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STORAGE RELIABILITY
OF
MISSILE MATERIEL PROGRAM

U.S. ARMY
MISSILE
RESEARCH
AND
DEVELOPMENT
COMMAND

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STORAGE RELIABILITY SUMMARY REPORT

VOLUME III

HYDRAULIC & PNEUMATIC DEVICES

LC-78-2 JANUARY 1978

Redstone Arsenal, Alabama 35809

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This summary documents findings on the non-operating reliability of valves, accumulators, actuators, pumps, cylinders, compressors, filters, gaskets and seals, bearings and regulators. The failure rates for each of the parts are given with one sided 90% confidence limits calculated. Elements of design are reviewed and predominant failure modes and mechanisms are discussed. This information is part of a research program being conducted by the U. S. Army Missile R&D Command, Redstone Arsenal, Alabama. The objective of this

20. Abstract (continued)

is the development of non-operating (storage) reliability prediction and assurance techniques for missile materiel. This report updates and replaces report LC-76-2 dated May 1976.

6

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ABSTRACT

This report summarizes analyses on the non-operating reliability of missile materiel. Long term non-operating data has been analyzed together with accelerated storage life test data. Reliability prediction models have been developed for various classes of devices.

This report is a result of a program whose objective is the development of non-operating (storage) reliability prediction and assurance techniques for missile materiel. The analysis results will be used by U. S. Army personnel and contractors in evaluating current missile programs and in the design of future missile systems.

The storage reliability research program consists of a country wide data survey and collection effort, accelerated testing, special test programs and development of a non-operating reliability data bank at the U. S. Army Missile Research & Development Command, Redstone Arsenal, Alabama. The Army plans a continuing effort to maintain the data bank and analysis reports.

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A

TABLE OF CONTENTS

TABLE OF CONTENTS (cont'd)

TABLE OF CONTENTS (cont'd)

TABLES

TABLES (cont'd)

1.0 INTRODUCTION

1.1 Missile Reliability Considerations

Materiel in the Army inventory must withstand long periods of storage and "launch ready" non-activated or dormant time as well as perform operationally in severe launch and flight environments. In addition to the stress of temperature soaks and aging, they must often endure the abuse of frequent transportation and handling and the climatic extremes of the forward area battlefield environment.

Missiles spend the majority of the time in this non-operating environment. In newer missile systems, complexity is increasing significantly, longer service lives are being required, and periodic maintenance and checkouts are being reduced. The combination of these factors places great importance on selecting missile materiels which are capable of performing reliably in each of the environments.

The inclusion of storage reliability requirements in the initial system specifications has also placed an importance on maintaining non-operating reliability prediction data for evaluating the design and mechanization of new systems.

1.2 Storage Reliability Research Program

An extensive effort is being conducted by the U. S. Army Missile Command to provide detailed analyses of missile materiel and to generate reliability prediction data. A missile material reliability parts count prediction handbook, LC-78-1, has been developed from this effort.

This report is an update to report LC-76-2, dated May 1976. It provides a summary of the analyses performed under the storage reliability research program and background information for the predictions in LC-78-1. Included are summaries of real time and test data, failure modes and mechanisms, and conclusions and recommendations include special design, packaging and product assurance data and information on specific part types and part construction.

For a number of the part types, detailed analysis reports are also available. These reports present details on part construction, failure modes and mechanisms, parameter drift and aging trends, applications, and other considerations for the selection of materiel and reliability prediction of missile systems.

The U. S. Army Missile Research & Development Command also maintains a Storage Reliability Data Bank. This data bank consists of a computerized data base with generic part storage reliability data and a storage reliability report library containing available research and test reports of non-operating reliability research efforts.

For the operational data contained in this report, the user should refer to the following sources: MIL-HDBK-217B, Military Standardization Handbook, Reliability Prediction of Electronic Equipment; Reliability Analysis Center (RAC) Microcircuit Failure Rates; RADC-TR-69-458, Revision to the Nonelectronic Reliability Handbook; and the Government-Industry Data Exchange Program (GIDEP) Summaries of Failure Rate Data.

1.3 Missile Environments

A missile system may be subjected to various modes of transportation and handling, temperature soaks, climatic extremes, and activated test time and "launch ready" time in addition to a controlled storage environment. Some studies have been performed on missile systems to measure these environments. A summary of several studies is presented in Report BR-7811, "The Environmental Conditions Experienced by Rockets and Missiles in Storage, Transit and Operations" prepared by the Raytheon Company, dated December 1973.

In this report, skin temperatures of missiles in containers were recorded in dump (or open) storage at a maximum of 165°F (74°C) and a minimum of -44°F (-42°C). In non-earth covered bunkers temperatures have been measured at a maximum of 116°F (47°C) to a minimum of -31°F (-35°C). In earth covered bunkers, temperatures have been measured at a maximum of 103°F (39°C) to a minimum of 23°F (-5°C).

Acceleration extremes during transportation have been measured for track, rail, aircraft and ship transportation. Up to 7 G's at 300 hertz have been measured on trucks; 1 G at 300 hertz by rail; 7 G's at 1100 hertz on aircraft; and 1 G at 70 hertz on shipboard.

Maximum shock stresses for truck transportation have been measured at 10 G's and by rail at 300 G's.

Although field data does not record these levels, where available, the type and approximate character of storage and transportation are identified and used to classify the devices.

1.4 System Level Analysis

The primary effort in the Storage Reliability Research Program is on analysis of the non-operating characteristics of parts. In the data collection effort, however, some data has been made available on system characteristics.

This data indicates that a reliability prediction for the system based on part level data will not accurately project maintenance actions if the missile is checked and maintained periodically. Factors contributing to this disparity include test equipment reliability, design problems, and general handling problems. In many cases, these problems are assigned to the system and not reflected in the part level analysis.

In general, a factor of 2 should be multiplied by the device failure rate to obtain the maintenance rate. Three system examples are described below:

1.4.1 System A

For system A, a check of 874 missiles in the field indicates 142 failed missiles. These failed missiles were taken to a maintenance facility. At the maintenance facility, no fault could be found in 51 of the missiles. Two missiles faults were corrected by adjustments. This left 89 failures which could be attributed to part failure. The parts were failure analyzed and the analysis indicated 19 failures to be a result of electrical overstress. These failures were designated design problems.

Therefore only 70 (49%) of the original 142 failures were designated as non-operating part failures.

1.4.2 System B

For system B, 26 missile failures were analyzed. Of these no fault was found in 2 missiles; adjustments were required for 2; external electrical overstress or handling damage was found in 10; a circuit design problem was assigned to 1, and component failures were assigned to 11.

1.4.3 Gyro Assemblies

An analysis of gyro assembly returns indicated that two thirds of the returns were attributed to design defects, mishandling, conditions outside design requirements, and to erroneous attribution of system problems.

Therefore, only 33 percent of the returns were designated as non-operating part failures.

1.5 Limitations of Reliability Prediction

Practical limitations are placed in any reliability analysis effort in gathering and analyzing data. Field data is generated at various levels of detail and reported in varying manners. Often data on environments, applications, part classes and part construction are not available. Even more often, failure analyses are non-existant. Data on low use devices and new technology devices is also difficult to obtain. Finally in the storage environment, the very low occurrence of failures in many devices requires extensive storage time to generate any meaningful statistics.

These difficulties lead to prediction of conservative or pessimistic failure rates. The user may review the existing data in the backup analyses reports in any case where design or program decision is necessary.

1.6 Life Cycle Reliability Prediction Modeling

Developing missile reliability predictions requires several tasks. The first tasks include defining the system, its mission, environments and life cycle operation or deployment scenario.

The system and mission definitions provide the basis for constructing reliability success models. The modeling can incorporate reliability block diagrams, truth tables and logic diagrams. Descriptions of these methods are not included here but can be studied in detail in MIL-HDBK-217B or other texts listed in the bibliography.

After the reliability success modeling is completed, reliability life cycle prediction modeling for each block or unit in the success model is performed based on the definitions of the system environment and deployment scenario. This reliability life cycle modeling is based on a "wooden round" concept in order to assess the missile's capability of performing in a no-maintenance environment. The general equation for this modeling is:

$$R_{LC} = R_{T/H} \times R_{STOR} \times R_{TEST} \times R_{LR/D} \times R_{LR/O} \times R_L \times R_F$$

where:

R_{LC} is the unit's life cycle reliability

$R_{T/H}$ is the unit's reliability during handling and transportation

R_{STOR} is the reliability during storage

R_{TEST} is the unit's reliability during check out and test

$R_{LR/D}$ is the unit's reliability during dormant launch ready time

$R_{LR/O}$ is the unit's reliability during operational (>10% electronic stress) launch ready time

R_L is the unit's reliability during powered launch and flight

R_F is the unit's reliability during unpowered flight

The extent of the data to date does not provide a capability of separately estimating the reliability of transportation and storage for missile materiel. Also data has indicated no difference between dormant (>0 and <10% electrical stress) and non-operating time. Therefore, the general equation can be simplified as follows:

$$R_{LC}(t) = R_{NO}(t_{NO}) \times R_O(t_O) \times R_L(t_L) \times R_F(T_F)$$

where:

R_{NO} is the unit's reliability during transportation and handling, storage and dormant time (non-operating time)

t_{NO} is the sum of all non-operating and dormant time

R_O is the unit's reliability during checkout, test or system exercise during which components have electrical power applied (operating).

t_O is the sum of all operating time excluding launch and flight

R_L is the unit's reliability during powered launch and flight (Propulsion System Active)

t_L is the powered launch and flight time

R_F is the unit's reliability during unpowered flight

t_F is the unpowered flight time

t is the sum of t_{NO}, t_O, t_L and t_F

The values R_{NO}, R_O, R_F are calculated using several methods. The primary method is to assume exponential distributions as follows:

$$R_{NO}(t_{NO}) = e^{-\lambda_{NO} t_{NO}}$$

$$R_O(t_O) = e^{-\lambda_O t_O}$$

$$R_L(t_L) = e^{-\lambda_L t_L}$$

$$R_F(t_F) = e^{-\lambda_F t_F}$$

The failure rates λ_{NO}, λ_O, λ_L and λ_F are calculated from the models in the following sections. λ_{NO} is calculated from the non-operating failure rate models. The remaining failure rates are calculated from the operational failure rate models using the appropriate environmental adjustment factors. Each prediction model is based on part stress factors which may include part quality, complexity, construction, derating, and other characteristics of the device.

Other methods for calculating the reliability include wearout or aging reliability models and cyclic or one shot reliability models. For each of these cases, the device section will specify the method for calculating the reliability.

1.7 Reliability Predictions During Early Design

Frequently during early design phases, reliability predictions are required with an insufficient system definition to utilize the stress level failure rate models. Therefore, a "parts count" prediction technique has been prepared. It provides average base failure rates for various part types and provides K factors for various phases of the system deployment scenario to generate a first estimate of system reliability. This prediction is presented in Report LC-78-1.

1.8 Summary of Report Contents

The report is divided into five volumes which break out major component or part classifications: Volume I, Electrical and Electronic Devices; Volume II, Electromechanical Devices; Volume III, Hydraulic and Pneumatic Devices; Volume IV, Ordnance Devices; and Volume V, Optical and Electro Optical Devices. Table 1-1 provides a listing of the major part types included in each volume.

1.9 Summary of Volume III, Hydraulic, Pneumatic and Electrical Control System Components

Table 1-2 summarizes the storage reliability failure rates of components used in missile control systems.

1.10 Missile Control System Studies

A study of missile control systems is being conducted and is expected to be published in the Summer 1978. A summary of the preliminary analyses is given in Tables 1-3, 1-4 and 1-5.

Table 1-3 gives the storage reliability of the various missile control systems grouped by the guidance type; Table 1-4 ranks the systems and Table 1-5 groups the control systems by type. These show the variation of storage reliability between hydraulic, pneumatic and electric.

TABLE 1-1. REPORT CONTENTS

		Detailed Rept. Number & Date
Volume I	Electrical and Electronic Devices	
Section		
2.0	Microelectronic Devices	LC-78-IC1, 1/78
3.0	Discrete Semiconductor Devices	-
4.0	Electronic Vacuum Tubes	LC-78-VT1, 1/78
5.0	Resistors	-
6.0	Capacitors	-
7.0	Inductive Devices	-
8.0	Crystals	-
9.0	Miscellaneous Electrical Devices	-
10.0	Connectors and Connections	-
11.0	Printed Wiring Boards	-
Volume II	Electromechanical Devices	
Section		
2.0	Gyros	LC-78-EM1, 2/78
3.0	Accelerometers	LC-78-EM2, 2/78
4.0	Switches	LC-78-EM4, 2/78
5.0	Relays	LC-78-EM3, 2/78
6.0	Electromechanical Rotating Devices	-
7.0	Miscellaenous Electromechanical Devices	-
Volume III	Hydraulic and Pneumatic Devices	
Section		
2.0	Accumulators	LC-76-HP2, 5/76
3.0	Actuators	LC-76-HP3, 5/76
4.0	Batteries	LC-78-B1, 2/78
5.0	Bearings	-
6.0	Compressors	-
7.0	Cylinders	-
8.0	Filters	-
9.0	Fittings/Connections	-
10.0	Gaskets	-
11.0	O-Rings	-
12.0	Pistons	-
13.0	Pumps	LC-76-HP4, 5/76
14.0	Regulators	-
15.0	Reservoirs	-
16.0	Valves	LC-76-HP1, 5/76
Volume IV	Ordnance Devices	
Section		
2.0	Solid Propellant Motors	LC-76-OR1, 5/76
3.0	Igniters and Safe & Arm Devices	LC-76-OR2, 5/76
4.0	Solid Propellant Gas Generators	LC-76-OR3, 5/76
5.0	Misc. Ordnance Devices	-
Volume V	Optical and Electro Optical Devices	

TABLE 1-2.
STORAGE RELIABILITY FAILURE RATES

SECTION	COMPONENT	M.T.B.F. 10^6	FAILURE RATE (FITS)	90% CONFIDENCE LIMIT
2.0	ACCUMULATORS	31.949	31.3	52.7
3.0	ACTUATOR			
	Hydraulic	5.025	199.	269.
	Pneumatic	11.364	88.	188.
4.0	BATTERIES			
	Silver-Zinc	23.8	42.0	97.9
	Thermal	10.526	94.3	367.
5.0	BEARINGS			
	Ball	200.0	5.0	19.5
6.0	COMPRESSORS	.245	<4,085.0	9434.
7.0	CYLINDERS	13.334	75.0	111.4
8.0	FILTERS	714.235	<1.4	5.2
	Filter (fine, 100 micron)	161.290	<6.2	-
	Filter (coarse, 100 micron)	476.190	<2.1	-
	Other	56.497	<017.7	-
9.0	FITTING & CONNECTIONS	2.25	444.4	888.5
10.0	GASKETS & SEALS	101.0	(<9.9)	23.0
11.0	O'RINGS	1070.0	4.0	-
12.0	PISTON	-	-	-
13.0	PUMPS			
	A. FIXED DISPL.	2.6	380.2	408.0
	1. Gear	2.27	439.9	486.4
	2. Piston	2.857	350.0	396.2
	3. Vane	3.333	300.0	363.4
	B. VARIABLE	2.105	475.0	528.0
	1. Piston	1.839	543.7	626.1
	2. Vane	2.462	406.2	477.1
	C. KINETIC	5.000	200.0	253.0
	1. Centrifugal	5.000	200.0	253.0
	D. HYDRAULIC (General)	10.493	95.3	105.9
	E. FUEL PUMP	8.071	123.9	286.5

TABLE 1-2.
STORAGE RELIABILITY FAILURE RATES
(cont'd.)

SECTION	COMPONENT	M.T.B.F. 10^6	FAILURE RATE (FITS)	90% CONFIDENCE LIMIT
14.0	REGULATORS	5.777	173.1	673.5
	Temperature	5.025	(<199.0)	459.7
	Pressure	.752	1330.0	5167.4
15.0	RESERVOIR			
16.0	VALVES			
	A. SOLENOID	117.647	8.5	14.4
	B. HYDRAULIC	357.142	2.8	5.6
	Bleeder	208.333	<4.8	11.0
	Check	43.668	22.9	51.0
	Control	149.253	<6.7	15.0
	Relief	714.285	1.4	5.5
	Shutoff	217.39	<4.6	11.0
	C. SHUTTLE/ SERVO	6.859	145.8	205.0
	D. PNEUMATIC	57.143	17.5	68.2
	General	4.673	214.0	833.0
	Check	4.149	<241.0	558.0
	Pressure	.6289	<1590.0	3680.0
	Manifold	47.619	21.0	48.5
	E. MOTOR OPERATED	8.849	113.2	301.0
	Sequence	5.155	194.0	755.0
	Freon	.667	<1500.0	3500.0
	Fuel	11.863	84.3	328.0
17.0	FLUIDS, HYDR.	-	-	-

TABLE 1-3.

