THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Final Report

DYNAMOMETER EVALUATION OF THE SPUR-GEAR FINAL DRIVES AS DESIGNED FOR THE LVTP-5 TYPE OF TRACKED VEHICLE

W. O. Hermanson
F. L. Schwartz

UMRI Project 2785

under contract with:

BORG-WARNER CORPORATION
INGERSOLL KALAMAZOO DIVISION
SUBCONTRACT NO. 2000
U. S. NAVY CONTRACT NO. NObs 3855-58-142
KALAMAZOO, MICHIGAN

administered by:

THE UNIVERSITY OF MICHIGAN RESEARCH INSTITUTE ANN ARBOR

January 1960

TABLE OF CONTENTS

LIST OF FIGURES

ABSTRACT

This report covers the evaluation of the spur-gear final drives designed for the LVTP-5 type of tracked amphibious vehicle.

The supply of lubrication oil furnished by the pump cluster, when operated at 2100 rpm, is adequate. The wear patterns indicate nonuniform contacts between the pinions and gears. The drives will operate without external cooling at 265°F, when loaded at 3500 ft-lb and 800 rpm. The input pinion wear rate is zero at 5000 ft-lb and 650 rpm once break-in has occurred (these data were obtained through use of the radioactive tracer technique). If the gear specifications are maintained, the expected life of the final drives, at the maximum load of 5000 ft-lb and 650 rpm, is better than 20 hours.

OBJECTIVE

The objective of this study was to evaluate, on the dynamometer, the spur-gear final drives for the LVTP-5 type of tracked amphibious vehicle.

INTRODUCTION

This report presents a dynamometer evaluation of the spur-gear final drives as redesigned by the Ingersoll Kalamazoo Division of the Borg-Warner Corporation for application to the LVTP-5 type of tracked amphibious vehicle. The original design of the spur-gear final drives was evaluated by The University of Michigan Engineering Research Institute (now The University of Michigan Research Institute) in 1955 and 1956, an evaluation reported in Final Report No. 2385-45-F, Dynamometer Testing and Evaluation of the Performance of the Spur-Gear Final Drives as Designed for the LVTP-5 Type of Vehicle, dated July, 1956.

Project 2785 originated through authorization of Ingersoll Kalamazoo Division's Subcontract No. 2000, under United States Navy Contract No. NObs 3855-58-142.

GENERAL DESCRIPTION OF EQUIPMENT AND TEST PROCEDURE

DESCRIPTION OF DRIVES EVALUATED

The general design of the spur-gear final drives submitted by Ingersoll Kalamazoo for evaluation combines the final drive and the drop gear into a single unit, utilizing spur gears, and a hydraulically disengaged, spring-actuated clutch for water steering. An overall gear reduction of 5.36:1 is accomplished through two sets of spur gears and pinions. The final drive terminates at the sprocket hub, 22 in. below the input yoke. The arrangement of components is shown in Fig. 1.

The two major differences between the redisigned drives and the drives evaluated four years ago are:

(1) Housings are now cast steel rather than ductile iron.

(2) The two bearings on the intermediate shaft are now self-aligning SKF spherical roller bearings rather than Bower straight roller bearings. This change allowed the intermediate shaft, which was found to be weak in the original design, to be strengthened by both increasing the diameter and reducing the length of the hubs.

TEST EQUIPMENT

The final drives were mounted, with output sprocket hubs connected together, in fixtures on a cast iron bed plate for rigidity. The final drives were set up in this manner so that the final output speed would be high enough to allow absorption by the dynamometer. Also, two drives may be evaluated at the same time. The port drive was connected to a T-1200 transmission which was driven by an Allison V-1710 engine. The input shaft of the starboard drive was connected to a 2000-hp eddy-current dynamometer (Fig. 2).

Lubrication and clutch pressure oil was supplied by the standard LVT pump cluster, which was driven at 2100 rpm by an electric motor (Fig. 3). During the gear-wear phase of the evaluation, lubrication oil was supplied and scavenged by separate, electrically driven pumps.

