant that has been observed on the surface and which caused the filter media to load during the course of the LHS testing at Rockwell.

2. Procedure

The filter element was put into the Degree of Filtration Test Stand and reverse flushed with MIL-H-5606 fluid, approximating a 15 GPM flow rate, so that the fluid flowed through the element from inside to outside (reverse of normal flow) in order to detach as many contaminant particles from the filter media as possible. The total fluid flowing through the element (2000 ML), with the contaminant suspended was collected in a beaker. The color of the fluid in the beaker was black.

The residue of black particles which settled in the bottom of the beaker was analyzed in our laboratory. An emission spectrographic analysis did indicate the probable presence of carbon in the residue. This was further substantiated by a thermal test.

The sample of black residue weighed 11.5 mg. It was heated for approximately 2 hours at 1400°F (700°C). The amount of residue that remained was 4.5 mg resulting in a weight loss of 7.0 mg or a 60% loss. A certain portion of this weight loss was probably due to oxidation of the carbon. The unanswered question is where is this carbon coming from, which was also identified on samples from a previous test program analyzed by the Rockwell Science Center. Their analysis at that time indicated that it was carbon 95% (as graphite) (reference (a)).

550 hour pump endurance test recently completed at NADC on the Abex 8000 PSI pump which had three filters in the system (case drain, pressure and return) and which was subjected to temperatures similar to the simulator temperatures, i.e., 240°F in case drain and approximately 170°F in the pump outlet. An examination of these filters does not indicate the presence of a black residue caused by small black particles.

They also did not show a premature loading as evidenced with the Simulator filter. There was no report of this condition by Grumman or Vought in their 8000 PSI testing with the Abex pumps at similar temperatures. It could therefore be isolated to conditions in either the Vickers pump or the simulator system itself.

The spectrographic analysis of the residue also revealed the following elements:

Al	Si	Sb	Sn	Fe	Bi	B	P	Pb	Ag	Zn	V
Vs	Vs	Vs	s	s	m	m	m	w	w	w	tr

Bi	Ba
tr	tr

- Order of decreasing intensity

Vs - Very strong
s - strong
m - medium
w - weak
tr - trace

Samples of Rockwell MIL-H-83282 hydraulic fluid tested to the various specification parameters showed the following results:

Property	Specification Requirement	Unused Fluid	100 Hr. Sample
Viscosity cs			
40°C	14.0 (Min)	14.04	13.84
100°C	3.5 (Min)	3.41	3.36
4-Ball Wear Scar Dia (mm) 75°C, 1200 RPM, 40 Kg	0.65 (Max)	0.56	0.51
Thermal Stability 204°C, 100 hr. Δ Vis. @ 40°C	± 5%	0	8.6%
Particle Count (Microns)			
5-15	10,000 (Max)	1683	---
16-25	1,000 (Max)	61	---
26-50	150 (Max)	18	---
51-100	20 (Max)	8	---
Over 100	5 (Max)	1	---

An analysis of the 100 hour fluid sample does not indicate a drastic change in the fluid properties. It is not apparent and NADC does not

6061
8 Mar 1984

believe that this condition is caused by a fluid breakdown. A closer look into Vickers pump or the Simulator components would be warranted.

J. Dever

J. DEVER

Prepared by D. R. Uhr, Jr., Ph.D.
Staff Scientist
PALL CORPORATION
February 12, 1985

NAS 1638-AEROSPACE INDUSTRIES ASSOCIATION OF AMERICA, INC.
CLEANLINESS REQUIREMENTS OF PARTS USED IN HYDRAULIC SYSTEMS
CLASSES (MAXIMUM CONTAMINATION LIMITS)