STORAGE RELIABILITY FOR VARIOUS MISSILE CONTROL SYSTEMS

(Grouped by Guidance Type)

MISSILE	GUIDANCE	CONTROL	NO. OF COMPONENTS	10^6 MTBF	STORAGE FAILURE (FITS)
A	Infrared Homing	Pneu/Elec	174	4.54	220.08
B	" "	Electric	1	1.148	871.0
C	" "	Electric	7	1.2	824.4
D	Command	Hydraulic	15	2.169	460.9
P	Command Semi-Act. Radar	Hydr/Pneu	20	.451	2215.0
E	Semi-Act. Radar	Hydraulic	174	.291	3435.5
H	Semi-Act. Radar	Hydraulic	233*	.328	3047.6
G	Inertial	Hydr/Elec	24	1.0	998.0
M	"	Hydraulic	39	.267	3741.0
I	TV	Hydraulic	79	.207	4827.1
N	TV	Hydraulic	40	.259	3866.5

*209 connections

TABLE 1-4.

STORAGE RELIABILITY ORDER BY FAILURE RATE (FITS)

MISSILE	GUIDANCE	CONTROL	FAILURE RATE (FITS)	MTBF (10^6)
A	Infrared Homing	Pneu. (Elect.)	220.08	4.54
D	Command	Hydr. (Elect.)	460.90	2.17
C	Infrared Homing	Electric	824.40	1.20
B	Infrared Homing	Electric	871.00	1.14
G	Inertial	Hydr. (Elect.)	998.00	1.00
P	Command/Semi-Act. Radar	Hydr. (Pneu.)	2215.0	0.53
H	Semi-Act. Radar	Hydraulic	3047.60	0.33
E	Semi-Act. Radar	Hydraulic	3435.50	0.29
M	Inertial	Hydraulic	3741.00	0.26
N	T.V.	Hydraulic	3866.50	0.25
I	T.V.	Hydraulic	4827.10	0.21

TABLE 1-5.

STORAGE RELIABILITY GROUPED BY CONTROL TYPES

Missile	Guidance	Failure Rate (Fits)	MTBF x 10^6
HYDRAULIC (Total)			
H	Semi-Active Radar	3,047.6	.33
E	Semi-Active Radar	3,435.5	.29
M	Inertial	3,741.0	.26
N	T.V.	3.866.5	.25
I	T.V.	4,827.1	.21
	AVERAGE	3,783.54	.26
HYDRAULIC ELECTRIC (Combination)			
D	Command	460.90	2.17
G	Inertial	998.00	1.00
	AVERAGE	729.45	1.17
HYDRAULIC PNEUMATIC (Combination)			
P	Command (Semi-Act. Radar)	2215.0	.45
PNEUMATIC ELECTRIC (Combination)			
A	Infrared Homing	220.08	4.54
ELECTRIC (Total)			
C	Electric	824.4	1.20
B	Infrared Homing	871.6	1.14
	AVERAGE	847.7	1.20

2.0 Accumulators

Accumulators are devices that store energy, and subsequently supply peak demands in a system having an intermittent duty cycle. They can also be used to provide hydraulic shock suppression. Accumulators may store energy by means of gravitational force, mechanical springs, or the compressibility of gases. Data was collected on accumulators that store energy by the compressibility of gases. Three types of separators are used in these accumulators: 1) bladder, 2) diaphragm, and 3) piston.

2.1 Storage Reliability Analysis

2.1.1 Failure Mechanisms and Modes

Accumulators of all types generally have similar failure characteristics. Failure mechanisms for stored accumulators are (1) contamination, (2) damaged parts (cracked), (3) blemishes, (4) misalignment problems or swelling. The storage failure modes are usually (1) internal leakage, (2) external leakage, and (3) swelling.

A summary of failure modes, causative mechanisms, detection methods and measures to minimize them are presented in Table 2.1-1.

2.1.2 Accumulator Failure Rates

Over 334 million part hours of storage data are included in this report. Table 2.1-2 shows data sources with their functional application and environment as well as failure information. For purposes of this table, "environment" is defined as the conditions for which the equipment was designed and intended to operate.

The data did not always contain specific information as to accumulator type or descriptions of failure modes and mechanisms. Quality grades were not defined and therefore failure rates derived in this section reflect the entire quality range defined for accumulators.

TABLE 2.1-1. FAILURE MECHANISM ANALYSIS - ACCUMULATORS

PART & FUNCTION	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHOD	HOW TO ELIMINATE MINIMIZE FAILURE MODE
Accumulator Seat	Internal Leakage	1	Internal Leakage is cause by: 1) contamination 2) damaged physical properties 3) aging O-rings 4) blemishes 5) pressure 6) bending 7) swelling material	PRE-INSTALLATION Testing & run-in will detect this type of failure. POST-INSTALLATION System pressure measurements can be used to determine if accumulator is leaking.	Control cleaniness level for parts, components & systems. Areas that can trap contaminants should be eliminated from accumulator design.
Separator Material	Swelling & Bending	2	Separator stuck in intermediate position due to: 1) contamination 2) misalignment uniformity 3) swelling material 4) aging material 5) bending (excess) material	PRE-INSTALLATION Testing will determine this type of failure prior to accumulator installation. POST-INSTALLATION Accumulator positon indicator or system pressure measurements.	Allow conservative force margins for opening and closing of seat. Run-in test or margin tests should reveal this type of failure.
Accumulator Body (support element & contain media)	External Leakage	3	Leakage through: 1) static seals, O-rings 2) plumbing connections 3) accumulator body (porosity) 4) blemishes 5) pressure	PRE-INSTALLATION Component tests will reveal this failure mode. POST-INSTALLATION System pressure (fluid) loss or visual detection of leaks.	Methods to control these failures include: 1) welded external body construction 2) install accumulator into system with permanent mechanical connections. 3) impregnate castings with sealants 4) use of vacuum melt metals to control inclusions or stringers.

TABLE 2.1-2. ACCUMULATOR STORAGE DATA

	PART DESCRIPTION	FUNCTIONAL APPLICATION	OPER. ENV.	PART POP.	NO. OF FAILURES	PART HOURS x 10^6	FAILURE RATE	UPPER 90% CONFIDENCE	YEAR OF REPORT
1.	Hydraulic	Storage	MSL	-	-	-	.01	-	65
2.	Hydr. Fluid	Storage	AIR	-	1	.14892	6.715	26.119	63
3.	Hydr. Fluid Piston	Storage	GRD	600	600	10.512	57.077	60.178	64
4.	Hydraulic	-	GRD	-	0	3.051	<0.327	.757	74
5.	Accumulator	Hydropak	MSL		2	50.	.04	.106	-
6.	Accumulator	Aircraft	AIR		20	100.	.2	.270	-
7.	Accumulator	Hydr. Field Data	MSL		3	110.	.027	.061	-
8.	Hydraulic Accumulator	Dormant	MSL		1	9.351	.106	.985	75
9.	Accumulator	Storage	MSL		0	21.516	<.0464	.180	58
10.	Accumulator Diaphragm	Ball. Missile (Storage)	MSL	30	13	.5256	24.733	36.069	64
11.	Hydraulic Accumulator	Missile Storage	MSL		1	17.0	.0588	.229	77
12.	Hydraulic Accumulator	Air-to-air missile	MSL	874	0	12.76	<.0784	.181	

2.1.3 Analysis of Storage Data

The combined failure rate for all of the entries in Table 2.1-2 is 1914 fits. However, close examination of the individual entries shows wide discrepancies in failure rate among the different sources. For programs reporting at least one failure, the failure rate ranges from a low of 27 fits to 57078 fits. In an attempt to reconcile these differences, analyses of the discordant data points were made as follows:

a) Data Point No. 2 - This was an accumulator on board an aircraft which crashed in the desert. Seventeen years later the equipment was recovered and analyzed. The accumulator was found to have failed although the analysis showed it held air pressure for "a few years." The failure rate shown in Table 2.1-2 shows a number of hours equal to 17 years. It was not possible to determine the time of failure, therefore this data is invalid.

b) Data Point No. 3 - A total of 600 accumulators were stored at the manufacturers' plants for two years. At the end of this period all of the accumulators had leaked. The accumulators were stored with the piston O-rings installed. This is not a recommended procedure and the manufacturer does not store accumulators with O-ring seals in place any more. Therefore, the information in data point 3 is no longer valid and will not be used for prediction.

c) Data Points No. 7 and No. 11 - This is the lowest failure rate source shown in Table 2.1-2. All of the accumulators in this source were submitted to a "run in" for six hours. This was accomplished by charging the unit at very high pressure for a few minutes and at nominal pressure for the rest of the time. It was estimated that the run in eliminated from 75 to 80 percent of all the potential problems in the field.

The accumulators in data point 11 were also submitted to a high pressure run in prior to storage. The combined failure rates of data points 7 and 11 is consistent with that of other accumulators in Table 2.1-2. In view of this and despite the fact that these accumulators went through a preconditioning process, data points 7 and 11 will be included in the prediction process.

d) Data Point No. 6 - The information in this source represents an estimate based on ratioing the operational failure rate of accumulators. Since it does not represent actual storage experience, this data will not be used for prediction.

e) Data Point No. 10 - This point represents data on a number of accumulators in a hydraulic thrust vector control system. A number of the failures were attributed to improper shipping and filling procedures and to inadequate accumulator capacity. Since most of the failures were attributed to improper procedures and design defects the data will not be used for prediction.

After eliminating four of the six data points discussed above, there are seven valid points left. Six of them are for a missile environment and one for a ground environment. The ground accumulator has 3.051 million hours of storage with no failures. This is consistent with the failure rate of missile accumulators and therefore, they will be grouped together.

The resultant data shows 223.68 hours of storage with seven failures for a failure rate of 31.3 fits. The one-sided 90% confidence limit is 52.7 fits. Of those sources reporting at least one failure, a range of failure rates from 27 fits to 106 fits was observed. Some of the differences still remaining may be due to the pressurization state in which the accumulators were stored. For example, the devices in data point 11 were stored in an unpressurized state. Similar information does not exist on the rest of the valid data points. Therefore, the effects of the pressurization state of the devices could not be quantified. However, this is recognized as possible reliability factor.

2.2 Operational/Non-Operational Failure Rate Comparison

The ratio of operating to non operating failure rate was computed as shown below. The operational failure rate was obtained from the RADC Nonelectronic Reliability Handbook.

Environment	λ(fits)	$\frac{\lambda_{op}}{\lambda_s}$
Operational	54000	
Storage	31	1742

2.3 Conclusions and Recommendations

2.3.1 Conclusions

Comparison between dormant and storage reliability data indicates no significant difference between the two. This agrees with previous studies (reference no. 35). Therefore, the dormant and storage data were combined in all analyses.

Quality grades were not well defined for the accumulator data collected. To determine quality grades extensive searching through component specifications and drawings would be required. It was therefore impossible to determine the effect, if any, of quality levels. The results presented in this report represent failure rate averages over the quality grade spectrum.

2.3.2 Recommendations

Record keeping for accumulators kept on storage should be improved, specifically the identification of quality grades and accumulator description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented.

2.4 Reference

The information in Section 2 is a summary of document number LC-76-HP2, "Hydraulic and Pneumatic Systems Accumulator Analysis," dated May 1976. Refer to that document for details of data collection and analysis, as well as, technical descriptions of accumulators.

3.0 Actuators

Four major types of actuators were analyzed; Hydraulic, Pneumatic, Motor Controlled, and Solenoid types. Each type has several types of actuating mechanisms, however the data has been grouped in accordance with major types.

3.1 Storage Reliability Analysis

3.1.1 Failure Mechanisms

The primary failure modes that could materially affect the functional reliability of actuators are individually discussed. Failure modes most likely to occur listed in their most probable order are:

1) Internal leakage (excessive)
2) Hysteresis
3) External leakage (excessive)

The actuator failure modes are often the cause of valve failure and are often difficult to determine.

3.1.1.1 Internal Leakage

As with a valve, the actuator has a leakage problem which is most serious particularly for long term stored actuators. This is a problem because a small leakage rate can deplete the supply of the flowing medium. The flowing medium may be corrosive or explosive and damage to equipment or personnel can result. Excessive internal leakage is attributed primarily to failed piston seals; however, contamination can also cause increased wear and leakage.

3.1.1.2 Hysteresis

Hysteresis, is a result of excessive friction between moving parts. Packing contributes to this effect because it must create a seal sufficient to hold line fluid within the body. Additional friction occurs in the guide, and very fine stem finishes are employed. Piston seal rings in cylinder actuators also offer resistance to movement and cause some hysteresis.

Other moving parts such as, the plug, diaphragm plate and stem are possible problems due to contamination and wear. After long periods of storage, sticking or sliding or contacting can be caused by (1) cold welding, (2) inadequate lubrication, (3) contamination, or (4) incorrect design.

3.1.1.3 External actuator leakage

External leakage is caused by leakage through or around seals. This is due to aging of elastomeric seals or static seals. To eliminate this failure mode, welded body construction is preferred and permanent connections such as brazed, welded or swaged should be made when the components are installed into the system.

3.1.2 Actuator Storage Failure Rates

The actuator storage data, Table 3.1-1, did not identify specific types other than whether they were used in a hydraulic or pneumatic system. Accordingly, storage failure rates were derived for these two categories as shown in Table 3.1-1.

TABLE 3.1-1. ACTUATOR STORAGE DATA SUMMARY

Type	Storage Hrs. x 10^6	Failures	λ_s (Fits)	90% one-sided confidence limit
Hydraulic	608.6	121	(9.8) 199 (40769)	269
Pneumatic	239.0	21	(63) 88 (256)	118

The numbers in parenthesis indicate the range of failure rate computed from individual data sources showing at least one failure. The 90% one sided confidence limit is also shown.

3.1.3 Hydraulic Actuator Storage Data

Storage data on hydraulic accumulators consisted of over 608 million hours with 125 failures for an overall failure rate of 199.7 fits. Examination of Table 3.1-2 reveals a wide variation in failure rates among the individual sources. From sources showing at least one failure, the failure rate varies from a low of 9.8 fits to a high of 40,769 fits. In an attempt to determine the reasons for this variance, the individual sources were reviewed for clues. Although the information on actuator types, storage environment, quality grades, length of storage and types of failures was not sufficient to reach absolute conclusions, several possibilities for the variance were identified.

TABLE 3.1-2. ACTUATOR STORAGE DATA

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. x 10^6	FAILURE RATE x 10^{-6}	UPPER 90% CONFIDENCE	YR. OF REPORT
1. Hydraulic Actuator	Linear	GND	-	0	31.000	<0.030	.074	74
2. Hydraulic Actuator	Linear	SUB	-	5	6.012	0.832	1.554	74
3. Hydraulic	Air-to-Air MSL (Storage)	MSL GND	46	0	2.11	<.474	1.091	64
4. Hydraulic	Ballistic MSL (Storage)	MSL GND	90	43	1.5768	27.270	33.413	64
5. Hydraulic Actuator	Dormant	GND	-	-	-	0.50	-	69
6. Actuator	Hot Gas	GND	-	-	-	0.175	-	69
7. Actuator	-	MSL	-	29	440.2	.0659	.084	69
8. Actuator	Storage	-	-	-	-	0.07	-	65
9. Actuator	Electrohy-draulic (Storage)	MSL	-	-	-	.01	-	65
10. Hydraulic Actuator	-	-	147	4	18.028	0.221	.444	63
11. Hydraulic Actuator	Desert Storage	-	5	0	0.7446	<1.343	3.102	63
12. Hydraulic Actuator	Aircraft	-	7	5	0.12264	40.769	76.205	63
13. Hydraulic Actuator	Aircraft	AIR	-	-	0.4555	-	-	63
14. Hydraulic Actuator	Storage	-	108	27	1.419	19.027	24.641	63
15. Hydraulic Actuator	Storage	-	11	8	.9636	8.302	13.495	63
16. Hydraulic Actuator			117	0	3.9288	<0.254	.588	63

TABLE 3.1-2. ACTUATOR STORAGE DATA (cont'd)

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. $\times 10^6$	FAILURE RATE $\times 10^{-6}$	UPPER 90% CONFIDENCE	YR. OF REPORT
17. Hydraulic Actuator	Piston	-	6992	0	102.003	<.0098	.022	71
18. Actuator	Linear	GND	-	0	0.628	<1.592	3.677	74
19. General Actuator	Explosive	GND	-	13	207.100	0.063	0.100	74
20. Explosive	Storage Explosive	GND	-	8	31.260	0.255918	0.415	66

a) Periodic exercising - Some of the equipments in Table 3.1-2 were exercised periodically during storage (data samples 15 and 16). Others (data samples 4, 11, 12 and 14) were never exercised throughout the storage period. A third class of data did not specify whether or not the equipment had been exercised. The overall failure rate for hydraulic actuators is 199 fits. The failure rate of those equipments known to have not been exercised is almost 100 times the overall failure rate. If the data from non-exercised systems is removed from the overall failure rate computation, the ratio of non-exercised to the overall failure rate increases to over 250. Although it is difficult to ascertain the storage conditions of the third class of data, its overall failure rate is consistent with that of exercised systems. Based on this data, a trend toward higher failure rates for equipment which remains not exercised during storage is apparent and may partially account for the higher failure rates in Table 3.1-2.

b) Changes in technology - As time passes, advancements in technology are sometimes reflected in improved failure rates. For example, welded body construction and improved elastomeric seals result in improved reliability. The data sources in Table 3.1-2 vary in age from 1963 to 1974. The lowest failure rates are obtained from data samples 1 and 7 taken from reports dated 1974 and 1969 respectively. Those two sources, by virtue of their large number of hours (over 75% of all the hours), bias the overall failure rate toward the low end. All of the high failure rate items were taken from reports dated 1963 and 1964. Therefore, it is probable that a decrease of failure rate with time (and improved technology) is shown by the data.

c) Complexity - The complexity of an actuator can vary from a simple linear piston type to a complex reciprocating vane type. Since the data did not call out specific types, it is possible that the variance is due to differing degrees of complexity.

d) Length of storage - The length of storage for individual sources varied from a few years to as many as 17 years. Some of the failures were attributed to aging of seals. In this case, the failure rate of equipments stored for longer times would tend to be higher. In spite of this, an exponential failure rate was still assumed since the data was not conclusive enough to establish an aging mechanism.

e) Reporting methods - Failures were counted on the basis of failure statements on failure reports. With the exception of the 6% of the reports concerned with corrosion, the balance of the failures may not have been caused by the storage environment.

