AD-785 580

TURBINE ENGINE FUEL CONTROL RELIABILITY TEST AND EVALUATION

R. D. Zagranski, et al

Colt Industries, Incorporated

Prepared for:

Army Air Mobility Research and Development Laboratory

July 1974

EUSTIS DIRECTORATE POSITION STATEMENT

The work performed by the Chandler Evans Control Systems Division and reported herein is considered to be thorough and comprehensive. All three facets of this fuel control failure mode investigative effort are considered to be successful. While the induced vibration and contaminated fuel tests failed to establish these factors as failure inducements, they served to virtually eliminate them as significant contributors to the high percentage of control failures for unknown causes. The seal endurance test phase was very thorough and quite successful in a more positive sense by establishing new seal compounds as very promising to eliminate wear-induced leakage modes.

It should be noted that the vibrational and fuel contaminate test results are applicable only to the particular fuel control model evaluated, and they would not necessarily apply to control models with different functional mechanizations, materials, and/or fabrication tolerances. However, the seal test results can have generic applications.

The results of this effort can be used in the design review and optimization of future turbine fuel controls, and as a source of information for review of adequacy of specifications governing fuel seals.

The U. S. Army technical monitor for this effort was Mr. R. L. Campbell, Sr., of the Military Operations Technology Division.

Unclassified
SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER USAAMRDL-TR-74-53	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER AD-785-580
4. TITLE (and Subtitle) TURBINE ENGINE FUEL CONTROL RELIABILITY TEST AND EVALUATION		5. TYPE OF REPORT & PERIOD COVERED Final - 30 June 1973 to 22 March 1974
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) R. D. Zagranski R. M. Lamart A. H. White		8. CONTRACT OR GRANT NUMBER(s) DAAJ02-73-C-0104
9. PERFORMING ORGANIZATION NAME AND ADDRESS Colt Industries, Inc. Chandler Evans Control Systems Division West Hartford, Connecticut 06101		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory Fort Eustis, Virginia 23604		12. REPORT DATE July 1974
		13. NUMBER OF PAGES 94
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report)
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Reproduced from best available copy.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Fuels
Fuel Contamination
Filtration
Fuel Seals
Vibration Effects
Control

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

This report describes the results of an 8-month fuel control reliability test program to evaluate new dynamic fuel seals, ultrafine fuel filtration and the vibration susceptibility of the Chandler Evans TA-2S hydromechanical fuel control. Previous studies conducted under Army Contract DAAJ02-72-C-0110 identified these areas as potential generic design deficiencies in hydromechanical controls.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

Unclassified
SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

Block 20 (cont.)

Two TA-2S controls were subjected to vibration for 5 hours each over a 10- to 2000-Hz spectrum at g levels based on actual engine vibration data from a UH-1H helicopter. No significant performance degradation was noted during or after the vibration testing.

These same TA-2S controls were run for 70 hours each on MIL-E-5007C contaminated fuel with a 25-micron filter versus a 3-micron absolute filter, and the data indicates that the ultra-fine filtration will not significantly improve the performance and wear-out life of hydromechanical fuel controls.

Fifteen different dynamic seals selected by various seal manufacturers were tested in an engine-simulated environment for 100,000 cycles. Test results indicate that a single O-ring is superior to various type combination seals. A Nitrile C-69 O-ring was the only seal to successfully complete all testing. It also exhibited the most consistent changes in different physical properties that were measured.

PREFACE

This is the final report covering work completed under contract DAAJ02-73-C-0104 (DA Task 1F162205A11902) during the period June 30, 1973 to March 22, 1974.

The program was conducted under the technical direction of Mr. R. L. Campbell, Sr., Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia. The work was conducted by Colt Industries, Inc., Chandler Evans Control System Division. Mr. A. H. White was program manager, and Mr. R. M. Lamart was the principal engineer. Acknowledgment is given to the following seal manufacturers for their contributions to this program:

Federal Mogul Corp., Detroit, MI
Minnesota Rubber Co., Minneapolis, MN
Nichols Engineering, Inc., Shelton, CT
Parker Seal Co., Newington, CT
Royal Industries, El Segundo, CA

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont)

LIST OF TABLES

INTRODUCTION

Reliability and maintainability (R&M) studies previously conducted under Army Contract DAAJ02-72-C-0110, and reported in Reference 1, identified various R&M related problem areas on Army gas turbine engine fuel controls. The objective of the work reported herein was to investigate potential solutions for three of these problem areas:

1. Susceptibility to Vibration
2. Fuel Contamination
3. Fuel Leaks

A significant finding of the study completed during the original contract was that the cause for approximately 25 percent of the justified removals could not be defined, from available data, in sufficient detail to establish the cause for removal for the given symptom. It was noted that this class of removals was primarily on controls which were on an engine for a relatively short time. The one factor which is common to all controls and was thought could be responsible for the high rate of unscheduled removal is vibration. Therefore, two fuel controls were subjected to a vibration spectrum based on engine vibration data taken from tests on a UH-1H helicopter. The objective of this testing was to determine the effects of vibration on fuel control performance before, during and after vibration.

The previous study also showed that fuel and air contamination were responsible for about 10 percent of the justified fuel control removals. Furthermore, contamination accelerates the wear-out life of the fuel control, and 12 percent of the justified removals were caused by excessive wear. The only known solution to fuel and air contamination is filtration, and recent reports in the literature based on service data indicate that a phenomenal increase in the wear-out life of hydraulic systems components is attributed to the use of ultrafine filtration. The objective of the fuel filtration test reported herein was to compare and evaluate the potential increase in the wear-out life and performance of fuel control components when operated with 25-micron versus 3-micron (absolute) barrier filters. Each control was subjected to 70 hours (50 hours room temperature, 20 hours cold fuel) of contaminated fuel operation.

The previous study further showed that both external and internal fuel leaks at rotary power lever shafts and translational piston and piston shafts are a problem common to all fuel controls. Data indicated that approximately 15 percent of the justified fuel control removals were due to fuel leaks. The fuel leakage problem is basically a materials limitation. Therefore, the seal test program reported herein was undertaken to evaluate recently introduced seal materials and configurations. Six each of fifteen different seals were subjected to hot, cold and room temperature and contaminated air cycling for 100,000 cycles. For a design life of 5,000-hour for a control, a dynamic seal would be subjected to an estimated five cycles per hour. Therefore, a 100,000-cycle test for each seal was specified.

VIBRATION TEST

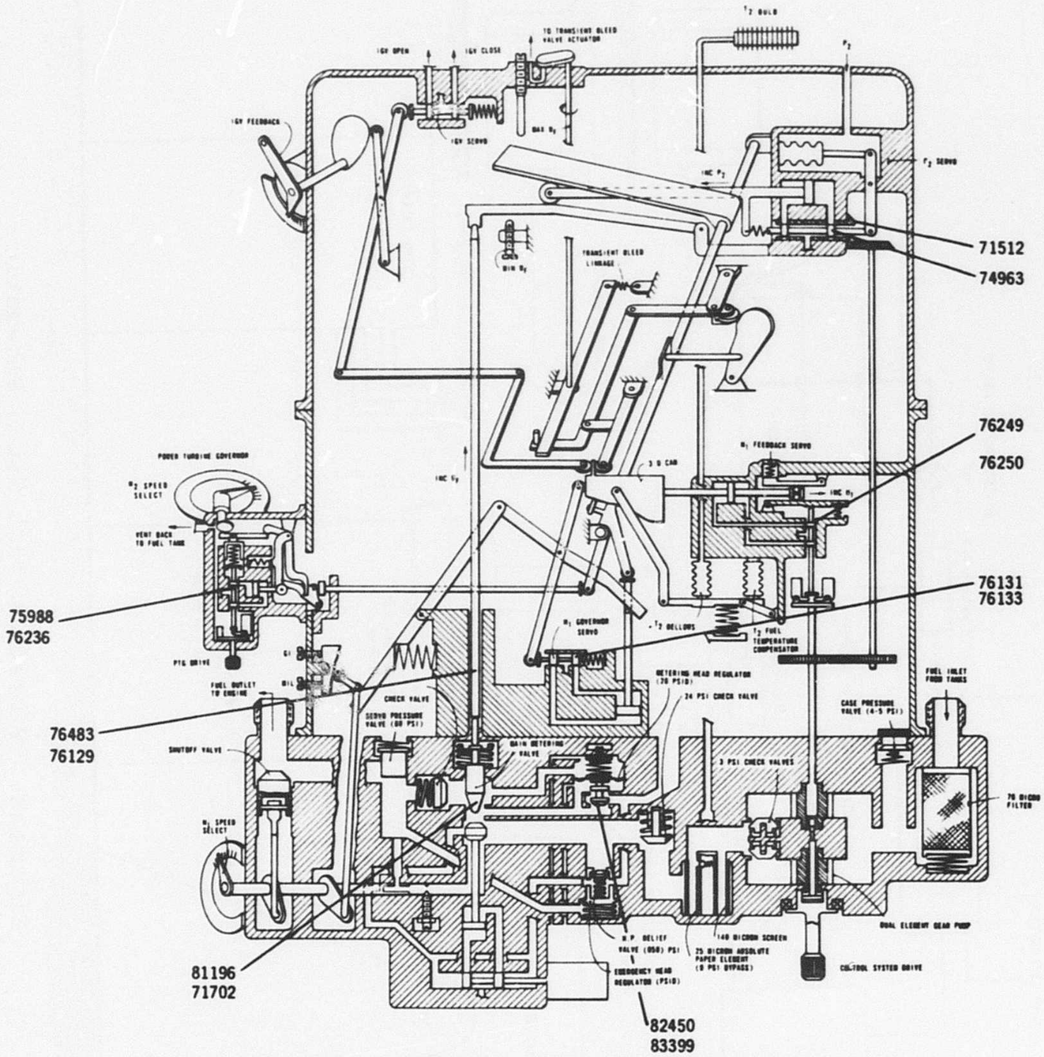
Two overhauled Chandler Evans Model TA-2S fuel systems (S/N 692AS8403 and 692AS7472) of the type used on the Lycoming T53-L-13 gas turbine engine were subjected to sinusoidal vibration during normal operating conditions. The control is a unitized, hydromechanical design consisting of an integral dual element gear pump, gas producer control and power turbine governor, as shown schematically in Figure 1. Metered fuel flow was continuously monitored during vibration tests in an effort to detect flow variations which could induce engine performance disturbances and thereby result in the rejection of a bench-tested and previously acceptable unit. In over 10 hours of testing on the two fuel systems, some fuel flow shifts and oscillations were observed. However, the magnitude of flow variation was within the operating flow limits of the control. Furthermore, the largest steady metered flow shifts would yield only a +1.5-percent N_2 and a $\pm$ 0.3-percent N_1 variation when operating closed loop on the power turbine and gas producer governors, respectively. These engine disturbances are relatively minor, and in all probability would not be cause for removing the fuel control.

The following is a detailed account of the vibration test program and its derived findings.

TEST DESCRIPTION

A schematic of the test setup is shown in Figure 2. The fuel control was supplied with relatively constant temperature (70°-100°F) JP-4 fuel. Flexible shafts coupled fixed speed motors to the N_1 and N_2 drive splines of the control. High-speed pressure and fuel flow transducers, accelerometers, and a light beam strip-chart recorder monitored critical functional parameters, including fuel flow, metering head and discharge pressures, and structural accelerations at various locations.

The fuel system was vibrated in each of three mutually perpendicular planes as defined in Figure 3. These planes were coincidental to the orthogonal planes in which frequency and g level measurements were made on a UH-1H helicopter by the



NUMBERS SHOWN REFER TO PART NUMBERS LISTED IN TABLES IV AND V

Figure 1. TA-2S Schematic.

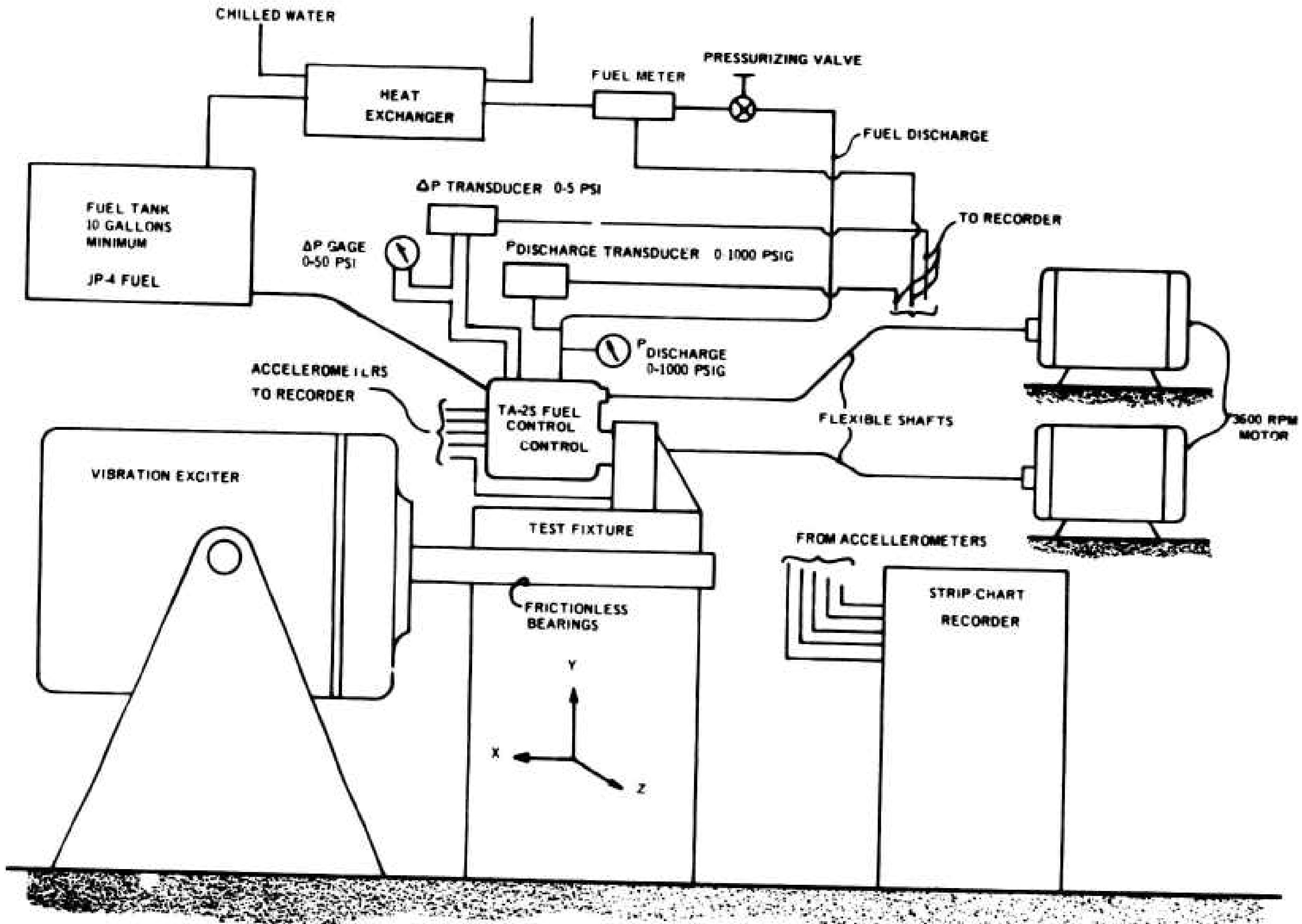


Figure 2. Vibration Test Setup Schematic.

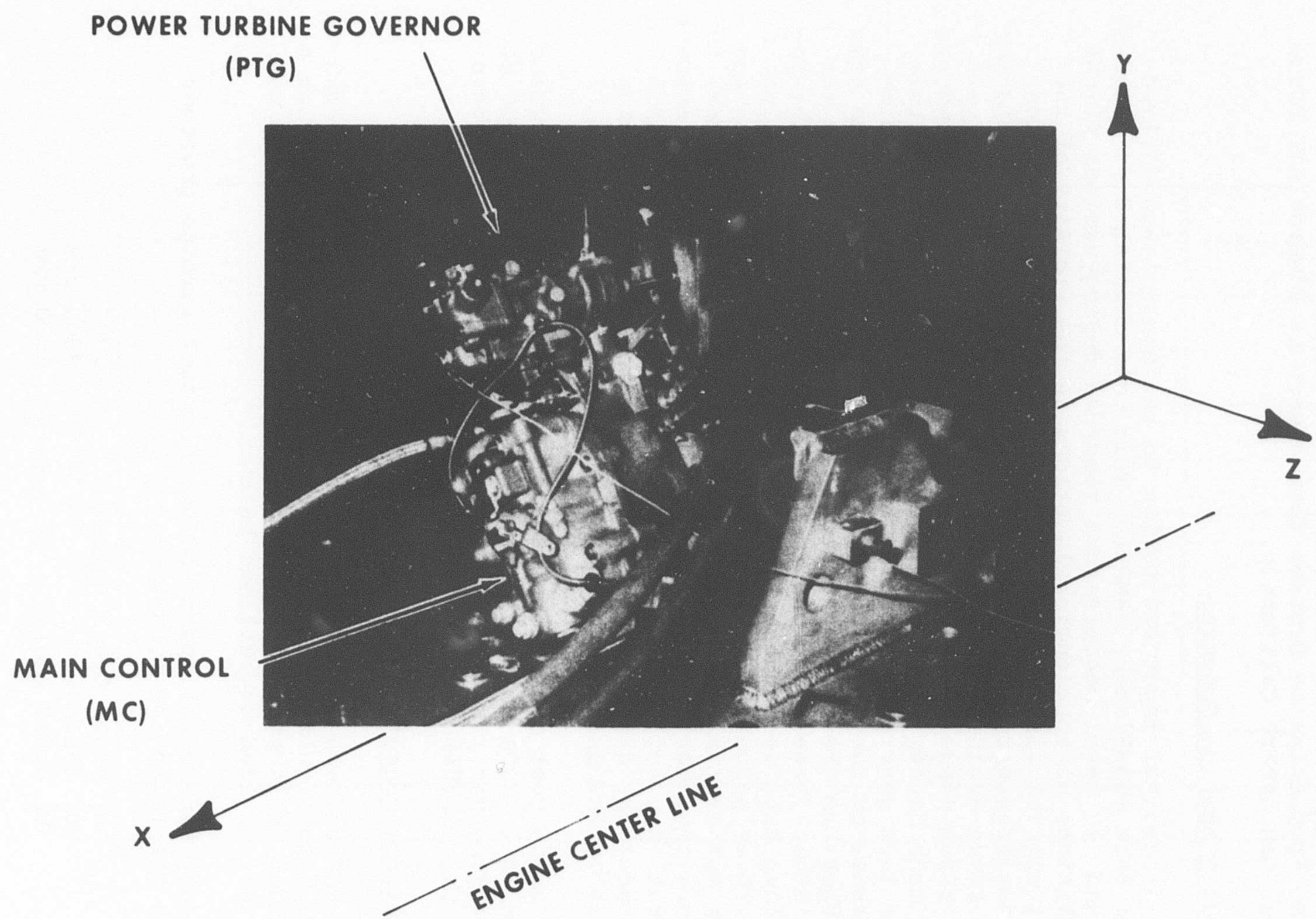


Figure 3. Axes of Vibration.

U. S. Army Aviation Systems Test Agency at Edwards Air Force Base, Lancaster, California.

Input Vibration Spectrum

The sinusoidal input excitation to the control was derived from data supplied by AAMRDL. This data consisted of frequency and g level measurements recorded during various flight conditions at four engine locations in the vicinity of the fuel control. Figure 4 shows an envelope of acceleration versus frequency which encompasses all of the data points for the seven engine operating conditions (takeoff, cruise, etc.) and three planes of vibration. Included in this envelope are the much higher vibration levels recorded at the power turbine tach generator location even though the power turbine governor on the TA-2S control is not hard mounted to the same gearbox as the tach generator. The vibration envelope for the other three locations is also shown in Figure 4. The vibration data from these locations showed close correlation in all three planes, and no one plane exhibited a vibration spectrum substantially different from this envelope. However, to insure that the test control would be subjected to the extreme vibration conditions, a composite vibration envelope which includes the data from the tach generator vibration pickup location was used in the test.

The basic test cycle consisted of varying the input frequency from 10 to 2000 Hz in a time frame per Figure 5. At 2000 Hz, the cycle was reset by reducing the input g level to zero and the frequency of oscillation back to 10 Hz.

Operating Conditions

The control was vibrated in each plane for at least 100 minutes. To simulate the duty cycle of a fuel control during a typical helicopter mission, the test time in each plane was further divided as outlined below.

Seventy minutes of operation on the power turbine governor with the following conditions:

N_1 Power Lever Angle =	100°
N_1 Speed =	3600 rpm
N_2 Speed =	3600 rpm

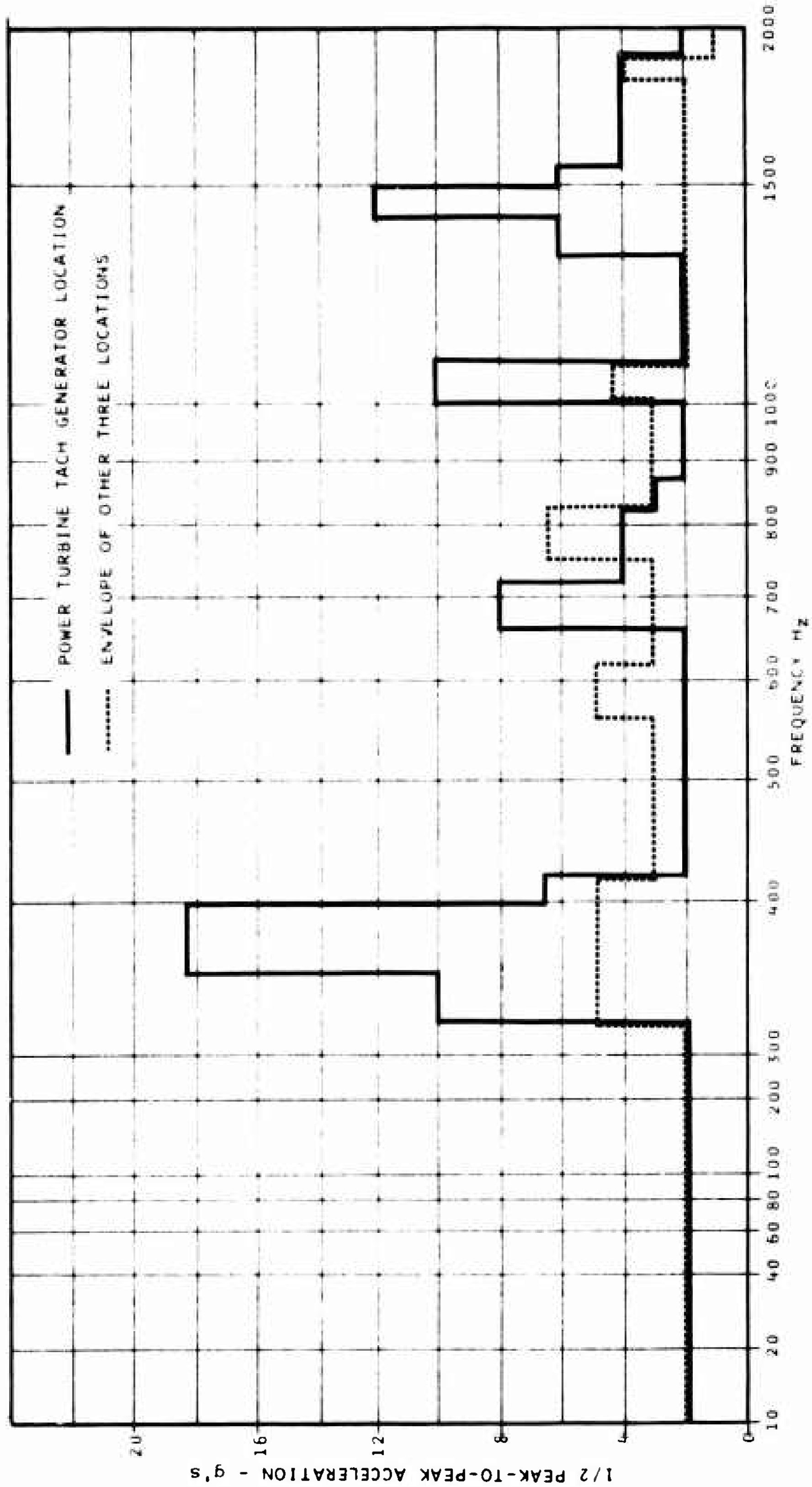


Figure 4. Vibration Envelope.