SIZE RANGE, MICRONS	00	0	1	2	3	4	5	6	7	8	9	10	11	12
Current NAS 1638 Table per 100 mls														
2-5*	625	1250	2500	5000	10000	20000	40000	80000	160000	320000	640000	1280000	2560000	5120000
5-15	125	250	500	1000	2000	4000	8000	16000	32000	64000	128000	256000	512000	1024000
15-25	22	44	89	178	356	712	1425	2850	5700	11400	22800	45600	91200	182000
25-50	4	8	16	32	63	126	253	506	1012	2025	4050	8100	16200	32400
50-100	1	2	3	6	11	22	45	90	180	360	720	1440	2880	5760
>100	0	0	1	1	2	4	8	16	32	64	120	256	512	1024
Current NAS 1638 Table recalculated as greater than size per 100 mls														
>2	777	1554	3109	6217	12432	24864	48731	99462	198924	397849	795690	1591396	3182792	6365184
>5	152	304	609	1217	2432	4864	8731	19462	38924	77849	155690	311396	622792	1245184
>15	27	54	109	217	432	864	731	3462	6924	13849	27690	55396	110792	221184
>25	5	10	20	39	76	152	306	612	1224	2449	4890	9796	19592	39184
>50	1	2	4	7	13	26	53	106	212	424	840	1696	3392	6784
>100	0	0	1	1	2	4	8	16	32	64	120	256	512	1024

*proposed

SIZE RANGE, MICRONS	00	0	1	2	3	4	5	6	7	8	9	10	11	12
Current NAS 1638 Table recalculated as greater than size per one (1) ml														
>2	7.77	15.54	31.09	62.17	124.32	248.64	487.31	994.62	1989.24	3978.49	7956.90	159131.96	31827.92	63651.84
>5	1.52	3.04	6.09	12.17	24.32	48.64	87.31	194.62	389.24	778.49	1556.90	3113.96	6227.92	12451.84
>15	0.27	0.54	1.09	2.17	4.32	8.64	7.31	34.62	69.24	138.49	276.90	553.96	1107.92	2211.84
>25	0.05	0.10	0.20	0.39	0.76	1.52	3.06	6.12	12.24	24.49	48.90	97.96	195.92	391.84
>50	0.01	0.02	0.04	0.07	0.13	0.26	0.53	1.06	2.12	4.24	8.40	16.96	33.92	67.84
>100	0	0	0.01	0.01	0.02	0.04	0.08	0.16	0.32	0.64	1.20	2.56	5.12	10.24

APPENDIX F

RELIABILITY AND MAINTAINABILITY

ANALYSIS UPDATE OF

HYDRAULIC SYSTEM DATA

ROCKWELL INTERNATIONAL
NORTH AMERICAN AIRCRAFT
4300 East Fifth Avenue
P. O. Box 1259
Columbus, Ohio 43216

4 May 1988

FINAL REPORT FOR PERIOD JANUARY 1988 TO MAY 1988

Approved for Public Release
Distribution Unlimited

Prepared for

Eagle-Technology
2300 South Ninth Street
Arlington, VA 22204

Naval Air Development Center
Warminster, PA 18974

UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

REPORT SECURITY CLASSIFICATION Unclassified	1b. RESTRICTIVE MARKINGS N/A
2a. SECURITY CLASSIFICATION AUTHORITY	3. DISTRIBUTION/AVAILABILITY OF REPORT Approved for Public Release; Distribution Unlimited
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE	
4. PERFORMING ORGANIZATION REPORT NUMBER(S) NA-88-0010	5. MONITORING ORGANIZATION REPORT NUMBER(S) -
6a. NAME OF PERFORMING ORGANIZATION Rockwell International North American Aircraft 6b. OFFICE SYMBOL (If applicable) --	7a. NAME OF MONITORING ORGANIZATION Naval Air Development Center (6061)
6c. ADDRESS (City, State and ZIP Code) P. O. Box 1259 Columbus, OH 43216-1259	7b. ADDRESS (City, State and ZIP Code) Warminster, PA 18974
8a. NAME OF FUNDING/SPONSORING ORGANIZATION Eagle-Technology 8b. OFFICE SYMBOL (If applicable)	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER Eagle Technology P.O. 006318-00
8c. ADDRESS (City, State and ZIP Code) 2300 South Ninth Street Arlington, VA 22204	10. SOURCE OF FUNDING NOS PROGRAM ELEMENT NO / PROJECT NO / TASK NO / WORK UNIT NO
11. TITLE (Include Security Classification) Reliability and Maintainability Analysis Update of Hydraulic System Data	
12. PERSONAL AUTHOR(S) Haning, Robert K. and Kauffman, Ellis E.	
13a. TYPE OF REPORT Final 13b. TIME COVERED FROM Jan 88 TO _____	14. DATE OF REPORT (Yr., Mo., Day) 4 May 1988 15. PAGE COUNT 7
16. SUPPLEMENTARY NOTATION	
17. COSATI CODES FIELD / GROUP / SUB GR	18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number) Aircraft Hydraulic Systems Lightweight Hydraulic Systems LHS, Reliability and Maintainability