All instruments and controls were mounted remotely in the control room. Two electronic counters were used to display the speed of both final drives. This instrumentation gave an accurate determination of speed and of any possible clutch slippage. Dynamometer torque was measured with a pneumatic balanced-diaphragm device, with the dial-type indicator located on the control panel. Thermocouples were installed at seven critical locations on each drive. The locations are shown in Fig. 1.

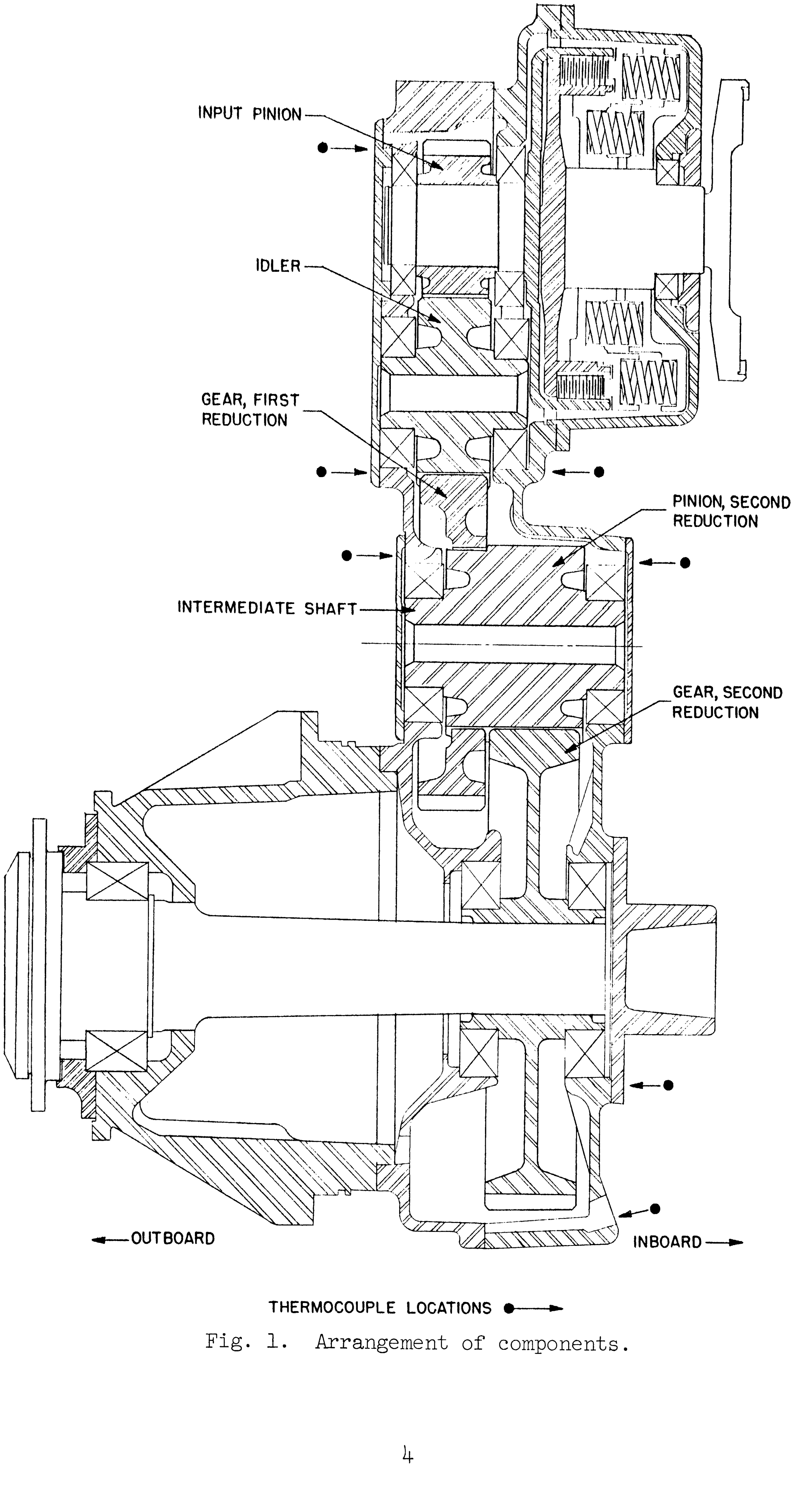

Fig. 1. Arrangement of components.