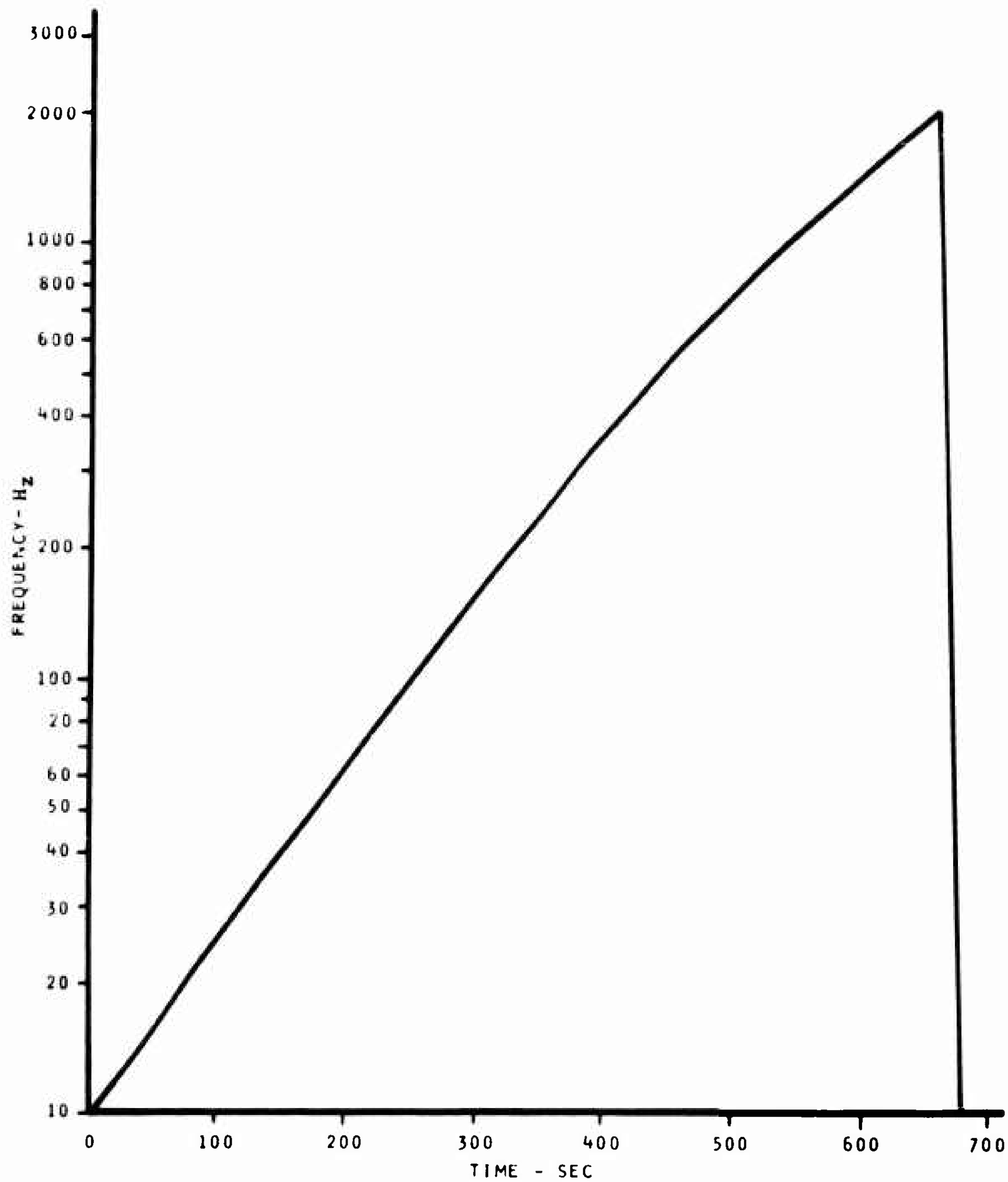


Figure 5. Vibration Test Cycle.

P_1 Atmospheric Pressure =	Ambient
T_A Atmospheric Temperature =	Ambient
Fuel Temperature =	70° to 100°F
Fuel Flow =	550 ± 200 pph
Control Outlet Pressure =	205 ± 10 psig

Ten minutes of operation on the N_1 governor under the following conditions:

N_2 Power Lever Angle =	60°
N_1 Speed =	3600 rpm
N_2 Speed =	3600 rpm
P_1 Atmospheric Pressure =	Ambient
T_A Atmospheric Temperature =	Ambient
Fuel Temperature =	70° to 100°F
Fuel Flow =	550 ± 200 pph
Control Outlet Pressure =	205 ± 10 psig

Ten minutes of operation on the acceleration control mode under the following conditions:

N_1 Power Lever Angle =	100°
N_2 Power Lever Angle =	60°
N_1 Speed =	3600 rpm
N_2 Speed =	3600 rpm
P_1 Atmospheric Pressure =	Ambient
T_A Atmospheric Temperature =	Ambient
Fuel Temperature =	70° to 100°F
Fuel Flow =	650 ± 50 pph
Control Outlet Pressure =	250 ± 10 psig

Ten minutes of operation on the deceleration control mode under the following conditions:

N_2 Power Lever Angle =	60°
N_1 Speed =	3600 rpm
N_2 Speed =	3600 rpm
P_1 Atmospheric Pressure =	Ambient
T_A Atmospheric Temperature =	Ambient
Fuel Flow =	220 ± 10 pph
Fuel Temperature =	70° to 100°F
Control Outlet Pressure =	30 ± 5 psig

TEST RESULTS

The vibration test was conducted to determine the magnitude of:

1. Metered flow shift during vibration.

2. Permanent calibration shift, calibration data taken before and after vibration.

The results of 10 hours of testing on two fuel systems follow:

Fuel Flow Shift During Vibration

The 10 hours of testing yielded approximately 300 feet of strip chart recordings. A section from these recordings is included in Figure 6. Considerable high-frequency noise is present in the fuel flow and pressure traces, which is probably induced by the gear pump. However, the noise frequency is at least as fast as the input excitation (10 Hz minimum) and is beyond the response capabilities of the engine. Therefore, variations in the average (rather than peak-to-peak value) of fuel flow was considered to be the item of interest since it could result in engine performance disturbances.

Power Turbine Governor

Table 1 contains a tabulation of the power turbine governor performance of both control systems during vibration testing. The major fuel flow deviations occurred at an excitation frequency of 400 Hz. As is shown in Figure 4, the input vibration at this frequency level is at maximum amplitude, and fuel flow shifts of 55-70 pph out of 550 pph were experienced at this condition. A plot of a typical worst-case governor response is given in Figure 7. Fuel flow excursions generally followed the magnitude of the power turbine governor housing acceleration and the peak fuel flow variations occurred at the two structural resonant frequencies of 230 and 400 Hz. The lower frequency structural resonant point was dominant; however, low g levels of excitation exist at this frequency which result in a maximum governor acceleration of 24 g's.

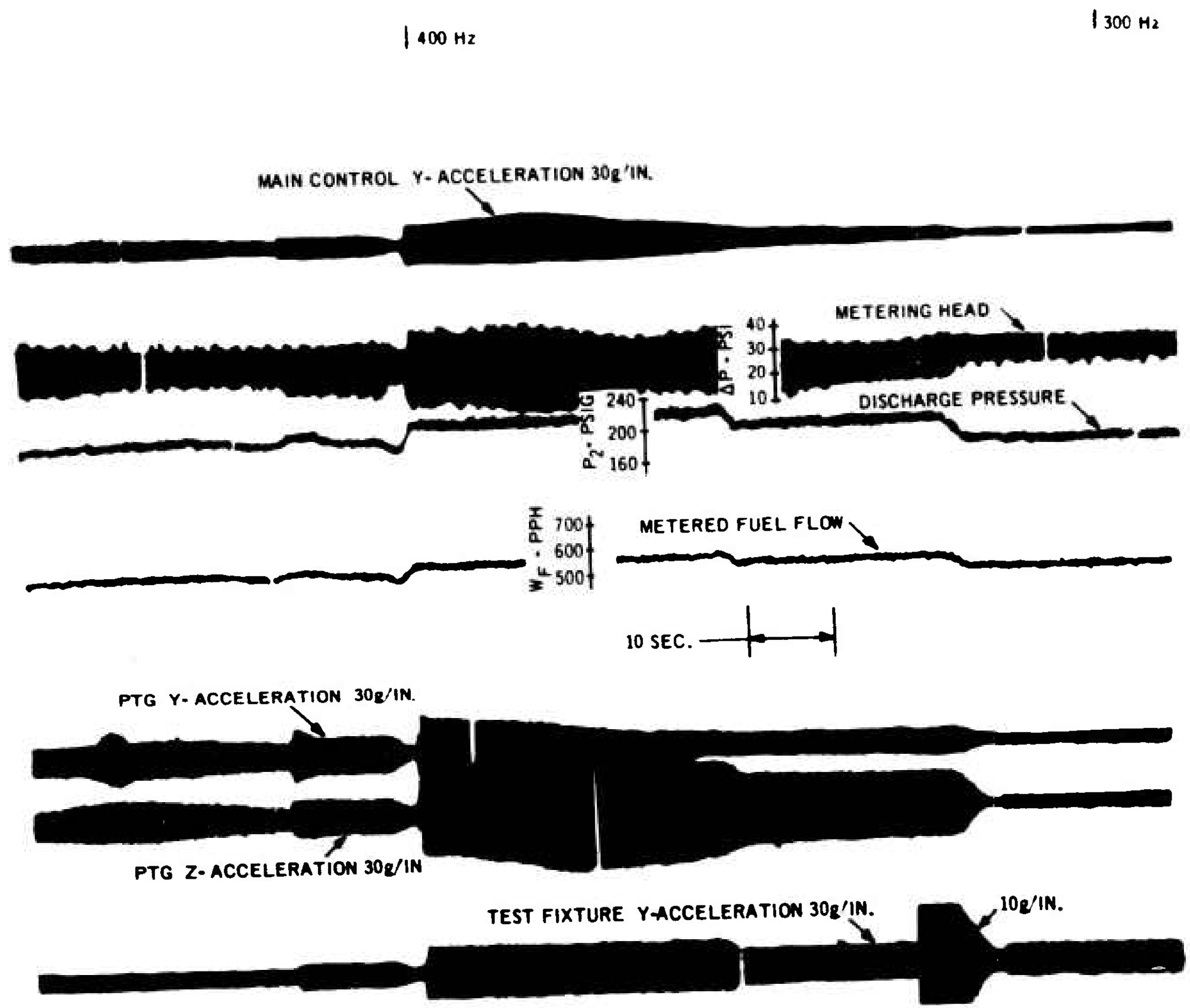


Figure 6. Sample Strip Chart Recording.

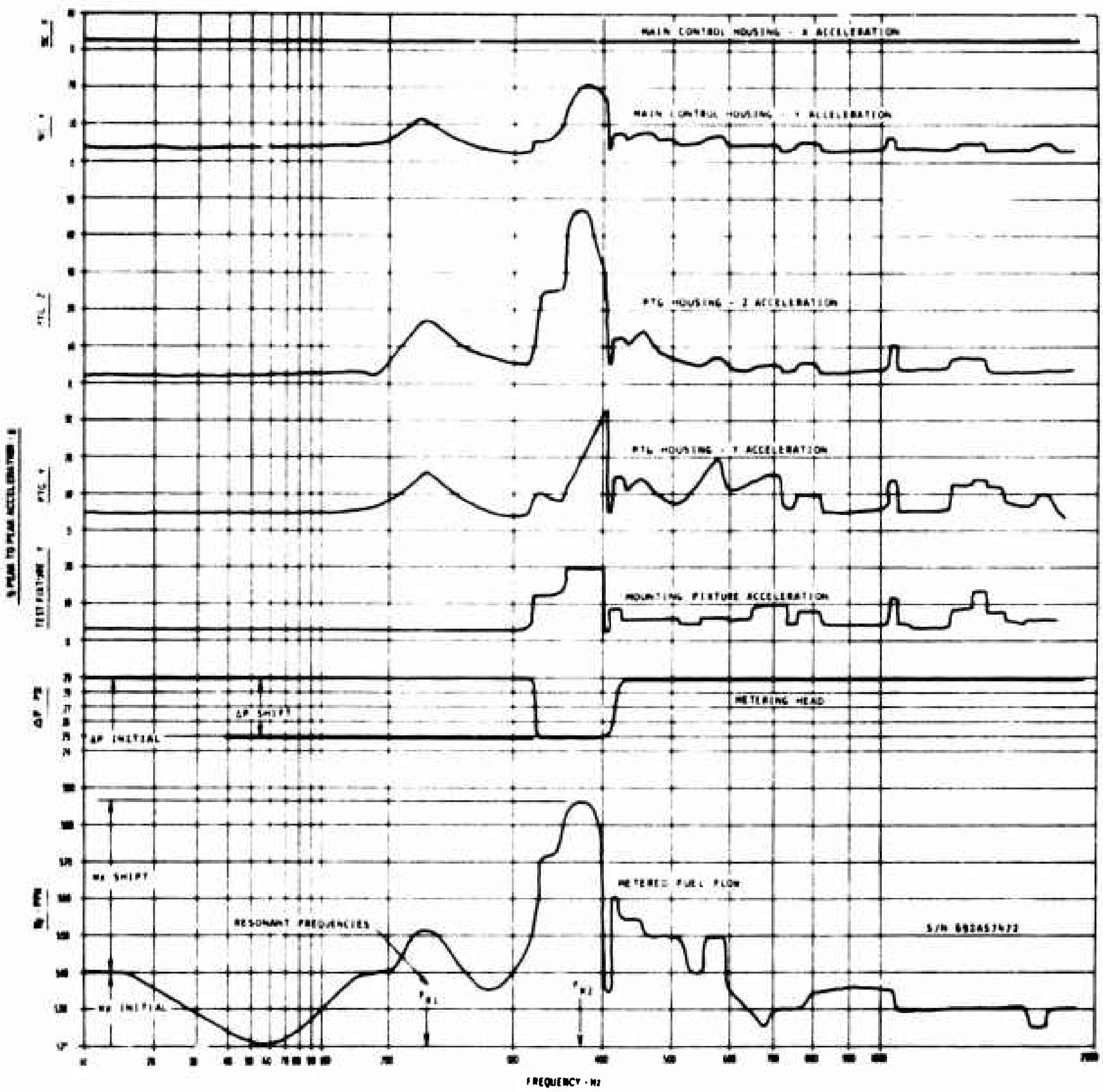


Figure 7. Power Turbine Governor Performance During Vibration in Y Plane.

At 400 Hz, although the structural transmissibility is significantly lower, the high level of input excitation resulted in accelerations of the PTG housing of approximately 50 g's. A plot of governor performance of both control systems at a constant, low level of excitation is given in Figure 8. This figure shows operation at the two major structural resonant points and reveals their relative magnitudes and effect on metered flow.

In addition to resonant fuel flow shifts as described above, the units exhibited some steady-state shifts between vibration cycles. Fuel Control S/N 692AS7472 exhibited a 92-pph shift in W_f (initial) (see Table 1, test cycles 10-18). Also, 20 pph peak-to-peak fuel flow oscillations were induced on one occasion (see Table 1, test cycle 6).

The above magnitude of fuel flow excursions can be put in perspective by referring to Figure 9, which shows the production acceptance limits for the control. A flow deviation of 80 pph is allowed at the operating condition of the vibration test. Thus, the resonant flow variation and random oscillations fall within limits and the steady-state flow shift is just outside the edge of the specified performance limits of the control. Since field service limits normally allow some expansion of the acceptance limits, the above performance demonstrates that the fuel system remained within the operating limits of the device during vibration. Also, when operating closed loop with the engine and helicopter rotor system, a vibration induced fuel flow excursion of 92 pph would be reduced to 18 pph by the loop gain. Figure 10 demonstrates this closed-loop effect and shows that, at worst, a 1.5-percent N_2 deviation would be the result. This speed error is minor and, if detected, could be trimmed out along with other load variations such as wind gusts, helicopter air speed and altitude, etc., with the pilot's beeper trim control.

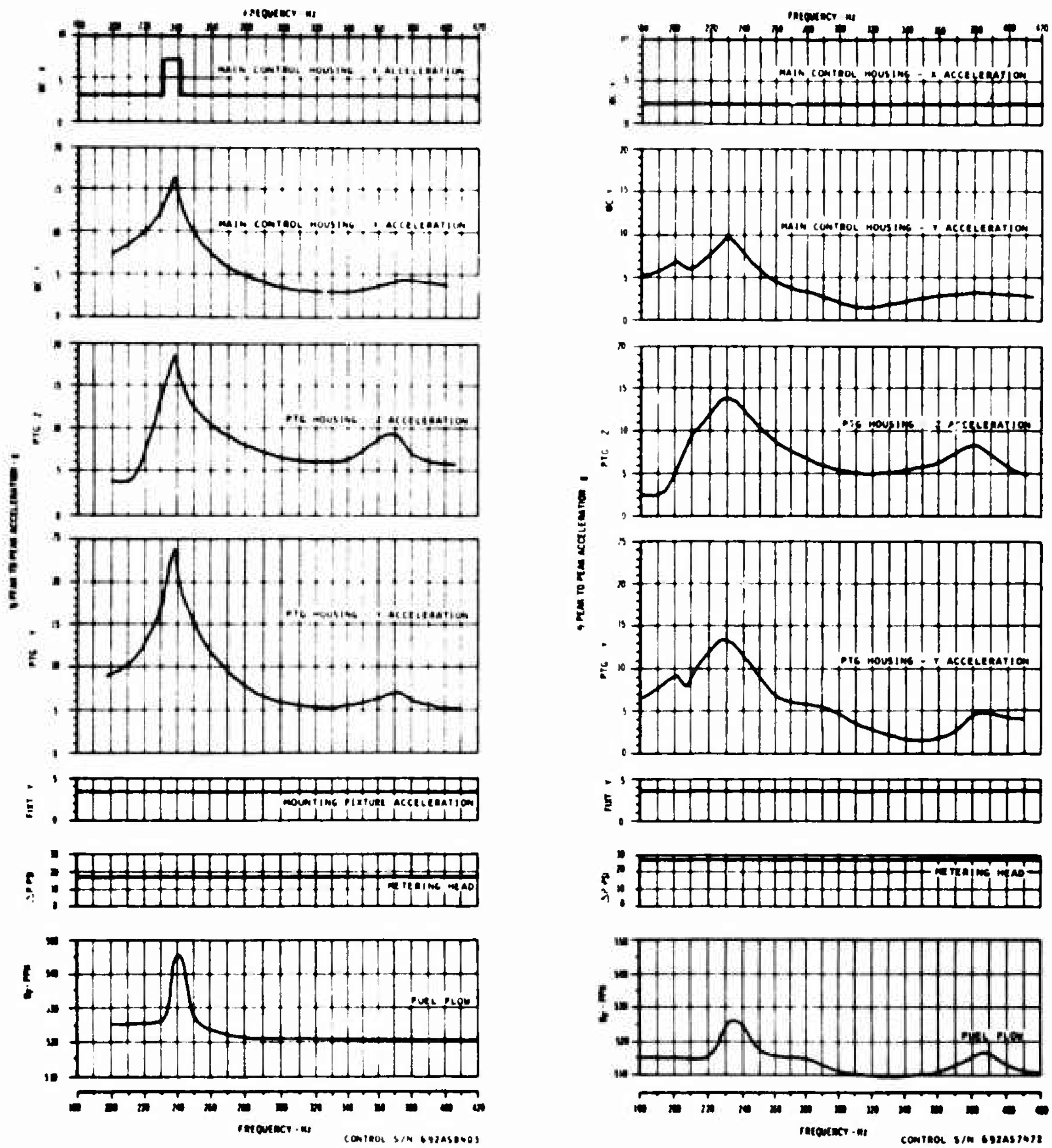


Figure 8. Power Turbine Governor Performance During Vibration in X Plane.

TABLE 1 VIBRATION TEST DATA SUMMARY, POWER TURBINE GOVERNOR CONTROL

INPUT EXCITATION				RESONANT POINTS											
										SIGNIFICANT STRUCTURAL TRANSMISSIBILITY					
TEST CYCLE	PLANE	FREQUENCY RANGE Hz	g LEVEL ±g	F_R Hz	FIXTURE INPUT ±g	W_F INITIAL PPH	W_F SHIFT PPH	(1) ΔP INITIAL PSI	ΔP SHIFT PSI	MC X	MC Y	MC Z	PTG X	PTG Y	PTG Z
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF Y PLANE TESTING															
1	Y	200-320 320-400	3.6 5.4	230 400	3.6 5.4	525 525	+20 +20	17 17	0 0	1.7	3.0 1.0			5.0 2.0	4.4 3.1
2		200-320 320-400	3.3 2.2	230 400	3.3 2.2	525 525	+20 +35	17.5 17.5	0 0		4.1 1.0			6.1 2.3	4.0 3.6
3		200-400	3.3	230 400	3.3 3.3	520 520	+25 0	17.5 17.5	0 0	3.2	4.0			7.1 2.3	5.6 3.1
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF X PLANE TESTING															
4	X	100-200	4.5	100	4.5	555	+20	22.5	0				2.4	1.9	1.5
5		20-2000	FIG 4 OR 6 MAX	100	4.0	555	+10	22.5	0				1.0	1.9	1.0
6		20-2000	FIG 4 OR 12 MAX	50/70	4.5	550	W_F OSCILLATIONS AT .6 HZ 540-560 PPH PEAK TO PEAK								
7		20-2000	FIG 4	80 400	4.5 19.0	565 565	-15 +25	22.8 22.8	0 0				1.0 .4	1.1 .7	.9
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF Z PLANE TESTING FIG 4															
8	Z	20-2000	3.6	100	3.6	595	+25	24	0		4.7		3.0	4.0	9.0
9		20-2000	FIG	100 400	4.1 22.5	585 585	+15 +30	23 23	0 0		4.4 1.0		3.1 .8	4.6 1.6	8.5 2.3
S / N 692AS8403															

LEGEND (SEE FIGURE 7)

F_R -- RESONANT FREQUENCY
FIXTURE INPUT -- ACCELERATION OF TEST FIXTURE IN THE INPUT PLANE
W_F - INITIAL -- FUEL FLOW AT START OF TEST CYCLE
W_F - SHIFT -- DEVIATION FROM INITIAL FLOW AT RESONANCE
ΔP - INITIAL -- METERING HEAD AT START OF TEST CYCLE
ΔP - SHIFT -- DEVIATION FROM INITIAL PRESSURE AT RESONANCE
ACCELEROMETER LOCATIONS
MC_X -- MAIN CONTROL (GPC) X PLANE
PTG_Y -- POWER TURBINE GOVERNOR Y PLANE
TRANSMISSIBILITY $\frac{PTG}{Y}$ -- RATIO OF CONTROL TO INPUT ACCELERATIONS

NOTES:
(1) ΔP METERING HEAD CHANNEL SHIFTED DURING THE TEST - HOWEVER A GAGE IN THE LINE, READ BETWEEN 24-26 PSI. THE MAGNITUDE OF ΔP SHIFT SHOULD BE ACCURATE

INPUT EXCITATION				RESONANT POINTS											
										SIGNIFICANT STRUCTURAL TRANSMISSIBILITY					
TEST CYCLE	PLANE	FREQUENCY RANGE Hz	g LEVEL ±g	F_R Hz	FIXTURE INPUT ±g	W_F INITIAL PPH	W_F SHIFT PPH	(1) ΔP INITIAL PSI	ΔP SHIFT PSI	MC X	MC Y	MC Z	PTG X	PTG Y	PTG Z
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF Y PLANE TESTING															
10		10-2000	FIG. 4	230	3.5	540	-10	29	-1		2.2			3.4	3.2
11		10-320 320-2000	3.5 5.5	230	3.5	515	+10	29	0		2.7			3.9	3.7
12		200-2000	FIG 4 OR 7.2 MAX	230 400	3.5 7.2	510 510	+10 +15	29 29	0 0		2.8 .9			3.5 1.4	4.5 2.6
13		200-2000	FIG 4 OR 9.0 MAX	400	9.0	560	+15	28.5	-4.5		.9			1.2	2.5
14		200-2000	FIG. OR 11 MAX	400	11	550	+35	29	-5		1.0			1.3	2.5
15		200-2000	FIG 4 OR 13.4 MAX	400	13.4	535	+55	28.5	-4.5		1.0			1.0	2.6
16		200-700	FIG 4 OR 17.1 MAX	400	17.1	525	+55	29	-5		1.0			.9	2.5
17		200-700	FIG 4 OR 22.8 MAX	400	22.8	602	-17	30	-5		.9			1.0	2.4
18		200-2000 (EXCEPT) 320--350 12 SEC	3.5 12	400	3.5	590	TRANSIENT W_F UNDERSHOOT OF -70 PPH PEAK EXCITED BY 12 g INPUT BURST AND LASTING FOR 41 SECONDS								
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF X PLANE TESTING															
19	X	20-2000	4.0	100	4.0	520	-10	21.5	0	.6			.5	.8	.8
20		20-2000	FIG 4 OR 6 MAX	170	4.0	510	+25	22	0				2.4	2.4	2.9
21		20-2000	FIG 4 OR 8 MAX	170	4.0	505	+30	22	0				2.6	2.4	3.0
OPERATING POINT WAS SET UP AND NOT ALTERED FOR THE DURATION OF Z PLANE TESTING															
22	Z	20-2000	4	170	4.2	540	-20	11	0			1.6	.7	1.3	2.2
23		20-2000	FIG 4 OR 20.4 MAX	400	20.4	510	+25	9	0			1.1	.9	1.0	2.5
S/N 692A57472															

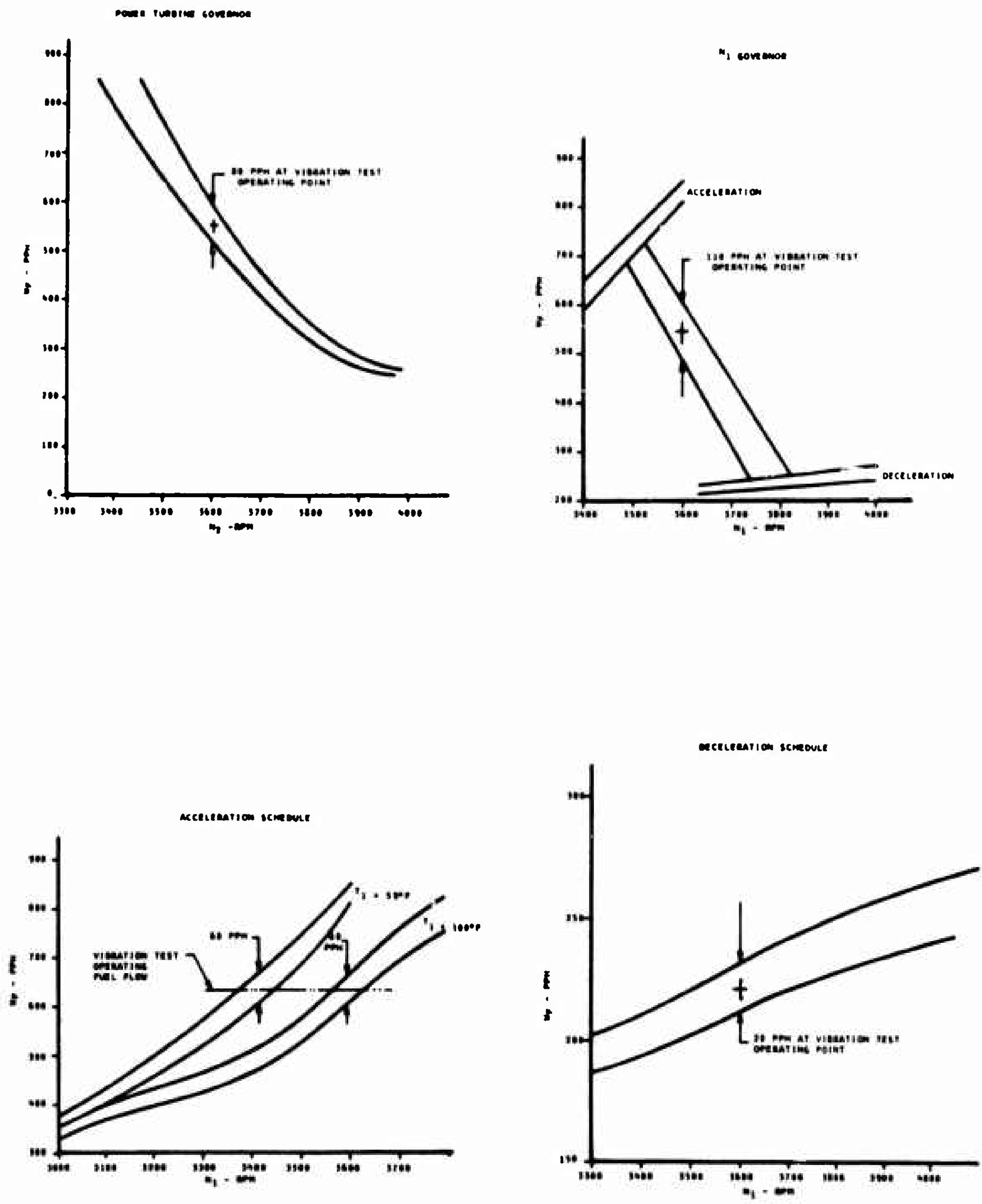


Figure 9. TA-2S Fuel System Operating Limits.