3.2 Pneumatic Actuator Storage Data

Storage data for pneumatic actuators consists of 238.988 million part hours with 21 failures for a failure rate of 87.9 fits. For programs reporting at least one failure, a range of failure rates from 9.8 fits to 256 fits was observed. Although this range is not as wide as that for hydraulic actuators, some of the reasons discussed in Section 3.1.1. responsible for the variation.

3.3 Operational/Non-Operational Failure Rate Comparison

Operational to non operational failure rates ratios (K factors) were computed for hydraulic and pneumatic actuators. Operating data was taken from the RADC Nonelectronic Reliability Notebook. The data and K factors are shown below.

Type	λ_{op} (fits)	λ_s (fits)	λ_{op}/λ_s
Hydraulic	15288	199	77
Pneumatic	1507	88	17

3.4 Conclusions and Recommendations

3.4.1 Conclusions

In general, actuator types could not be identified except for the system type in which they were installed. Quality grades were not well defined for the actuator data collected. To determine quality grades extensive searching through component specifications and drawings would be required. Hence, effects of quality levels, if any, could not be determined.

There was no significant difference between dormant and storage reliability data. Dormant and storage data were combined in all analyses.

Storage data collected for each generic actuator type was not plentiful. Therefore, failure rates derived at this high level have higher statistical confidence than those of the sub-categories and should be utilized unless specific information is available to further define the type of actuator under consideration.

3.4.2 Recommendations

Storage failure rates for hydraulic and pneumatic actuators are as follows:

Type	λ(fits)
Hydraulic	199
Pneumatic	88

Although a wide variance was observed among the different sources, the above failure rate is recommended for prediction. This failure rate is representative of missile actuators stored under varying conditions and of different quality grades.

Record keeping for actuators kept on storage should be improved, specifically the identification of quality grades and actuator description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented for all missile hydraulic and pneumatic systems.

3.5 Reference

The information in Section 3 is a summary of document number LC-76-HP3, "Hydraulic and Pneumatic Systems Actuator Analysis," dated May 1976. Refer to that document for details of data collection and analysis as well as technical descriptions of actuators.

4.0 BATTERIES

Two types of batteries are used in missiles, the silver-zinc primary batteries and the thermal batteries.

4.1 Failure Mechanisms and Modes

The primary failure modes are listed in Table 4.1-1.

TABLE 4.1-1. BATTERY FAILURE MODE DISTRIBUTION

FAILURE MODE

1. Discharge Duration
2. Rise Time Spec.
3. Leakage
4. Corrosive

4.2 Storage Reliability Analysis

4.2.1 Storage Failure Rates

Non-operating (storage) failure rates for batteries are summarized in Table 4.2-1.

TABLE 4.2-1. BATTERY STORAGE FAILURE RATES

BATTERY	ENVIRONMENT	FAILURE RATE (FITS)
Silver Zinc	Shelf	<72.5
Silver Zinc	Service	<102.0
TOTAL		<42.4
Thermal (A)	Field	<110.0
Thermal (B)	Field	<666.0
TOTAL		<94.0

The failure rates for silver zinc batteries are from data collected from shelf life and from service life. The shelf life data represents a failure rate of <72.5 fits and service life of <102.0 fits. Combining this data yields an overall failure rate for silver-zinc batteries of <42.4 fits.

The failure rates depicted in Table 2-2 represent data collected on thermal batteries stored in the field environment. Battery A and Battery B represent two different manufacturers.

Battery Type A tests yield a failure rate of 110.0 fits and Battery Type B tests resulted in a failure rate of <666.0 fits. Combining the data give an overall failure rate of <94.0 fits.

4.2.2 Storage Reliability Data

Non-operating (storage) data for batteries is summarized in Table 4.2-2.

TABLE 4.2-2. BATTERY STORAGE DATA

BATTERY TYPE	ENVIRONMENT	QTY.	FAILURES	STORAGE HOURS (10^6)
Silver Zinc	Shelf	483	0	13.8
Silver Zinc	Service	510	0	9.8
TOTAL		993	0	23.6
Thermal				
A	Field	163	0	9.1
B	Field	37	1	1.5
TOTAL		200	1	10.6

Data collected for silver-zinc batteries was from two data sources. The first data source is given in the primary battery document and is not represented in this summary. Data from that source was collected prior to 1963 and considered obsolete.

Data collected from data source two is depicted above for silver zinc batteries. As shown, overall 993 batteries were tested. Zero (0) failures result from these tests. The batteries had accumulated 23.6 million storage hours.

Note that 483 of the silver zinc batteries were shelf life for a total of 13.8 million storage hours and zero failures. Of the total 993, 510 were service life batteries which were stored for 9.8 million hours and resulted in zero (0) failures.

Battery types A and B in Table 4.2-2 represent field storage data for 200 thermal batteries. The Types A and B represent two different manufacturers. The Type A battery was stored under field condition for a cumulative total of 9.1 million hours with zero failures. There were 163 batteries of this type.

Of the 200 batteries, only 37 were of Type B. These were stored for 1.5 million hours. As shown, one (1) failure occurred during tests of these batteries.

The overall storage hours for the thermal battery is 10.6 million hours.

4.3 Recommendations

4.3.1 Silver-Zinc Batteries

The results of the analysis for shelf and service life of silver zinc batteries depicting remaining battery service life after known shelf life should be used to obtain optimum utilization of batteries. Where definite shelf life is uncertain for a period of time, total service life should be assumed for this period.

Further surveillance testing of these batteries should be continued at periodic intervals. Present plans have surveillance testing to be conducted on silver zinc batteries up to an age of 20 years at intervals of less than two years.

The following failure rates should be used for prediction:

BATTERY	ENVIRONMENT	FAILURE RATE (FITS)
Silver Zinc	Shelf Life	<72.47
Silver Zinc	Service Life	<102.0

4.3.2 Thermal Batteries

It has been shown that for both Types A and B batteries, that aging of the battery is reducing its electrical output and thus it is considered extremely important that surveillance tests be continued in order to carefully monitor this reduction so that the limiting life of these batteries can be determined.

The batteries in stockpiles with bar connected brackets should be examined for clearance between the firing pin and the brackets and that all batteries that may result in a firing pin hang-up be further spread to prevent this occurrence.

The following failure rates for prediction of thermal battery reliability should be used:

BATTERY TYPE	ENVIRONMENT	FAILURE RATE (FITS)
A	Shelf	<110.0
B	Shelf	<666.0
OVERALL		<94.3

4.4 Reference

Information in this document is from LC-78-B1, Missile Systems Battery Analysis, February 1978. Refer to that document for details of data collection and analysis.

5.0 Bearings

This section contains reliability information and analysis on bearings. Since bearings and lubrication are the acknowledged and proven life limiting elements of motors, emphasis has been placed on the examination of bearing fatigue life and reliability and the types of lubrication systems which enhance long life. Wear must be virtually eliminated for bearings to achieve their ultimate life capability, i.e., their fatigue life. For storage and operation, fluid lubrication offers this opportunity.

The primary bearing used in military systems is the ball bearing. Other bearing types are specified in the detailed bearing report.

5.1 Failure Mechanisms and Modes

The primary bearing failure modes are listed in Table 5.1-1.

TABLE 5.1-1. BEARING FAILURE MODE DISTRIBUTION

FAILURE MODES

1. Excessive wear
2. Mechanical binding
3. Sticking
4. Dented
5. Jammed
6. Pitted
7. Friction Excessive
8. Chattering
9. Excessive vibration
10. Frozen
11. Clogged
12. Scored
13. Bent
14. Cracked
15. Lack of Lubricant

Table 5.1-2 lists the most common failure modes for bearings by their frequency of occurrence. Given for each failure mode is the reason of failure, failure mechanism, possible causes and suggestions as how to eliminate/minimize the failure mechanisms.

5.2 Storage Reliability Analysis

5.2.1 Storage Failure Rates

Non-operating (storage) failure rates for bearings are summarized in table 5.2-1.

TABLE 5.2-1. STORAGE FAILURE RATES

Type	Environment	λ (10^{-9})
Ball Bearing	MSL	5.0
	GND	11.4
	SUB	2994.0
	OVERALL ---------------	14.48

5.2.2 Storage Failure Data

Storage failure data was collected from three environments. Storage hours and numbers of failures for each environment are shown in Table 5.2-2.

TABLE 5.2-2. STORAGE FAILURE DATA

Bearing Type	Environment	Failures	Hours (10^6)
Ball	MSL	1	200.0
	GND	4	351.52
	SUB	3	1.002
TOTAL		8	552.522

5.3 Operational/Non-Operational Failure Rate Comparison

Table 5.3-1 presents operational to non-operational failure rate ratios for the two bearing types in an air environment. For comparison purposes, the types are combined in the storage data.

TABLE 5.1-2 FAILURE MECHANISM ANALYSIS

MOTORS AND/OR MOTOR GEARHEADS - BEARINGS

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
A. SOLID TRANSFER LUBRICATED BALL BEARINGS with MOS_2 in synthetic ball retainer Loss of Power (Complete or partial)	Excessive losses due to high bearing friction or failure	I	1. Excessive wear of retainer with deb debris fouling ball tracks.	i. Excessive bearing loads.	(a) Balance gear tooth loads. (b) Use spline or coupling for the output drive. (c) Utilize larger bearings. (d) Use MOS_2 coatings on raceways.
				ii. Excessive ball/ball pocket interface loads.	(a) Utilize preload springs or adopt duplex preloaded bearing pair to unify ball velocity and load.
				iii. Excessive temperature.	(a) Use more efficient motor. (b) Use motor concept wherein motor heat flux path bypasses the gearhead. (c) Improve heat flux path from motor mounting.

TABLE 5.1-2 FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iv. Skidding of balls during acceleration due to frequent stop/start duty cycles.	(a) Utilize preload to provide ball traction.
				v. Bearing misalignment	(a) Improper housing tolerances or out of roundness at interface with bearing.
3. DRY FILM LUBRICATED BALL BEARINGS i.e., MoS_2 dry film lube on ball races.					
Loss of Power (Complete or partial)	Excessive losses due to bearing friction or failure	II	1. Fouling of ball raceways with dry film lube detritus.	i. Unit did not receive a preliminary run-in followed by a disassembly and cleansing operation.	(a) Improve process control. (b) Specify the exact run-in procedures needed. (c) Have the bearings lubed and run in prior to assembly of shields possibly by the manufacturer, then cleaned prior to final assembly.
C. ALL BALL BEARING TYPES					
Loss of Power (Complete or partial)	Excessive losses due to bearing friction of failure	III	1. False brinnelling (fretting or fretting corrosion) of raceways and balls.	i. Vibration environment without resonance; i.e., motor experiences only input energy.	(a) Utilize duplex preloaded bearing. (b) Operate motor during launch phase (often not feasible).

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
					(c) Use bearing with better static load capacity. (d) Use vibration isolators. (e) Use sleeve bearings (and probably larger motor). (f) Exclude air and oxygen (this eliminates the corrosion aspect).
				ii. Vibration environment with motor resonance.	(a) Redesign motor rotating assembly and bearing stiffnesses to prevent excitation. (b) Operate motor during vibration environment (often not feasible). (c) Utilize duplex preloaded bearing. (d) Use larger capacity bearings. (e) Use vibration isolation. (f) Use sleeve bearings.
				iii. Vibration environment with resonance of supporting structure.	(a) Modify structure to prevent excitation. (b) Any of the solutions (b) to (f) in Item ii above.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MIN-IMIZE FAILURE MECHANISMS
			2. Brinnelling (plastic deformation of raceways.)	i. Excessive loads from shock environment.	(a) Use larger capacity bearings. (b) Incorporate bearing preloading. (c) Mount motor on shock isolators. (d) Use sleeve bearings.
				ii. Excessive radial load due to incipient gear failure or to overload.	(a) Remedy the gear or overload problem.
				iii. Excessive thrust load due to differential radial interference between housing and bearing (at low temp) restricting relative motion of bearing.	(a) Use housing material with compatible thermal expansion coefficient. (b) Use insert in housing of compatible material to house bearing. (c) Reduce thermal gradient by providing better heat flux path to heat sink.
				iv. Excessive thrust load due to differential linear expansion between housing and rotor combined with inadequate bearing end float allowance.	(a) Use housing material with compatible thermal expansion coefficient. (b) Allow greater bearing and float. (c) Use wet lube instead of solid lube for better heat conduction.
				v. Improper installation (installation loads imposed through balls.)	(a) Implement stricter assembly procedures and quality control measure.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
			3. Fretting corrosion at interface between bearing and housing.	i. Excess clearance between bearing and housing reacting to bearing loads (motor operating).	(a) Lessen the clearance between housing and ball race; i.e. use closer tolerance. (b) Use housing material that has similar coefficient of thermal expansion to that of the bearing.
				ii. Resonance of motor structure acting on clearance between bearing and housing (motor operating or not operating).	(a) Redesign motor rotation assembly and bearing stiffnesses to prevent excitation. (b) Use vibration isolation. (c) Use sleeve bearings.
				iii. Resonance of supporting structure acting on clearance between bearing and housing (motor operating or not operating).	(a) Modify structure to prevent excitation. (b) Use vibration isolation. (c) Use sleeve bearings.
			4. Contamination of raceways with gear wear detritus.	i. Unshielded type bearings used.	(a) Utilize shielded bearings.
				ii. Gearing undergoing a failure mode.	(a) Improve gear lubrication. (b) Change to a harder gear material and/or surface treatment of both faces. (c) Eliminate gear misalignment.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Detritus deposited in bearing as internal pressure is bled to vacuum (pertains to output shaft bearings).	(a) Use vent (with filter) in motor housing to avoid airflow through bearings.
			5. Corrosion of raceways and balls.	i. Unit is subject to humid atmospheres.	(a) Use a sealed motor either hermetically sealed, or sealed with shaft seals or labrinth. (b) Locate motor in sealed section of instrument or spacecraft. (c) Use stainless steel ball bearings.
				ii. Chemical reaction between bearing materials and lubricant.	(a) Change to corrosion inhibiting lubricant. (b) Utilize 440C ball bearings.
				iii. Oil migration loss during storage.	(a) Use barrier film adjacent to lubed area. (b) Use ball retainer that acts as oil resevoirs. (c) Use interference fit bearing shields. (d) Use supplementary oil reservoirs.
			6. Premature failure accompanied by wear.	i. Inadequate lubrication - bearing operates in severe boundary regime.	(a) Use more viscous lubricant and/or better surface finish bearing to achieve partial EH D conditions.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
					(b) Use lubricant with better lubricating characteristics. (c) Increase rpm to achieve partial EHD conditions. (d) Chemically contaminated balls &/or raceways causing non-wettable surfaces.
				ii. Inadequate lubrication due to evaporation loss.	(a) Use sealed motor either hermetically sealed or sealed with shaft seals or labrinth. (b) Utilize oil or grease with lower evaporation rate. (c) Locate motor in sealed section of instrument or spacecraft. (d) Use ball retainers that can act as oil reservoir and add supplementary reservoir. (e) Reduce extent of preflight testing. (f) More frequent lubrication refurbishments.
				iii. Inadequate lubrication due to migration loss.	(a) Use barrier films adjacent to lubed area. (b) Use ball retainers that act as oil reservoirs.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
					(c) Use interference fit bearing shields. (d) Use supplementary oil reservoir.
				iv. Excessive temperature causing degradation of wet lube (polymerization) followed by severe wear.	(a) Use more efficient motor. (b) Use motor concept wherein heat flux path bypasses gearhead. (c) Improve heat flux path from motor mounting. (d) Use lube with higher temperature capability. (e) Reduce normal load or preload on bearings. (f) Change from grease to oil lubrication.
			7. High oil viscosity.	i. Actual operating temperature is lower than anticipated.	(a) Change to an oil or grease with lower temperature/viscosity coefficient and less viscous at the low temperature. (b) Change to dry film lube. (c) Use more powerful motor. (d) Use heater to avoid the low temperature condition.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				ii. Oxidation of lubricant.	(a) Use an oxidation inhibitor. (b) Use a more powerful motor.
				iii. Lubricant properties deviate from specification.	(a) Impose stricter quality control procedures.
			8. Contamination of raceways with brush debris.	i. Vibration or maneuver forces disturbs debris causing it to enter bearing.	(a) Use shielded or sealed bearing. (b) Use flange/slinger on shaft adjacent to bearing.
				ii. Brush debris deposited in bearing as internal pressure is bled to vacuum (pertains to output shaft bearings).	(a) Use vent (with filter) in motor housing to avoid airflow through bearings. (b) Use contact type shaft seal (and inert gas pressurization). (c) Hermetically seal. (d) Avoid use of brush type motor.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
			9. Ball retainer wear, combined with race contamination leading to fatigue and rupture of retainer	i. Loss of lubricant	(a) Change from a ball riding retainer to a land riding retainer, oil impregnated. (b) Use supplementary oil reservoirs.
				ii. Excessive ball/ball pocket interface forces due to excessive bearing loads.	(a) Balance gear tooth-load. (b) Reduce bearing pre-load.
				iii. Excessive ball/ball pocket interface forces due to widely varying ball velocities and loads.	(a) Add preload - but check hertz stress and fatigue life.
				iv. Excessive friction between land riding ball retainer and land (particularly at extremes of temperature).	(a) Increase clearance between retainer and raceway. (b) Use lower viscosity oil, or change from grease to oil. (c) Use a more dimensionally stable retainer material.