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

A reliability and maintainability analysis was performed on the failure rate data generated by 1200 hours endurance operation of the Navy's Lightweight Hydraulic System (LHS) simulator. One set of data analysis was generated for 0-600 hours and another for 0-1200 hours of operation. The data confirms that 8000 psi hydraulic systems can be designed, fabricated, and operated without unusual problems occurring. The 123 hours MTBF after 1200 simulator hours of operation at 8000 psi is a 58% MTBF improvement over the A-7E 3000 psi hydraulic system (77.8 MTBF) as reported by 3M data.

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT ☐ DTIC USERS ☐	21. ABSTRACT SECURITY CLASSIFICATION Unclassified
22a. NAME OF RESPONSIBLE INDIVIDUAL	22b. TELEPHONE NUMBER (Include Area Code) 22c. OFFICE SYMBOL

DD FORM 1473, 83 APR EDITION OF 1 JAN 73 IS OBSOLETE

UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

NA-88-0010

RELIABIITY AND MAINTAINABILITY ANALYSIS UPDATE OF HYDRAULIC SYSTEM DATA

INTRODUCTION

The effort reported herein was performed under Eagle Techology Purchase Order 006318-00 - Technical Services for Evaluation of Hydraulic Equipment. The task provides an analysis of the failure rate data for a hydraulic system. The candidate hydraulic system selected is the Navy Lightweight Hydraulic System (LHS) located at Rockwell International, North American Aircraft, Columbus, Ohio. The LHS hydraulic system is representative of a Vought A7-E aircraft using 8000 psi hydraulic systems and components in lieu of the 3000 psi hydraulic systems and components typically used.

BACKGROUND

The LHS simulator, hydraulic systems, and components are described in detail in the following reports:

- Design, Development, and Evaluation of Lightweight Hydraulic System Hardware - Phase I, Report Number NADC-77108-30. Performed for the Naval Air Development Center under contract N62269-78-C-0363.

Abstract

The Lightweight Hydraulic System (LHS) program assesses the advantages of using an 8000 psi operating pressure level in Navy aircraft instead of the conventional 3000 psi level. This report presented the results of Phase I of a program to design, fabricate, and test a full scale 8000 psi system in a ground simulator and A-7E flight test aircraft. Two independent lightweight hydraulic systems, powered by variable delivery 8000 psi pumps, utilize twenty 8000 psi actuators and fourteen types of LHS minor hardware items. A steel framework ground simulator was designed with LHS component installation and hydraulic distribution systems similar to the A-7E aircraft. Laboratory tests conducted on components fabricated in Phase I include rod seal development, servo valve erosion, compatibility, pressure impulse, and endurance. A math model of the compatibility system was verified. Test results demonstrated that the Phase II simulator will function as designed. Weight and space analyses made on LHS components projected that the 30% weight and 40% space saving goals can be achieved. Based on preliminary R&M asessments of the development hardware, the MFHBF and MMH/FH improvements goal of 15% will be obtained.

- Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware - Phase II, Report Number NADC-79024-60. Performed for the Naval Air Development Center under contract N62269-80-C-0261.