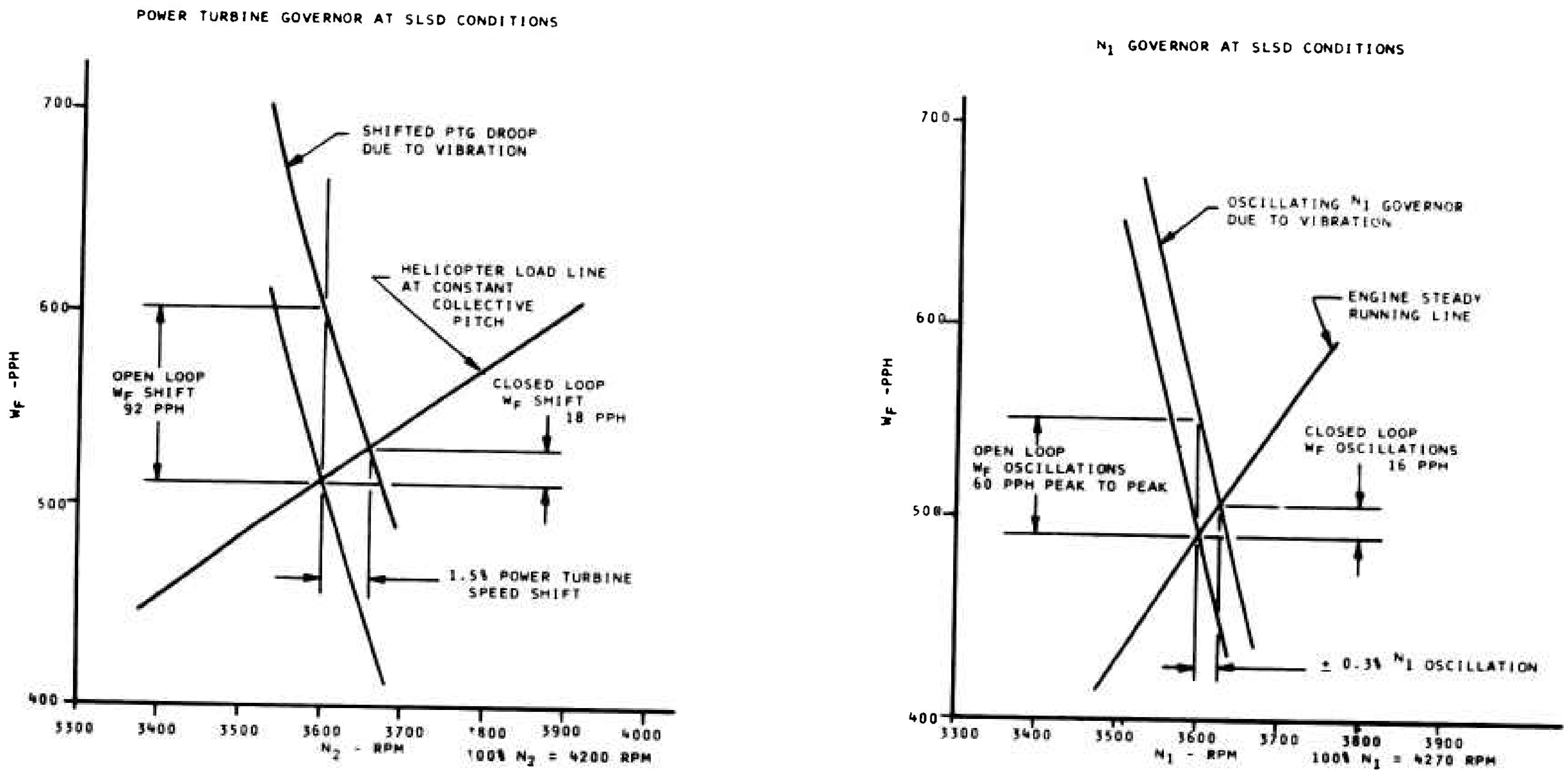


Figure 10. Open- vs. Closed-Loop Operation Gas Producer and Power Turbine Governor.

Gas Producer Control

Table 2 contains a tabulation of the gas producer control performance of both controls operating in the acceleration, deceleration and N_1 governing mode.

Acceleration

A maximum fuel flow resonant shift of 30 pph was recorded. Compared to the allowable production test limit of 60 pph, as shown in Figure 9, the unit performed satisfactorily.

Deceleration

The deceleration mode of control resulted in the steadiest metered flow of all operating conditions. Negligible fuel flow variations were observed in all test cycles except for one test (see Table II, test cycle 6). In this test, the level of fuel flow was inadvertently mis-set to an abnormally high fuel flow level and a shift of 25 pph was recorded. At this condition, the unit was operating on the N_1 governor and for this condition the flow shift is within operating limits.

N_1 Governor

The N_1 governor of both control systems exhibited nervous behavior in all planes of vibration. The governor speed sensor is a second-order hydromechanical servo system which has, on occasion, yielded open-loop fuel flow oscillations due to inherent low damping. Vibration seemed to excite the two units tested such that a maximum open-loop fuel flow oscillation of 60 pph peak-to-peak was observed. This variation is within the acceptance limits of 110 pph as shown in Figure 9. Furthermore, when operating closed loop through the engine, the steep N_1 governor gain would reduce the 60-pph oscillation to about 16 pph peak-to-peak. The resultant gas producer speed drift would be an acceptable ± 0.3-percent N_1 as shown in Figure 10. Moreover, on the helicopter the control operates on the N_1 governor only during idle and maximum power or

TABLE 2 VIBRATION TEST DATA SUMMARY, GAS PRODUCER CONTROL

INPUT EXCITATION				RESONANT POINTS											
TEST CYCLE	PLANE	FREQUECY RANGE HZ	g LEVEL ± g	F_R HZ	FIXTURE INPUT ± g	W_F INITIAL PPH	W_F SHIFT PPH	ΔP INITIAL PPH	ΔP SHIFT PSI	SIGNIFICANT STRUCTURAL TRANSMISSIBILITY MC X	MC Y	MC Z	PTG X	PTG Y	PTG Z
ACCELERATION MODE															
1	Y	10 2000	FIG 4	400	20.4	670	-30	17.5	0		1.0			1.8	2.5
2	X	10 2000	FIG 4	50	4.8	615	+20	22.5	0				1.0	1.0	
3	Z	10 2000	FIG 4	W_F = 625 PPH THROUGHTOUT TEST											
DECELERATION MODE *															
4	Y	10 2000	FIG 4	W_F = 220 PPH THROUGHOUT TEST											
5	X	10 2000	FIG 4	W_F = 200 PPH THROUGHOUT TEST											
6	Z *	10 2000	FIG 4	180	3.9	265	-25	24	-1			4.9	3.8	2.5	10.5
N_1 GOVERNOR MODE															
7	Y	10 2000	FIG 4	W_F STEADY BUT SHIFTS 130 PPH DUE TO THROTTLE LEVER MOVEMENT - NOT A VALID TEST											
8	X	10 2000	FIG 4	50-70	2 W_F OSCILLATIONS AT .09HZ - 500 - 530 PPH PEAK TO PEAK										
9	Z	10 2000	FIG 4	RANDOM OSCILLATIONS AT .16HZ 565 - 585 PPH PEAK TO PEAK											
S / N 692A58403															

LEGEND (SEE FIGURE 7)

F_R -- RESONANT FREQUENCY
FIXTURE INPUT -- ACCELERATION OF TEST FIXTURE IN THE INPUT PLANE
W_F - INITIAL -- FUEL FLOW AT START OF TEST CYCLE
W_F - SHIFT -- DEVIATION FROM INITIAL FLOW AT RESONANCE
ΔP - INITIAL -- METERING HEAD AT START OF TEST CYCLE
ΔP - SHIFT -- DEVIATION FROM INITIAL PRESSURE AT RESONANCE
ACCELEROMETER LOCATIONS
MC_x -- MAIN CONTROL X PLANE
PTG_Y -- POWER TURBINE GOVERNOR Y PLANE
ETC --
TRANSMISSIBILITY -- RATIO OF CONTROL TO INPUT ACCELERATIONS

NOTE:
OPERATING POINT WAS RESET BETWEEN EACH PLANE CHANGE THUS VARIATIONS IN INITIAL CONDITIONS ARE CAUSED BY HUMAN ERROR

* Z PLANE OPERATING POINT WAS MISSET SUCH THAT THE CONTROL WAS OPERATING ON THE N_1 GOVERNOR

INPUT EXCITATION				RESONANT POINTS											
TEST CYCLE	PLANE	FREQUENCY RANGE HZ	g LEVEL ± g	F_R HZ	FIXTURE INITIAL ± g	W_F INITIAL PPH	W_F SHIFT PPH	P INITIAL PPH	P SHIFT PSI	SIGNIFICANT STRUCTURAL TRANSMISSIBILITY MC X	MC Y	MC Z	PTG X	PTG Y	PTG Z
ACCELERATION MODE															
10	Y	10 2000	FIG 4	400	22.8	600	-25	28.5	-2.5		1.1			1.1	2.3
11	X	10 2000	FIG 4	W_F = 585 PPH THROUGHOUT TEST											
12	Z	10 2000	FIG 4	W_F = 585 PPH THROUGHOUT TEST											
DECELERATION MODE															
13	Y	10 2000	FIG 4	W_F = 200 PPH THROUGHOUT TEST											
14	X	10 2000	FIG 4	W_F = 190 PPH THROUGHOUT TEST											
15	Z	10 2000	FIG 4	W_F = 190 PPH THROUGHOUT TEST											
N_1 GOVERNOR MODE															
16	Y	10 2000	FIG 4	20/100	3.9	W_F OSCILLATIONS AT .2 HZ 490-550 PPH PEAK TO PEAK									
				1500	9	W_F OSCILLATIONS AT .1 - .2 HZ 440-500 PPH PEAK TO PEAK									
17	X	10 2000	FIG 4	50/70	3.9	W_F GLITCH OF 20 SEC. DURATION -20 TO +25 PPH									
18	Z	10 2000	FIG 4	RANDOM OSCILLATIONS AT .18 HZ 520 - 540 PPH PEAK TO PEAK											
S/N 692A57472															

temperature limiting conditions. These conditions occur during only a small percentage of the flight time; therefore the N_1 governor vibration problem would be easily identified.

Calibration Data

The two TA-2S fuel control systems were completely overhauled, inspected and bench calibrated prior to the vibration test. Upon completion of the 5 hours of vibration testing on each unit, the controls were again bench tested to determine the extent of any calibration shifts. Unit S/N 692AS7472 repeated its pretest calibration with all check points falling within acceptance limits. However, control S/N 692AS8403 exhibited a rich fuel flow shift of between 4.5 and 7.8 percent which fell outside the acceptance limits at four of the thirteen acceleration check points. This schedule shift was found to be caused by the improper assembly of the three-dimensional acceleration cam. A 0.040-in. shim which locates the cam in relation to the speed computer actuator piston had been inadvertently left out. By replacing the missing shim, the unit fell within calibration limits. This could have happened because the nonlinearities in the speed computer and the inherent variations in the performance of the fuel metering system and P_1 multiplier make it possible to calibrate a single control to within limits with the cam shims missing. However, the control will not repeatedly meet acceptance limits, as was the case in this test. Moreover, there is no reason that vibration would cause a calibration shift with the shims missing because the assembly is just as secure as it is with the shims. Furthermore, at the conclusion of the post-test calibration, both controls were disassembled and inspected without detection of any damage or evidence of vibration-induced problems such as loose bolts, etc.

Conclusions

The results of the analysis of vibration test and calibration data indicate that vibration is not the underlying cause of the high percentage of justified control removals for which the causes are unknown.

It is concluded that the effect of vibration on the repeatability of the control was negligible. Neglecting the human error of improper assembly, the two controls maintained their calibration settings within the operating limits of the device.

FUEL FILTER TEST

Contaminated fuel limits the wear life of components by producing abrasive wear, and degrading the performance of the control. Reports from various sources indicate that benefits could be gained by using filtration finer than the 25μ absolute filter presently used in fuel controls. A 3μ absolute filter had been recommended based on experience gained from engine and commercial hydraulic systems. To determine the validity of this theory, the two TA-2S controls previously vibration-tested were disassembled and their critical parts were dimensionally inspected. The two controls were subsequently preset and flow bench calibrated, and run on fuel contaminated per MIL-E-5007C as outlined in Table III. The complete test for each control consisted of 50 hours with fuel at room temperature and 20 hours with fuel at -65°F. To eliminate double filtration, the control inlet screen and servo wash flow filter were removed, and thereby all filtration was by the single external barrier filter. One control was run with a 3μ absolute filter, and the other control was run with a 25μ absolute filter. Following the 70-hour test, each control was calibrated and disassembled for inspection. The critical parts were remeasured to establish the amount of wear.

TEST DESCRIPTION

A schematic and a photograph of the test setup are shown in Figures 11 and 12. The contaminated fuel supply tank consisted of an outer shell with an internal conical shell perforated near its upper rim. The returning fuel entering the annulus between the shells spilled into the conical section through the perforations. The solid contaminant was distributed on a belt conveyor driven by a motor and speed reduction gearing. As the belt rotated, the contaminant fell into the conical section of the fuel tank. The belt advanced at a speed such that specified contaminant fell evenly in the tank at a rate of 40.1 grams of contaminant per 1000 gallons of fuel. The average fuel flow was used to calculate the weight of the contaminant. The crude naphtenic acid listed in Table III was added to the fuel tank at the start of each test based on the capacity of the tank including piping, heat exchanger, etc. Specification MIL-E-5007C lists salt water as one ingredient of the fuel contaminant to be added. Because of the -65°F phase of testing, no salt water was added to the fuel. Past

TABLE 3. FUEL CONTAMINANT

CONTAMINANT	PARTICLE SIZE	QUANTITY
Iron oxide	0- 5 microns	28.5 gm/1,000 gal.
Iron oxide	5-10 microns	1.5 gm/1,000 gal.
Sharp silica sand	300-420 microns	1.0 gm/1,000 gal.
Sharp silica sand	150-300 microns	1.0 gm/1,000 gal.
Prepared dirt conforming to A. C. Spark Plug Co. Part No. 1543637 (coarse Arizona road dust).	Mixture as follows: 0- 5 microns (12 percent) 5-10 microns (12 percent) 10-20 microns (14 percent) 20-40 microns (23 percent) 40-80 microns (30 percent) 80-200 microns (9 percent)	8.0 gm/1,000 gal.
Cotton linters	Staple below 7 (U.S. Department of Agriculture Grading Standards)	0.1 gm/1,000 gal.
Crude naphthenic acid		0.03 percent by volume

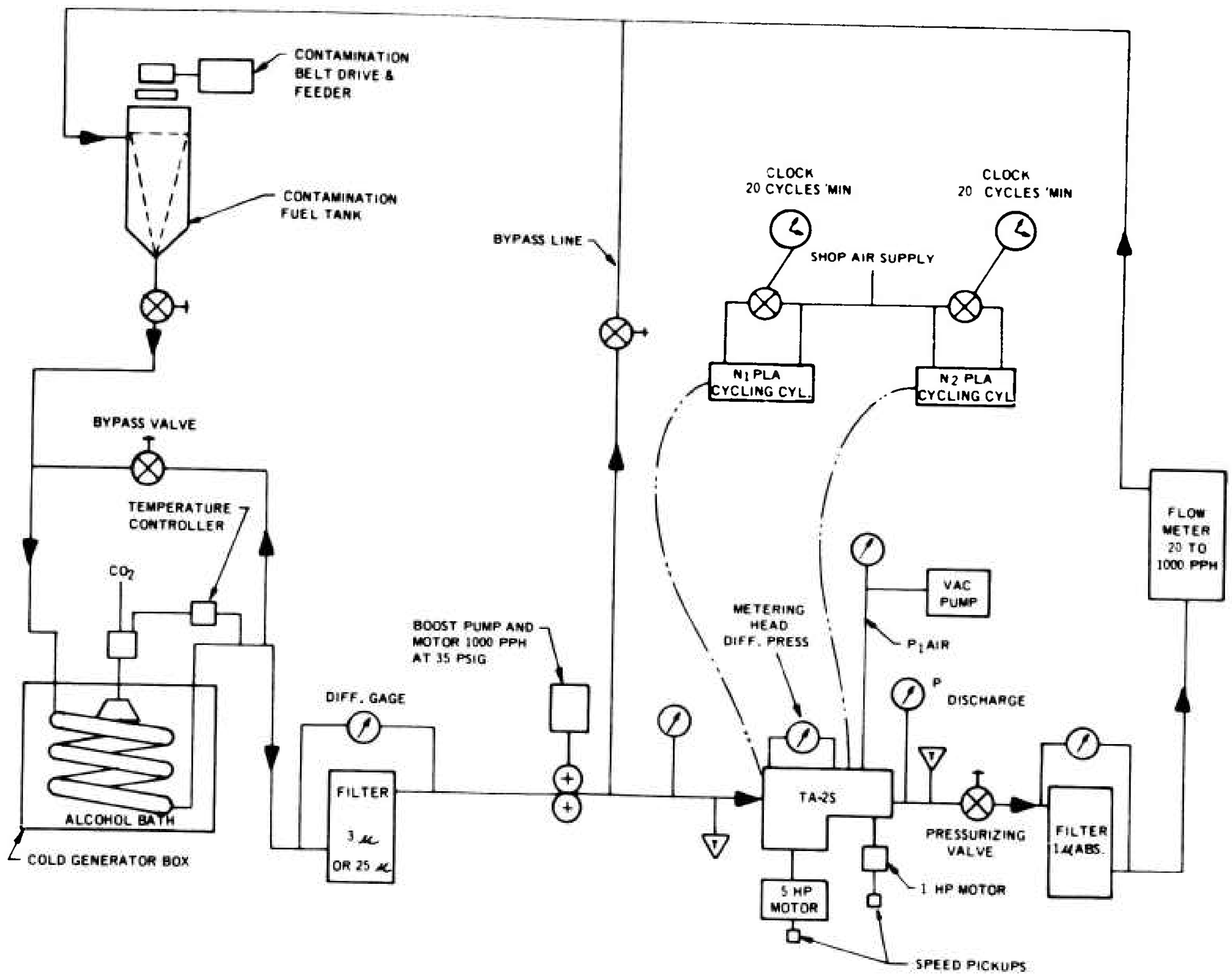


Figure 11. Contamination Test Schematic.

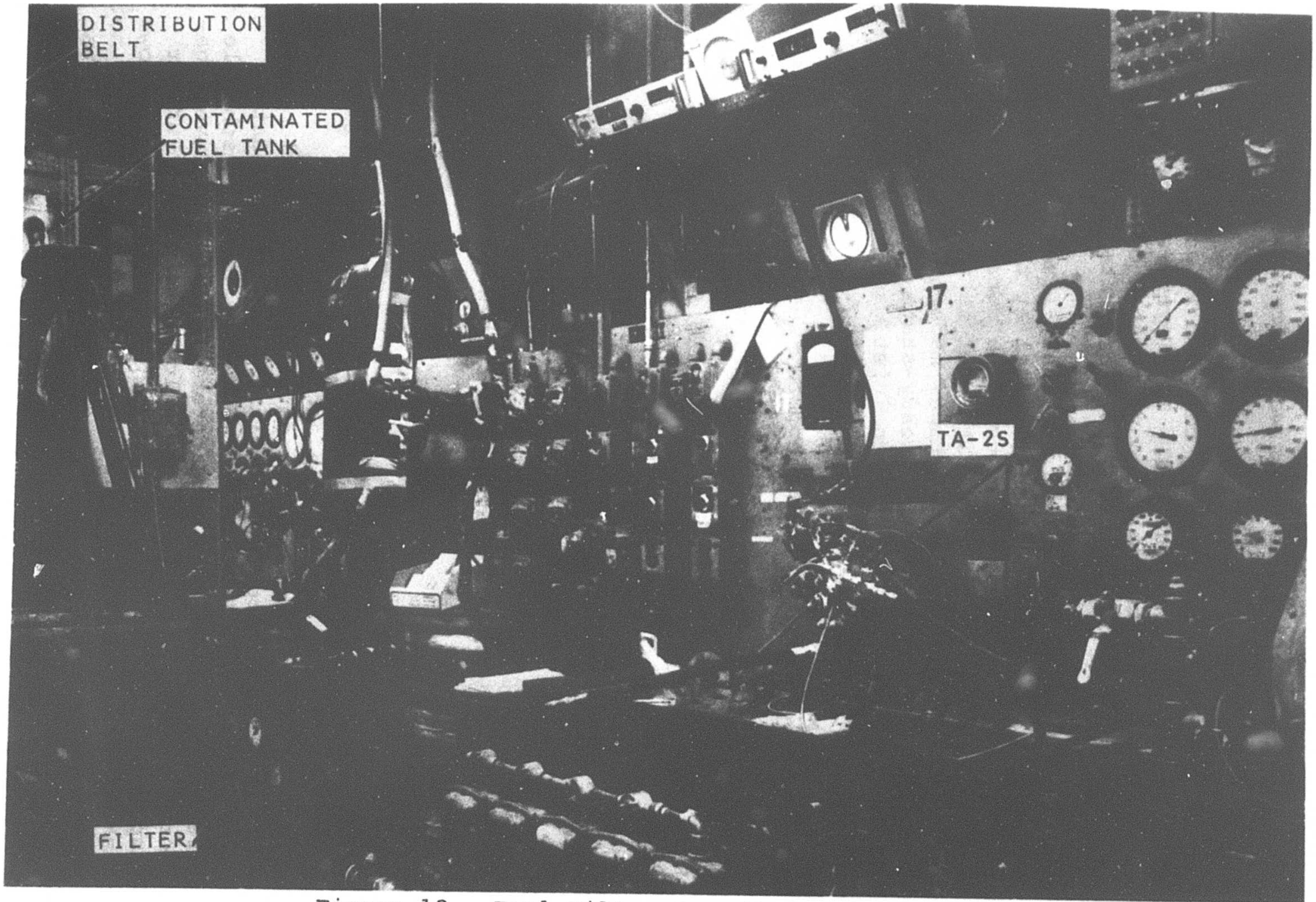


Figure 12. Fuel Filter Room Temperature Test.

experience dictated that the water would freeze in the heat exchanger and prevent testing.

The controls were operated during the test in accordance with the test cycle shown in Figure 13. This cycle exposes one rotating and one nonrotating servo valve to a constant 20 cycles/minute throughout testing, to accelerate valve wear.

Fifty hours, which is equivalent to twenty-five cycles, were accumulated on each control with the fuel at the inlet maintained at room temperature. Twenty hours corresponding to ten cycles were run on each control with fuel at the inlet to the filter at -65°F ± 5°F. No attempt was made to control the ambient temperature; however, the steady-state temperature at the control was noted at -55°F when inlet was -65°F. Figure 14 shows the control during -65°F fuel testing.

Following the 70-hour test, each control was run for at least one hour with clean fuel prior to being calibrated.

TEST RESULTS

Calibration

Both controls were calibrated prior to and following the contamination testing.

Fuel control S/N 692AS7472 was tested with the 3μ filter, and the S/N 692AS8402 control was tested with the 25μ filter.