TABLE 5.1-2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				v. Skidding at ball during acceleration (particularly in frequent start/stop duty cycles).	(a) Utilize preload to provide ball traction
				vi. Bearing misalignment due to improper housing tolerances.	(a) Remedy faulty Q.C. procedures.
				vii. Bearing misalignment due to out-of-roundness at housing bore.	(a) Improve machining expertise. (b) Use material with better dimensional stability.
				viii. Improper fit of shaft onto inner race of bearing with consequent creep, scoring and cracking.	(a) Use correct tolerance and size code.
				ix. Shaft bending (due to gear load).	(a) Use larger shaft or shorter bearing span. (b) Balance gear lead.
			10. Spalling of raceways (and sometimes balls); classical subsurface fatigue initiated by subsurface imperfections.	i. Hertz stresses much higher than anticipated due to improper choice of bearing.	(a) Retrofit larger capacity bearings.
				ii. Inadequate determination of statistical reliability - failed bearing is a low life outcome of the Weibull distribution.	(a) Retrofit bearings, adequately derated for load, to provide the required life at 99.9% to 100% reliability.

TABLE 5.1.2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Excessive hertz stresses due to large race radius.	(a) Utilize bearing with smaller raceway radius (but higher friction).
				iv. Excessive portion of fatigue life expended during vibration environment.	(a) See Item iii above.
				v. Bearing not operating in favorable lubrication regime. Hertz stresses are therefore higher than otherwise.	(a) Use more viscous lube, higher speed, better surface finishes and precision grade bearings.
				vi. Inferior metal (high inclusion density)	(a) Improve quality of steel. (b) Use selective portion heat; i.e. impose stricter quality control.
			ii. Spalling of raceways (and sometimes balls) due to surface initiated fatigue.	i. Non-metallic inclusions coincident with surface of ball race, causing pits (micro-spalling).	(a) Enforce quality control standards on steel supplier, or improve quality control standards. (b) Select a better quality steel. (c) Impose a more rigid finished part inspection routine.
				ii. Wear debris from asperity interaction imbedding or causing dents in ball path, from which spalling nucleates.	(a) See Item C 9 above.

TABLE 5.1.2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
				iii. Lubricant contaminant causing dents and stress concentrations in ball path which initiates spalling.	(a) Use stricter quality control on lubricant. (b) Improve cleanliness of assembly site. (c) Prevent ingress of foreign matter through labrinth seal by installing filter/breather in motor housing.
				iv. Grinding and housing imperfections - nicks and furrows, causing stress concentrations.	(a) Use stricter quality control measure. (b) Demand high quality (more expensive) bearings.
			12. Contaminated lubricant.	i. Inadequate cleanliness during fabrication and assembly.	(a) Impose more rigerous cleanliness provisions on motor and bearing manufacturers.
				ii. Inadequate lubricant cleanliness.	(a) Impose more rigerous cleanliness provisions on lubricant supplier, and/or; (b) Microscopically select lubricant for cleanliness for each bearing.
			13. Encroachment of ball track on race land.	i. High thrust on ordinary (conrad) type bearing (due possibly to differential thermal expansion between casing and rotor).	(a) Use precision grade bearing. (b) Use angular contact type bearing. (c) Allow more bearing end float.

TABLE 5.1.2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANISMS
			13. Encroachment of ball track on race land (cont)	ii. High thrust on ordinary, conrad type bearings (due possibly to differential radial interferences between housing and bearing - at low temperature - restricting relative motion of bearing).	(a) Use bearing housing material with compatible thermal coefficient of expansion. (b) Perform worse case analysis and correct errors in dimensioning.
			14. Retainer instability.	i. Inadequate ball/ball pocket lubrication.	(a) Utilize porous oil impregnated ball separator.
				ii. Inadequate clearance or lubrication at separator/race-land interface.	(a) Utilize porous oil impregnated ball separator. (b) Adjust separator/race land clearance to provide hydrodynamic operation. (c) Use oil with more suitable viscosity range. (d) Change from outer race to inner race riding separator, or vice versa. (e) Use preloading to unify ball velocity and ball/ball pocket forces.

TABLE 5.1.2. FAILURE MECHANISM ANALYSIS (cont'd)

FAILURE MODE	FAILURE REASON	REL RANK	FAILURE MECHANISM	POSSIBLE CAUSES	HOW TO ELIMINATE/MINIMIZE FAILURE MECHANSIMS
D. PLAIN BEARINGS (as used for planet gears of some gearheads). Loss of Power (complete or partial)	Excessive losses due to bearing friction	IV	1. Excessive wear or galling of bearing interfaces.	i. Degraded dry film lubrication	(a) Use a more tenacious dry film lube. (b) Change to a wet lube (with output shaft seals). (c) Utilize ball bearings in planet gears.
				ii. Loss of wet lubricant	(a) Use sealed motor either hermetically sealed with shaft seals or labrinth. (b) Locate motor in sealed environment. (c) Use grease lubricant instead of oil. (d) Use oil or grease with lower evaporation rate.
			2. High lubricant viscosity.	i. Actual operating environment is colder than anticipated.	(a) Use oil or grease with lower temperature/viscosity coefficient and less viscous at low temperature. (b) Change to dry film lube. (c) Use a heater to avoid the low temperature condition.

TABLE 5.3-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE RATIOS

Environment	$\lambda(10^{-9})$	OP/NON-OP Ratio
Non-operational (overall)	14.48	-
Operational (air) (ball)	3902.0	269.47
Operational (air) (roller/rod)	58137.0	4014.98

6.0 Compressors

This section contains reliability information and analysis on compressors. Although there are several types of compressors, the scarcity of data made it necessary to combine it into one general category of pneumatic compressors.

6.1 Storage Reliability Analysis

6.1.1 Failure Mechanisms

Failure mechanisms and modes, detection methods and corrective measures are summarized in Table 6.1-1.

6.1.2 Design for Long Life Assurance

Design factors to insure long life of compressors are summarized in Table 6.1-2.

6.1.3 Non-Operational Failure Rate

No failures were reported in the available storage data. The non-operational failure rate is estimated to be less than 4085.0 failures per billion hours.

6.1.4 Non-Operational Failure Rate Data

Only one source of storage data was located for pneumatic compressors. This contained 0.2448 million hours of storage with no failures giving a failure rate of 4.085 failures per million hours assuming one failure.

6.2 Operational/Non-Operational Failure Rate Comparison

Ratios of operational failure rate (under both environments) to non-operational failure rate are summarized in Table 6.2-1.

TABLE 6.2-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE RATIOS

ENVIRONMENT	FAILURE RATE (10^{-9})	OPERATING/NON-OPERATING RATIO
Non-Operating	4085.0	-
Operating (Ground)	12700.0	3.1
Operating (Air, Helicopter)	297560.0	72.8

TABLE 6.1-1. FAILURE MECHANISM ANALYSIS - COMPRESSORS

PART AND FUNCTION	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODE
Bearings - Maintain shaft or rotor alignment under load.	Loss of power due to high starting or running torque	1	Dry film lubrication deterioration.	Slow starts and high motor currents.	1) Select stable dry lubricant 2) Change to wet lubricant. 3) Reduce rpm for pump, compressor, or fan. 4) Utilize ball bearings, or
			Loss of wet lubrication.		1) Use grease lubricant instead of oil. 2) Use lubricant with low evaporation rate. 3) Change ball retainer to land type oil impregnated.
			Contamination.	Visual examination.	1) Assemble on laminar flow work bench. 2) Prevent foreign particles from entering component.
			Vibration	Erratic torque requirements.	1) Operate units during periods of high vibration (if possible). 2) Use sleeve bearings. 3) Employ vibration isolators.
Seals, static and dynamic - contain flow medium	Leakage	2	Wear or incompatibility of fluids.	Visual observation or inefficiency of unit	1) Dielectric coolant fluids. 2) Magnetic coupling. 3) Use redundant static seals. 4) Positive seal housing with welded construction.
Housing - support moving parts and contain flow medium.	External leakage	3	Housing porosity, stress corrosion, poor workmanship, cracked housings.	Visual observation, proof pressure tests for housings used in fluid systems.	1) Inpregnate castings with sealant. 2) Use bar stock that has vacuum melt treatment. 3) Weld housing parts. 4) Increase housing design margins.

TABLE 6.1-1. FAILURE MECHANISM ANALYSIS -COMPRESSORS (cont'd)

PART AND FUNCTION	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHODS	HOW TO ELIMINATE/ MINIMIZE FAILURE MODE
Failure of rotating or sliding part (impellers, fan blades, etc.) pass flow medium through	Structural failure	4	Fatigue, overstressed part.	Visual observation.	1) Use large safety margins. 2) Eliminate stress concentration points

TABLE 6.1-2. DESIGN FACTORS FOR LONG-LIFE ASSURANCE PART/ COMPONENT: COMPRESSORS

Design Factors	Remarks
Bearing Material	Flow media and lubricant are the most important considerations (see text for particular recommendations). Ceramic bearings may be required for long life. Metallic bearings must be free of inclusions or stringers.
Dynamic Seals	Avoid dynamic seals which wear and cause contamination by using: 1) Magnetic coupling system. 2) Submerged pump/rotor assembly. 3) Totally wet pump/motor.
Static Seals	Brazed or welded housing joints are preferred to captive types of seals for fluid systems.
Materials Compatibility	Inert fluids are recommended, such as Coolanol or Oronite for coolant systems.
Housing	Housing leakage in fluid systems can be solved by: 1) Impregnation of castings with sealant substance. 2) Using vacuum melt material to eliminate stringers or inclusions.
Noise Suppression	Methods to suppress noise include: 1) Mechanical isolation. 2) Sound suppressor/acoustical insulation material. 3) Non-metallic duct and connectors.
Journal Bearings	Advantages: simple, inexpensive, and can be used in small spaces. Disadvantages: higher coefficient of friction than ball bearings
Ball Bearings	Advantages: low friction, short length, can accept both radial and thrust loads. Disadvantages: diametrically large, costs more than journal bearings.
Roller Bearings	Advantages: higher load capacity than ball bearings, diametrically smaller than ball bearings. Disadvantages: longer than ball bearings, costs more than journal bearings.

6.3 Conclusions & Recommendations

6.3.1 Conclusions

Quality grades were not well defined for the compressor data collected. The results represent failure rate averages over the quality spectrum. Generally compressor types within each class of hydraulic or pneumatic system could not be identified. Therefore, failure rates derived at this high level have higher statistical confidence than those of the sub-categories and should be utilized unless specific information is available to further define the type of compressor under consideration.

6.3.2 Recommendations

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented.

7.0 Cylinders

Cylinders have been classified in accordance with the hydraulic or pneumatic system rating described under MIL-H-24475B. The missile hydraulic systems are of the following types and class, as specified.

Type I	-	-65° to 160°F Temperature range
Type II	-	-65° to 275°F " "
Type III	-	-65° to 450°F " "
Type IV	-	Temperature ranges extending above +450°F
Class	-	3,000 pounds per square inch (psi).

7.1 Storage Reliability Analysis

7.1.1 Failure Modes and Mechanisms

Table 7.1-1 gives a summary of the major failure modes, failure mechanisms, detection methods and methods to minimize each failure mode.

7.1.2 Non-Operational Failure Rates

The available failure data did not contain information about specific cylinder types. Therefore the resulting failure rate is viewed as that for a general class of cylinders. Based on this, the storage failure rate for cylinders is 75.0 failures per billion hours.

7.1.3 Non-Operational Failure Rate Data

The failure rate was based on storage data consisting of 160 million hours with 12 failures reported. No information as to the specific types of cylinders was available.

7.2 Operational/Non-Operational Failure Rate Comparisons

Rates of operational to non-operational failure rates for the different environments are shown in Table 7.2-1. The overall ratio of operating (under all environments) to non-operating failure rate is 2819.12. The operational failure rate was obtained from the RADC Nonelectronic Reliability Handbook. The failure rate comparison is shown using the

TABLE 7.1-1. FAILURE MECHANISM ANALYSIS - CYLINDERS

Part & Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Cylinder piston seat	Internal leakage	1	Internal leakage is caused by: 1) contamination 2) damaged physical properties 3) aging O-rings 4) blemishes 5) pressure 6) bending 7) swelling material	Pre-Installation Testing & run-in will detect this type of failure. Post-Installation System pressure measurements can be used to determine if cylinder is leaking.	Control cleanliness level for parts, components & systems. Areas that can trap contaminants should be eliminated from cylinder designs.
Separator material	Swelling & Bending	2	Separator stuck in intermediate position due to: 1) contamination 2) misalignment uniformity 3) swelling material 4) aging material 5) bending (excess) material	Pre-Installation Testing will determine this type of failure prior to cylinder installation into a system. Post-Installation Cylinder position indicator or system pressure measurements.	Allow conservative force margins for opening and closing of seat. Run-in test or margin tests should reveal this type of failure.
Cylinder body (support elements & contain media)	External leakage	3	Leakage through: 1) static seals, O-rings 2) plumbing connections 3) cylinder body (porosity) 4) blemishes 5) pressure	Pre-Installation Component tests will reveal this failure mode. Post-Installation System pressure (fluid) loss or visual detection of leaks.	Methods to control these failures include: 1) welded external body construction 2) install cylinder into system with permanent mechanical connections 3) impregnate castings with sealants 4) use of vacuum melt metals to control inclusions or stringers.

operating failure rate from three environments: ground, air and helicopter.

TABLE 7.2-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE COMPARISON

ENVIRONMENT	$\lambda \times 10^{-9}$	OPERATING TO NON-OPER. RATIO
Non-Operating	75.	-
Ground, Operating	33.154	442.05
Air, Operating	212.168	2828.91
Helicopter, Operating	973.269	12976.92

7.3 Conclusions and Recommendations

7.3.1 Conclusions

Comparison between dormant and storage reliability data indicates no significant difference between the two. Therefore, the dormant and storage data were combined in all analyses.