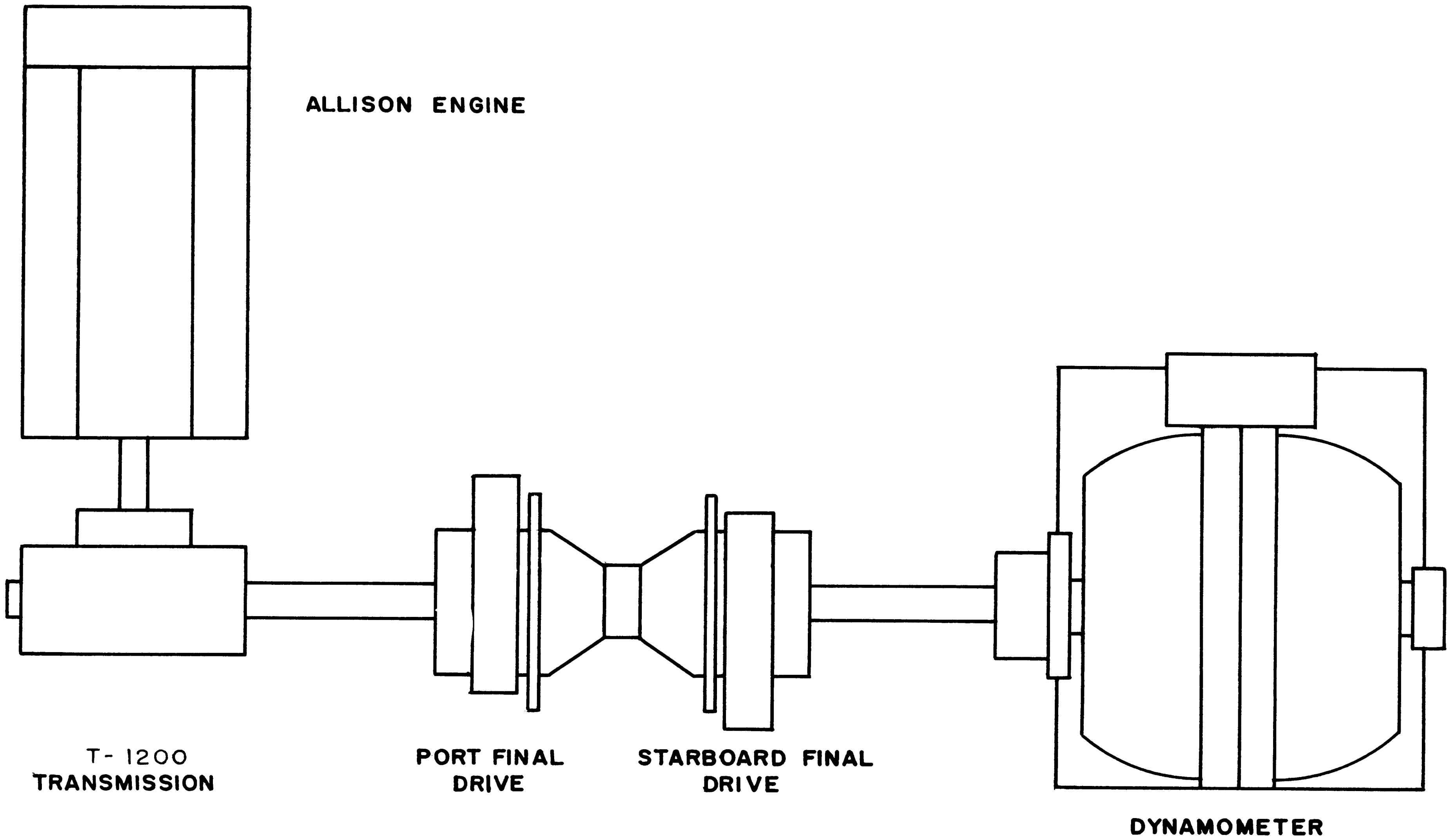

Fig. 2. Schematic layout of test components.