Calibration Data

All pretest and post-test calibration points on S/N 692AS7472 control were within acceptable limits.
S/N 692AS8403 control pretest data showed that all points were within the acceptable limits; the post-test data showed that two points on the -65°F acceleration schedule were out of limits by a considerable value. An investigation indicated a defective T_1 bellows assembly. Further inspection determined that the liquid-filled motor bellows leaked at a brazed junction. Because the T_1 sensor was not supported during the vibration test, it was theorized that a fatigue crack started to develop during that test, and

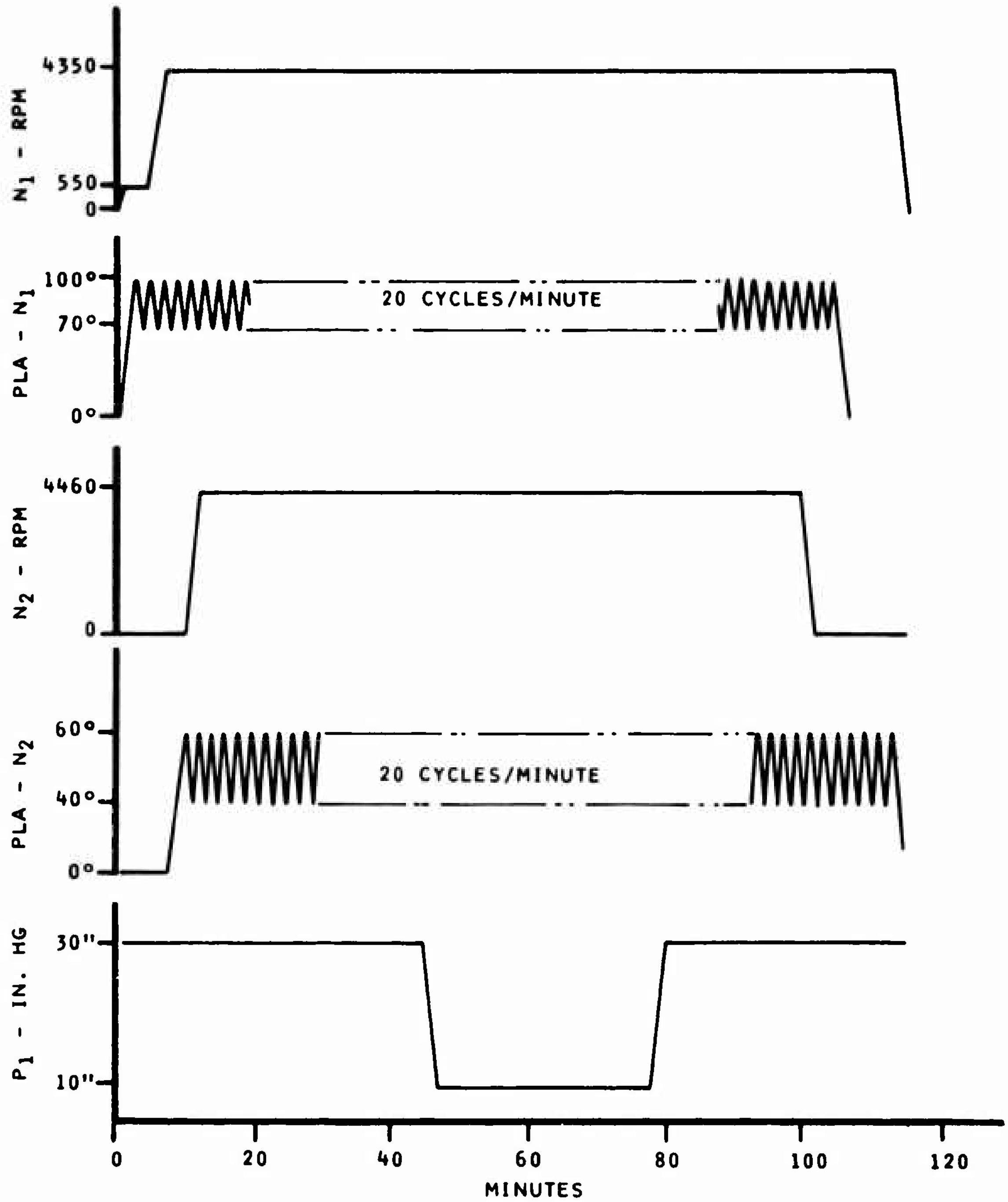


Figure 13. Fuel Filter Test Cycle.

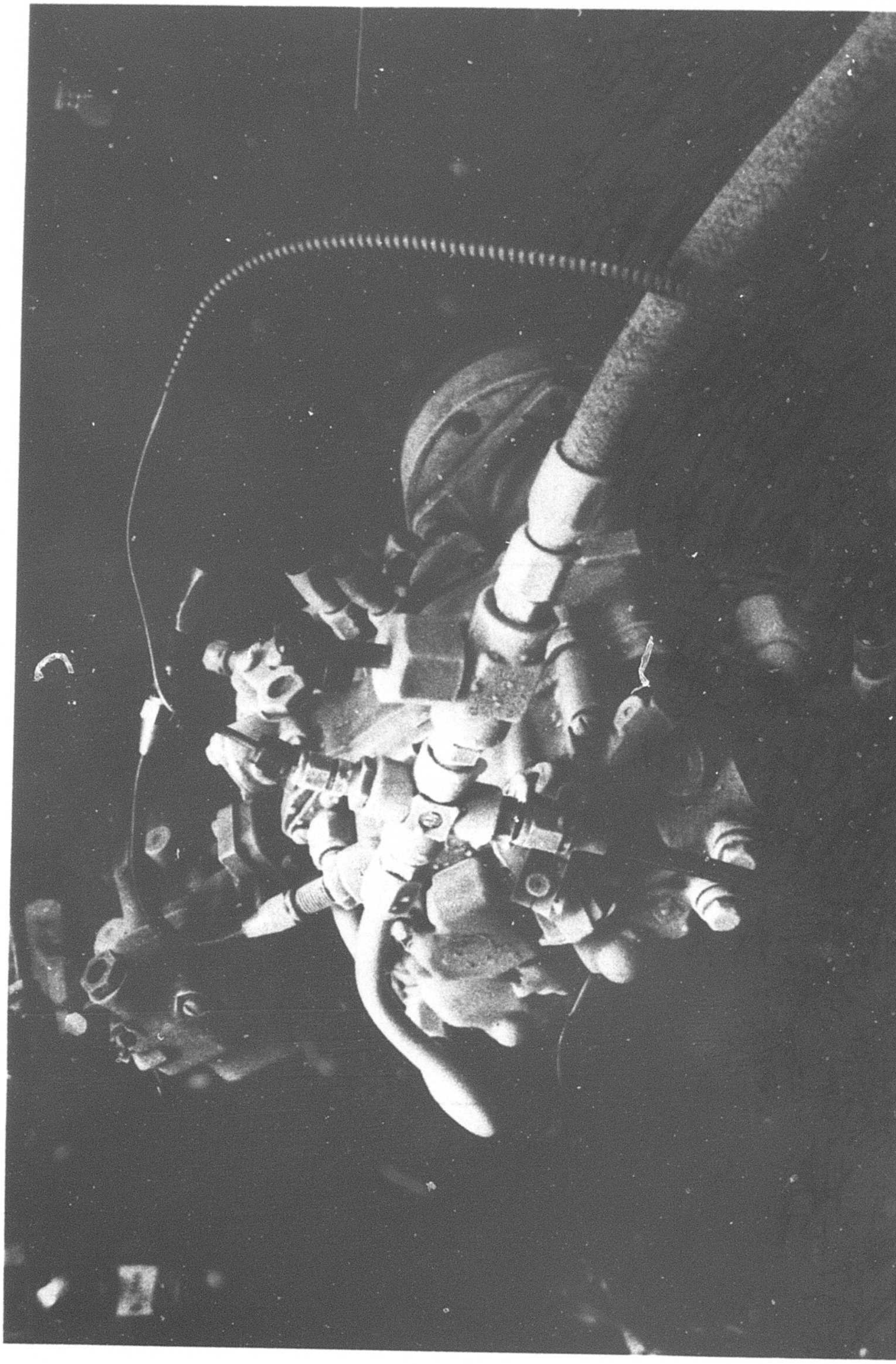


Figure 14. TA-2S at -65°F.

eventually cracked from normal handling during the contamination testing. However, this type of failure has not been reported from the field, and since it is so easily identified, it could not have been occurring without being detected. Therefore, the failure is attributed to the test conditions.

The pump in each control system was tested before and after contamination testing. The test results are shown plotted in Figures 15 and 16. The results show that after the contamination test, fuel flow increases slightly at low discharge pressures and decreases at the high discharge pressure. There is no clear explanation for the slight flow increase but it is probably due to test equipment inaccuracies, and operator-to-operator variations in setting and recording pressures, speeds and flows. The decrease in fuel flow at high discharge pressure is of the order of 100 pph. This loss in performance can be attributed to normal wear and settling of the bearings resulting in additional clearance between the gears and housing.

Inspection

Before and after the contamination tests were conducted, both controls were disassembled and critical parts, such as servo valves, pump gears and bearing, etc., were dimensionally inspected. The results of the inspection on the fuel control components are summarized in Tables 4 and 5, and the parts listed in these tables are designated on the fuel control schematic shown in Figure 1. The results of the inspection of the pumping components for both controls are summarized in Tables 6 through 9. A review of the inspection data on both systems indicates that there was no significant wear of the control or the pump components in either system. In some cases the dimensional data indicates that a part increased in size rather than decreased due to wear. This is most likely due to measurement accuracy, but in any case the differences in dimensions are insignificant. The pump bearing did show a slight wear; however, the bearings in the 25μ filter test showed less wear than the bearing in the 3μ filter test. Therefore, this wear cannot be attributed to fuel contamination.

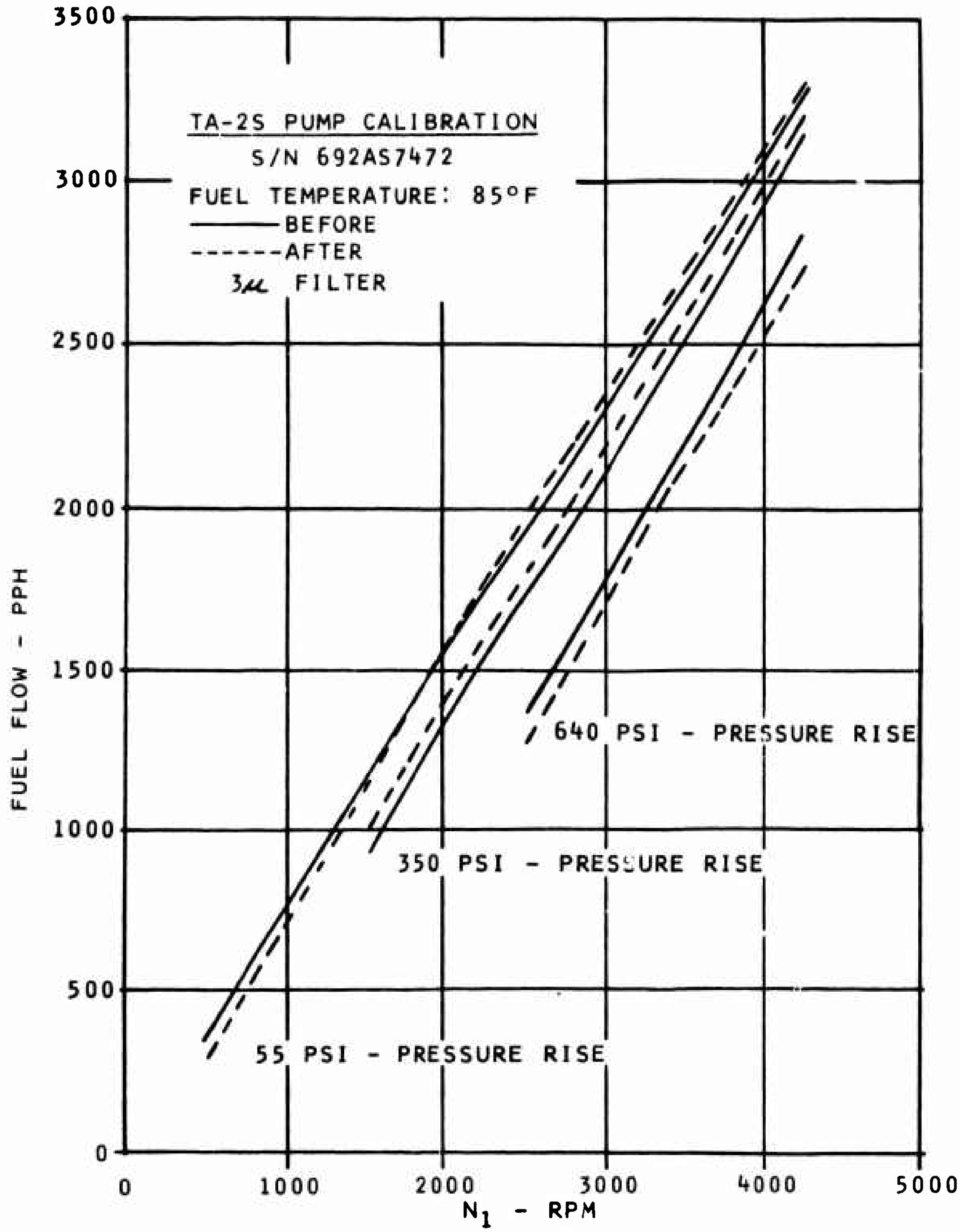


Figure 15. Pump Performance, S/N 692AS7472.

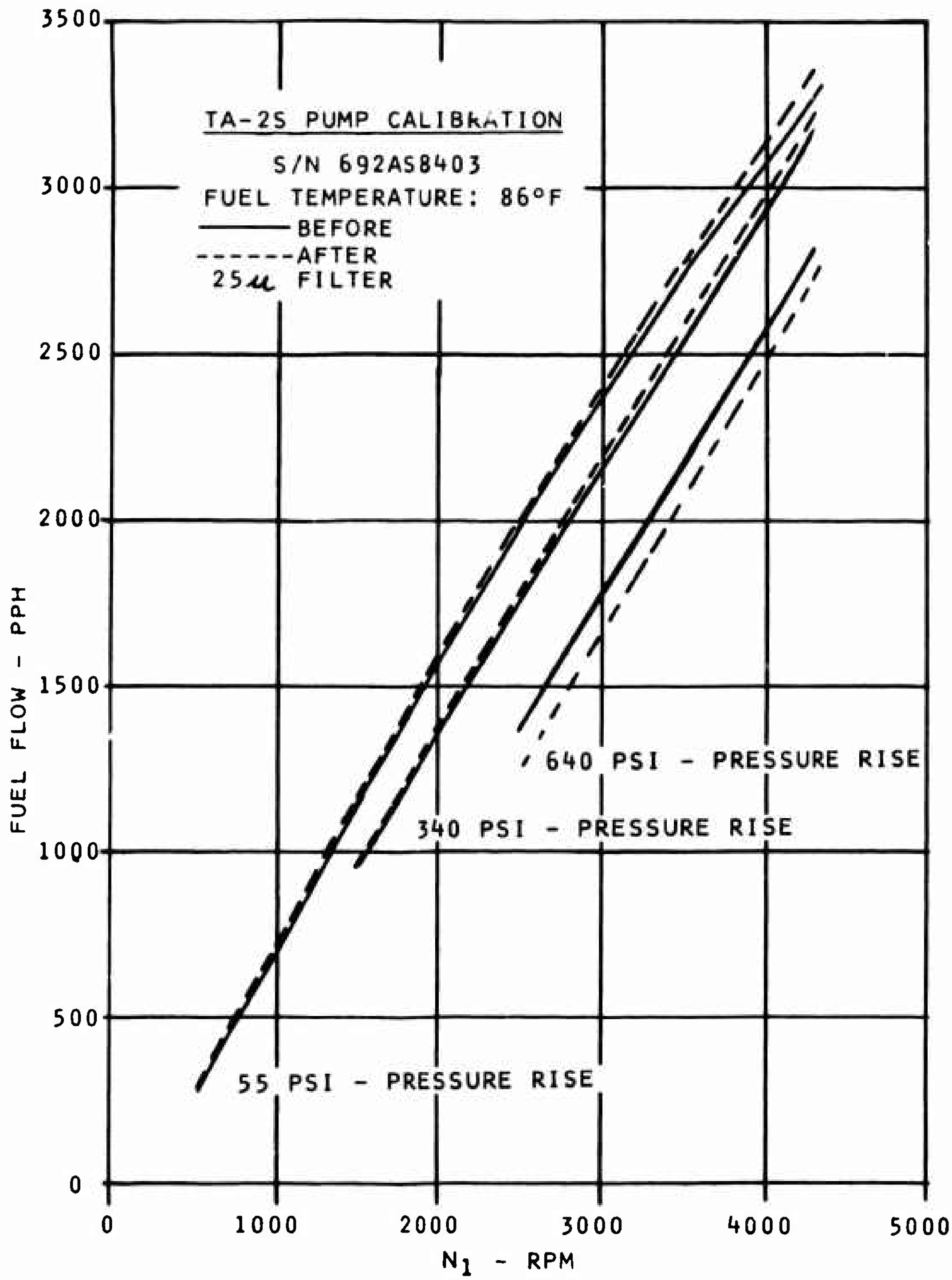


Figure 16. Pump Performance, S/N 692AS8403.

TABLE 4. DIMENSIONAL INSPECTION DATA-S/N: 692AS7472
3μ FILTER

QUAN	PART NO.	PART NAME	MEASUREMENTS IN.			AMOUNT OF WEAR IN.
				BEFORE TEST	AFTER TEST	
1	81196	MAIN METERING VALVE	O.D.	.31220 .31225	.31219 .31225	0.00001 -
1	71702	MAIN METERING SLEEVE	I.D.	.31261 .31264	.31261 .31264	- -
1	76250	R.P.M. SERVO VALVE	O.D.	.2496	.2496	-
1	76249	R.P.M. SERVO SLEEVE	I.D.	.25000 .25006	.25000 .25006	- -
1	76483	PLUNGER (MAIN METERING VALVE)	O.D.	.0925	.0925	-
1	76129	GUIDE - PLUNGER	I.D.	.09282 .09285	.09282 .09285	- -
1	74963	P_1 SERVO VALVE	O.D.	.24965 .24967	.24965 .24967	- -
1	71512	P_1 INNER SLEEVE	I.D. O.D.	.40617 .40662 .40590 .40595	.40617 .40622 .40593 .40596	- - 0.00003 0.00002
1	76131	GOVERNOR SERVO VALVE	O.D.	.24958 .2496	.2496	-
1	76133	GOVERNOR SERVO VALVE SLEEVE	I.D.	.24998 .25003	.25004 .25007	0.00006 0.00004
1	75988	P.T.G. GOVERNOR VALVE	O.D.	.2497	.2497	-
1	76236	P.T.G. GOVERNOR VALVE SLEEVE	I.D.	.25004 .25007	.25004 .25007	- -
1	82450	MAIN PRESSURE REGULATOR VALVE	O.D.	.31245 .31246	.31244 .31251	0.00001 0.00005
1	83399	MAIN PRESSURE REGULATOR SLEEVE	I.D.	.31280 .31282	.31280 .31282	- -

TABLE 5. DIMENSIONAL INSPECTION DATA-S/N: 692AS8403
25μ FILTER

QUAN	PART NO.	PART NAME	MEASUREMENTS IN.			AMOUNT OF WEAR IN.
				BEFORE TEST	AFTER TEST	
1	81196	MAIN METERING VALVE	O.D.	.31218 .31225	.31215 .31223	0.00003 0.00002
1	71702	MAIN METERING SLEEVE	I.D.	.31268	.31270	0.00002
1	76250	R.P.M. SERVO VALVE	O.D.	.24956 .24958	.24956 .2496	- 0.00002
1	76249	R.P.M. SERVO SLEEVE	I.D.	.25001 .25005	.25001 .25005	- -
1	76483	PLUNGER (MAIN METERING VALVE)	O.D.	.09253	.09253 .09254	- 0.00001
1	76129	GUIDE - PLUNGER	I.D.	.09280 .09282	.09280 .09282	- -
1	74963	P_1 SERVO VALVE	O.D.	.24957 .24964	.24957 .24961	- 0.00003
1	71512	P_1 INNER SLEEVE	I.D.	.40620 .40621	.40620 .40624	- 0.00003
			O.D.	.40586 .40590	.40586 .40590	- -
1	76131	GOVERNOR SERVO VALVE	O.D.	.24958 .2496	.24958 .24961	- -
1	76133	GOVERNOR SERVO VALVE SLEEVE	I.D.	.24995 .25005	.25000 .25005	0.00005 -
1	75988	P.T.G. GOVERNOR VALVE	O.D.	.2497	.24972 .24973	- -
1	76236	P.T.G. GOVERNOR VALVE SLEEVE	I.D.	.25001 .25007	.25001 .25007	- -
1	82450	MAIN PRESSURE REGULATOR VALVE	O.D.	.31245 .31246	.31245 .31248	- -
1	83399	MAIN PRESSURE REGULATOR SLEEVE	I.D.	.31277 .31280	.31277 .31282	- 0.00002

TABLE 6. DIMENSIONAL INSPECTION DATA
3μ FILTER
PRIMARY PUMP ELEMENT-S/N: 692AS7472

SHEET 1 OF 2

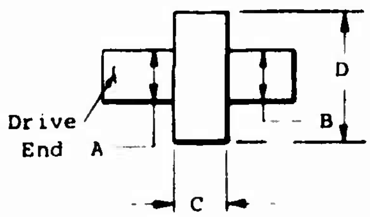
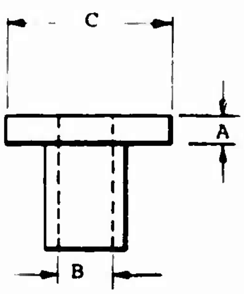
CONFIGURATION	PART	DIMENSION IN.			
			INITIAL	FINAL	WEAR
	DRIVE GEAR	A	.5008 .5009	.5008 .5008	- -
		B	.5006 .5007	.5006 .5007	- -
		C	.4503 .4503	.4502 .4502	- -
		D	1.0911	1.0911	-
	DRIVEN GEAR	A	.5007	.5007 .5008	- -
		B	.5006 .5008	.5006 .5008	- -
		C	.4503 .4504	.4503 .4503	- -
		D	1.0911	1.0911	-
	DRIVE BEARING (PUMP BODY)	A	.2985 .2995	.2984 .2995	.0001 -
		B	.5033 .5037	.5032 .5038	- .0001
		C	1.0927	1.0926 1.0927	.0001 -
	DRIVEN BEARING (PUMP BODY)	A	.2985 .2989	.2984 .2988	.0001 .0001
		B	.5032 .5035	.5032 .5035	- -
		C	1.0928	1.0927 1.0928	.0001 -

TABLE 6. CONTINUED

SHEET 2 OF 2

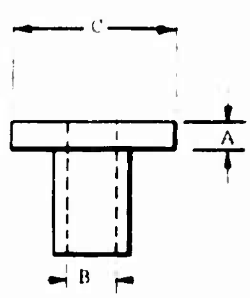
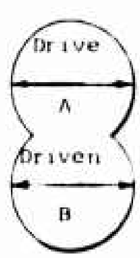
CONFIGURATION	PART	DIMENSION IN.			
			INITIAL	FINAL	WEAR
	DRIVE BEARING (PUMP COVER)	A	.2197	.2195 .2196	.0002 .0001
		B	.5032 .5037	.5031 .5040	- .0003
		C	1.0927 1.0928	1.0927 1.0928	- -
	DRIVEN BEARING (PUMP COVER)	A	.2200 .2204	.2195 .2200	.0005 .0004
		B	.5031 .5032	.5031 .5032	- -
		C	1.0929	1.0927 1.0928	.0002 .0001
	BODY BORE	A	1.0934 1.0935	1.0934 1.0935	- -
		B	1.0935	1.0934 1.0935	- -

TABLE 7. DIMENSIONAL INSPECTION DATA
3μ FILTER
SECONDARY PUMP ELEMENT-S/N: 692AS7472

SHEET 1 OF 2

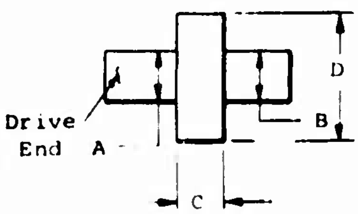
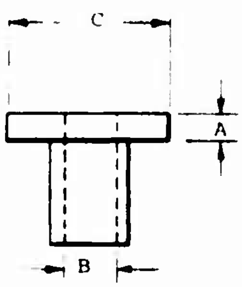
CONFIGURATION	PART	DIMENSION IN.	INITIAL	FINAL	WEAR
	DRIVE GEAR	A	.5006 .5007	.5006 .5007	- -
		B	.5007 .5008	.5007 .5008	- -
		C	.4503 .4503	.4503 .4502	- .0001
		D	1.0911	1.0911	-
	DRIVEN GEAR	A	.5006 .5007	.5006 .5007	- -
		B	.5005 .5007	.5006 .5007	- -
		C	.4503	.4503	-
		D	1.0911	1.0911	-
	DRIVE BEARING (PUMP BODY)	A	.2911 .2995	.2988 .2995	.0003 -
		B	.5031 .5033	.5032 .5035	.0001 .0002
		C	1.0927 1.0929	1.0927 1.0928	- .0001
	DRIVEN BEARING (PUMP BODY)	A	.2993 .2995	.2992 .2995	.0001 -
		B	.5031 .5033	.5031 .5033	- -
		C	1.0927 1.0929	1.0927 1.0928	- .0001