Quality grades were not well defined for the cylinder data collected. To determine quality grades extensive searching through component specifications and drawings would be required. It was therefore impossible to determine the effect, if any, of quality levels. The results presented in this report represent failure rate averages over the quality grade spectrum.

In general, cylinder types within each class (Type I, Type II, etc.) could not be identified. Therefore, failure rates derived at this high level have higher statistical confidence than those of the sub-categories and should be utilized unless specific information is available to further define the type of accumulator under consideration.

7.3.2 Recommendations

Record keeping for cylinders kept on storage should be improved, specifically the identification of quality grades and cylinder description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented.

8.0 Filters

This section contains reliability information and analysis on hydraulic filters. The purpose of a hydraulic filter is to remove contamination from a fluid. A filter may be thought of as a series of small orifices or a semipermeable membrane. As the fluid flows through it, contaminants are removed by being mechanically blocked without harm to the useful characteristics of the fluid.

Since a filter will pass small particles and stop larger ones, a means of designating the degree of filtration must be used. Filters are rated in terms of "nominal" and absolute" filtration. A filter with a particular nominal rating is one that will remove 98 percent of all particles of that particular size and larger. A filter is given an absolute rating when it is said to stop all particles above a particular size; it has often been said that there are no absolute filters. It is extremely difficult to contain all fibers that are greater in length than the absolute rating of the filter, since they may also be many times smaller in diameter. In order to define the capabilities of a filter completely, both a nominal and an absolute rating should be specified.

The main hydraulic filter elements used in aerospace weapon systems are of two basic classifications: cleanable and non-cleanable. Elements classified as cleanable are usually sintered bronze or woven wire mesh. Sintered bronze types are susceptible to media migration, require unusually refined manufacturing techniques and quality control, and are virtually impossible to clean completely. Woven wire mesh filter elements are the most common cleanable type, and are not as susceptible to media migration. Equipment that will adequately clean a so called "cleanable" filter is not often available to maintenance personnel. Such filters are usually either discarded or rinsed in fluid and returned to service; the latter procedure adds to the contamination in the element, and can lead to a clogged filter.

In the recent past, noncleanable filter elements have been considered the least desirable of the elements available. They are usually made of resin-impregnated pressed paper, and are susceptible to media migration, aging embrittlement, and collapse. It has been shown that storage time directly affects the capability of paper filter elements to withstand their rated differential pressure. With the advent of woven wire elements, paper filters were not considered adequate for high-performance systems. Paper filters are highly regarded at the present state-of-the-art, however. Recent investigations of the contamination levels in naval aircraft hydraulic systems indicate that paper elements provide cleaner systems than wire mesh in that application. Another recent test, conducted by commercial airlines, has resulted in the conclusion that paper elements used in series with woven wire mesh filters provide significantly better contamination control than either wire mesh or sintered metal filters.

Present thinking on the subject, then, stems from two considerations: paper filters are actually depth filters and tend to remove fine as well as coarse particles, whereas woven wire elements tend to pass most particles finer than their nominal rating; also, paper filters are inexpensive and can be discarded. The trend now appears to be in the direction of a paper element backed up by a wire mesh element to protect against mechanical failure of the paper element. (A limiting factor, for missile systems in particular, is the fact that paper filters cannot be used at temperatures above 275°F.)

8.1 Failure Modes and Mechanisms

Principal filter failure modes are listed in Table 8.1-1.

TABLE 8.1-1.

FAILURE MODES FOR FILTERS

1. Leaking
2. Cracked or Broken
3. Out of Tolerance
4. Broken
5. Excessive Wear
6. Cracked
7. Clogged
8. Burst
9. Warped
10. Damaged-mishandling
11. Structural Failure
12. Bubble Test

The failure modes are primarily operational or quality problem. However, storage of the filters over long periods have similar problems; such as leaking, cracks and clogging.

8.1.1 Storage Failure Rate

The storage or non-operational data for filters are summarized in Table 8.1-2.

TABLE 8.1-2 STORAGE FAILURE RATE - FILTERS

Type	Failure Rate 10^{-9}
Filter (fine, 100 micron)	<6.2
Filter (coarse, 100 micron)	<2.1
Other types	<017.7
TOTAL (overall)	<1.4

8.1.2 Non-Operational Storage Data

Non-operational storage data was collected from eight programs. Storage hours and number of failures for each are shown in Table 8.1-3.

8.2 Operational/Non-Operational Failure Rate Comparisons

Table 8.2-1 presents operational to non-operational failure rate ratios for the different operating environments. For comparison purposes, the combined filter failure rate is used since information on specific types was not available in the operational data.

TABLE 8.2-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE RATIOS

Environment	$\lambda(10^{-9})$	Operational/Non-Operational Ratio
Non-operational	1.4	-
Operational, ground	94640.	67600
Operational, heli.	214790.	153421
Operational, air	9860.	7043

8.3 Conclusions and Recommendations

8.3.1 Conclusions

The specific filter type could not be identified and quality grades were not well defined. To determine the specific filter types used and quality grades, extensive searching through component specification and drawings would be required. It was therefore impossible to determine the effect, if any, of quality level.

8.3.2 Recommendations

Record keeping for filters kept on storage should be improved, specifically the identification of quality grades and filter description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only be their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented.

TABLE 8.1-3. STORAGE DATA

PART CLASSIFICATION: FILTERS NUMBER: ________

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HOURS (10^6)	FAILURE RATE (10^{-9})	UPPER 90% CONFIDENCE	YEAR OF REPORT
Gas, 25-Micron Absolute	Storage	MSL	1897	0	20.397	-	.112889	67
.65 Hydraulic fluid	Mechanical Storage	MSL	2506	0	9.3	-	.247591	65
Filter, Liquid	Non-electric	GRD	-	0	26.562	37.64	-	-
Filter, (fine, 100 micron	Stora_ _	MSL	1	0	110.05	9.1		
Filter, (fine, 100 micron)	Storage	MSL	1	0	50.0	20.0		
Filter (coarse, 100 micron)	Storage	MSL	4	0	440.2	2.3	-	-
Filter (coarse, 100 micron)	Storage	MSL	4	0	50.	20.		
TOTAL			10	0	706.509	1.41		

9.0 FITTINGS/CONNECTIONS

Fittings and connections are commonly found in hydraulic and pneumatic systems. They range from metal connections to rubber pipes and hoses. Included within this report are two types of connections and two types of fittings.

9.1 Storage Reliability Analysis

9.1.1 Failure Mechanisms and Modes

Principal failure mechanisms and modes are summarized in Table 9.1-1.

TABLE 9.1-1. FITTINGS, CONNECTIONS FAILURE MECHANISMS

1. Leaking
2. Physical Damage
3. Deteriorated
4. Cracked
5. Broken

These failure modes are experienced for storage and operational storage failure modes. Lack of quality control may be responsible for some failures.

9.1.2 Non-Operational Failure Rates

Non-Operational failure rates for connections and fittings are given in Table 9.1-2.

TABLE 9.1-2. STORAGE FAILURE RATES - CONNECTIONS AND FITTINGS

TYPE	ENVIRONMENT	$\lambda(10^{-9})$ FITS
Connections		
Solder	Ground	(<.029)
General	Ground	1.5
	Submarine	(<158.7)
Fittings		
Quick Disconnect (Liq.)	Ground	(<1510.6)
	Submarine	499.0
Hydraulic	Ground	(<3030.3)

9.1.3 Non-Operational Data

Non-operational data was collected from two sources. Storage hours and number of failures for each are shown in Table 9.1-3.

TABLE 9.1-3. DATA - CONNECTIONS & FITTINGS

TYPE	ENVIRONMENT	FAILURES	PART HOURS ($x10^6$)
Connections			
Solder	Ground	0	34,900.0
General	Ground	17	11,603.4
	Sub	0	6.3
	TOTAL	17	46,509.7
Fittings			
Quick Disconnect			
Liquid	Ground	0	.662
	Sub	4	8.012
Hydraulic	Ground	0	.330
	TOTAL	4	9.004

9.2 Conclusions and Recommendations

9.2.1 Conclusions

Quality grades and descriptions were not well defined for connections or fittings. The failure rates presented are significant and represent a cross section of connections and fittings.

9.2.2 Recommendations

The failure rates calculated should be used for reliability prediction until additional data can be collected on specific types of fittings and connections.

10.0 Gaskets and Seals

This section contains reliability information and analysis on gaskets and seals. Data for gaskets (general types), static seals and dynamic seals are included.

10.1 Storage Reliability Analysis

10.1.1 Failure Mechanisms and Modes

Principal failure mechanisms and modes are summarized in Table 10.1-1.

TABLE 10.1-1. SEALS, GASKETS FAILURE MODES

1. Leaking
2. Physical Damage
3. Torn
4. Cut
5. Deteriorated
6. Forged
7. Cracked
8. Broken
9. Excessive Wear
10. Distorted

These failure modes are for operation and storage. The primary storage failure modes are leaking and deteriorated. Many of the operational failures are caused by lack of quality control.

10.1.2 Non-Operational Failure Rates

Non-operational failure rates for gaskets and seals are summarized in table 10.1-2.

TABLE 10.1-2. STORAGE FAILURE RATES - SEALS

Type	λ (x 10^{-9})
Seals, Dynamic	(<5000.0)
Seals, Static	(<10.0)
Overall	(< 9.9)

10.1.3 Non-Operational Failure Data

Non-operational failure data was collected from two sources. Storage hours and number of failures for each are shown in table 10.1-3.

TABLE 10.1-3. STORAGE RELIABILITY DATA - SEALS

Type	Failures	Hours (10^6)
Dynamic	0	.2
Static	0	100.0
TOTAL	0	100.2

10.2 Operational/Non-Operational Failure Rate Comparison

Table 10.2-1 presents operational to non-operational failure rate ratio for the air operational environment. For comparison purposes, dynamic seals and static seal data (storage) was combined.

TABLE 10.2-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE RATIO

Environment	$\lambda(10^{-9})$	Ratio
Non-operational	9.9	-
Operational, air	48996.0	4,949.09

10.3 Conclusions and Recommendations

10.3.1 Conclusions

Quality grades and descriptions of gaskets and seal types were not well defined for the data collected. To determine quality grades, extensive searching through component specifications and drawings would be required. It was therefore impossible to determine effect. The failure rates derived are significant and represent a cross section of gasket and seal types.

10.3.1 Recommendations

The failure rates calculated should be used for reliability prediction until additional data can be collected on specific types of gaskets and seals.

Record keeping for gaskets and seals kept on storage should be improved, specifically the identification of quality grades and gasket and seal description. This should be done within existing data collection systems.

Additional research and data collection to attain a better definition of the data already on hand. More detailed identification of those units classified only in their generic names should be attempted.

11.0 O'RINGS

The reliability data for O'rings is incomplete at this time.

The best estimate for reliability prediction for O'rings range from 4.0 fits to 78 fits.

12.0 PISTONS

Reliability data for non-operating prediction for pistons is being investigated. Prediction will be made as soon as data is available.

13.0 Pumps

Pumps, defined as devices for transferring energy to a liquid to cause it to flow in a duct, opposing gravity and other forces, are normally classified into two principal categories, namely positive-displacement and kinetic. These are defined as:

A. Positive displacement: A pump wherein liquid is caused to flow in volumetric proportion to an alternating increase and decrease of the volume in the pump body. The principal classes of positive-displacement pumps are reciprocating, rotary, and flow case.

B. Kinetic: A pump wherein the energy transferred to the liquid is mainly kinetic energy nonuniformly imparted to the liquid, with subsequent distribution throughout the liquid. The principal classes of kinetic pumps are centrifugal and peripheral plus some special types such as jet (ejector), jet-boosted, gas-lift, and ram.

Pumps are further defined in five classes.

A. Reciprocating: A positive-displacement pump wherein liquid is moved by displacement as the result of an alternating mechanical increase and decrease in the volume of the pump body through reciprocating pumps,i.e.piston, plunger, and diaphragm.

B. Rotary: A positive-displacement pump wherein liquid is caused to flow by displacement induced by a rotating device which creates cavities that move from suction to discharge, forcing the liquid along. The principal types of rotary pumps are gear, screw, vane, and cam.

C. Blow case: A positive-displacement pump wherein liquid is displaced from a container by an immiscible liquid or a gas.

D. Centrifugal: A kinetic pump wherein energy is imparted to the liquid principally by the action of centrifugal force. The principal types of centrifugal pumps are radial flow, axial flow, and mixed flow.

E. Peripheral: A kinetic pump wherein energy is imparted to the liquid by a combination of centrifugal and tangential shear forces.

The classes of pumps are defined by fourteen separate types.

A. Axial flow: A centrifugal pump using a propeller or screw in which most of the head is developed by the propelling action of the vanes on the liquid in an axial direction.

B. Cam: A rotary pump wherein the rotating device consists of an eccentrically bored cam mounted concentrically within a cylindrical casing with a radial seal vane.

C: Diaphragm: A pump using a reciprocating flexible sheet or disk which comprises a portion of the walls of the pump body and which is flexed to effect an alternating increase and decrease of the volume.

D. Gear: A rotary pump wherein the rotating device consists of two or more gears.

E. Jet (ejector): A kinetic pump without moving parts, in which a jet of fluid (usually steam or water) is discharged at high velocity into a venturi-shaped diffuser, together with entrained fluid surrounding the jet, the velocity energy of the total effluent being converted to pressure energy.

F. Jet-boosted: A single- or multistage centrifugal or peripheral pump having an ejector booster in series with it.

G. Mixed flow: A centrifugal pump in which the head is developed by a combination of the action of the radial- and axial-flow types of pumps.

H. Piston: A pump using a reciprocating member (with self-adjusting packing), both the member and packing moving within a chamber. Motion is imparted to the reciprocating member by an affixed rod of smaller diameter.

I. Plunger: A pump using a reciprocating member moving within a chamber fitted with stationary packing.

J. Radial flow: A pump wherein the head is developed by the sum of centrifugal force and the kinetic energy imparted to the liquid by impeller vanes and wherein the liquid enters at or near the axis of rotation of the impeller.

K. Screw: A rotary pump wherein the rotating device consists of two or more screw rotors.

L. Vane: A rotary pump wherein the rotating device consists of a hub fitted with a series of vanes, blades, or buckets.

13.1 Pump Storage Reliability Analysis

13.1.1 Failure Mechanisms

The principal pump classifications are depicted in Table 13.1-1.

TABLE 13.1-1. PUMP CLASSIFICATIONS

Positive Displacement	Kinetic
Reciprocating	Centrifugal
Piston	Radial Flow (Cent)
Plunger	Axial Flow (Prop)
Diaphragm	Mixed Flow
Rotary	Peripheral (Reg. Turbine)
Gear	Special
Screw	Jet (Ejector)
Vane	Jet-Boosted
Cam	
Blow Case	

Table 13.1-2 reflects the failure modes, mechanisms, detection methods and comments as to how to minimize the failure modes. The table only reflects the most predominant failure modes. Also specified are whether the failure mechanisms are storage, operational, or quality control problems.

13.1.2 Design Factors for Long Storage and Operational Life

Long life assurance design factors for the basic pump components are summarized in Table 13.1-3.

13.1.3 Pump Failure Rates

Over 1.2 billion part hours of storage data are included in this report. The individual data points, containing pump classification, environments and failure data are summarized in Table 13.1-3. Identification of pump type was not uniform. Sometimes specific types were identified (e.g., fixed displacement-vane) and in some other cases

TABLE 13.1-2. FAILURE MECHANISM ANALYSIS - PUMPS

PART	FAILURE MODE	REL. RANK	FAILURE MECHANISM	DETECTION METHOD	HOW TO ELIMINATE/ MINIMIZE FAILURE MODE
Bearings	Loss of power due to high starting or running torque.	1	Dry film lubrication deterioration (storage)	Slow starts and high motor currents.	1) Select stable dry lubricant. 2) Change to wet lubricant. 3) Reduce rpm for pump, compressor, or fan. 4) Utilize ball bearings.
			Loss of wet lubrication (storage)		1) Use grease lubricant instead of oil. 2) Use lubricant with low evaporation rate. 3) Change ball retainer to land type oil impregnated.
			Contamination	Visual examination.	1) Assemble on laminar flow work bench. 2) Prevent foreign particles from entering component.
			Vibration (operational)	Erratic torque requirements	1) Operate units during periods of high vibration (if possible). 2) Use sleeve bearings. 3) Employ vibration isolators.
Seals-static & dynamic	Leakage	2	Wear or incompatibility of fluids (operation and storage)	Visual observation or inefficiency of unit.	1) Dielectric coolant fluids. 2) Magnetic coupling. 3) Use redundant static seals. 4) Positive seal housing with welded construction.
Housing	External Leakage	3	Housing porosity, stress corrosion, poor workmanship, cracked housings (operation, storage & QC)	Visual observation, proof pressure tests for housing used in fluid systems.	1) Impregnate castings with sealant. 2) Use bar stock that has vacuum melt treatment. 3) Weld housing parts. 4) Increase housing design margins.