Abstract

The Lightweight Hydraulic System (LHS) Advanced Development Program is a multi-phase investigation of the concept of using an 8000 psi operating pressure level to achieve smaller and lighter weight hydraulic components than those used in aircraft with conventional 3000 psi systems. This report presents the results of Phase II in which a full scale A-7E 8000 psi dual system hydraulic simulator was fabricated and tested. Tests conducted were proof pressure, system integration, baseline, dynamic performance, and 600 hours of endurance cycling. No major technological problems were encountered. Four flight control actuators accumulated over 3,000,000 cycles; one pump accumulated over 1000 hours of operation (Phase I + Phase II). Hydraulic system math models were corroborated by test data. A weight and space analysis update projected 33.1% and 36.3% savings, respectively, over an equivalent 3000 psi system. A study of simulator operating experience indicated a 23% improvement in reliability over a comparable 3000 psi system. An additional 600 hours of simulator endurance cycling (1200 hours total) was completed in FY 86 and 87. An addendum to this report will be published which documents the results of this follow-on program.

DISCUSSION

The tasks which were performed under the Technical Services Program for Evaluation of Hydraulic Equipment under the Eagle Technology Purchase Order were as follows:

I. Identify and Collect Available Data - The data for this effort was derived from the 600 to 1200-hour portion of LHS simulator endurance operation performed under the Phase II effort. This data will be included in an addendum to the Phase II final report.

II. Perform Analysis of Failures - Each failure was analyzed, the cause and effect determined, and a corrective action was recommended. Corrective actions were not initiated in all cases depending on the effect of the failure on simulator operation and the availability of resources. This data is included in an addendum to the Phase II final report.

III. Statistical Summary and Reliability Trend Analysis - A reliability growth trend line had been generated for the first 600 hours of simulator operation and is shown on Figure 1. This is based upon