TABLE 7. CONTINUED

SHEET 2 OF 2

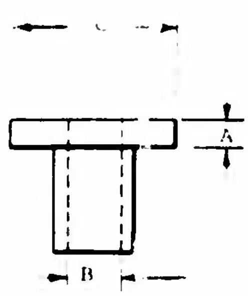
CONFIGURATION	PART		DIMENSION IN. INITIAL	FINAL	WEAR
	DRIVE BEARING (PUMP COVER)	A	.2193 .2203	.2195 .2200	- .0003
		B	.5031 .5032	.5029 .5033	- .0001
		C	1.0928 1.0930	1.0927 1.0928	.0001 .0002
	DRIVEN BEARING (PUMP COVER)	A	.2200 .2205	.2201 .2207	- -
		B	.5031 .5032	.5030 .5035	- .0003
		C	1.0928	1.0927 1.0928	.0001 -
	BODY BORE	A	1.0933 1.0937	1.0936 1.0937	.0003 -
		B	1.0935 1.0937	1.0937 1.0938	.0002 .0001

TABLE 8. DIMENSIONAL INSPECTION DATA
25μ FILTER
PRIMARY PUMP ELEMENT-S/N: 692AS8403

SHEET 1 OF 2

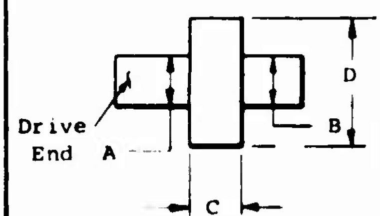
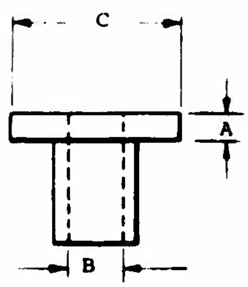
CONFIGURATION	PART	DIMENSION IN.			
			INITIAL	FINAL	WEAR
	DRIVE GEAR	A	.5006 .5008	.5006 .5008	- -
		B	.5007 .5008	.5007 .5009	- -
		C	.4503 .4504	.4503 .4504	- -
		D	1.0910	1.0910	-
	DRIVEN GEAR	A	.5005 .5006	.5005 .5006	- -
		B	.5005 .5006	.5005 .5006	- -
		C	.4503 .4504	.4503 .4503	- -
		D	1.0910	1.0910	-
	DRIVE BEARING (PUMP BODY)	A	.2995 .3000	.2986 .3000	.0009 -
		B	.5033 .5041	.5033 .5040	- -
		C	1.0929 1.0930	1.0927 1.0928	.0002 .0002
	DRIVEN BEARING (PUMP BODY)	A	.2993 .3000	.2993 .3000	- -
		B	.5033 .5038	.5034 .5038	.0001 -
		C	1.0929 1.0930	1.0928 1.0929	- -

TABLE 8. CONTINUED

SHEET 2 OF 2

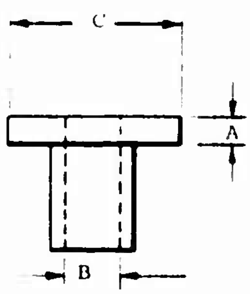
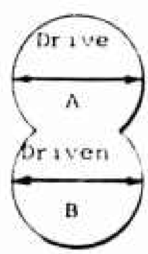
CONFIGURATION	PART		DIMENSION IN.		
			INITIAL	FINAL	WEAR
	DRIVE BEARING (PUMP COVER)	A	.2193 .2195	.2193 .2195	- -
		B	.5035 .5041	.5034 .5037	- -
		C	1.0926 1.0927	1.0925 1.0926	.0001 .0001
	DRIVEN BEARING (PUMP COVER)	A	.2190 .2195	.2190 .2195	- -
		B	.5035 .5037	.5035 .5040	- .0003
		C	1.0927 1.0928	1.0927 1.0928	- -
	BODY BORE	A	1.0933 1.0935	1.0933 1.0935	- -
		B	1.0933 1.0935	1.0932 1.0935	- -

TABLE 9. DIMENSIONAL INSPECTION DATA
25μ FILTER
SECONDARY PUMP ELEMENT-S/N: 692AS8403

SHEET 1 OF 2

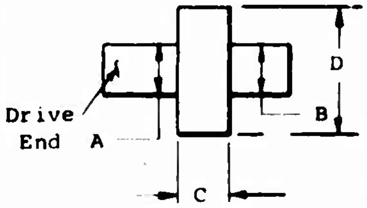
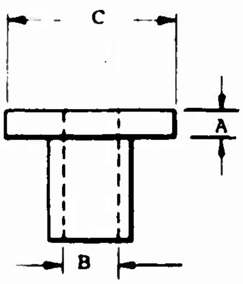
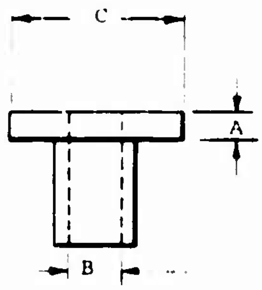
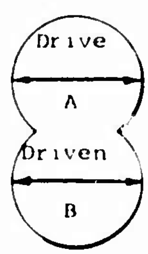
CONFIGURATION	PART	DIMENSION IN.			
			INITIAL	FINAL	WEAR
	DRIVE GEAR	A	.5007 .5007	.5006 .5008	.0001 -
		B	.5006 .5007	.5006 .5007	- -
		C	.4503 .4503	.4503 .4503	- -
		D	1.0912	1.0912	-
	DRIVEN GEAR	A	.5006 .5007	.5006 .5007	- -
		B	.5006 .5007	.5006 .5007	- -
		C	.4503	.4503	-
		D	1.0912	1.0912	-
	DRIVE BEARING (PUMP BODY)	A	.2993 .3000	.2990 .2995	.0003 .0005
		B	.5033 .5039	.5033 .5041	- .0002
		C	1.0929 1.0930	1.0928 1.0929	.0001 .0001
	DRIVEN BEARING (PUMP BODY)	A	.2995 .3000	.2995 .3000	- -
		B	.5031 .5036	.5033 .5038	.0002 .0002
		C	1.0928 1.0929	1.0927 1.0928	.0001 .0001

TABLE 9. CONTINUED

SHEET 2 OF 2

CONFIGURATION	PART	DIMENSION IN.			
			INITIAL	FINAL	WEAR
	DRIVE BEARING (PUMP COVER)	A	.2195 .2201	.2190 .2200	.0005 .0001
		B	.5033 .5038	.5032 .5038	- -
		C	1.0928	1.0927 1.0928	.0001 -
	DRIVEN BEARING (PUMP COVER)	A	.2195 .2205	.2195 .2203	- .0002
		B	.5032 .5037	.5031 .5038	- .0001
		C	1.0929 1.0930	1.0929 1.0930	- -
	BODY BORE	A	1.0934 1.0935	1.0934 1.0936	- .0001
		B	1.0933 1.0935	1.0934 1.0936	.0001 .0001

In addition to the dimensional inspections, visual inspection of all parts of the controls did not show any significant signs of contaminant. The only visible sign of contaminant was found on the flyweight of the power turbine governor of the control used in the 25μ filter test.

Conclusion

The results of the calibration testing for both the controls and pumps did not reveal any indications of performance degradation that could be attributed to the contamination test. Moreover, there were no discernible differences in performance due to the levels of filtration.

FUEL SEAL TEST

The procurement of fuel seals for testing was initiated by contacting seal manufacturers and informing them of the test plan for environmental evaluation of rotary and translational fuel seals. Each was asked to recommend a seal and describe its properties so that we could evaluate and select at least nine seals for environmental testing. Of the 22 manufacturers contacted, eight did not reply, eight indicated that they did not have a seal that would meet the requirements, and six manufacturers made recommendations which offered for testing a total of ten different seals, six rotary and four translational. All of these six manufacturers provided enough seal samples so that six of each type of the ten types of seals could be tested. In actuality, three of these ten types of seals were of the same material and were selected for evaluation in both the rotary and translational configuration.

The material composition of the various seals that were tested was considered proprietary information by the manufacturers. Therefore, the seals are identified only by the commercial designation.

At the completion of testing these ten types of seals, a program extension was initiated primarily to test one of the more successful rotary seals as a translational seal. Four additional rotary seals were included in this program extension for an overall total of 15 configurations of seals.

DESCRIPTION OF TEST

Figure 17 depicts the test setup used to conduct the seal testing. Test fixture T-7971 provided the device for testing six sample rotary seals on three different shafts. Two of these fixtures were used in the test setup allowing 12 rotary seal samples to be tested simultaneously. In addition, test fixture T-8222 provided the means for testing four translational seals. All three fixtures mounted in the same environmental box.

A complete test cycle consisted of 30,000 cycles at room temperature, 30,000 cycles at -65°F ambient air and fuel temperatures, 30,000 cycles at 250°F ambient air and fuel temperatures,

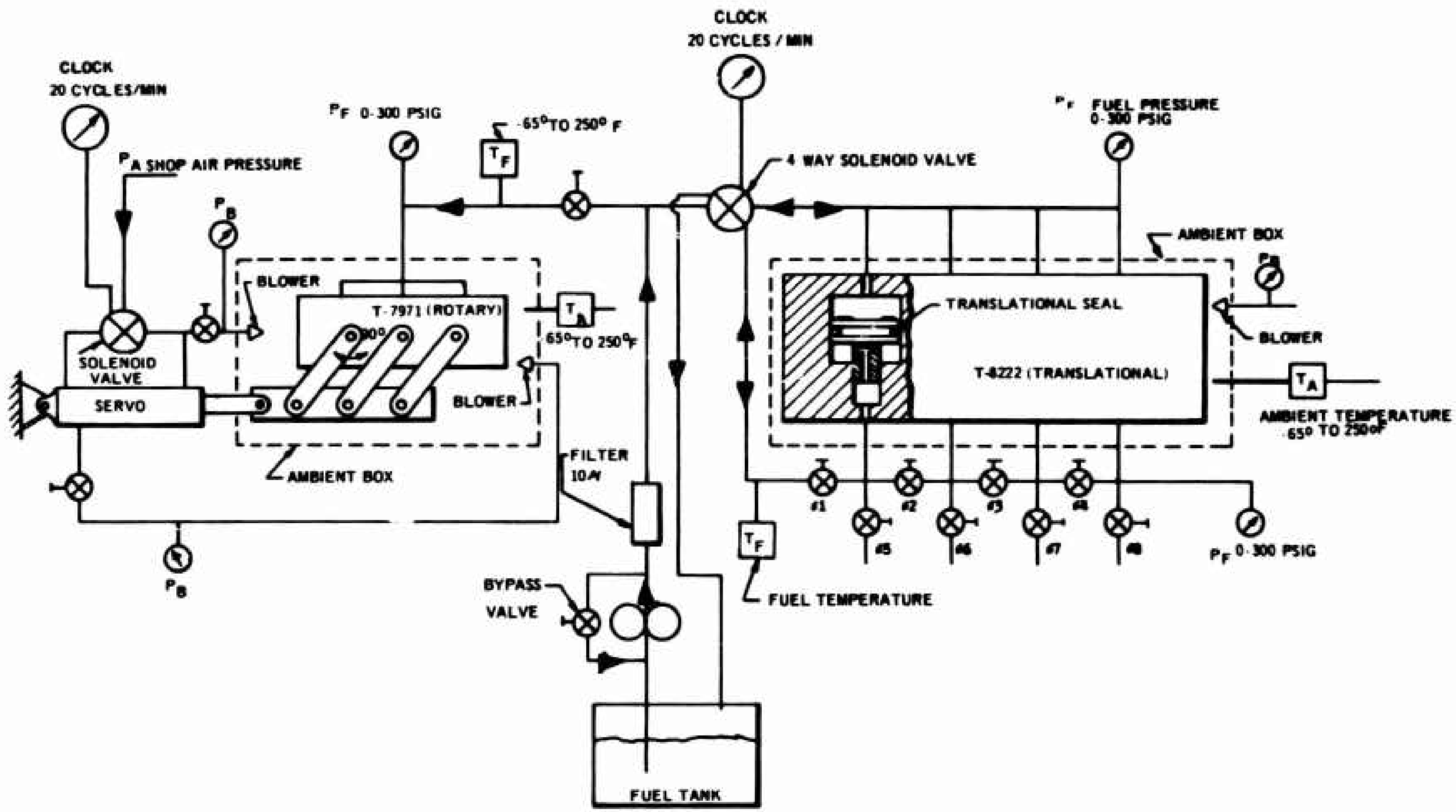


Figure 17. Seal Test Schematic.

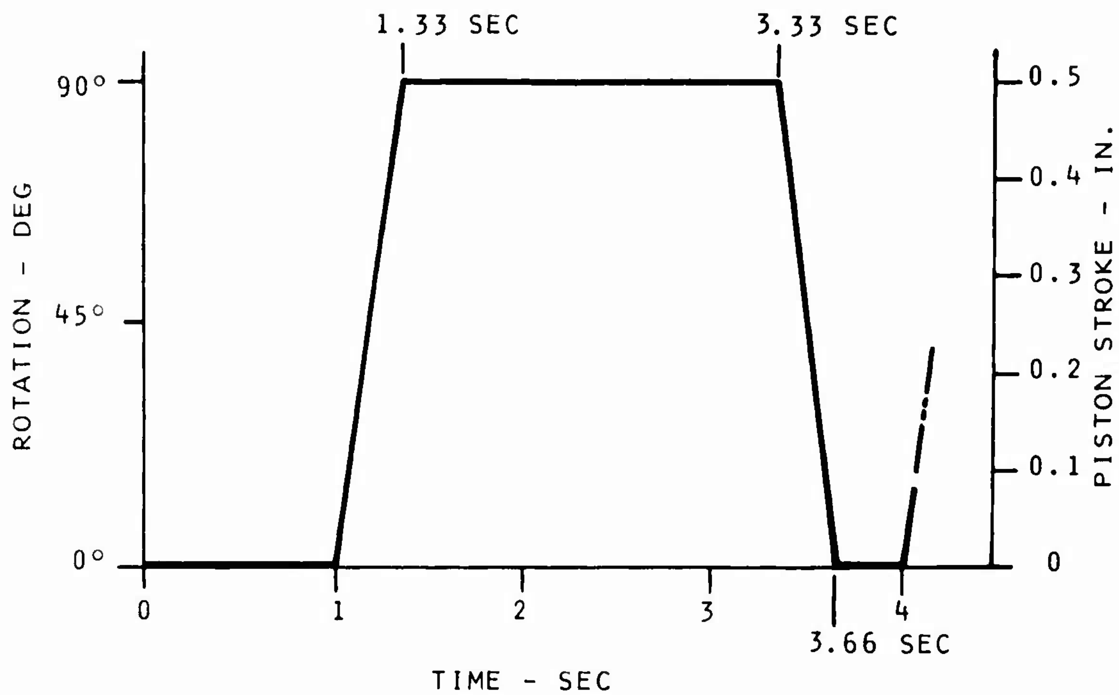


Figure 18. Seal Test Cycle.

and 10,000 cycles at room temperature in a sand and dust environment. The mechanical cycle is illustrated in Figure 18 and corresponds to a surface velocity of 6 ft/min., which is typical of actual operating velocities. The fuel pressure supply to the fixtures was held at 300 psig ± 10 psig. Fuel was MIL-J-5624, grade JP-4.

Room temperature testing was done with the test fixtures at 70°F ± 10°F, and fuel temperature was maintained at 70°F ± 10°F.

Hot temperature testing was achieved by blowing hot air into the environmental box, thereby heating the test fixtures as well as the fuel in contact with the seals. Fuel flow was calculated to be approximately 10 gal./hr. in the translational test fixture. No fuel flow was required for the rotational fixtures. Because of the low fuel flow rate, it was not necessary to heat the fuel with a heat exchanger. Fuel and ambient temperatures were maintained at 250°F. Figure 19 shows the 250°F test setup.

Cold temperature testing was achieved by cooling the fuel to -65°F in heat exchangers located in an alcohol bath. Carbon dioxide was blown into the alcohol bath and into the environmental box to maintain the ambient and fuel temperatures at -65°F. Figure 20 shows the environmental box at -65°F.

Sand and dust environment testing was achieved by closing the environmental box hermetically after introducing enough sand and dust to maintain a minimum concentration of 1.5 ± 0.4 gram of sand and dust per cubic foot of box volume. The mixture of air and sand and dust was kept in continuous motion by blowing air intermittently inside the box. Figure 21 shows the sand and dust test setup. The sand and dust mixture consisted of 90 percent silicon dioxide, was angular in structure and screened as follows:

100	mesh	screen	U.S.	standard	sieve	series	passing	100	percent
140	"	"	"	"	"	"	"	98±2	"
200	"	"	"	"	"	"	"	90±2	"
325	"	"	"	"	"	"	"	75±2	"

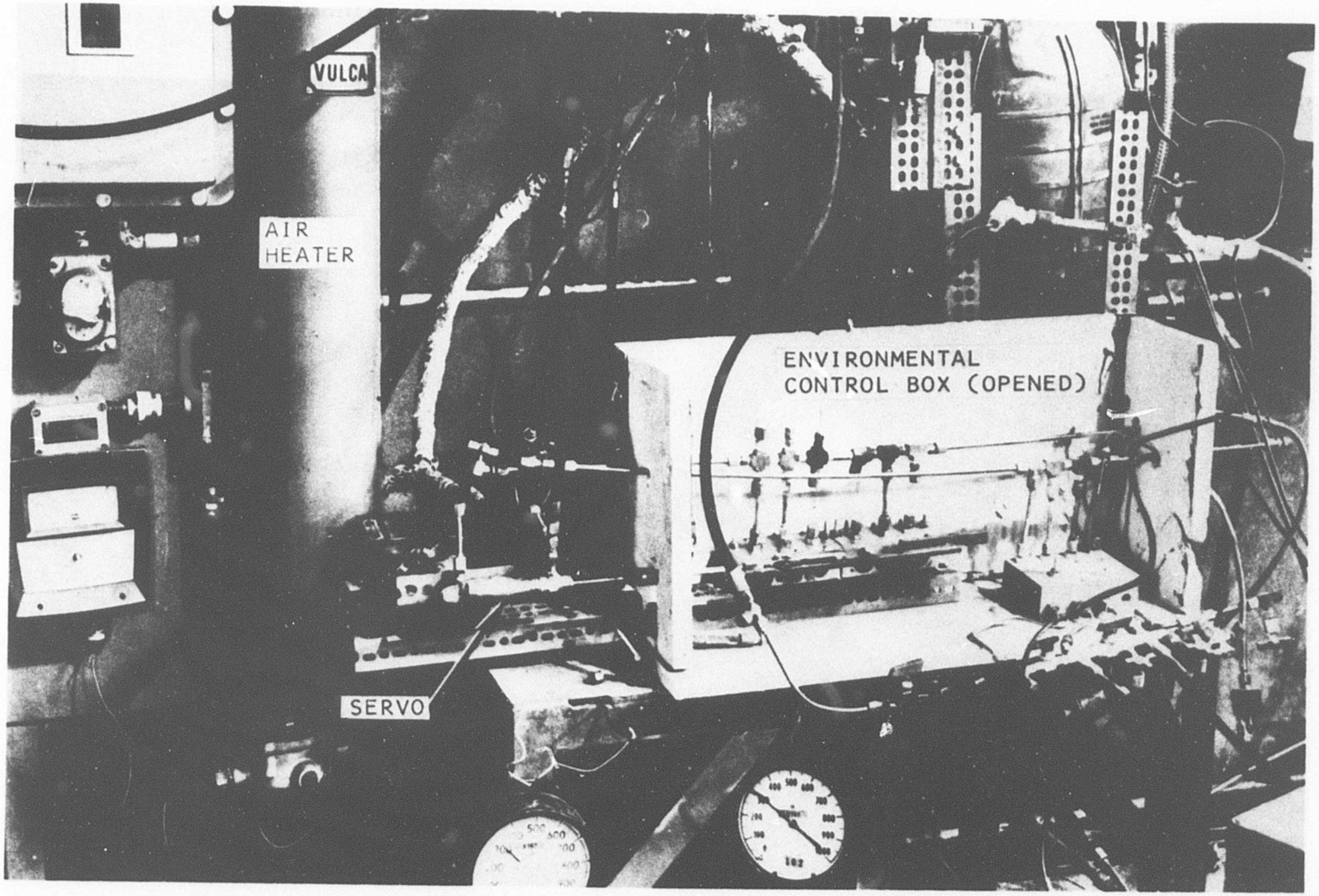


Figure 19. Seal Test Setup for 250°F Test.

Figure 20. Seal Test Environmental Box at -65°F.

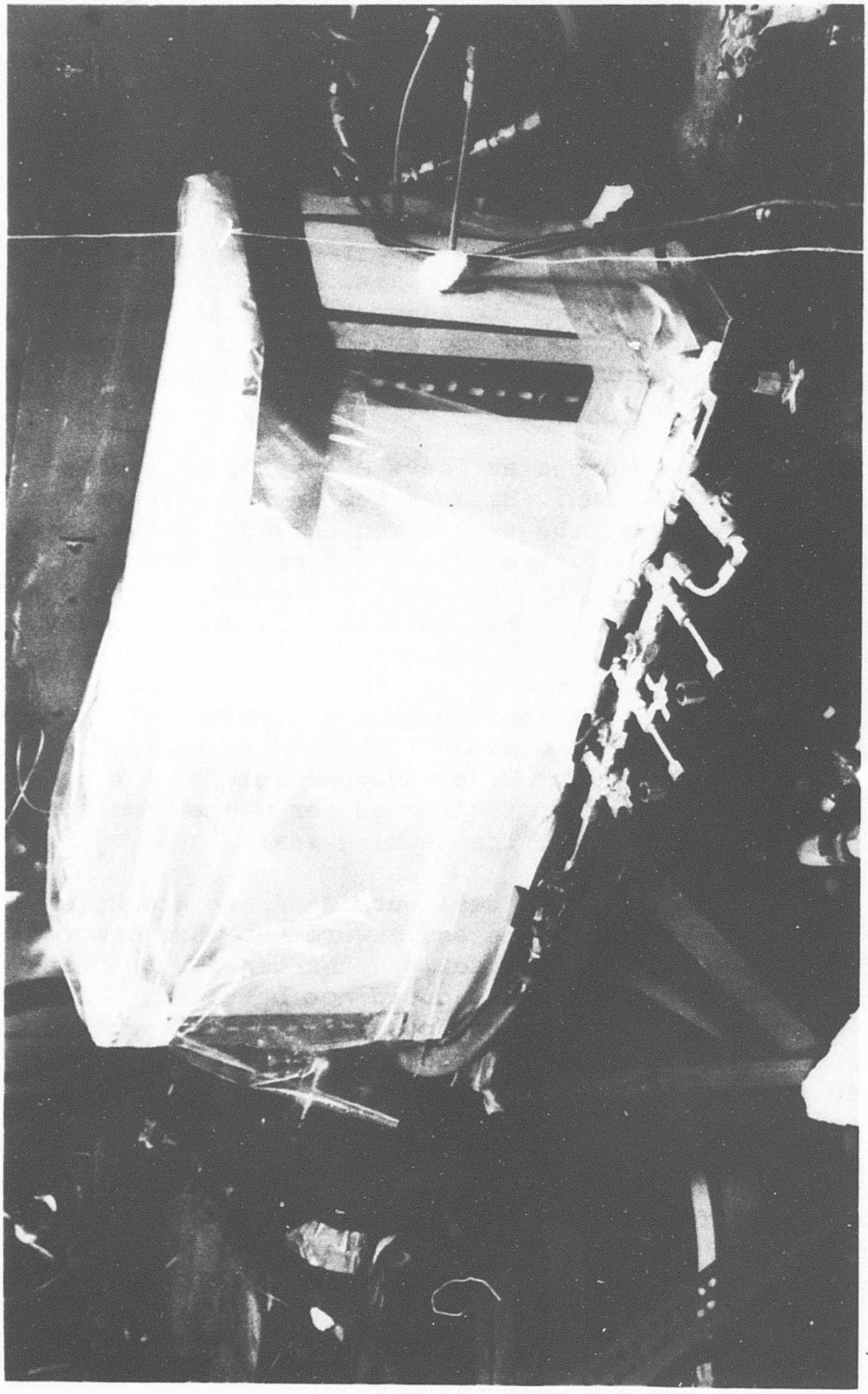


Figure 21. Seal Test Setup for Sand and Dust Environment Test.

Inspection

One seal from each batch received from the manufacturers was inspected. The outside diameter of the seal was measured with a shadowgraph instrument. A microphotograph of that seal was taken showing the surface which would most likely wear during the test. Durometer hardness was measured as well as the specific gravity of the seal. Each seal tested was weighed before and after the test. In some cases, however, the nature of the seal precluded its complete inspection. For example, the combination seals tend to be too out-of-round for outside and inside diameter measurement and too thin for hardness measurement.

Test seal leakage was recorded at least every 10,000 cycles during all phases of testing. Because testing was stopped at least once every 24 hours, the test fixtures and the fuel were not kept at constant temperature during hot or cold testing while testing was stopped. Leakage was, therefore, checked at room temperature as well as at the test temperature. Rotary seal leakage was observed but not measured. Paper-towel-wetting was judged to be sufficient cause for rejection of a leaking seal. Translational seal leakage was measured in drops per minute by pressurizing one side of the piston and collecting fuel dripping at the other side independently of the other pistons. Leakage in excess of five drops per minute was sufficient cause for rejection of the leaking seal.