TABLE 13.1-3 DESIGN FACTORS FOR LONG-LIFE ASSURANCE PART/COMPONENT: PUMPS

Design Factors	Remarks
Bearing Material	Flow media and lubricant are the most important considerations. Ceramic bearings may be required for long life. Metallic bearings must be free of inclusions or stringers.
Dynamic Seals	Avoid dynamic seals which wear and cause contamination by using: 1) Magnetic coupling system. 2) Submerged pump/rotor assembly. 3) Totally wet pump/motor.
Static Seals	Brazed or welded housing joints are preferred to captive types of seals for fluid systems.
Materials Compatibility	Inert fluids are recommended, such as Coolanol or Oronite for coolant systems.
Housing	Housing leakage in fluid systems can be solved by: 1) Impregnation of castings with sealant substance. 2) Using vacuum melt material to eliminate stringers or inclusions.
Noise Suppression	Methods to suppress noise include: 1) Mechanical isolation. 2) Sound suppressor/acoustical insulation material. 3) Non-metallic duct and connectors.
Journal Bearings	Advantages: simple, inexpensive, and can be used in small spaces Disadvantages: higher coefficient of friction than ball bearings
Ball Bearings	Advantages: low friction, short length, can accept both radial and thrust loads Disadvantages: diametrically large, costs more than journal bearings
Roller Bearings	Advantages: higher load capacity than ball bearings, diametrically smaller than ball bearings Disadvantages: longer than ball bearings, costs more than journal bearings

TABLE 13.1-4. STORAGE DATA FOR PUMPS

	PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. $\times 10^6$	FAILURE RATE $\times 10^{-6}$	UPPER 90% CONFIDENCE	YR. OF REPORT/ SOURCE
1.	PUMP, PISTON	Variable Dis-placement	MSL B		77	110.0	.7	.812	
2.	PUMP, VANE	Variable Dis-placement	MSL B		55	110.0	.5	.598	
3.	PUMP, HYDRAULIC	Desert	Air-craft	2	1	.2978	3.3	7.760	69/3
4.	PUMP, HYDRAULIC	Ambient	-	121	0	2.1353	<.468	-	69/3
5.	PUMP, HYDRAULIC	70°F at 55% relative humidity	STR AGE	2	5	.1752	28.5380	53.5	69/3
6.	PUMP, HYDRAULIC	unknown	-	30	0	.7884	<1.268	2.93	69/3
7.	PUMP, HYDRAULIC	Ambient	STR	630	0	7.3146	<0.1367	.316	69/3
8.	PUMP, HYDRAULIC	Ambient	STR	2538	1	34.5494	0.0289	.113	69/3
9.	PUMP, HYDRAULIC	Ambient	STR	2600	6	46.9536	0.12778	.225	69/3
10.	PUMP	Fuel	GND	-	0	8.070	<0.114	.286	74
11.	PUMP	Hydraulic	GND	-	0	21.375	<0.043	.108	74
12.	PUMP	Hydraulic	SUB	-	3	2.004	1.497	3.33	74
13.	PUMP, GEAR	Fixed Dis-placement	MSL	1	64	110.05	.5815	.688	-
14.	PUMP, PISTON	Fixed Dis-placement	MSL A		25	50.0	.50	.654	
15.	PUMP, VANE	Fixed Dis-placement	MSL A		15	50.0	.30	.449	
16.	PUMP, GEAR	Fixed Dis-placement	MSL A		15	50.0	.3	.426	

TABLE 13.1-4. STORAGE DATA FOR PUMPS (cont'd)

PART DESCRIPTION	FUNCTIONAL APPLICATION	ENV.	PART POP.	NO. OF FAILURES	PART HRS. x 10^6	FAILURE RATE x 10^{-6}	UPPER 90% CONFIDENCE	YR. OF REPORT/ SOURCE
17. PUMP, CENTRIFUGAL	Fixed Displacement	MSL A		15	50.0	.3	.426	
18. PUMP, PISTON	Fixed Displacement	MSL A		10	50.0	.2	.309	
19. PUMP, PISTON	Variable Displacement	MSL A		10	50.0	.2	.309	
20. PUMP, VANE	Variable Displacement	MSL A		10	50.0	.2	.309	
21. PUMP, PISTON	Fixed Displacement	MSL B		55	110.0	.5	.598	
22. PUMP, VANE	Fixed Displacement	MSL B		33	110.0	.3	.379	
23. PUMP, GEAR	Fixed Displacement 12000 rpm	MSL B		55	110.0	.5	.598	
24. PUMP, GEAR	Fixed Displacement	MSL B		33	110.0	.3	.397	
25. PUMP, CENTRIFUGAL	Fixed Displacement	MSL		22	110.0	.2	.266	
26. PUMP, PISTON	Fixed Displacement	MSL B		22	110.0	.2	.266	

only the general category of "hydraulic pump" was available. In a large number of cases, failure modes and mechanisms were not disclosed. Quality grades were not defined for any of the pump data. Therefore, failure rates derived in this section reflect average values over all of the quality grades specified for pumps.

13.1.4 Analysis of Storage Data

The individual data points in Table 13.1-4 show a wide variance in failure rates. This is due to the differences in pump types, differences in sample sized, and the fact that some of the data proved to be invalid as discussed below.

a) Data Point 3 - This was a pump in an aircraft which crashed in the desert and was recovered 17 years later. Although the pump was found in a failed condition, a failure analysis showed it to be a result of the crash. Therefore, this data point is invalid for prediction of storage failure rates.

b) Data Point 4 - This represents data on 121 pumps stored for five years. After this period the pumps were disassembled to replace the seals. Although the pumps were classified as "usable" no tests for leakage were performed. Since leakage is one of the primary failure modes for pumps, and since they were not tested for this, the assumption of zero failures is questionable.

c) Data Point 5 - Two aircraft hydraulic pumps were stored for ten years. Throughout most of this period, the hydraulic system containing these pumps was exercised weekly. Of the five failures reported, two were caused by related failures elsewhere in the hydraulic system. The other three failures were seal leakage. The original seals were neoprene and then were replaced with Buna-N seals. Since two of the failures were caused by other components of the hydraulic system, and the other three failures were considered design defects, this data was declared invalid for storage failure rate prediction.

d) Data Points 8 and 9 - These data points reported one and six failures respectively. However, the failure reports specify these as "hydraulic system failures" rather than specific pump failures. Since there is no way to tell whether these were indeed pump failures, these data will not be further considered.

Although all of the data sources did not contain specific description of the pumps, the information is sufficient to group failure data by pump type. This data, arranged by pump type, is presented in Table 13.1-5.

TABLE 13.1-5. STORAGE FAILURE RATES BY PUMP TYPE

TYPE	FAILURES	STORAGE HRS. x 10^6	FAILURE RATE (fits)	90% CONF. LIMIT
I. Positive Displacement				
A. Fixed Displacement	327	860.05	380.2	408.0
1. Gear	167	380.05	439.9	486.4
2. Piston	112	320.0	350.0	396.2
3. Vane	48	160.0	300.0	363.4
B. Variable Displacement	152	320.0	475.0	528.0
1. Piston	87	160.0	543.7	626.1
2. Vane	65	160.0	406.2	477.1
II. Kinetic	32	160.0	200.0	253.0
A. Centrifugal	32	160.0	200.0	253.0
III. Hydraulic (General)	3	31.482	95.3	105.9
IV. Fuel Pump	0	8.070	<123.9	<286.5

The classification "positive displacement" includes fixed and variable displacement pumps. Sufficient information, in terms of failures and storage hours, was collected for the different subtypes (piston, gear, vane) to allow development of failure rates at this level. These failure rates are shown in Table 13.1-5. Failure rates applicable to the general categories of "fixed" and "variable" displacement were developed by pooling the data of individual types under each category. These rates are to be used when the specific type is unspecified. A failure rate for the general category of "positive displacement" pumps was not developed. The reason for this is the different construction between fixed and variable displacement categories as well as a statistical analysis which showed that the two failure rates (for fixed and variable displacement) did not belong to the same population and should not be combined.

Only one type of kinetic pump was identified - centrifugal. The storage failure rate for this pump is 200 fits as shown in Table 13.1-5.

There were several sources which did not provide any identification other than hydraulic pump. This data is represented by data points 6, 7, 11, and 12. Their combined failure rate is 95.3 fits which is not compatible with the rate of any of the other categories. This is possibly due to the fact these sources represent a relatively small number of hours and three of the four sources did not report a single failure. For these reasons, this data was not used in conjunction with any of the other categories and is included for comparison purposes only.

A fourth category, fuel pump, is included in Table 13.1-5. The operation and construction of a fuel pump differs significantly from the other hydraulic pumps included in the analysis. Notice that the failure rate is based on a relatively small number of hours and no failures. This failure rate is therefore pessimistic.

Of the valid data points reporting at least one failure, the range of failure rate (regardless of pump type) varied from 200 fits to 1497 fits. A failure rate of 200 fits was observed on several data points and on different types of pumps. The source reporting 1497 fits did not specify pump type and it was the only pump designed for submarine use.

13.2 Operational/Non-Operational Failure Rate Comparison

For comparison purposes, the ratios of ground environment operating to non operating failure rate were computed for the different pump types. This is shown in Table 13.1-6. The operational data was obtained from the RADC Nonelectronic Reliability Handbook. The handbook does not differentiate the specific pump types such as gear, piston, etc. Therefore, the numbers with an asterisk devote the operational failure rate for a general category of pumps. Specific failure rates for centrifugal and fuel pumps were listed in the Reliability Handbook and were included in Table 13.1-6.

TABLE 13.1-6. OPERATIONAL/NON OPERATIONAL FAILURE RATE RATIOS

PUMP TYPE	λ_{op}(fits)	$\lambda_{storage}$(fits)	$\lambda_{op}/\lambda_{st.}$
Positive Displacement			
Fixed Displacement	4219*	380.2	11
Gear	4219*	439.9	10
Piston	4219*	350.0	12
Vane	4219*	300.0	14
Variable Displacement	4219*	475.0	9
Piston	4219*	543.7	8
Vane	4219*	406.2	10
Kinetic	12058	200.0	60
Centrifugal	12058	200.0	60
Hydraulic (General)	4219*	95.3	44
Fuel Pump	24390	<123.9	>197

13.3 Conclusions and Recommendations

13.3.1 Conclusions

Sufficient data was collected to permit development of failure rates for specific pump types. In addition, failure rates for four general types were developed. These are variable displacement, fixed displacement, centrifugal, and fuel pumps.

Quality grades were not well defined for the pump data collected. To determine quality grades extensive searching through component specifications and drawings would be required. Hence, the effect of quality levels, if any, could not be determined.

13.3.2 Recommendations

The pump storage failure rates shown in Table 13.1-5 are recommended for predicting the storage reliability of missile equipment.

Record keeping for pumps in storage should be improved, especially in the areas of quality grade identification and equipment description. This should be done within the existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented for all missile hydraulic and pneumatic systems.

13.4 Reference

The information in Section 13 is a summary of document LC-76-HP4, "Hydraulic and Pneumatic Systems Pump Analysis," dated May 1976. Refer to that document for details of data collection and analysis as well as technical descriptions of pumps.

14.0 Regulators

This section contains reliability information and analysis on regulators. Data for temperature and pressure regulators is included.

14.1 Storage Reliability Analysis

14.1.1 Failure Mechanisms

Principal regulator failure mechanisms are summarized in table 14.1-1.

14.1.2 Non-Operational Failure Rates

Non-operational failure rates for regulators are summarized in Table 14.1-2.

TABLE 14.1-2. STORAGE FAILURE RATES - REGULATORS

Type	$\lambda (x10^{-9})$
Temperature	(<199.)
Pressure	1330.
Combined (temp & press)	173.1

14.1.3 Non-Operational Failure Data

Non-operational data was collected from three programs. Storage hours and number of failures for each are shown in table 14.1-3.

TABLE 14.1-3. NON-OPERATIONAL RELIABILITY DATA - REGULATORS

Type	Storage Hours (10^6)	Failures
Temperature	5.024	0
Pressure	.37	1
Pressure	.383	0
	5.777	1

TABLE 14.1-1. FAILURE MECHANISM ANALYSIS - PRESSURE REGULATOR

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Poppet/seat assembly - (seal flow media)	Internal leakage	1	Internal leakage is caused by: 1) Contamination 2) Damaged seats 3) Seat wear	<u>Pre-Installation</u> Part inspection and run-in tests. <u>Post-Installation</u> Audible or by pressure sensor.	Provide cleanliness control for parts, components, and systems. Use design features which control plastic seat cold flow.
Sensing and actuation element - (control pressure)	Regulator fails open or regulates high	2	Contamination between moving parts can slow the actuation mechanism resulting in loss of pressure (possibly out a relief valve).	<u>Pre-Installation</u> Run-in tests and log of calibration. <u>Post-Installation</u> Failure is detected by on-board displays or pressure downstream of the regulator.	Provide cleanliness control for parts, components, and systems. Areas that can trap contaminants should be eliminated from regulator design. Install a filter upstream of pressure regulator.
	Regulator fails closed or regulates low	3	Spring creep or relaxation resulting in loss of pressure regulatior path.		Provide 100% inspection of parts, and provide run-in test to measure if the sensing element has a tendency to drift out of limits.
Regulator housing - (support actuation element and contain flow media)	External leakage	4	Leakage through: 1) Static seals 2) Plumbing connections 3) Regulator housing (porosity)	<u>Pre-Installation</u> Component pressure tests reveal this type of failure mode. <u>Post-Installation</u> System pressure loss or visual detection for manned flights.	Methods to control these failures include: 1) Welded housing construction 2) Install regulator package into the system with permanent mechanical connections 3) Impregnate castings with sealant

14.2 Operational/Non-Operational Failure Rate Comparisons

Table 14.2-1 presents operational to non-operational failure rate ratios for the different operating environments. For comparison purposes, the combined pressure and temperature regulator failure rate is used since information on specific type was not available in the operational data.

TABLE 14.2-1. OPERATIONAL/NON-OPERATIONAL FAILURE RATE RATIOS

Environment	$\lambda(10^{-9})$	Operational/Non-Operational Ratio
Non-operational	173.1	-
Operational, ground	393856.0	2275
Operational, air	367306.0	2122
Operational, helicopter	126930.0	733

14.3 Conclusions and Recommendations

14.3.1 Conclusions

The storage failure rate for the combined (temperature and pressure) regulator types should be used for reliability prediction as shown in Table 14.1-2, i.e., 173.1 failures per billion part hours.

The specific regulator type could not be identified and quality grades were not well defined. To determine the specific regulator types and quality grades, extensive searching through component specifications and drawings would be required.

14.3.2 Recommendations

Record keeping for regulators on storage should be improved, specifically the identification of quality grades and regulator descriptions.

Additional research and data collection should be performed to attain a better definition of the data already on hand.

15.0 Reservoir

Non-operating data for reservoirs used in hydraulic systems is unavailable. Reliability prediction will be made as soon as this type data is available.

16.0 VALVES

Valves are classified by actuating device: solenoid, hydraulic, pneumatic, or motor operated valves.

A solenoid valve is a combination of two basic functional units: a solenoid (electromagnet) with its plunger, and a valve containing an orifice in which a disc or plug is positioned to stop or allow flow of gas or fluid. Solenoid valves may be direct acting, internal pilot operated, external pilot operating, or a latching type valve which requires either manual or electrical resetting.

Basic hydraulic valve types for missile applications are: bleeder, check, control, relief, servovalve, and shutoff.

Bleeder valves allow manual draining of hydraulic lines or components.

Check valves allow flow in one direction and, if the system pressure reverses, close quickly to stop flow in the opposite direction. Check valves are self-contained devices, requiring no external actuation signals or sources of power. The valving elements are activated by the pressure forces of the flow media. The types of check valves include the poppet, swing, and flapper.

Relief valves automatically discharge fluid to relieve pressure. They are generally used in applications with non-compressible fluids such as water, oil, etc. Immediate full-flow discharge is not needed since a small flow reduces the pressure.

An electrohydraulic servovalve is controlled electrically by a magnetic coil, which, in the majority of designs, positions a sliding flow-control spool either directly or indirectly in response to a remote electric signal. The name "servovalve" results from its primary use as the fluid-control element in hydraulic servo systems.