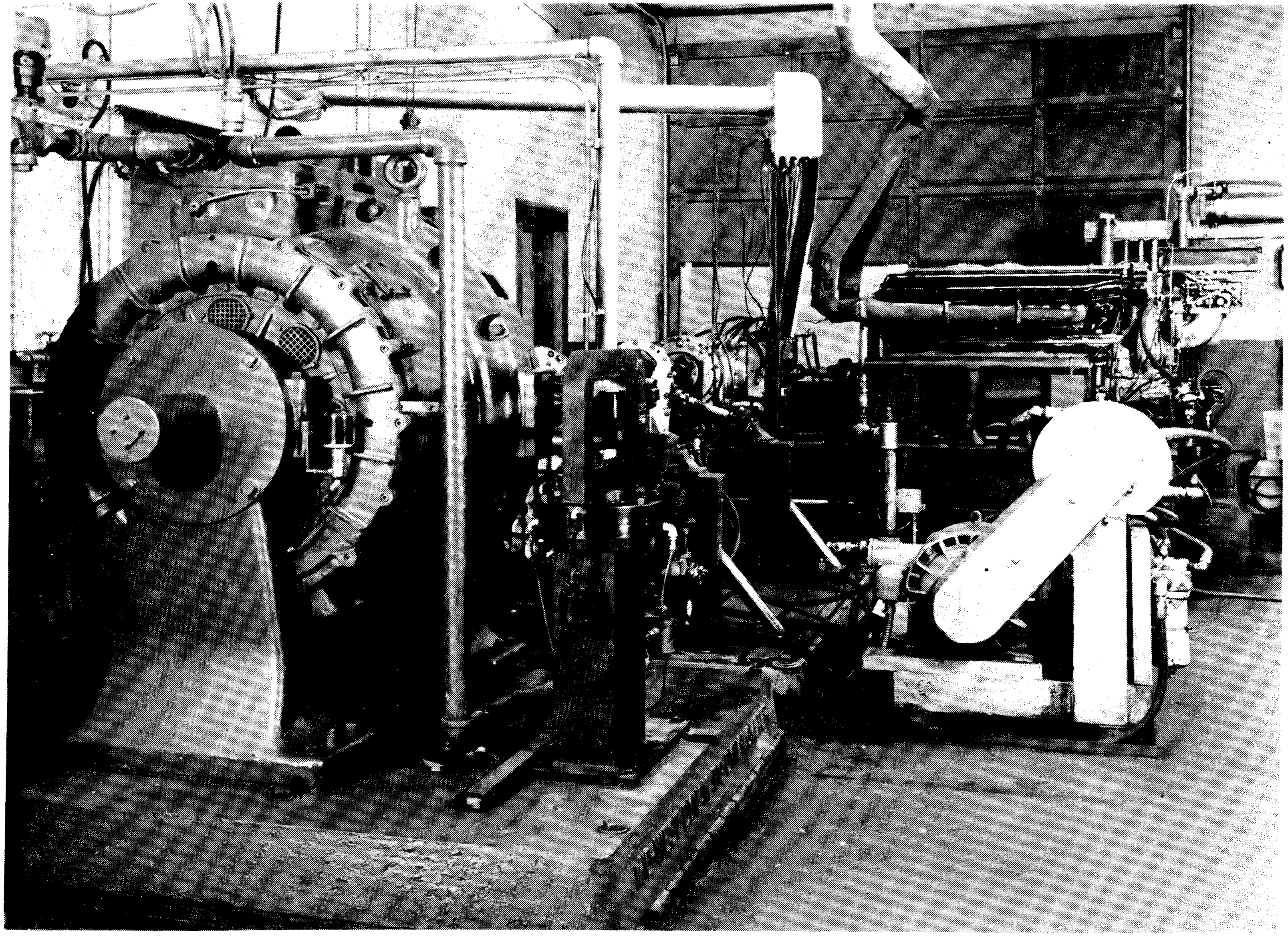

Fig. 3. Overall view of test setup. Dynamometer and pump cluster in foreground Allison engine and T-1200 transmission in background.