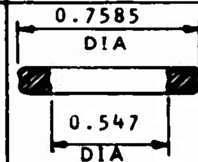
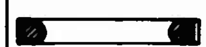
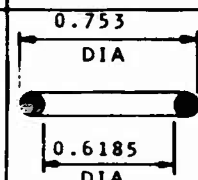
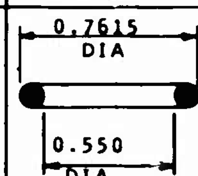
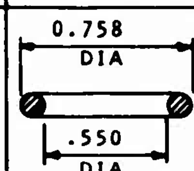
Upon completion of testing, all seal outside diameters were measured as before testing. Some seals were cut when removed because of the tendency of those seals to harden during testing. In these cases, measurements could not be taken. All seals were weighed, checked for hardness, and their specific gravity was measured. One tested seal of each type was microphotographed side by side with a new one. Figures 22 through 36 show these seals magnified 30 times.

The surface finish of the shaft and piston grooves and of the fixture bores used for testing conformed to the surface finish recommended by seal manufacturers and specified in MIL-P-5514, General Requirements for Hydraulic Packings Gland Design.

TABLE 10. ROTARY SEAL PHYSICAL PROPERTIES

MATERIAL	CONFIGURATION	SEAL NO.	CYCLES (1000'S)	WEIGHT CHANGE %	SPEC. GRAV. CHANGE %	HARDNESS CHANGE %	OUTSIDE DIA. CHANGE %	INSIDE DIA. CHANGE %	THICKNESS CHANGE %
POLYURETHANE A-1086	0.5125 DIA; 0.364 DIA	1	0.6	-0.24	+1.7	-1.3	-1.95	+0.8	+5.4
		2	30.5	ALL SEALS FAILED AT HOT TEST					
		3	30.5						
		4	0.8	AND FELL APART AT [illegible]ASSEMBLY					
		5	0.8						
		6	0.6	NO MEASUREMENTS POSSIBLE					
VITON 523A Q4113	0.757 DIA; 0.5475 DIA	1	6.5	-5	+0.77	+ 7.7	-1.3	-1.8	-0.49
		2	20	-9.1	+1.54	+10.7	-	-	-1.47
		3	11	-5.2	+1.54	+10.7	-4.1	-3.9	+0.49
		4	6.5	-3.3	+0.77	+ 6.1	-1.3	-0.5	+0.49
		5	4.6	-5.5	+0.77	+ 4.6	-1.7	-1.2	+0.49
		6	15.5	-9.5	+2.3	+18.0	-4.3	-5.7	-2.5
TETRALON 720 AND FLUORO-SILICONE O-RING		1	20	-0.17	-2.4	-	-	-	+1.1
		2	20	-0.93	-1.5	-	-	-	0
		3	20	-0.43	-1.0	-	-	-	+9.1
		4	20	-11.7	0	-	-	-	+14.8
BUNA XN 1996-2 WITH ULTRA LUBE	0.502 DIA; 0.366 DIA	1	100	-10	+2.3	+ 6.8	-1.5	-0.5	-2.2
		2	100	-17.5	+3.1	+ 5.9	-	-	-2.2
		3	92	-30	+3	+16.5	-	-	-13.5
		4	92	-31	+1.5	+13.7	-	-	-10.8
		5	100	-22.4	+2.3	+ 9.6	-3	+0.2	-13.7
		6	100	-20	+3	+ 2.7	-2.4	-1.1	-6.5
		7	91	-11	+1.5	+ 9.6	-2.4	+0.2	-3.6
BUNA N 756-75 WITH ULTRA LUBE	0.4975 DIA; 0.356 DIA	1	100	-16	+4.1	+15	-2.3	+0.5	-7 0
		2	100	-14	+4.1	+17.5	-	-	-8.5
		3	100	-14	+4.1	+12	-	-	-2.8
		4	10	-17	+3.3	+ 9.5	-1.5	-0.3	-2.8
		5	100	-15	+9.9	+ 5.4	-	-	-8.[illegible]
		6	100	-14	+9.1	+ 4.1	-3.1	+1.0	-7 0
		7	100	-15.5	+5.8	+15	-	-	-3.5
		8	90	-16.5	+5.0	+1[illegible]	-	-	-4.2
NITRILE C-69	0.5045 DIA; 0.3605 DIA	1	100	-16	+4.6	+15	-	-	-3.5
		2	100	-15	+4.1	+19	-5.2	-1.0	-2.1
		3	[illegible]	-15.8	+4.9	+15	-5.9	-1.0	-3.5
		4	100	-16.7	+4.9	+12	-5.6	-1.0	-3.5
		5	100	-16.3	+4.0	+16	-4.3	-1.0	-3.5
		6	100	-13.3	+4.9	+12	-3.9	-1.0	-0.7
TETRALON 729 AND BUNA N9021 O-RING		1	48	- 1.8	+ 0.5	-	-	-	+ 5.7
		2	48	-	+ 0.5	-	-	-	+ 1.1
		3	36	- 3.2	+ 2.8	-	-	-	+ 5.7
		4	48	- 1.5	+ 2.5	-	-	-	+ 6.9
BUNA N 9021	0.500 DIA; 0.360 DIA	1	100	-32.1	+11.0	+15.4	-14.0	-3.1	-25.0
		2	100	-13.4	+ 7.7	+ 1.5	- 4.8	-2.5	- 2.9
		3	100	-12.0	+ 8.3	+ 7.7	- 5.0	-0.7	- 1.4
		4	100	-14.0	+10.1	+21.5	- 7 7	-4.5	- 2.9
		5	30	-40.0	+10.9	+15.4	-	-	- 4.3
		6	100	-21.0	+ 9.4	+15.4	- 3.9	-2.5	+ 8.7
BUNA N 9021 IN TFE-COATED GROOVE	0.566 DIA; 0.427 DIA	1	100	-20.6	+ 1.6	0	- 3.9	-1.6	- 7.3
		2	74	NO MEASUREMENTS POSSIBLE					
		3	36	- 5.9	+ 0.8	0	- 4.0	-3.5	+ 1.4
		4	100	-15.4	+ 1.6	0	- 4.0	-2.3	+ 1.4
		5	100	-14.4	+ 5.0	0	- 3.5	-1.8	- 1.4
		6	100	-20.0	+ 5.0	0	- 6.5	-2.6	- 1.4
BUNA N 9021 WITH BACK UP RING MS 28774	0.500 DIA; 0.360 DIA	1	20	- 5.0	+ 8.3	+ 3.0	- 1.9	- 2.2	- 1.4
		2	100	-11.3	+10.0	+ 7.7	- 5.3	- 4.6	0
		3	100	-11.7	+11.2	+13.8	- 6.0	-4.7	- 1.4
		4	32	- 8.6	+10.1	+ 6.1	- 2.2	-2.2	- 2.8
		5	32	- 4.0	+ 8.6	0	-10.8	-1.3	- 1.4
		6	20	NOT AVAILABLE					

TABLE 11. TRANSLATIONAL SEAL PHYSICAL PROPERTIES

MATERIAL	CONFIGURATION	SEAL NO.	CYCLES (1000'S)	WEIGHT CHANGE %	SPEC. GRAV. CHANGE %	HARDNESS CHANGE %	OUTSIDE DIA CHANGE %	INSIDE DIA CHANGE %	THICKNESS CHANGE %
VITON 523A Q4113	 0.7585 DIA; 0.547 DIA	1	7	- 1.7	0	+ 3	-0.46	0	0
		2	24	-10.7	+2.3	+23	-5.5	- 4.9	-3.9
		3	30	- 9.5	+3.0	+20	-5.0	-13.0	-3.9
		4	30	- 7.9	+2.3	+17	-2.6	0	-2.0
		5	10	-10.4	+3.0	+21.5	-4.1	- 3.1	-2.9
		6	80	-10.4	+3.0	+23	-4.7	- 3.8	-2.9
TETRALON 720 AND FLUORO-SILICONE O-RING		1	50	- 1.6	-0.5	-	-	-	+ 7.8
		2	12	- 1.2	-0.5	-	-	-	+12.0
		3	100*	- 2.8	-0.5	-	-	-	+ 7.8
		4	10	- 0.5	+1.0	-	-	-	+ 2.1
		5	8.7	- 1.2	+1.0	-	-	-	+ 5.0
		6	90	- 7.9	0	-	-	-	+ 6.4
BUNA N 756-75 + ULTRA LUBE	 0.753 DIA; 0.6185 DIA	1	100	-11.1	+5	+15.7	-6.6	- 2.8	-2.1
		2	7	- 1.3	+2.5	+ 5.7	+1.3	- 1.7	-0.7
		3	100*	-10.5	+5.8	+17.1	-4.5	- 4	-0.7
		4	43.7	-15.7	+5.8	+14.3	-4.4	- 3.8	-3.5
		5	100*	-12.3	+3.3	+18.5	-	-	-3.5
		6	100	-10.5	+4.1	+18.5	-	-	-3.5
VITON V-135	 0.7615 DIA; 0.550 DIA	1	100	+0.59	+1.6	+ 2.9	+3.3	+ 0.5	0
		2	100	-0.1	-1.1	+ 2.9	+0.6	+ 0.2	0
		3	24	+0.52	-1.6	+ 1.5	+3.3	- 0.3	0
		4	20	+0.37	-1.1	0	+3.3	- 0.5	0
		5	7	-0.33	0	+ 2.9	+4.6	- 1.1	0
		6	100	+0.7	-2.2	+ 2.9	-0.6	- 0.3	-1.0
NITRILE COMPOUND C-69	 0.758 DIA; .550 DIA	1	100	-13.2	+4.0	+ 6.1	-5.0	- 4.0	-3.9
		2	100	-14.0	+4.8	+20.0	-5.1	- 4.0	-3.9
		3	100	-13.0	+4.8	+18.5	-5.3	- 4.3	-3.9
		4	100	-13.6	+4.8	+20.0	-6.1	- 4.1	-2.9
		5	100	-13.2	+4.8	+20.0	-6.2	- 7.5	-2.9
		6	100	-13.2	+4.8	+20.0	-6.1	- 4.0	-3.9

*SLIGHT LEAKAGE AT END OF TEST

TABLE 12 R&M TEST AND EVALUATION ROTARY SEAL PROGRAM

SEAL TYPE		TESTING COMPLETED				COMMENTS
CONFIGURATION	MATERIAL	HOT (250°F)	COLD (-65°F)	ROOM TEMP.	DUST	
O-RING	BUNA XN 1996-2 + ULTRA LUBE					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE SEAL CUT - POSSIBLE HUMAN FAILURE REMOVED BECAUSE OF FAILURE OF COMPLEMENTARY SEAL TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE REMOVED BECAUSE OF FAILURE OF COMPLEMENTARY SEAL
O-RING	BUNA N 756-75 + ULTRA LUBE					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE SEAL CUT - POSSIBLE HUMAN FAILURE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE FAILED AT ROOM TEMPERATURE
O-RING	POLYURETHANE COMPOUND A-1086					ALL SIX SEALS FAILED AT HOT TEST AFTER FEWER THAN 1000 CYCLES EACH
QUAD RING 	VITON COMPOUND 523A Q4113					FAILED AT COLD TEST FAILED AT COLD TEST FAILED AT HOT TEST REMOVED BECAUSE OF FAILURE OF COMPLEMENTARY SEAL FAILED AT COLD TEST FAILED AT HOT TEST
	TETRALON 720 + FLUORO-SILICONE 'O' RING					FAILED AT COLD TEST FAILED AT COLD TEST FAILED AT HOT TEST FAILED AT HOT TEST
O-RING	NITRILE COMPOUND C 69					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE
O-RING	BUNA N 9021					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE FAILED AT ROOM TEMPERATURE TEST COMPLETED - NO LEAKAGE
O-RING & BACKUP RING 	BUNA N 9021 & TFE BACKUP RING					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE FAILED AT -65°F FAILED AT -65°F FAILED AT -65°F FAILED AT -65°F
TEFLON COATED 	BUNA N 9021 & TFE COATED GROOVE					TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE FAILED AT ROOM TEMPERATURE FAILED AT ROOM TEMPERATURE TEST COMPLETED - NO LEAKAGE TEST COMPLETED - NO LEAKAGE
	TETRALON 720 + BUNA N 9021 O-RING 2000CYCLES →					FAILED AT -65°F FAILED AT -65°F FAILED AT -65°F FAILED AT ROOM TEMPERATURE

TABLE 13. R&M TEST AND EVALUATION TRANSLATIONAL SEAL PROGRAM

SEAL TYPE		TESTING COMPLETED				COMMENTS
CONFIGURATION	MATERIAL	HOT (250°F)	COLD (-65°F)	ROOM TEMP.	DUST	
O-RING	BUNA N 756-75 + ULTRA LUBE					TEST COMPLETED - NO LEAKAGE
						FAILED AT COLD TEST
						FAILED AT COLD TEST
						TEST COMPLETED - 4D/MIN LEAKAGE
						TEST COMPLETED - 2D/MIN LEAKAGE
						TEST COMPLETED - NO LEAKAGE
O-RING	FLUOROCARBON COMPOUND VITON V-135					TEST COMPLETED - NO LEAKAGE
						FAILED AT COLD TEST
						FAILED AT HOT TEST
						FAILED AT COLD TEST
						TEST COMPLETED - NO LEAKAGE
						TEST COMPLETED - NO LEAKAGE
QUAD RING 	VITON COMPOUND 523A Q4113					FAILED AT HOT TEST
						FAILED AT HOT TEST
						FAILED AT ROOM TEMPERATURE TEST
						FAILED AT HOT TEST
						FAILED AT COLD TEST
						FAILED AT COLD TEST
	TETRALON 720 + FLUORO SILICONE 'O' RING					FAILED AT HOT TEST
						FAILED AT HOT TEST
						FAILED AT HOT TEST
						FAILED AT DUST TEST
						TEST COMPLETED 1D/MIN
						FAILED AT COLD TEST
O-RING	NITRILE COMPOUND C-69					TEST COMPLETED - NO LEAKAGE
						TEST COMPLETED - NO LEAKAGE
						TEST COMPLETED - NO LEAKAGE
						TEST COMPLETED - NO LEAKAGE
						TEST COMPLETED - NO LEAKAGE
	2000 CYCLES					TEST COMPLETED - NO LEAKAGE

TEST RESULT ANALYSIS

Tables 10 and 11 list the physical properties of each type of seal as well as the changes that occurred in those properties during cycling. Tables 12 and 13 list the various types of seals and the performance of each seal.

In general, all seals experienced loss of weight due to wear and drying up. However, this weight loss was not proportional to the number of cycles. The specific gravity of most seals increased, indicative of shrinkage and/or loss of lighter components of the seal material. This is also accompanied by an increase in the hardness of the seals. It should be noted that Tetralon seals were not tested for hardness because of their thin wall, nor were their diameters measured because they are stretched out of round at disassembly. The diameters of the other seals, both outside and inside, showed a decrease in dimension, except for one translational seal, Viton V-135, which appeared to be fairly stable. Thickness of the seals decreased except for the Tetralon seal, which increased due to the flowing characteristic of the material.

Rotary Seals

The first seal listed in Table 10 is a polyurethane compound A-1086. All six seals of that type failed at hot test. Two of them lasted 30,000 cycles at -65°F but failed after only 600 cycles at 250°F. All seals except one showed a severe breakdown of the material at disassembly. Figure 22 shows a new and a used seal side-by-side, magnified 30 times. Although not revealed by the photo, the used seal was considerably flattened at the outer circumference after only 600 cycles, which is the reason it leaked.

The next type of seal is a Viton, compound 523A Q4113. All seals experienced excessive wear as shown by Figure 23. It should be noted that the quad ring mounts in the bore and therefore seals on the inside rather than outside diameter. Minute circumferential wear lines are visible after only 15,500 cycles, indicating the susceptibility of the seal material to tear and wear. It should be noted that most of these wear lines are located where the seal could have been caught in the clearance between the rotating shaft and the fixed sleeve. The decrease in weight of all six seals appears to

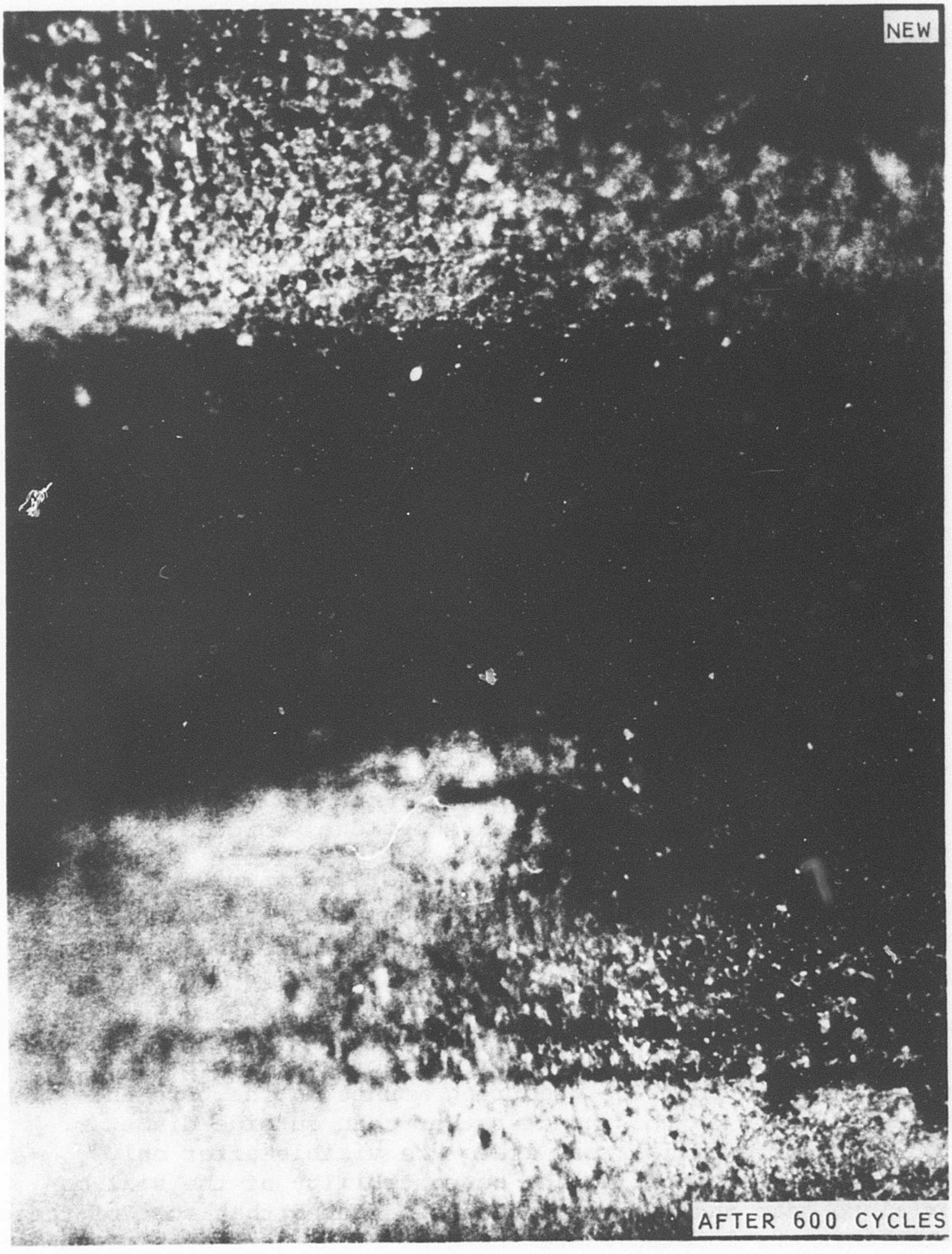


Figure 22. Rotary Seal Polyurethane A-1086.

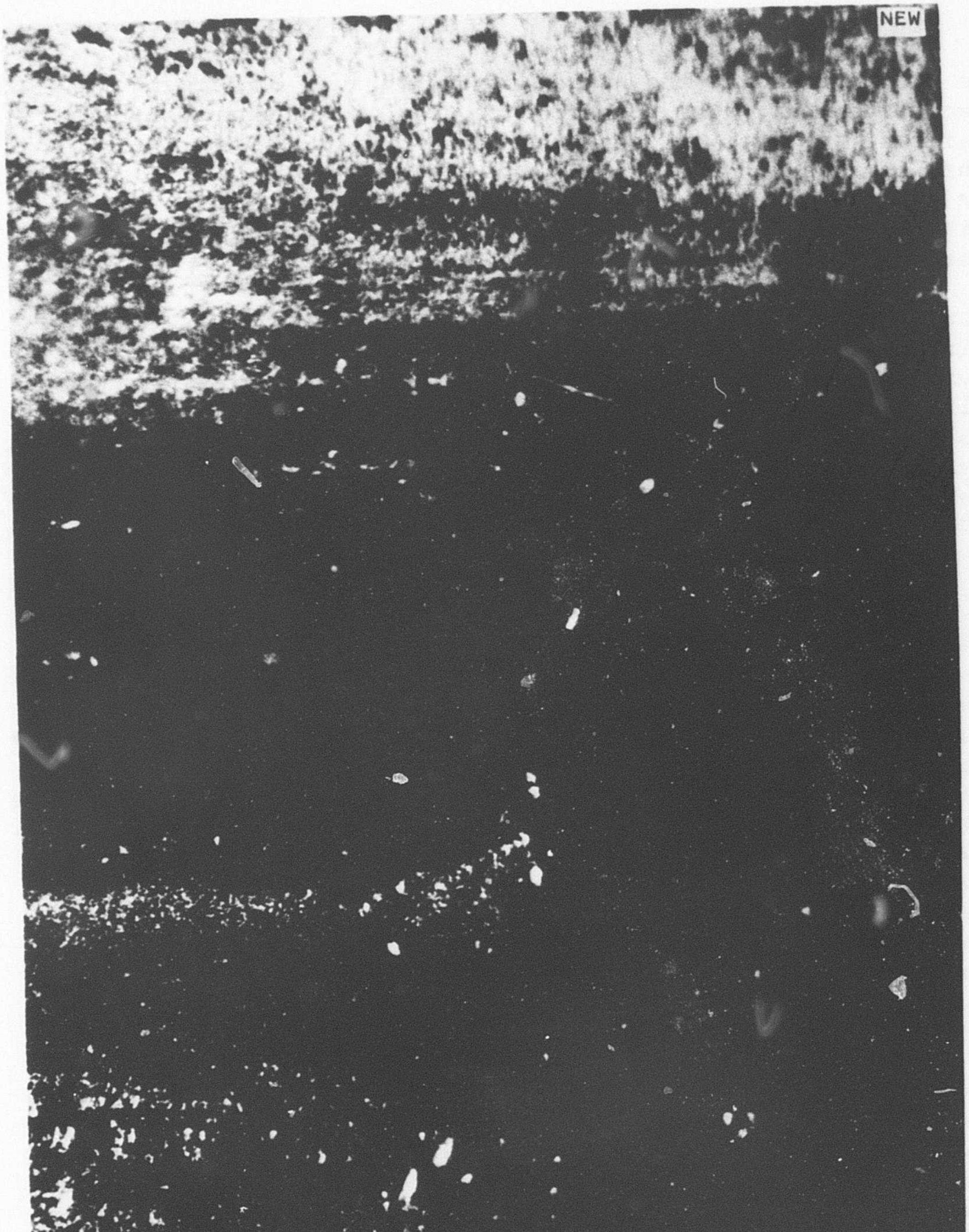


Figure 23. Rotary Seal Viton 523A Q4113.

be proportional to the number of cycles run, confirming the excessive wear. The specific gravity did not change by a significant amount. All seals hardened during cycling; however, this increase in hardness is unrelated to the number of cycles or the type of test. Both inside and outside diameters of five seals showed a decrease due to shrinkage but the thickness did not vary appreciably.

Four Tetralon 720 seals were run 20,000 cycles each. The first two were assembled with a smaller size O-ring than recommended by the manufacturer because of difficulties in assembling the seal on the test shaft. After leakage developed at cold test, the manufacturer suggested that a correct size O-ring be used as originally specified. The next two seals were assembled with the specified O-ring and failed after 20,000 cycles at 250°F. At disassembly, the fluorosilicone O-ring showed signs of being crushed, pointing to excessive squeeze. Unfortunately, because of the thin wall of the Tetralon ring, its hardness could not be measured. It is expected that the hardness would be much higher than that of rubber O-rings. Consequently, leakage becomes a function of the finish of the seal and that of the sleeve bore. Weight change for three seals was insignificant. The fourth seal lost 11.7 percent of its weight at the same time, increasing its width by 14.8 percent. This increase in width indicates an excessive squeeze caused by the O-ring which acts as a mechanical spring. The specific gravity tended to increase rather than decrease, pointing to the absorption characteristic of TFE. Diameters were not measured. Figure 24 shows a new seal next to a seal which ran for 20,000 cycles. The left side of the photo is out of focus, which is the reason for the blur. There is little evidence of wear; however, the surface of the used seal seemed to be somewhat smoother than that of the new one.

Four additional Tetralon 720 seals were tested during the seal test program extension. These seals were assembled on the shaft with a C-69 O-ring of the size called for by the manufacturer. However, it was impossible to insert the test shaft into its sleeve without damaging the Tetralon ring. A smaller size Buna N 9021 ring was tried because the C-69 seal was not available in the smaller size. Three seals failed at -65°F and one failed at room temperature. In all four cases, failure appeared to be caused by damage to the O-ring. A typical side view of a damaged O-ring is shown in Figure 25. The photo

Figure 24. Rotary Seal Tetralon 720.

Figure 25. Rotary Seal Buna N 9021 Used With Tetralon 720.

indicates that the corner of the Tetralon ring dug into the O-ring, leaving a step as well as minute lacerations. The configuration of that seal is such that the pressure forces the O-ring in the space left between the Tetralon ring and the bottom of the groove. Possibly there is a relative motion between the O-ring and the Tetralon ring causing the lacerations.

Only four Tetralon seals were tested during each phase because the other seals supplied by the manufacturer were damaged at assembly.