Shutoff valves are used as lock valves in many hydraulic systems. Lock or shutoff valves may be installed in subsystems to hydraulically lock the actuating cylinders where the unit must remain closed through extreme changes in temperature.

Pneumatic valves are like hydraulic valves except they are operated by air, gases or by the pressure or exhaustion of air.

Motor operated valves are similar to hydraulic valves. Their classification comes from the actuating element. Motor operated valves for missile design selection are classified as sequence valves, freon valves or fuel control valves.

16.1 Storage Reliability Analysis

16.1.1 Failure Mechanisms

The principal failure modes, the part affected, and the mechanism resulting in that mode of failure are presented in Table 16.1-1. The relative ranking, in accordance to the frequency of the mode, detection methods and techniques to minimize the effects are also listed.

16.1.2 Non-Operating Failure Rate Data

Over 2.4 billion hours of storage data are included within this report. The data is summarized and categorized by specific component type as shown in Table 16.1-2.

The data, only reflected the number of failures or part hours. Rarely did collected data contain failure modes or mechanisms. However, users' surveys show that the principal storage failure modes and mechanisms include: leakage (internal and external), failure to open or close due to contamination.

The quality control levels are not well defined for any of the valve data. The basic information used to calculate Table 16.1-2 reflects the average failure rates for valves over the quality grade spectrum specified for valves. Each valve failure rate is expressed in failures per billion part-hours (failures/10^9) of field experience.

16.1.3 Calculation of Failure Rates

The data collection effort includes all available data existing in a usable format for missile valves. The data tables show the number of failures and the part hours for each part type which was in a same environment. From this a failure rate was calculated. The data was combined first

Table 16.1-1. Failure Mechanism Analysis - Valves

Part and Function	Failure Mode	Rel. Rank	Failure Mechanism	Detection Method	How to Eliminate/ Minimize Failure Mode
Valve seat - (seal flow media)	Internal leakage	1	Internal leakage is caused by: 1) Contamination 2) Damaged seat 3) Seat wear	<u>Pre-Installation</u> Testing and valve run-in will detect this type of failure. <u>Post-Installation</u> System pressure measurements can be used to determine if valve is leaking.	Control cleanliness level for parts, components and systems. Areas that can trap contaminants should be eliminated from valve designs (use a "swept by" design where flow path is through the valve).
Poppet assembly - (control flow)	Failure to open or close	2	Poppet stuck in intermediate stroke position due to: 1) Contamination 2) Misalignment of poppet 3) Solenoid failure	<u>Pre-Installation</u> Testing will determine this type of failure prior to valve installation into a system. <u>Post-Installation</u> Valve position indicator or system pressure measurements.	Allow conservative force margins for opening and closing of valve seat. Run-in test or margin tests should reveal this type of failure.
Valve body - (support valving elements and contain media)	External leakage.	3	Leakage through: 1) Static valve seals 2) Plumbing connections 3) Valve body (porosity)	<u>Pre-Installation</u> Component tests will reveal this failure mode. <u>Post-Installation</u> System pressure (fluid) loss or visual detection of leaks for manned flight.	Methods to control these failures include: 1) Welded external body construction 2) Install valve into sys tem with permanent mechanical connections 3) Impregnate castings with sealants 4) Use of vacuum melt metals to control inclusions or stringers.

TABLE 16.1-2. OVERALL STORAGE FAILURE RATE RESULTS

($\lambda \times 10^{-9}$)

TYPE	STORAGE FAILURE RATE ($\lambda \times 10^{-9}$)
Solenoid	8.5
2-pos, 2 way	<100.0
Dual, 2 way	-
2-pos, 3 way	<500.0
2-pos, 4 way	<1800.0
Hydraulic	-
Bleeder	<4.8
Check	22.9
Control	<6.7
Relief	1.4
Shutoff	<4.6
Servo Shuttle	145.8
Pneumatic	17.5
Pneu. Oper.	17.5
Springless	-
Motor Op.	113.2
Sequence	194.0
Freon	<1500.0
Fuel	84.3

by type valve and environment and second by valve type disregarding environment.

16.1.4 Solenoid Valve Data

Data was identified for 3 of 4 valve types: 2 position-2 way, 2 position-3 way, and 2 position-4 way.

Storage data collected was from seven missile programs and three space programs. The valid data is summarized in Table 16.1-3 with a calculated 90% one-sided confidence limit.

TABLE 16.1-3. SOLENOID VALVE DATA SUMMARY

TYPE	MILLION STORAGE HOURS	FAILURES	λ_S IN FITS*	90% ONE-SIDED λ_S IN FITS
2 pos-2 way	10.	0	<100	231
2 pos-3 way	2.1	0	<500	1100
2 pos-4 way	.55	0	<1800	4200
Dual-2 way	-	-	-	-
General Sol.	808.	7	8.6	14.6
TOTAL	**820.65**	**7**	**8.5**	**14.4**

* Failures per billion hours

As shown, 820 million part storage hours with seven reported failures giving a failure rate of 8.5 failures per billion hours. For a 2 position-2 way solenoid valve, data was available from one missile program which included 10 million part storage hours with no failures reported. The failure rate of 100.0 failures per billion hours assumes one failure which makes the failure rate pessimistic.

For 2 position-4 way solenoid valves, data was available on a ground system with 0.55 million part hours. In this case, there were no failures reported. The failure rate of 1818.0 failures per billion hours is therefore pessimistic. For the 2 position-3 way solenoid valve, there was data from one ground system with 2.1389 million part hours with no failures reported. This makes the 500.0 failure rate per billion hours pessimistic for this tvpe valve.

There are many storage data points for solenoid valves which do not identify the specific valve type, such as, 2 position-2 way, 2 position-3 way, etc. This data was used for the overall failure rate for solenoid valves. There are points which are not used as shown because they were marked invalid by the original source or because information was incomplete.

From analysis of this data, it is recommended that a storage failure rate of 8.5 failures per billion hours be used for all solenoid valves including the 2 position-2 way, 2 position-3 way, and 2 position-4 way until more detailed data is collected on these types.

16.1.5 Hydraulic Valve Data

The data collected is summarized in Table 16.1-4 for hydraulic type valves. The servo/shuttle valve data is broken separately because of the difference in complexity of this valve type. The 90% one-sided confidence limit is calculated as shown.

TABLE 16.1-4 HYDRAULIC VALVE DATA

TYPE	MILLION HOURS	FAILURES	λ_S IN FITS	90% ONE-SIDED λ_S IN FITS
Bleeder	210.4	0	<4.8	11.
Check	131.0	3	22.9 (4250)	51.
Control	150.2	0	<6.7	15.
Relief	712.8	1	1.4 (499)	5.5
Shutoff	214.8	0	<4.6	11.
TOTAL	1419.2	4	2.8	5.6
Servo/ Shuttle	109.7	16	(68) 145.8 (3100)	205.

The numbers in parenthesis show the highest and lowest failure rates computed from individual sources showing at least one failure. This is a measure of the range of failure rates observed from all sources.

The three data points for bleeder valve storage data have 210.43 million part hours with no failures. Hence, the failure rate <4.8 failures per billion hours is pessimistic.

Check valve storage data has five data points containing 131.018 million part hours with three (3) failures. The check valve storage failure rate is 22.9 failure per billion hours.

Control valve data consists of three data points, 150.165 million part hours with no failures. The failure rate calculated is <6.7 failures per billion hours assuming one failure.

Relief valve data has five (5) data points with one (1) failure. With 712.78 million part storage hours, a failure rate of 1.4 failures per billion hours was calculated.

Five (5) data points for hydraulic "shut-off" valves reflect 214.8 million part storage hours with no failures. This calculates a <4.6 failure rate per billion hours for shutoff valves assuming one failure.

Fifteen (15) storage data points were collected for "servo or shuttle" valves. This resulted in 109.67 million part storage hours with 16 failures reported for a failure rate of 145.8 per billion hours.

Analysis of the bleeder, check, control, relief and shutoff valves show compatible failure rates. Therefore, these data were pooled for a failure rate of 2.8 fits applicable to these valves. The servo valve differs significantly in construction and mode of operation. The failure rate for this type of valve is estimated at 145.8 fits.

16.1.6 Pneumatic Valves

Data was collected for four types of pneumatic controlled valves: general, check, pressure, and manifold. The data was available from five (5) programs. It included 57.05 million part hours with one (1) reported failure giving a failure rate of 17.5 failures per billion hours. A summary of the data is shown in Table 16.1-5.

A failure rate is calculated for each pneumatic valve type with a 90% one-sided confidence limit. Note a significant higher failure rate for the pressure valve than for the other types. However, this was calculated with an assumed failure and a small amount of storage hours. Hence, it is recommended that the data be pooled together and that the overall failure rate (17.0 fits) be used for prediction.

TABLE 16.1-5. PNEUMATIC VALVES

TYPE	MILLION HOURS	FAILURES	λ_S IN FITS	90% ONE-SIDED λ_S IN FITS
Pneumatic Valves				
General	4.67	1	214.	833.
Check	4.14	0	<241.	558.
Pressure	.628	0	<1590.	3680.
Manifold	47.61	0	<21.	48.5
TOTAL	57.05	1	17.5	68.2

16.1.7 Motor Operated Valves

There are three types of motor operated valves for which data was found. These are the sequence, freon and fuel valves. A summary of the data is shown below, Table 16.1-6, with a 90% one-sided confidence limit.

TABLE 16.1-6. MOTOR OPERATED VALVES

TYPE	FAILURES	STORAGE HRS. x 10^6	FAILURE RATE (10^{-9})	90% ONE-SIDED λ_S IN FITS
Sequence	1	5.150	194.0	755
Freon	0	0.662	<1500.0	3500
Fuel	1	11.853	84.3	328
TOTAL	2	17.665	113.2	301.0

The sequence motor operated valve had 1 reported failure with 5.150 million part hours for a failure rate of 194.0 failures per billion hours.

The freon valve had one data point, no failures and 0.662 million part hours. This gives a failure rate of 1500.0 failures per billion hours.

The fuel valve had three data points with a total of 11.853 million part storage hours. They had one failure giving a failure rate of 84.5 failures per billion hours of storage.

It is recommended that a storage failure rate of 113.2 failures per billion hours be used for all motor operated valves including the sequence, freon and fuel valves.

16.2 Calculation of K-Factors

The K-factors have been calculated below based on operating failure rate data collected from the RADC Nonelectronic Reliability Notebook. The operating failure rates used were from ground-fixed data unless otherwise specified. The formula used was

$$K = \frac{\text{Ground-fixed } (\lambda_{op}) \text{ Operating}}{\text{Storage } (\lambda) \text{ Storage Data}}$$

TABLE 16.2-1. K-FACTORS

VALVE TYPE	λ_{op} (fits)	λ_s (fits)	$\frac{\lambda_{op}}{\lambda_s}$
Solenoid			
2 pos-2 way	15000	<100	150
2 pos-3 way	16000	<500	32
2 pos-4 way	14400	<1800	8
General Sol.	14620	8.6	1700
Hydraulic			
Bleeder	1008	<4.8	210
Check	12595	22.9	550
Control	10720	<6.7	1600
Relief	2052	1.4	1470
Shutoff	4646	<4.6	1010
Servo	65610	145.8	450
Pneumatic			
General	6420	214	30
Check	6409	<241	26
Pressure	6360	<1590	4
Manifold	6510	<21	310
Motor Operated			
Sequence	-	190	-
Freon	34500	<1500	23
Fuel	8232	84	98

16.3 Conclusions and Recommendations

16.3.1 Conclusions

Comparison between dormant and storage reliability data indicates no significant difference between the two. This agrees with previous studies (reference no. 35). Therefore, the dormant and storage data were combined in all analyses.

Quality grades were not well defined for the valve data collected. To determine quality grades extensive searching through component specifications and drawings would be required. It was therefore impossible to determine the effect, if any, of quality levels. The results presented in this report represent failure rate averages over the quality grade spectrum.

Data available for each generic valve type (solenoid, hydraulic, pneumatic, motor operated) was plentiful compared to that available for each sub-category. Therefore, failure rates derived at this level have higher statistical confidence than those of the sub-categories and should be utilized unless specific information is available to further define the type of valve under consideration.

16.3.2 Recommendations

Record keeping for valves kept on storage should be improved, specifically the identification of quality grades and valve description. This should be done within existing data collection systems.

Additional research and data collection should be performed to attain a better definition of the data already on hand. More detailed identification of those units classified only by their generic names should be attempted.

A more vigorous and better documented program of failure mode analysis should be implemented.

16.4 Reference

The information in Section 16 is a summary of document number LC-76-HP1, "Missile Hydraulic and Pneumatic Systems Valve Analysis," dated May 1976. Refer to that document for details of data collection and analysis as well as technical descriptions of valves.

17.0 Hydraulic Fluids

Long-term storage of hydraulic fluid inside or outside a hydraulic system leads to two primary causes for deterioration: 1) breakdown of the fluid because of inherent instability, heat or surface contact with air, and 2) failure of the fluid as a result of contamination. References 6 and 7 discuss these problem areas in considerable detail.

Several types of petroleum-base hydraulic fluids can be considered for applications in hydraulic thrust vector control systems. Synthetic oils are excluded from this study since none were found to have been utilized in system subject to long-term storage. They differ primarily by the presence or absence of an additive for a particular attribute. Typical attributes are as follows:

1. High-temperature protection. The flash point should be high enough so that combustion is precluded, and the oil should be chemically stable at operating temperatures.

2. Low-temperature protection. The pour point is generally required to be below -40°F (typical is -50°F). The oil should be stable at points slightly above its pour point even when chilled for over 72 hours.

3. Drag and hysteresis of hydraulic components. Frictional drag and hysteresis are related to the oil viscosity as well as the degree that additives in the oil wet or plate out on highly polished,close-fitting parts. This is especially true for the polymeric additive used for improving the viscosity index.

4. Corrosion protection. This is a necessary attribute where long-term storage is involved.

5. Flushing efficiency. This attribute is combined with corrosion protection when this protection comes from the use of certain detergent wetting agent. After manufacture most components and units are "checked out" and flushed. Several recently presented technical papers have stated that flushing with MIL-H-5606A is ineffectual. They have usually concluded that MIL-H-6083 (corrosion-preventive hydraulic oil) gives

more efficient results. It is felt that the removal of the viscosity-index-improver additives described above also contribute to this flushing efficiency by eliminating the adhesive film of deposited polymer that acts as a trap for particles. Detergent wetting agents may be added to improve flushing efficiency.

6. Particle dispersion. This attribute is related to the ability of the fluid to retain particles in suspension, without agglomeration taking place. In most cases, the larger suspended particles are removed as they pass through a filter.

7. Cleanliness. The particle count (in a 100-ml sample of fluid) is often expressed as the number of particles in each of the following size ranges:

5-15 microns, 1 micron = 10^{-6} meters
15-25 microns
25-50 microns
50-100 microns
100-300 microns
Over 300 microns

8. Effect of shear on viscosity. Shear forces tend to break down long chain polymers and cause a reduction in viscosity. Oils without viscosity-index improvers are not susceptible to this effect.

9. Long-term seal preservation. An additive may be added to oil that conditions rubber seals and O-rings and prevents extraction of plasticizer during seal-oil contact. The following processes normally occur when rubber compounds are placed in an environment of MIL-H-5606A:

a. The dissolution or leaching of some soluble comconent of the rubber (solid dissolves in a liquid).
b. The acceptance of hydraulic fluid into the structure of the rubber, with subsequent rejection of some components of the hydraulic fluid (liquid dissolves in a solid).

c. The chemical combination of some component of the fluid with the rubber.

All of the above processes take place simultaneously at consistent though varying rates. As a result, the value of any given physical property of the rubber as a function of time may vary through several maxima or minima, depending upon which process has accumulated the greatest effect at any given point in time. Process "b" is limited by swelling of the rubber compound when the rubber object is geometrically restrained. The swelling action closes off the pores through which the oil enters the structure of the rubber. Buna-N O-rings soften (unless overheated) in MIL-H-5606 hydraulic fluid. Their squeeze (sealing) reaction force is thereby lessened. At the same time they swell, which tends to compensate for the softening. This swelling is very probably a strong reason why installed O-rings continue to seal under long-term storage conditions. Laboratory investigations of percentage of squeeze (sealing) force retained versus immersion time have been conducted and show a leveling out of the decline of sealing force with time. The testing involved immersing O-rings, while held squeezed, in fluid and periodically measuring the squeeze force.

10. Oil interchangeability. It is desirable that an oil be completely miscible with commonly used hydraulic fluids in case a change of oil is made in a hydraulic system.

11. Viscosity index. "Viscosity index" denotes an arbitrary method of stating the rate of change of viscosity with change of temperature.