NA-88-0010

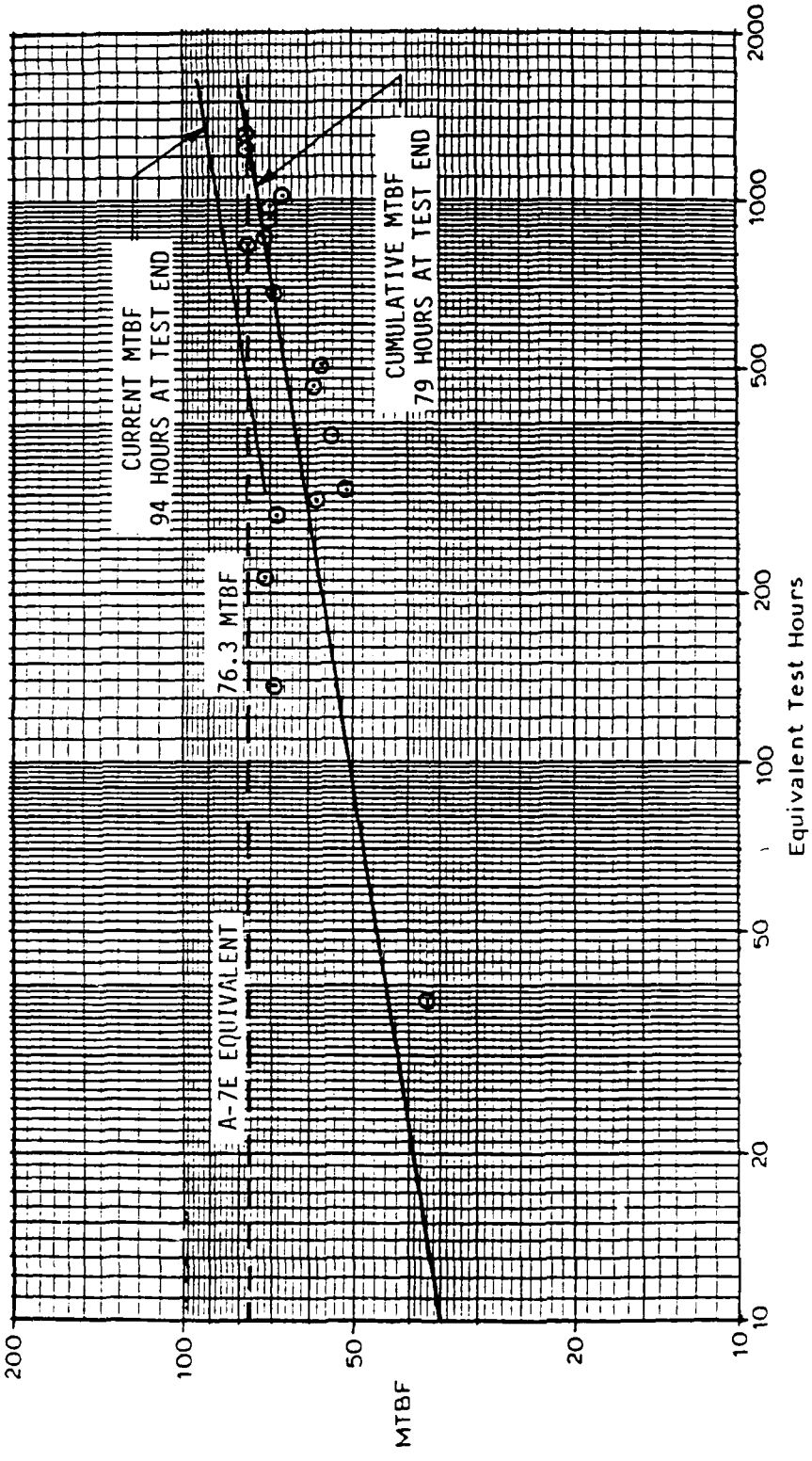


Figure 1. LHS Reliability Growth - 0 to 600 Hours of Simulator Operation

the failures that occurred during the first 600 hours of simulator operation. The data for that period is detailed in the Phase II final report number, NADC-79024-60, discussed above. A revised and updated reliability growth trend analysis was generated for this task by incorporating the data obtained from the 600 to 1200 hour LHS simulator operation.

Reliability Trend Analysis 0-600 Simulator Hours

The reliability trend analysis for the first 600 hours of simulator operation was developed in accordance with concepts for reliability growth given in MIL-HDBK-189, Reliability Growth Management. The growth concept projects that Mean-Time-Between-Failure (MTBF) plotted versus time will approximate a straight line (using log-log scaling) if corrections are incorporated following each failure. The slope of the line, showing cumulative MTBF as a function of time, is indicative of the aggressiveness and completeness of the corrective actions taken. The data was plotted on log-log scales as depicted in Figure 1. Operating time for the high pressure components was increased by a factor of 2.7 since cycles were accumulated at a rate 2.7 times faster than would be experienced in an operational aircraft. Actual simulator hours were used for low pressure components. A least square linear regression line was drawn through the data points. The end point of the curve at 1620 hours (600 simulator hours x 2.7 acceleration factor) yields a cumulative MTBF of 79 hours. Since this is based on accumulative data, and includes failures which have been corrected, it is not representative of the current MTBF. The principles of reliability growth development recognize this fact and provide an expression for determining current MTBF. Current (or instantaneous) MTBF is related to cumulative MTBF as follows:

$$\text{Current MTBF} = \frac{\text{Cumulative MTBF}}{1 - \alpha}$$

Where, α = slope of the line

The slope of the cumulative MTBF on Figure 1 is 0.162. Using the above equation, the current MTBF is found to be 94 hours. The current MTBF is shown on Figure 1 plotted as a line parallel to the cumulative MTBF.

A comparison with an operational hydraulic system was made using an analyzed sample of A-7E 3M data provided by Vought Corporation. The data were based on a 3.5 year period and 353,466 flight hours. Only those A-7E components that had equivalent counterparts on the LHS simulator were used to establish the A-7E MTBF base. A-7E pumps were not included since the 8000 psi pumps received special treatment during simulator testing. The MTBF for an A-7E hydraulic system containing equipment equivalent to that tested on the LHS

simulator is 76.3 hours. This is indicated by the horizontal dotted line on Figure 1. The apparent improvement of the LHS MTBF over the A-7E 3000 psi system is 23%. This exceeds the 15% improvement goal set of the LHS Advanced Development Program and is approaching the 25% goal set for a production program.