Four out of seven Buna XN 1996-2 seals completed 100,000 cycles. The decrease in weight varied between 10 and 31 percent, indicating substantial wear not visible on the photograph, Figure 26. This wear was confirmed when measuring the change in thickness. Specific gravity varied ± 3 percent. This variation did not indicate a trend leading to a conclusion. Hardness increased up to 16.5 percent but the increase is not constant from one seal to another. Three of the seven seals were cut when removed from the test shaft groove, indicating a loss of tensile strength which, when coupled with the increase in hardness, could lead to cracks while in operation, and eventually to leakage. This Buna XN 1996-2 seal was treated with an ultralube process designed to reduce friction and ultimately reduce wear. There are no indications that this ultralube process accomplished the goal it was intended to do since weight loss was 10 to 31 percent. All seven seals of this type completed 89,000 cycles or better and yet the changes in mechanical characteristics of all seals are spread over a wide range, pointing to an unstable material requiring further refinement. Figure 26 shows a new seal next to one that completed 100,000 cycles without leakage. A slight cut is visible on the side of the test seal, on the right side of the photo. This sliver, 0.002-inch wide, was most likely cut when the test pressure forced the seal in the clearance between the test shaft and the sleeve. This clearance is, by design, 0.0005 to 0.0035 inch. In addition, the photo shows the used seal outside diameter to reflect the light better than the new one, due to a smoother surface resulting from wear.

The Buna N 756-75 seal (shown in Figure 27), contrary to the preceding Buna XN 1996-2 seal, did not exhibit circumference wear (as can be seen in the photograph), although its weight

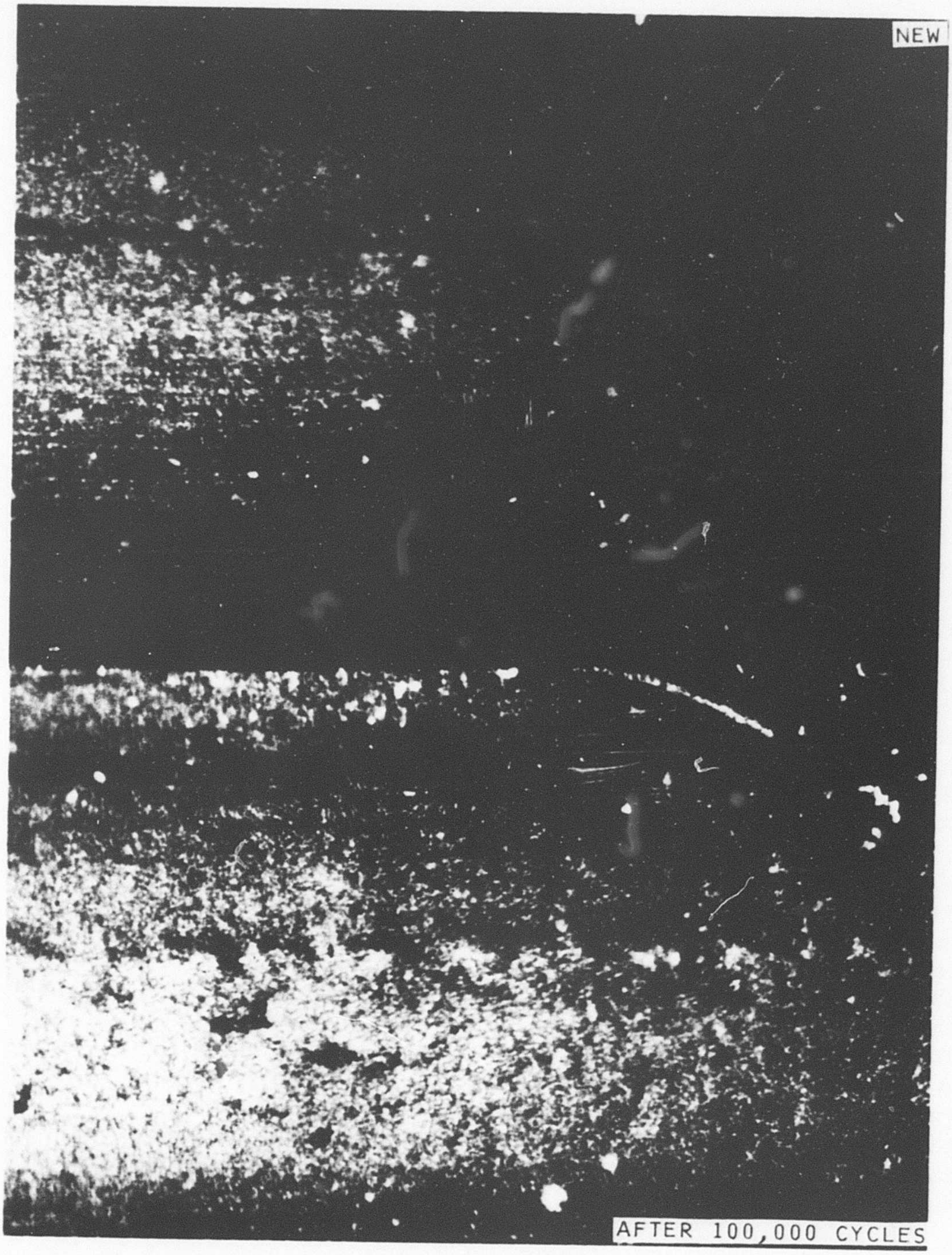


Figure 26. Rotary Seal Buna XN 1996-2.

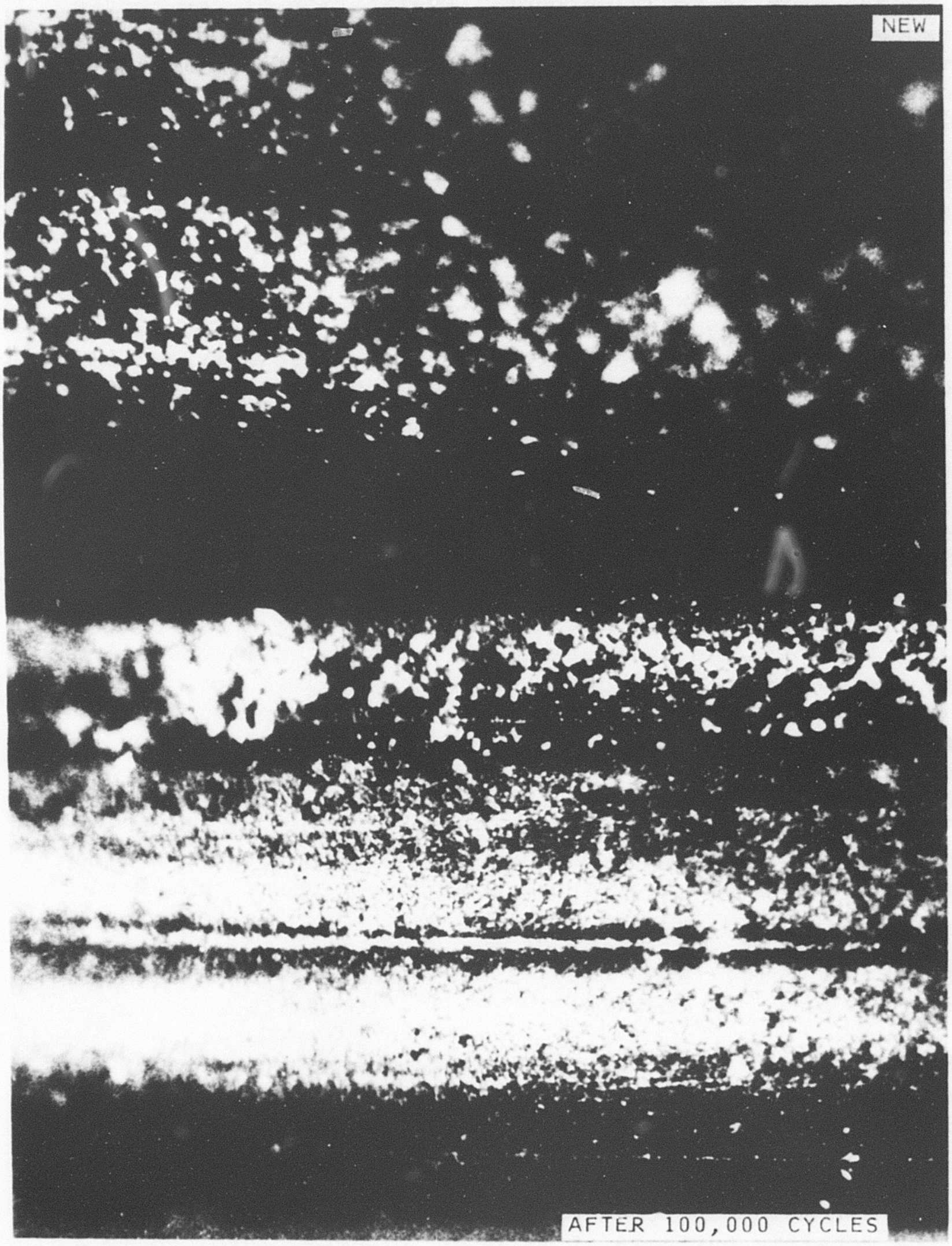


Figure 27. Rotary Seal Buna N 756-75.

loss ranged from 14 to 17 percent. Examination of the tested seal shows the mold flash still visible after 100,000 cycles; however, shiny surfaces on either side of that flash indicate a smoother surface than the new seal, revealing some wear. Specific gravity increased from 3.3 to 9.9 percent, pointing to some material instability also confirmed by the range of hardness increase from 4.1 to 19 percent. Six out of eight seals completed 100,000 cycles. It should be noted that one seal completed only 8700 cycles because of a possible erroneous assembly; it also experienced the highest weight loss of all the seals. This would indicate that weight loss occurs during the early stage of testing. Five seals were cut at disassembly, thereby preventing diameter measurements and also indicating loss of tensile strength as with the Buna XN 1996-2 seal.

The Nitrile C-69 seal is shown in Figure 28. The circumferential lines visible in the new seal are worn away in the 100,000 cycle seal; however, this wear is minimal because the mold flash of the tested seal is still visible. All six seals tested completed 100,000 cycles each without leaking. The changes in properties are not too different from the two Buna N seals; however, the changes are more consistent, indicating a more stable material, thus a better seal. This seal is qualified to MIL-P-5315. Table 14 lists the properties of the C-69 seal as well as the specified limits. Two properties stand out as being better in the C-69 seal: First, the tensile strength is highest of the group at 1740 psi. This explains why only one C-69 seal was cut at disassembly while being pulled out of its groove. Obviously, the higher the tensile strength of the seal, the better the protection of that seal will be against tearing and possibly pinching. Second, the compression set is the lowest of the group at 7 percent at 158°F for 70 hours. Note that 30,000 cycles at 250°F corresponds to 25 hours of testing.

It should be noted that MIL-P-5315 specifies a compression set between 0 and 50 percent. However, each compound must be held within ± 5 percentage points of the value at which it was qualified. For example, C-69 compound compression set must be held between 2 and 12 percent. Ideally, a seal with zero compression set maintains contact under all temperature conditions with the metallic surfaces it is being squeezed into, while a seal with a high compression set leaves a gap under cold

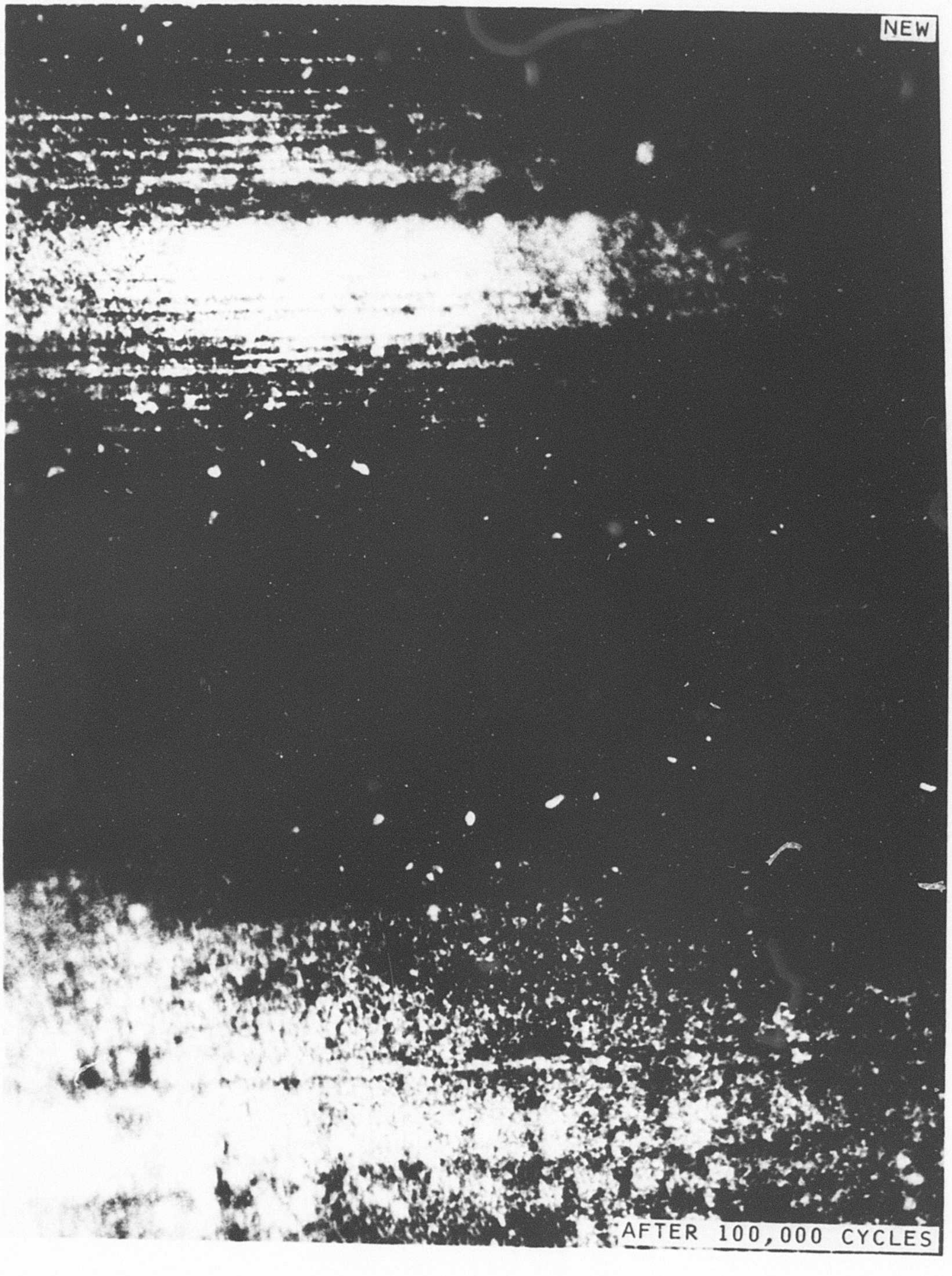


Figure 28. Rotary Seal Nitrile C-69.

TABLE 14. QUALIFICATION VALUES OF THE PHYSICAL PROPERTIES OF MIL-P-5315 COMPOUNDS

COMPOUND NO.	TENSILE STRESS AT 100% ELONGATION PSI	TENSILE STRENGTH PSI	ULTIMATE ELONGATION PERCENT	SPECIFIC GRAVITY POINTS	VOLUME CHANGE PERCENT		HARDNESS DUROMETER	COMPRESSION SET PERCENT	TEMPERATURE RETRACTION °F
					TYPE 1 FLUID	TYPE III FLUID			
SPEC LIMITS	AS DETERMINED	1000 MIN	200 MIN	AS DETERMINED	0, +10	0, +50	60-70	0-50	-50 OR COOLER
C-69	520	1740	210	1.27	+3.4	+35	68	7	-51.0
V-135	870	1720	184	1.85			78	6.1-11.8*	-10.0

* FIRST VALUE IS AT ROOM TEMPERATURE, THE SECOND VALUE IS AT 392°F

temperature after being subjected to high temperature. This is caused by the expansion of the seal material with high temperature causing a set to a different shape and subsequent shrinkage at low temperature resulting in a gap. The C-69 seal has demonstrated its superiority over most of the other seals tested, due partly to this characteristic.

Three Buna N 9021 O-rings were tested in three different configurations. Buna N 9021 is presently used in Chandler Evans MC-40 fuel control. The first configuration is a standard O-ring. Five out of six samples completed 100,000 cycles, and one failed after 30,000 cycles of room temperature testing. When removed from the fixture, the failed O-ring was found disintegrated and separated into two complete rings, which explains the high weight loss listed in Table 10. The other samples also experienced weight loss ranging from 12 to 32.1 percent. One sample which lost 32.1 percent weight also had a reduction in thickness of 25 percent, indicating excessive side wear, even though it completed 100,000 cycles without failure. Specific gravity of all six samples increased between 7.7 and 11 percent, indicating the loss of lighter components of the material. Hardness also increased somewhat in relation to the specific gravity increase. Figure 29 shows a new seal next to a seal tested for 100,000 cycles. The latter exhibits a darker area representative of a compression set groove caused by the test pressure forcing the seal against the groove wall. Note that the mold flash is still visible in both seals.

The second Buna N 9021 configuration is an O-ring assembled in a TFE-coated groove. Four out of six samples completed 100,000 cycles. One failed after 74,000 cycles and was so badly damaged before disassembly that no measurements were possible. Another failed after 36,000 cycles with a weight loss of 5.9 percent. The other four samples experienced weight losses of 14.4 to 20.6 percent, and one lost 7.3 percent of its thickness while the other three retained their thickness. The significant fact is that none of the samples examined increased its hardness and at the same time none increased its specific gravity by more than 5 percent. It can be theorized that the groove TFE coating transferred from the groove to the O-ring, thereby preventing the loss of lighter components as in other cases. The tested seal shown in Figure 30 exhibits the same darker area as the one shown in Figure 29. The object of a

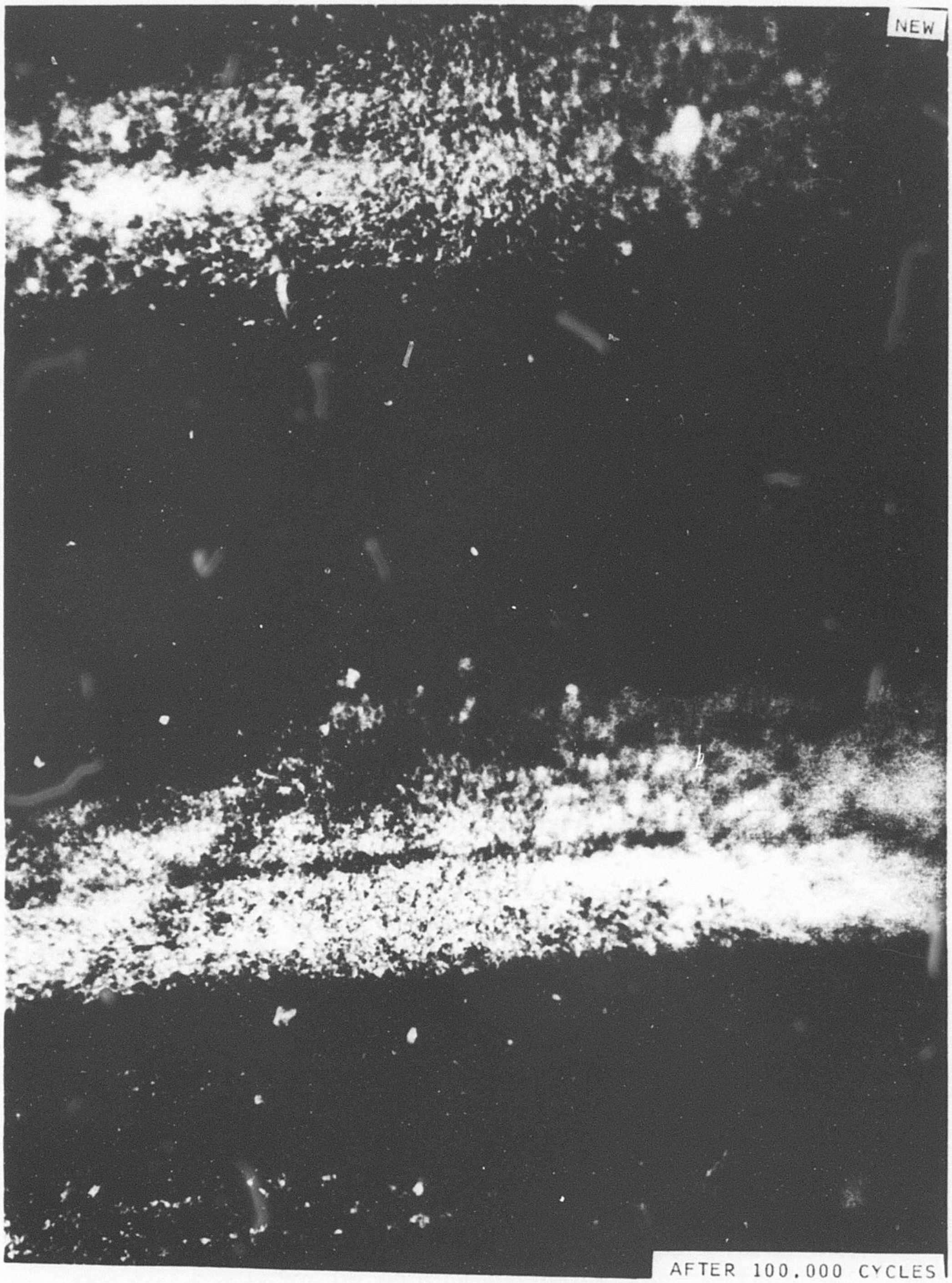


Figure 29. Rotary Seal Buna N 9021.

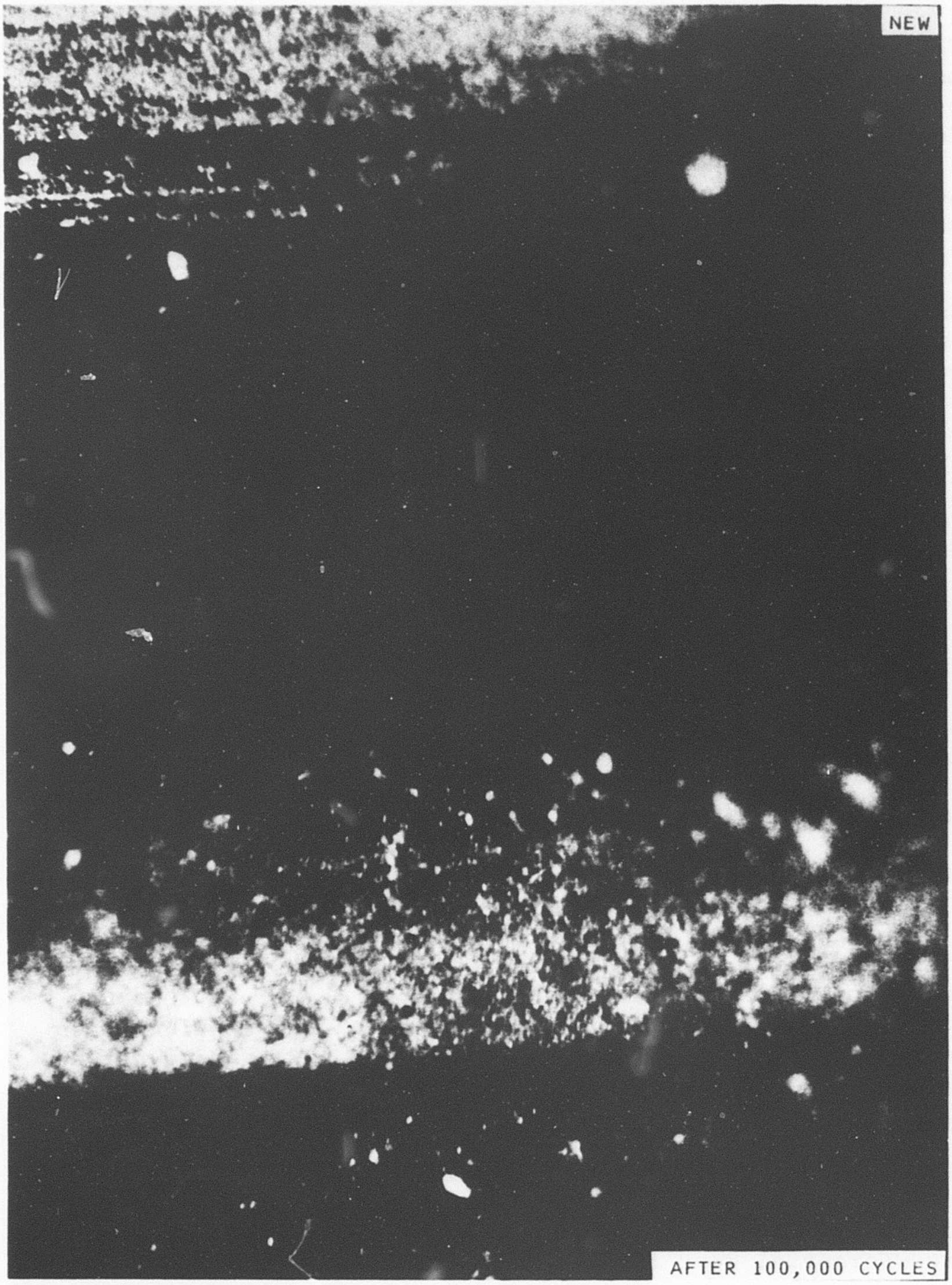


Figure 30. Rotary Seal Buna N 9021 in TFE-Coated Groove.