Table 17.0-1 presents the specifications for four hydraulic fluids suitable for use in missile hydraulic systems. MIL-H-5606A oil is the most commonly used in this application. MIL-H-6083 is used as a preservative oil and also as an operating hydraulic fluid. MIL-H-25598 and AB0145-001 have attributes that make them particularly suitable for long-term storage; they contain additives that give corrosion protection, dispersibility, and good flushing efficiency. Pour-depressing

TABLE 17.0-1. HYDRAULIC FLUID SPECIFICATIONS

Specification	MIL-H-6083	MIL-H-5606A	MIL-H-25598	AB0145-001
	Properties (Typical Tests Unless Otherwise Noted)			
Viscosity at 210° F, cs	4.5[a]	5.35[a]	1.28[a]	3.82
Viscosity at 170° F, cs	6.5	6.6	1.63	6.5
Viscosity at 130° F, cs	10.13 (10 min)	10.1 (10 min)	2.4 (2.3 min)	12.9
Viscosity at 100° F, cs	15.3	13.9	3.3	25.0
Viscosity at 0° F, cs	155	91	22	1185
Viscosity at -40° F, cs	765 (800 max)	486 (500 max)	95 (80 min)	
Viscosity at -65° F, cs		2577	380 (400 max)	
Flash point, °F	200 min	200 min	200 min	355
Pour point, °F	-75 max	-90 (-75 max)	-90 max[b]	-50[b]
Gravity, API degrees at 60° F	30.3	31.6	32	28.0
Gravity, specific 60°/60°	0.8745	0.8676	0.8654	0.8871
Corrosion and oxidation stability Airblown at 250° F for 168 hours with metal catalysts	No appreciable viscosity or neut. number change. No discernible corrosion to metals.			
Copper strip corrosion: 72 hours at 212° F	Slight brown stain permitted			
Synthetic "L" rubber swelling	24%	24%	24%	20%

Notes: (a) From extrapolated graph.

(b) Pour depressants not allowed.

TABLE 17.0-1. (cont'd)

Properties (Typical Tests Unless Otherwise Noted)

Specification	MIL-H-6083	MIL-H-5606A	MIL-H-25598	AB0145-001
Corrosion protection:				
Steel in 100% humidity	100 hrs min	None required	20 hrs min	100 hrs
Static water drop	100 hrs min	None required	100 hrs min	100 hrs min
Shear resistance, 5,000 cycles: Maximum viscosity decrease at 130° F	-25%	-25%	Nil	Nil
Contamination level of oil as delivered in container per ARP 598	No limit	No limit	Less than 2,000 particles over 5-micron size	
Composition:				
Oxidation inhibitor	Yes	Yes	Yes	Yes
Tricresyl phosphate (antiwear additive)	Yes	Yes	Yes	Yes
Wetting agent (corrosion inhibitor, system detergent, and particle dispersant)	Yes	None	Yes	Yes
Polymeric additive (viscosity-temperature corrector)	Yes	Yes	None allowed	None allowed
Elastomer preservative (for long-term storage)	None	None	Yes	Yes

and viscosity-temperature additives (polymeric additives) are specifically excluded. MIL-H-25598 is essentially the low-temperature counterpart of AB0145-001. AB0145-001 has been adopted as a system fluid by at least one missile and satellite manufacturer.

17.1 Hydraulic Fluid Storage Reliability Analysis

17.1.1 Failure Modes

The primary failure modes of hydraulic fluids are:

1. contamination
2. oxidation
3. foaming
4. loss of viscosity
5. air
6. vaporization
7. carbon formation

The most common is contamination which can be greatly reduced by careful handling and storage.

17.1.2 Failure Mechanisms

The failure mechanisms most commonly found to cause hydraulic system deterioration: 1) breakdown of the fluid because of inherent instability, heat or surface contact with air, and 2) failure of the fluid as a result of contamination.

Typical attributes to the failure mechanisms are:

1. high-temperature
2. low-temperature
3. drag and hysteresis of hydraulic components
4. corrosion
5. flushing
6. particle dispersion
7. cleanliness
8. effect of shear on viscosity
9. long-term seal preservation
10. oil interchangeability
11. viscosity index

17.1.3 Data Analysis

Ten data sources were investigated ranging from one to 400 samples each with storage time from 2 months to 17 years. A specific failure rate is not calculated for the fluid types. Data analysis reflects no failures attributable to the various hydraulic fluid types.

A summary of hydraulic fluid data is presented in Table 17.1-1.

Half of the data samples (i.e., Data Samples No. 1, 5, 6, 7 and 8) pertain to long-term storage of MIL-H-5606 hydraulic fluid in dormant aircraft hydraulic systems. In general, this fluid exhibited varying degrees of contamination at the beginning of the dormancy period. In all cases, the hydraulic system was operable just prior to the dormancy period.

Data Samples No. 2 and 3 pertain to long-term shelf storage of MIL-H-5606 in tin cans and in a drum.

Data Sample No. 4 pertains to the long-term storage of hydraulic fluid in dormant missile hydraulic power supply units that had been in the field. The hydraulic fluid conformed to MIL-H-25598.

Data Samples No. 9 and 10 pertain to loss of viscosity of MIL-H-5606 hydraulic fluid in hydraulic systems that impose considerable shear stress on the hydraulic fluid.

Based on the data pertinent to long-term storage of MIL-H-5606 hydraulic fluid in dormant aircraft hydraulic systems, the following generalizations can be made:

1. This fluid may become contaminated with dirt, pieces of elastomeric materials, and miscellaneous particles after a 1-year dormancy period.
2. In none of the cases studied was failure of hydraulic components attributed to the contamination referred to in (1) above.
3. No traces of corrosion, either external or internal, were reported. MIL-H-5606 provided adequate corrosion protection in all cases studied.

TABLE 17.1-1. - HYDRAULIC FLUID STORAGE DATA SUMMARY

Data Sample No.	Name of Part	No. of Parts Tested	Storage Time	Storage Environment	No. of Failures	Type of Failure	Remarks
1	Hydraulic fluid	400-ml sample	17 years	Libyan Desert	None	(Not applicable)	Fluid met Specification AN-VV-O-366b.
2	Hydraulic fluid MIL-H-5606A	3	1 year	Within tin cans	None	(Not applicable)	
3	Hydraulic fluid MIL-H-5606A	1	79 months	Within 55-gallon drum	None	(Not applicable)	
4	Hydraulic fluid MIL-H-25598	43	2 years	Field missiles	None	(Not applicable)	
5	Hydraulic fluid MIL-H-5606	2	2 years	Air temp., 50-110° F; humidity, 6-30%	None	(Not applicable)	Fluid was dirty and discolored in appearance; the actuators operated satisfactorily, however, except for three which leaked.
6	Hydraulic fluid MIL-H-5606	24	1 year	Air temp., 50-110° F; humidity, 6-30%	None	(Not applicable)	Oil was discolored and dirty.
7	Hydraulic fluid MIL-H-5606	12	12-14 months	Air temp., 50-110° F; humidity, 6-30%	None	(Not applicable)	Oil was discolored and dirty.
8	Hydraulic fluid MIL-H-5606	2	18 months	Average temp., 53.3-85.5° F; average humidity, 27-59%	None	(Not applicable)	Components tested; results satisfactory. No traces of corrosion.
9	Hydraulic fluid viscosity test MIL-H-5606	1					Viscosity reduction as a function of operating time.
10	Hydraulic fluid viscosity test MIL-H-5606	1					Viscosity reduction as a function of operating time.

4. The dormancy period was, in general, 1 to 2 years. In the case of Data Sample No. 1, the "Lady Be Good" aircraft lay dormant for 17 years on the Libyan Desert. The satisfactory condition of the hydraulic fluid similar to MIL-H-5606 after this period of time is undoubtedly strongly influenced by the extremely dry conditions on the desert.

The results of shelf storage tests of MIL-H-5606 hydraulic fluid indicate that, provided the oil is free of contamination to start with, the shelf life is a minimum of 6 years. Agglomeration of particles in this fluid due to agitation can be effectively prevented by holding the concentration of particles in the 5-15 micron range below 1,000 for a 100-ml sample. Hydraulic fluid is now being packaged at this low contamination level in accordance with NSM 60-20, Military Petroleum Supply Agency, Washington, D. C.

No significant contamination of hydraulic fluid was found in the 43 missile hydraulic power supply units stored in the field for 2 years. No failures of hydraulic components were attributable to contamination of the hydraulic fluid (fluid conforming to MIL-H-25598).

ss of viscosity of MIL-H-5606 hydraulic fluid occurs in hydraulic systems during operation. The loss rate depends on the shear stress imposed, the system fluid inventory, and the flow rate. Data Samples No. 9 and 10 demonstrate the above. The loss in viscosity is due to cracking of the polymeric additive, polyalkylmethacrylate, used for improvement of the viscosity index. The viscosity of MIL-H-5606 hydraulic fluid approaches that of the base stock asymptotically with operating time. In hydraulic systems operating at very high temperatures, thermal polymerization of the oil offsets some of the viscosity loss due to shearing.

The maximum shear stress is induced at a point of high turbulence, usually a point where a large pressure drop occurs, as in a valve or pump.

Specifications MIL-H-5606 and MIL-H-6083 limit viscosity changes to +5 to -20 percent after a specific shear breakdown test. These viscosity limits are considered stringent enough for hydraulic thrust vector control systems subject to long-term storage.

Each data sample is discussed in more detail below:

DATA SAMPLE NO. 1: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

Four hundred milliliters of hydraulic fluid from the "Lady Be Good" aircraft B-24D, AAC No. 124301, were examined after 17 years in the aircraft's hydraulic systems. The location of the aircraft was the Libyan Desert in North Africa. The air temperature varied from 26°F to 120°F, and the maximum aircraft skin temperature varied from 150°F to 200°F. It rained twice during the 17-year period. This fluid met Specification AN-VV-O-366b, which is similar to MIL-H-5606A. The sample was found to be in excellent condition, showing good stability.

DATA SAMPLE NO. 2: HYDRAULIC FLUID SHELF-LIFE TEST

Three samples of MIL-H-5606A hydraulic fluid were stored at ambient temperature in tin cans for one year. No significant changes occurred in the viscosity at 130°F, viscosity at -40°F, pour point, flash point, or precipitation number.

DATA SAMPLE NO. 3: HYDRAULIC FLUID SHELF-LIFE TEST

A sample of MIL-H-5606A hydraulic fluid was stored from July 1952 to December 1958 (6 years, 5 months) in a 55-gallon drum kept on its side in the open. Five-gallon samples were removed periodically, the volume being replaced by ambient air. No significant changes occurred in the viscosity at 130°F, viscosity at -40°F, pour point, flash point, precipitation number, or acid number. In addition, the sample passed the MIL-H-5606 specification for 250°F oxidation and corrosion, and for -65°F stability at the end of the storage period. There was no evidence of separation or gumming.

DATA SAMPLE NO. 4: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

Forty-three hydraulic power supply units were removed from missiles which had been in the field 24 months. As part of the test program, the hydraulic fluid was checked for particle size distribution. The hydraulic fluid was reported to contain numerous particles smaller than 50 microns and a few larger than 50 microns. Some of the particles were rubber, while most were a soft crystalline substance which was easily broken up with a probe.

No failures to fire properly were attributed to oil contamination. The oil contamination noted was not considered significant. The oil met Specification MIL-H-25598.

DATA SAMPLE NO. 5: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

Samples of MIL-H-5606 hydraulic fluid from two B-26C's, Serial Nos. 44-34606 and 44-35585, were examined after a 2-year storage period. The summer air temperature varied from 90°F to 110°F, and the winter air temperature varied from 50°F to 70°F at the storage facility. The average summer humidity was 6 percent, and the average winter humidity was 25 to 30 percent. Both samples showed that the fluid was discolored and dirty in appearance. The actuators on these aircraft operated satisfactorily, with the exception of three which had excess leakage. The oil was changed in both aircraft.

DATA SAMPLE NO. 6: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

Samples of MIL-H-5606 hydraulic fluid present in 24 additional stored B-26"s were found to be discolored and dirty and to have accumulated contamination after 1 year of storage. Several people observed that this oil must be replaced within 1 year if the system is to be maintained in an operational condition.

One missile manufacturer, who is conducting special research on fluid contamination levels using the Millipore method, has the same opinion of the storability of MIL-H-5606.

The summer air temperature varied from 90°F to 110°F, and the winter air temperature varied from 50°F to 70°F. The average summer humidity was 6 percent, and the average winter humidity was 25 to 30 percent.

DATA SAMPLE NO. 7: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

Twelve F-100 aircraft were reactivated after 12 to 14 months of storage. The hydraulic oil was not changed, although it was dirty. It was said that the oil required replenishment on all systems, but the basis for requiring replenishment was not explained.

The summer air temperature varied from 90°F to 110°F, and the winter air temperature varied from 50°F to 70°F at the storage facility. The average summer humidity was 6 percent, and the average winter humidity was 25 to 30 percent.

DATA SAMPLE NO. 8: HYDRAULIC FLUID STORAGE IN DORMANT SYSTEM

An examination was made of the hydraulic systems of F4U-4 stored aircraft to determine whether it was necessary to replace MIL-H-5606 hydraulic fluid with MIL-H-6083 preservation fluid for adequate preservation of the hydraulic system components. Environmental conditions at the storage facility were as follows: average relative humidity at 5:30 am, 50 percent; at 11:30 am, 32 percent; at 5:30 pm, 27 percent, at 11:30 pm, 52 percent; average daily maximum temperature 85.5°F, minimum 53.3°F; and average annual rainfall, 7.56 inches.

Data was compiled from two F4U-4 aircraft, Bu. Nos. 97259 and 97302, selected at random from four in outdoor storage. These aircraft were known to have MIL-H-5606 hydraulic fluid, contaminated within limits, in their systems for at least 18 months while in outdoor storage status. Various hydraulic system components of both aircraft were removed, brought to the laboratory, degreased, dissected, and examined. Extreme care was taken in selecting typical components containing the different materials in the system (i.e., anodized aluminum, steel, O-rings, and rubber diaphragms).

Since the entire sampling of typical components revealed no traces of corrosion, either externally or internally, it was concluded that MIL-H-5606 hydraulic fluid afforded adequate protection for these systems.

On the basis of this investigation, and in view of the favorable past history of corrosion control exhibited at this activity, it was recommended that MIL-H-5606 hydraulic fluid be left in aircraft hydraulic systems during periods of storage (providing contamination limits are not exceeded).

DATA SAMPLE NO. 9: HYDRAULIC FLUID VISCOSITY TEST

Starting in August 1956, the viscosity, at 130°F, of MIL-H-5606 in an operating hydraulic system was monitored once a week for 6 weeks. The hydraulic system was a test stand for hydraulic components. The system imposed considerable shear stress on the oil. The listing below gives the results.

Original Oil	Viscosity = 10.06 centistokes
1 week in system	Viscosity = 8.15 centistokes
2 weeks in system	Viscosity = 7.80 centistokes
3 weeks in system	Viscosity = 7.69 centistokes
4 weeks in system	Viscosity = 7.60 centistokes
5 weeks in system	Viscosity = 7.44 centistokes
6 weeks in system	Viscosity = 7.37 centistokes

DATA SAMPLE NO. 10: HYDRAULIC FLUID VISCOSITY TEST

In a system charged with two gallons of MIL-H-5606A running at approximately 2.5 gallons per minute, 160°F, 3,000 psi, an 18.5-percent loss in viscosity was reported after 8 hours.

Coincidentally, the fluid in a test stand operated by a manufacturer of aircraft valves showed the same viscosity decrease after 7 days' operation.

17.2 Conclusions & Recommendations

Hydraulic fluid petroleum-base stocks do not degrade significantly with storage time. Some of the additives, however (especially the acrylics), can have adverse effects during storage. Recommended fluids are MIL-H-25598 for low temperatures and AB0145-001 for normal temperatures.

Adequate filtration techniques in the manufacturing process, in system filling, and designed into the operating hydraulic system can greatly assist in maintaining hydraulic systems within acceptable contamination levels. Air filters and air dryers should be used in oil breather systems if such systems are necessary.

Particular care in the selection of compatible metals for the hydraulic system can greatly reduce the potential for dielectric corrosion, which, in addition to damaging hydraulic components, contaminates the oil.

From the data analyzed it is concluded that MIL-H-5606 hydraulic fluid is suitable for long-term storage in missile thrust vector control systems. However, MIL-H-25598 and AB0145-001 are inherently more suitable for this application due to the absence of the polymeric additive, thus eliminating the viscosity change and agglomeration problems. The inclusion of an elastomer preservative, and a wetting agent that gives corrosion protection, detergency , dispersibility, and minimum wear, also contributes to their desirability.

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