Reliability Trend Analysis Update - 0-1200 Simulator Hours

A summary of LHS simulator failures that occurred during the period of operation from 600 to 1200 hours is given on Table I. This data was used to generate the updated reliability growth trend line shown on Figure 2. The same method of trend analysis used for the first 600-hour period was used for the updated Figure 2. The MTBF for the first 600-hour period, Figure 1, was 94 hours. The MTBF calculated at the 1200 hour point, Figure 2, is 123 hours. This is an overall 31% growth of MTBF over the first 600 hours of test. The 123 hours MTBF is a 58% MTBF improvement over the A-7E 3000 psi hydraulic system (77.8 MTBF) as reported by 3M data.

The data confirms that 8000 psi hydraulic systems can be designed, fabricated, and operated successfully for extended periods of time without unusual problems occurring. The reliability growth as presented in Figure 2 has clearly demonstrated that design and quality factors that impact reliability can be identified through a test program, and corrections made to improve the system reliability. These same factors are the potential elements which may be used for predicting the reliability of similar mechanical components and systems.

The testing during this program did not include an important environmental influence on reliability; vibration as imposed upon the equipment from the airframe operating environment. Although the equipment is somewhat contributory to its own vibration environment, an additional applied vibration environment would likely have accelerated some of the failures. It would probably have induced additional failures, resulting in a lower overall cumulative MTBF. However, industry experience with environmental stress screening (ESS) suggests that quality defects are usually driven out within the first few hours of test. This being the case, failures of the LHS may have occurred earlier creating a steeper slope to the growth curve. The resultant of the increased slope (indicative of a more aggressive reliability developmental test) could be a calculated, current MTBF greater than the value achieved in Figure 2. This is academic and speculative; however, it does point out the need to consider program factors as an element of the mechanical reliability prediction methodology.

Table I. Summary of LHS Failures - 600-1200 Simulator Operating Hours

Component	Simulator Hours	Failure	Location/Remarks(1)(2)
RH UHT Actuator	619	Malfunction	Control valve sticking; faulty fabrication
O-Ring	696	Leaking	Spoiler actuator C1 port Rosan fitting loose; faulty fabrication.
Channel Seal	838	Leaking	L.E. flap solenoid valve; faulty fabrication
Speed Brake Actuator	850	Fatigue	Cylinder barrel split; under-design
'T' Seal	853	Leaking	FC-1 reservoir piston low pressure seal; normal wear
O-Ring	898	Leaking	RFI actuator center down bleed hole; normal wear
Restrictor	Approx. 900	Fatigue	L.E. flap circuit; poppet failed; under-design
Hose	904	Leaking	Aileron actuator FC-2 pressure; new design
Fitting	998	Leaking	Rudder AFCS solenoid valve return port; loose swage
LH UHT Actuator	1080	Fatigue	FC-2 piston edge cracked; under-design and faulty fabrication
L.E. Flap Actuator	1144	Fatigue	End cap fractured; lock nut worked loose permitting impact loading

(1) Details and recommended corrective action are discussed in the Phase II addendum report.

(2) Summary excludes coil tube failures, pump rework and non-hydraulic related mechanical failures.