The intensity of radiation (hence the amount of steel) in the oil was detected by a Nuclear-Chicago DS-1 Scintillation Detector, indicated on a Tracerlab SC-34 Precision Rate Meter, and recorded on an Esterline-Angus AW recording milliammeter.

TEST PROCEDURE

The procedure was divided into the following eight phases:

I. Oil Flow Through Final-Drive Orifices
II. Flush Lubrication System
III. Break-in Phase
IV. Wear Pattern due to Deflection
V. Wear Pattern at Maximum Load
VI. Torque Required to Stabilize Drives at a Temperature of 271°F
VII. Gear Capacity Before Excessive Surface Breakdown
VIII. Gear and Bearing Life

DISCUSSION OF TESTS AND RESULTS

I. OIL FLOW THROUGH FINAL-DRIVE ORIFICES

Description.—A gear mesh spray-jet and a bearing lube-jet were removed from a final drive and the time required for a measured quantity of oil to flow through each jet was recorded. The flow measurements were conducted with SAE 50 oil, a pump cluster input speed of 2100 rpm, and under the following temperature and pressure conditions:

Temperature, °F	Pressure, psi
73	75
160-180	24-40
220-240	19-42

The minimum pressure for each temperature was that pressure that exists when the standard pump is supplying both final drives. The higher pressures were obtained by throttling the oil that flowed to the final drive whose jets were not being measured.

Results.—The oil flow rates through the final drive jets are shown in Fig. 1

II. FLUSH LUBRICATION SYSTEM

Description.—

A. The oil reservoir was filled to the proper level with SAE 50 oil.
B. The drives were operated at an input speed of 600 rpm and at zero torque for two hours. The filters were rotated each hour.
C. The lubrication system was drained and the filters cleaned.
D. Steps A, B, and C were repeated three times.
E. The oil reservoir was filled to the proper level with SAE 50 oil.

Results.—After the first two-hour period, a very small amount of pipe-sealing compound was removed from the filters. The filters remained clean after the second and third two-hour period.

III. BREAK-IN PHASE

Description.—The drives were operated for one hour at each of the following load conditions:

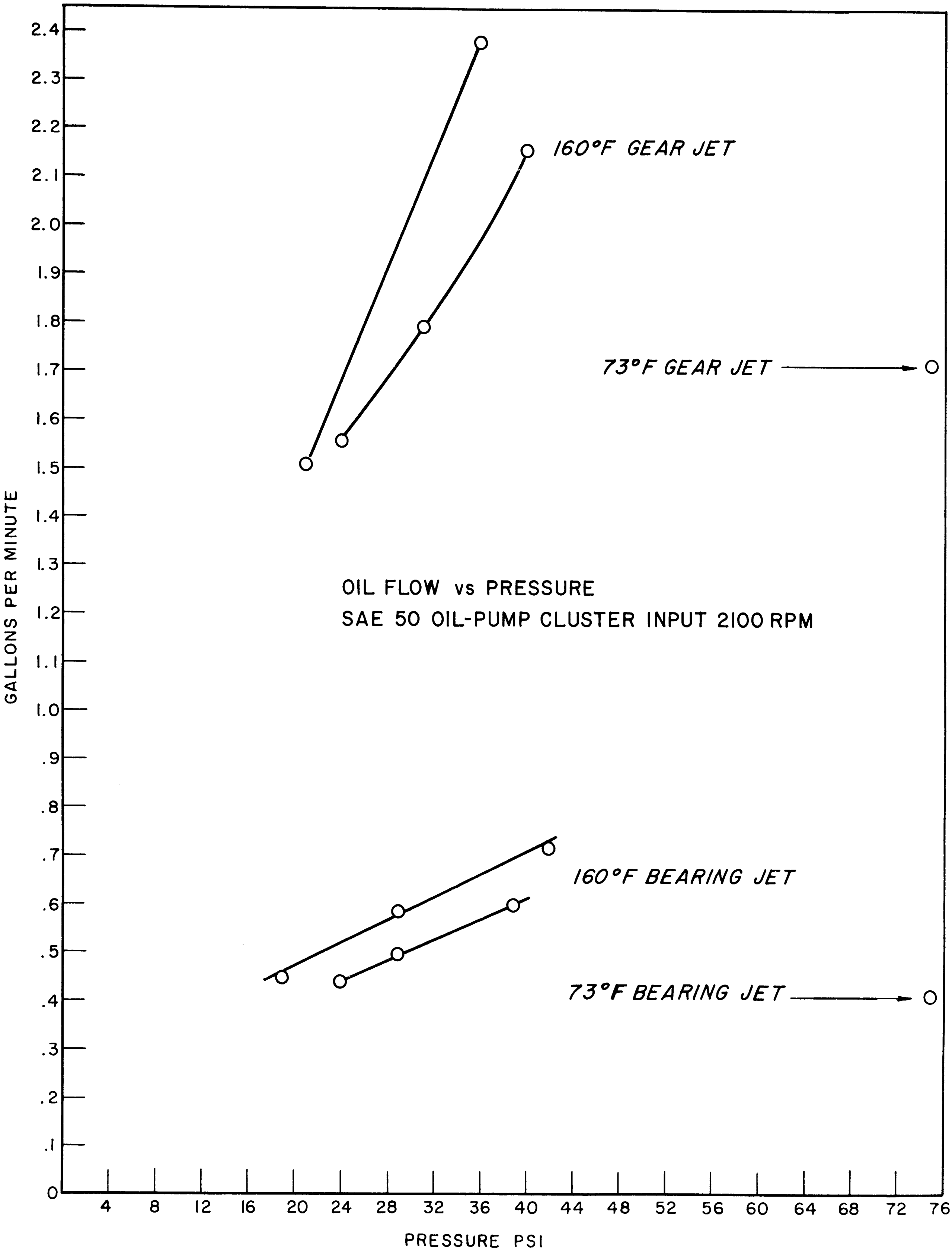

Fig. 4. Oil flow vs. pressure.

400-rpm input and 1000 ft-lb torque
600-rpm input and 1500 ft-lb torque

The slip torque of the disengaged clutches was measured at 600 rpm at various times during the flushing and breaking-in phases.

Results.—The drives operated satisfactorily during the above loads. The following slip torque data were obtained.

Final Drives	Slip Torque
Port (Serial No. 6)	72 ft-lb (new) 30 ft-lb (6-hr operation)
Starboard (Serial No. 5)	100 ft-lb (new) 30 ft-lb (6-hr operation)

IV. WEAR PATTERN DUE TO DEFLECTION

Description.—The drives were run at 600-rpm input and 3000 ft-lb torque for 30 minutes to determine the wear pattern due to deflection. After this run, both final drives were disassembled and all bearings, gears, and oil rings were inspected.

Results.—The following conditions existed in both units:

(1) All straight roller bearings in excellent condition.
(2) All spherical roller bearings slightly burnished.
(3) Uniform gear contact patterns between input and idler gears and between pinion and gear, second reduction.
(4) Nonuniform gear contact patterns between idler and gear, first reduction.
(5) Removal of the idler and pinion cover plates from both units showed that the cover projection (and its piston ring seal) which feeds oil to the clutch pack was badly scored. The clearance between this projection and the pilot hole in the input shaft was quite low while considerable play was possible between the cover pilot diameter and its mating hole in the housing. The diametral clearance was increased to 0.020 in. by grinding the outer diameter of the projections. No trouble was experienced after this modification.

V. WEAR PATTERN AT MAXIMUM LOAD

Description.—The drives were operated at 650 rpm and 5000 ft-lb torque. The gears and bearings were inspected after the above run.

Results.—All gears and bearings were in the same condition as after the previous run.

VI. TORQUE REQUIRED TO STABILIZE DRIVES AT A TEMPERATURE OF 271°F

Description.—The units were driven at an input speed of 800 rpm and the torque increased until a stabilized temperature was reached in the final-drive oil sump.