TFE-coated groove was to eliminate rubbing between the O-ring and the sleeve bore and to have rubbing occur instead between the O-ring and the shaft groove. However, this resulted in additional wear, although it is not clearly shown in the photo.

The third Buna N 9021 configuration consists of an O-ring assembled with a TFE backup ring. Two samples completed 100,000 cycles, two failed after 32,000 cycles, and two failed after 20,000 cycles. Weight loss ranges from 4 to 11.7 percent, and is lower than that of the two previous Buna N 9021 configurations. The increase in specific gravity and in hardness is similar to that of the Buna N O-ring used without TFE coating. The change in thickness is insignificant. Test results indicate that the TFE backup ring did not help in any way the performance of the O-ring. Figure 31 shows the side of a tested O-ring. Minute flakes of material were gouged from the O-ring, and were found stuck to other sections of the O-ring. It should be noted that the backup ring is split at an angle of 22° to permit assembly in the groove. If this split is responsible for removing the particles from the O-ring, relative motion between the O-ring and the split ring had to take place.

It appears that of the three Buna N 9021 configurations tested, the best one is the standard O-ring.

Translational Seals

The Viton 523A Q4113 is shown in Figure 32. Considerable wear is evident. This seal is a four-lobe configuration seal where only the top of the lobes is in contact with the sleeve bore. Axial wear lines are visible on the tested seal lobe shown in the photo. None of the six seals tested completed 100,000 cycles. This seal is of the same material and configuration as the rotary listed in Table 10. Weight loss varied between 7.9 and 10.7 percent except for one seal, which failed after 7000 cycles. This weight loss is not related to the number of cycles run. The specific gravity increased as well as the hardness. The diameters, both inside and outside, decreased, thereby showing some shrinkage. The thickness of most seals decreased. Failure of all seals was due to the excessive circumferential wear coupled with some increase in hardness causing surface cracks.

Figure 31. Rotary Seal Buna N 9021 With Backup Ring.

Figure 32. Translational Seal Viton 523A Q4113.

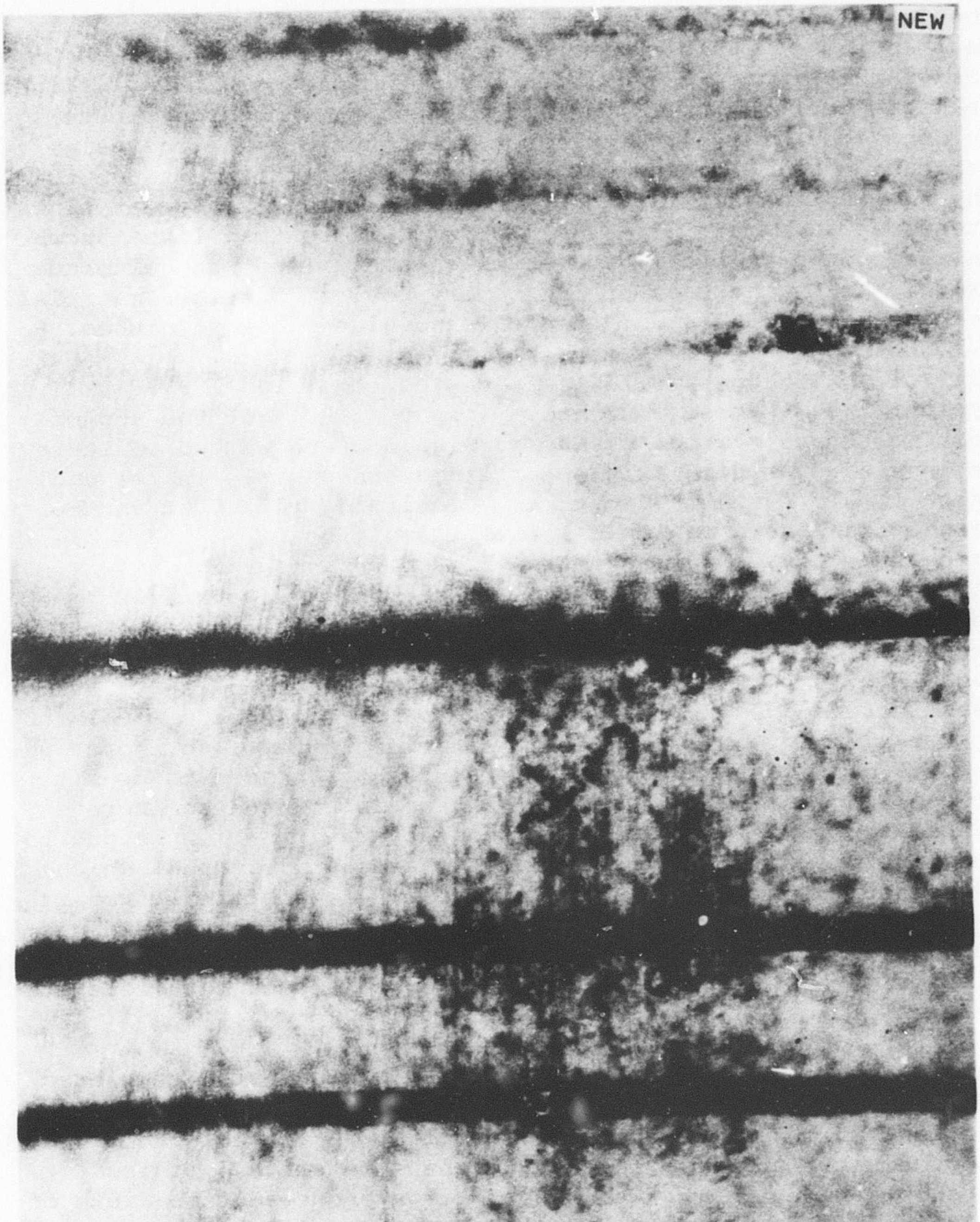


Figure 33. Translational Seal Tetralon 720.

The Tetralon 720 seals are of the same material as the rotary seal listed in Table 10 but of a different configuration. Only one of the six seals completed 100,000 cycles and then a slight leakage occurred toward the end of testing (one drop/minute). Hardness and diameters could not be measured. The only significant physical change is the increase in thickness due to the flowing characteristic of that material. Axial wear lines are shown in Figure 33. The dark circumferential lines shown on the tested seal are grooves which are filled with aluminum oxide. As in the case of the rotary seal of the same material, leakage is a function of the seal and sleeve bore finishes.

Four of six Buna N 756-75 seals tested completed 100,000 cycles; two of these four had a slight leakage at the end of testing (2 and 4 drops/minute). Figure 34 shows two of these seals side-by-side. Axial wear lines are visible on the seal which completed 100,000 cycles. Confirming this wear is the weight loss shown in Table 11. It should be noted that, although the tested seal shown in Figure 34 appears to have flattened out, none of the seals tested took a compression set. This seal was treated with an ultralube process designed to reduce friction and wear. As in the case of the rotary seal Buna XN 1996-2, this process failed to reduce the wear. Hardness increased as a function of the number of cycles completed to a maximum of 18.5 percent. Specific gravity also increased up to 5.8 percent and physical measurements tended to decrease, indicating shrinkage of the seals. Hardness increase is related to the increase in specific gravity which is due to compression set. This loss leads to a denser material which is harder to compress. It is interesting to note that rotary seals of the same material were fairly successful and did not show any wear, as can be seen in Figure 27. This leads to the conclusion that rolling motion of the seal tends to accelerate wear.

Three of six Viton V-135 translational seals tested completed 100,000 cycles without leakage. Figure 35 shows no wear at the seal which completed 100,000 cycles. Both seals, new and used, exhibit the mold flash, and the same surface finish. It should be noted that this seal has a cross-section width of 0.100 inch compared to 0.070 inch for most of the other seals. Table 11 shows that this seal experienced little weight change. Most of the physical changes were insignificant, indicating a

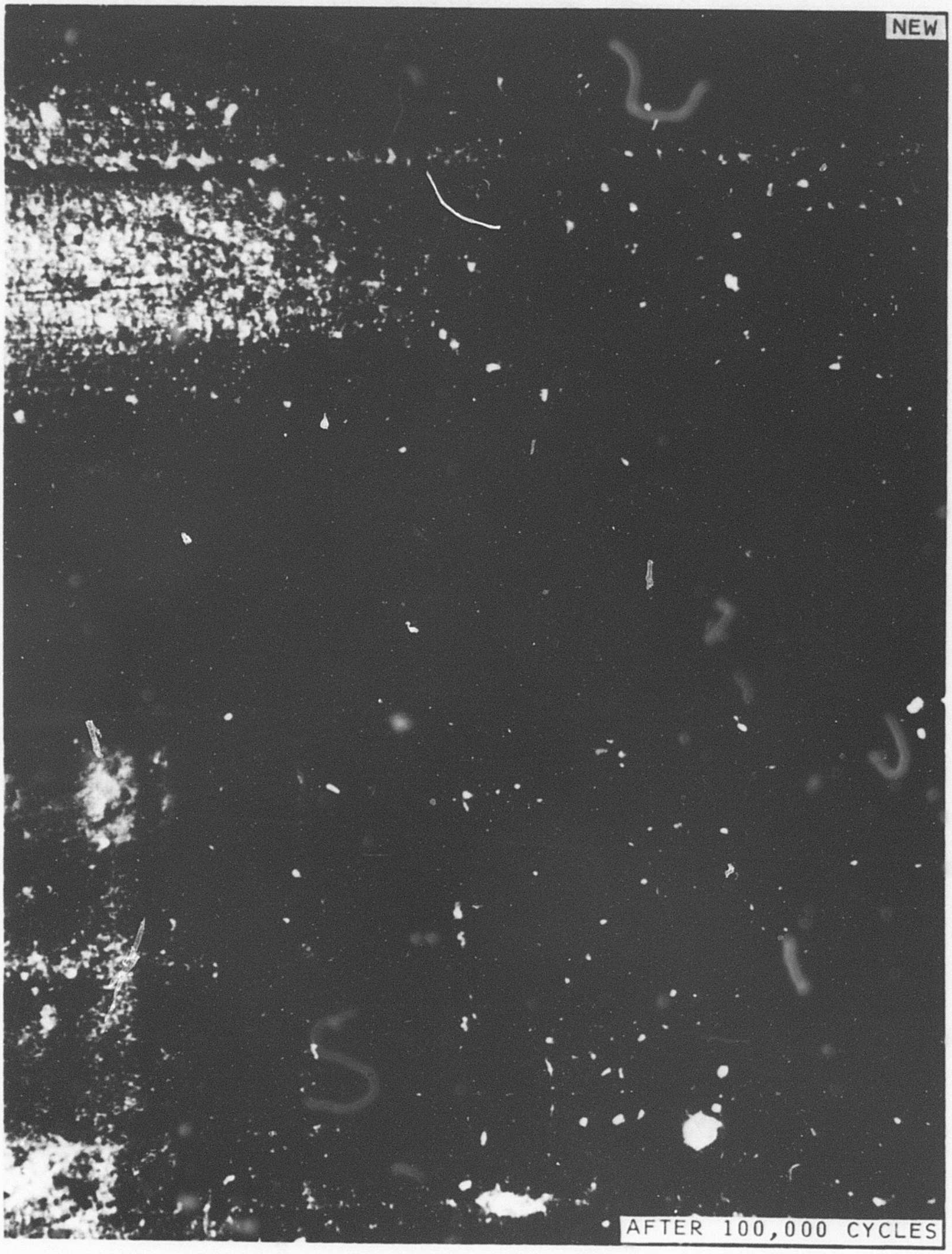


Figure 34. Translational Seal Buna N 756-75.

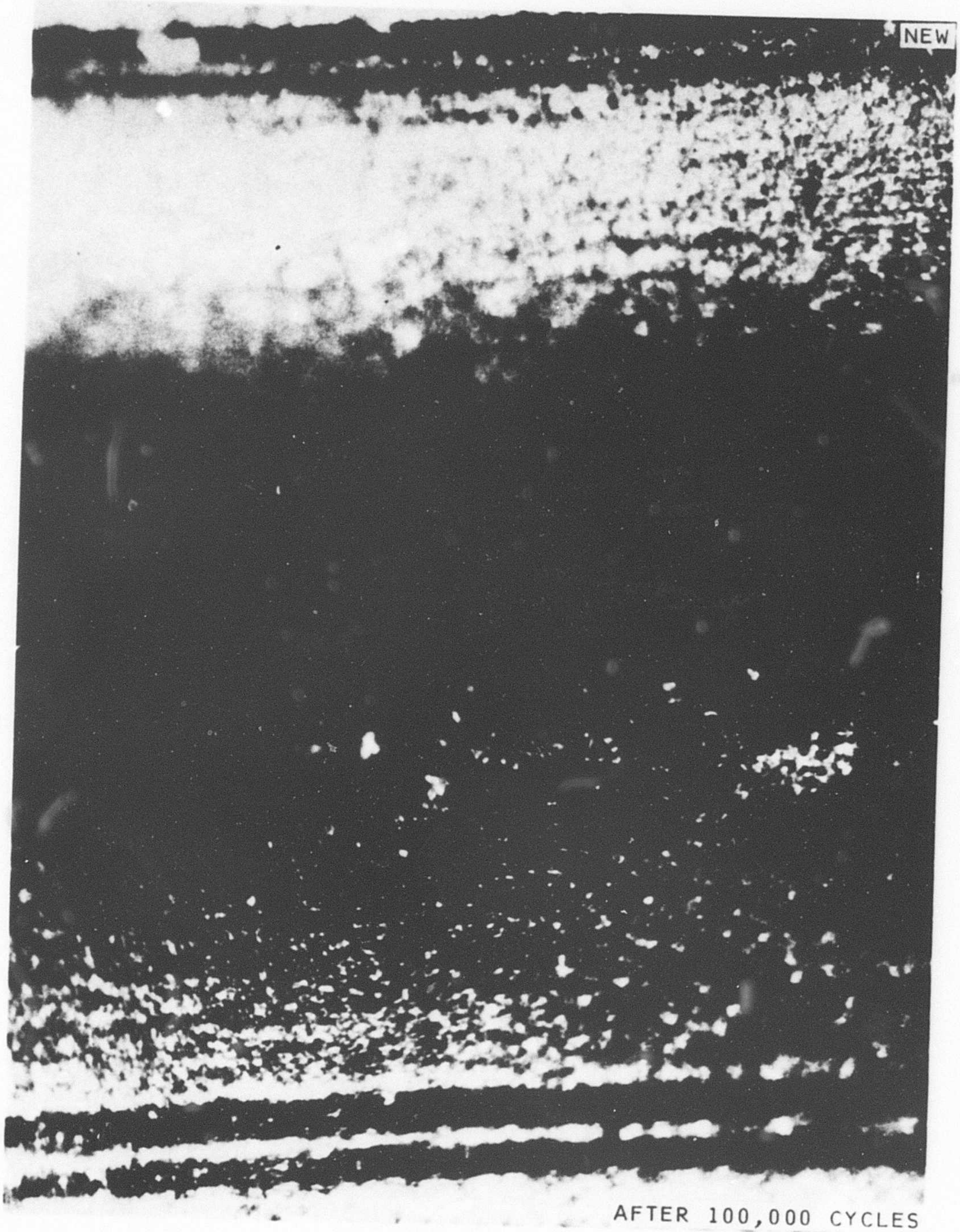


Figure 35. Translational Seal Viton V-135.

very stable material. The measurement which showed the greatest variation is the outside diameter. This varied from -4.6 to +3.3 percent. After making allowance for measuring errors, this small change in diameter confirms the lack of wear and indicates a low compression set of the material, which, as in the case of the C-69 rotary seal, results in a superior seal.

The last seal listed in Table 11 was a nitrile compound C-69. This is the same type of seal as the rotary listed in Table 10, although a different size was used. In each case, all six samples completed 100,000 cycles without leakage. This seal was reviewed as a rotary seal. The analysis of the rotary type applies as well to the translational type. Comparing the changes in physical properties listed in Tables 10 and 11, it is obvious that these changes were consistent, indicating again a stable material. More significant is the fact that the majority of the samples tested experienced changes contained within a very narrow band, further confirming the consistent quality from one seal to another. Figure 36 shows two seals side-by-side; one new, the other after 100,000 cycles. Axial wear lines are visible on the tested seal. These wear lines are less than 0.0005 inch wide and do not appear to be very deep. The mold flashes are visible in both seals, indicating little wear in the tested seal. There is no evidence of flatness or deformation.

Conclusion

Tables 10 and 11 list the 15 seals tested. In both cases, rotary and translational, the best seals were standard O-rings. The only seal to complete 100,000 cycles as a rotary and translational seal was the nitrile C-69, which is a Buna N. Other Buna N seals had good results; however, their physical properties had wide variations. It appears that Buna N O-rings are the best seals available. Additional development is required, however, to arrive at a Buna N material which will give consistently good performance. This performance can be achieved only if the physical properties do not vary from one seal to the next and if the limits of such properties as tensile strength and compression set are tightened.

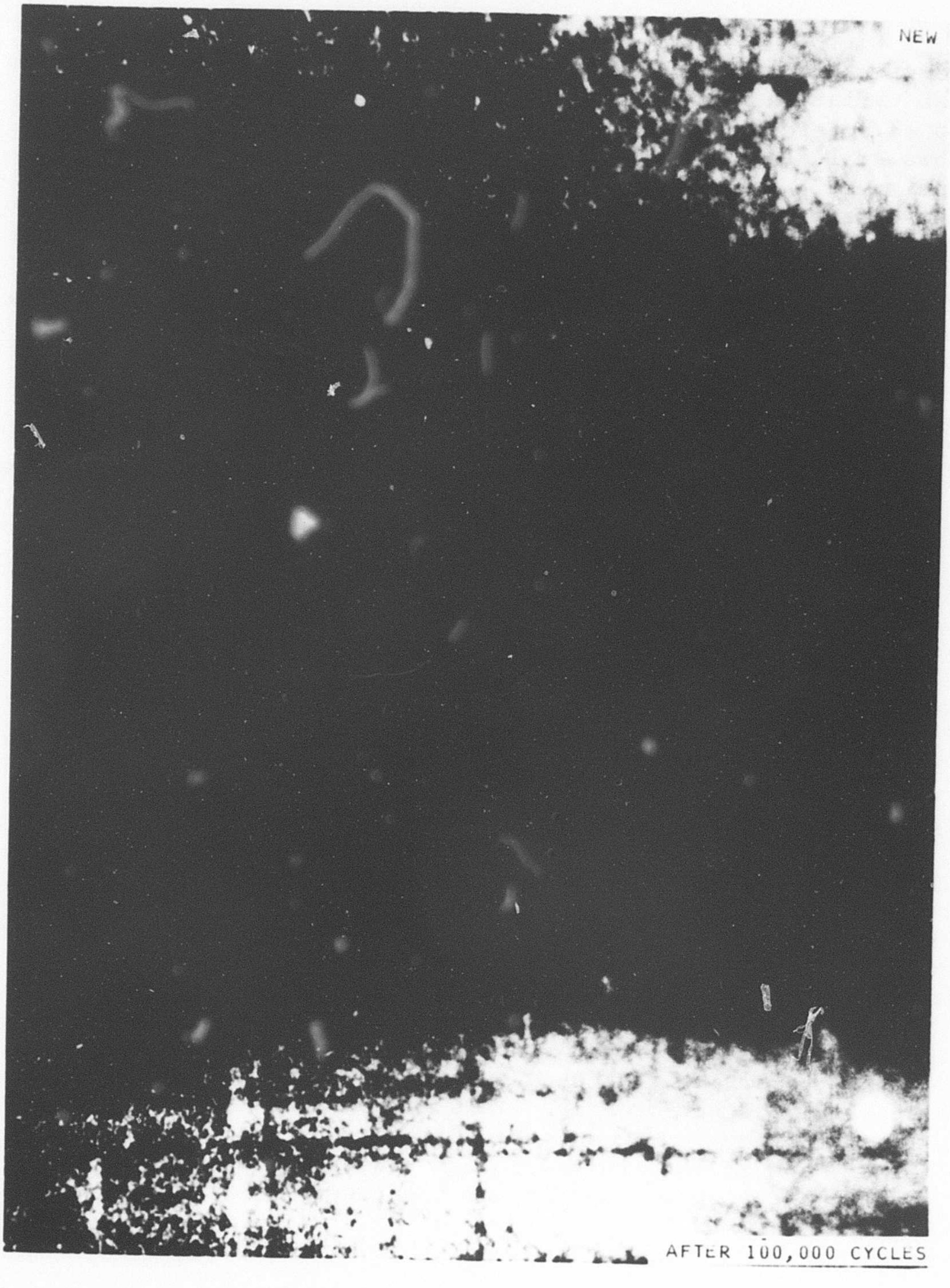


Figure 36. Translational Seal Nitrile C-69.

SUMMARY AND CONCLUSIONS

Vibration Test

Metered fuel flow was continuously monitored during vibration tests on two Chandler Evans Model TA-2S fuel controls to detect flow variations which could induce flight disturbances and thereby cause a control rejection. Also, bench calibrations were performed before and after the vibration test to detect any permanent shifts caused by the vibration testing. A complete disassembly and inspection of hardware was undertaken following the completion of all testing. The results of the above program follow.

During vibration, some fuel flow shifts and oscillations were detected; however, their magnitude remained within the operating limits of the control. Furthermore, the largest metered flow shifts should yield only a 1.5-percent N_2 and ± 0.3-percent N_1 variation when operating closed loop on the power turbine and gas producer governors.

The bench calibrations of the two controls were performed before and after 5 hours of vibration testing on each unit. These tests showed one unit to repeat its calibration within acceptance limits; however, an acceleration schedule flow shift of between 4.5 and 7.8 percent, which fell outside of bench acceptance limits at 4 of the 13 check points, was detected on the other fuel control. This calibration shift was explained when a teardown of the unit revealed an assembly error causing the control to be unrepeatable.

Detailed inspection of the disassembled controls gave no evidence of any hardware deterioration or damage. Therefore, the conclusion reached from the vibration test program is that the controls perform within their design limits and should not induce engine malperformance or shift their calibrations due to engine or airframe induced vibration so as to cause control rejections.

Fuel Filter Test

The two Chandler Evans Model TA-2S controls previously vibration-tested were run with fuel contaminated to MIL-E-5007C levels. One control was run with fuel filtered by a 25μ absolute filter; the other control was run with fuel filtered by a 3μ absolute filter. Testing consisted of 50 hours at room temperature and 20 hours with fuel at -65°F. Both controls were cycled to accelerate wear. The controls were calibrated before and after the contamination tests, and several control and fuel pump components were detail-inspected to obtain measures of wear.

Based on calibration test results, no evidence of any degradation in performance was determined for either control system. The amount of measured wear in both cases was negligible. Therefore, on the basis of these results, it is concluded that there is no justification for using filtration finer than the presently used 25-micron absolute filter.

Fuel Seal Test

A total of 15 seal types were supplied for environmental testing by various manufacturers. Ten seal types were cycled in a rotary oscillating motion, and five were cycled in a translational motion. Test cycling simulated extreme cold, hot, and dusty environmental conditions commensurate with Army aircraft operations. Various physical properties of each seal were measured before and after the cycling tests.

Only one rotary seal out of ten types tested and one translational seal out of five types tested, were successful in having all six samples complete 100,000 cycles without leakage or failure. Both of these seals were the same type and made from nitrile C-69 compound. However, different sizes were used for rotary and translational testing.

A second rotary seal, which was a Buna N compound with an ultralube additive, had six out of eight seals successfully complete 100,000 cycles. A standard Buna N compound rotary seal had four out of six seals complete 100,000 cycles. None of the other seals were successful; in most cases, the seals were structurally destroyed and/or sprung a severe leak.

In general, test results indicated that seal configurations composed of a single O-ring are better than seals composed of an O-ring and a TFE cap or backup ring. The TFE ring has better wear characteristics than the rubber O-ring; however, it tends to cut the O-ring. When the physical properties of other Buna N O-rings are compared to Nitrile C-69, the major difference is the lack of consistency in changes in physical properties experienced by other Buna N O-rings. These changes are spread over such a wide range that it indicates either better quality control or additional material development is necessary to arrive at a more stable material. Also, the Nitrile C-69 compound has the highest tensile strength and the lowest compression set of all the seals tested. These are judged to be desirable in an O-ring. Material development efforts should concentrate on raising the tensile strength and further lowering the compression set, while at the same time tightening the tolerance limits of these properties.

REFERENCES

1. Burnell, D. G., Morrison, T. B., and White, A. H., TURBINE ENGINE FUEL CONTROL RELIABILITY AND MAINTAINABILITY ANALYSIS, USAAMRDL Technical Report 73-60, U.S. Army Air Mobility Research and Development Laboratory, Ft. Eustis, Virginia, August 1973.

LIST OF SYMBOLS

Brg	bearing
D, Dia	diameter
ΔP	metering head differential pressure
F_R	resonant frequency
g	gravitational acceleration - 32 ft/sec^2
g/in	gravitational acceleration per inch
GPC	gas producer control (also, N_1 governor)
HP	horsepower
Hz	frequency, cycles per second
I.D.	inside diameter
MC	main control
N_1	gas producer control speed
N_2	power turbine governor speed
O.D.	outside diameter
P_1	atmospheric pressure
P_2	control discharge pressure
P_A	air pressure
P_B	blower air pressure
P_F	fuel pump pressure
PLA	power lever angle, deg

LIST OF SYMBOLS (Cont)

pph	pounds per hour
psi	pounds per square inch
PTG	power turbine governor
rpm	revolutions per minute
SLSD	sea level, standard day
S/N	serial number
T	temperature, °F
T_A	ambient temperature, °F
T_F	fuel temperature, °F
TFE	tetrafluoroethylene
W_F	metered fuel flow, pph
μ	micron

U.S. Government Printing Office: 1974—636-096/27 Region No. 3-11