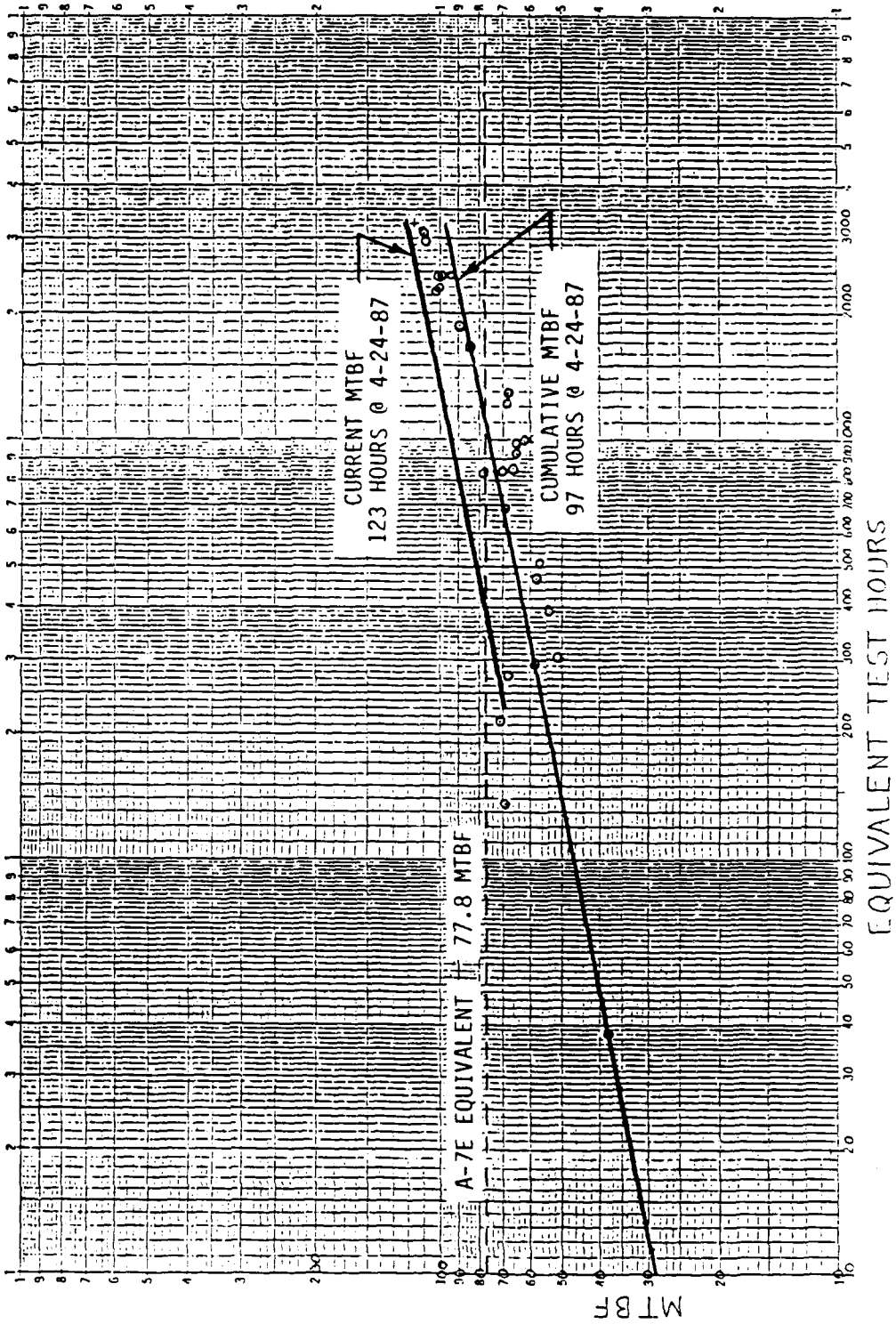


Figure 2. LHS Reliability Growth - 0 to 1200 Hours of Simulator Operation

DISTRIBUTION LIST
SUBMITTED FOR REPORT NO. **NADC-89006-60** PHASE II ADDENDUM

	No. of Copies
Naval Air Development Center Warminster, PA 18974-5000	
Code 6061	12/1
Code 6013	2/0
Code 8131	2/0
Naval Air Systems Command Washington, DC 20361	
(AIR-53031B)	2/0
(AIR-931)	2/0
Aeronautical Systems Division Wright-Patterson Air Force Base, OH 45433	1/0
(AFWAL/POOS)	1/0
(AFWAL/ENFEM)	1/0
Defense Documentation Center . . . . . . Cameron Station Alexandria, VA 22314	12/0
TOTAL	35/1

LMED

-8