Results.—After operating for 143 minutes at 800 rpm and 3000 ft-lb the final-drive sump temperature stabilized at 230°F. After an additional 127 minutes at 800 rpm and 3500 ft-lb, the sump temperature stabilized at 265°F. The ambient temperature, taken between the two drives, was 90°F. Since 3500 ft-lb is the maximum torque output of the third gear range of the T-1200 transmission, it was decided not to attempt to obtain a stable temperature of 271°F and risk possible damage to the T-1200. The originally designed spur-gear final drives, evaluated by our laboratory four years ago, stabilized at 271°F at only 2200 ft-lb torque. The probable reason for the large difference in torque is the replacement of the straight Bower roller bearings on the intermediate gear shaft with SKF spherical roller bearings. The straight roller bearings were a major source of heat in the originally designed final drives.

VII. GEAR CAPACITY BEFORE EXCESSIVE SURFACE BREAKDOWN

Description.—The radioactive-tracer technique was chosen for the gear-wear determination because the rate of wear can be measured continuously while the speeds and loads are varied, and also because very low wear rates may be measured. Wear rates for many different operating conditions may be determined in a relatively short time and without visual inspection of the gears between load conditions.

Two extra input pinions were obtained for irradiation. The pinions were manufactured to specifications shown on Drawing No. 1200618. The more important gear specifications are as follows:

Gear OD	6.67 in.
Gear ID	3.30 in.
Gear Width	2.75 in
Gear Weight	15.8 lb
Number of teeth	21

The gears were manufactured from either SAE 4620, 4820, 8620, or TS 8720 steels. The approximate chemical composition for this group of steels is as follows:

Carbon	0.17-0.23%	Silicone	0.20-0.35%
Manganese	0.45-0.90%	Sulphur	0.040% maximum
Nickel	0.40-3.75%	Phosphorus	0.040% maximum
Chromium	0-0.60%	Iron	Balance
Molybdenum	0.15-0.30%		

The principal contributors to the gamma ray radioactivity and their half-lives are $iron^{59}$, 46 days, and $chromium^{51}$, 26.5 days. The remaining elements have very short half-lives so that after two weeks the remaining activity is due mainly to the iron and chromium.

Steel powder was ground from the bevel ends of the gear teeth, separated from the grinding wheel material magnetically, weighed, and wrapped in aluminum foil. The sample packets were placed in the gear-tooth spaces so that the sample powder would receive, in the reactor, the same neutron bombardment as the gear teeth themselves.

The first of the two extra input pinions, labeled as pinion "A," was irradiated at the Brookhaven National Laboratories in Upton, Long Island, New York. The pinion was wrapped in aluminum foil and placed in the reactor under the following operating conditions:

Approximate flux level (before gear was loaded into reactor)	5×10^{11} nv
Time in reactor	
Operating	883.5 hr
Not operating	84 hr
Intensity of radiation at time of use (time of use was 68 days after removal from reactor)	1.3 roentgens/hr at 1 ft

The calibration system (Figs. 5 and 6) was filled with six gallons of oil and the smallest sample of steel powder was added to the oil. The activity level of the steel-powder and oil mixture was not stable with respect to time which indicated that the powder was apparently falling out of the mixture. We then decided to dissolve the powder in acid and form an iron compound which was soluble in the oil. The iron atoms will exhibit the same radioactive properties whether they exist as iron power or as some iron compound. One hundred milligrams of steel powder were dissolved in concentrated hydrochloric acid. Hydrogen peroxide was added to oxidize the ferrous iron to ferric iron, and the solution was evaporated to dryness. The residue was taken up in water and then carbon tetrachloride was added to facilitate separation in the next process. A solution of potassium naphthenate in isopropyl alcohol was added. The ferric naphthenate formed was extracted (by shaking) into the carbon tetrachloride phase. The solution of ferric naphthenate in carbon tetrachloride was separated and used to make a calibration standard of iron naphthenate in oil. The iron naphthenate standard was divided into four equal parts, each representing the activity of 25 mg of steel powder. The four parts of the calibration standard were added, one at a time, to known quantities of oil. The activity level now remained constant which indicated no separation of the iron naphthenate from the oil. The radiation intensity for 100 mg of steel as iron naphthenate in 6 gallons of oil was the same as that for 100 mg of steel powder in the oil before

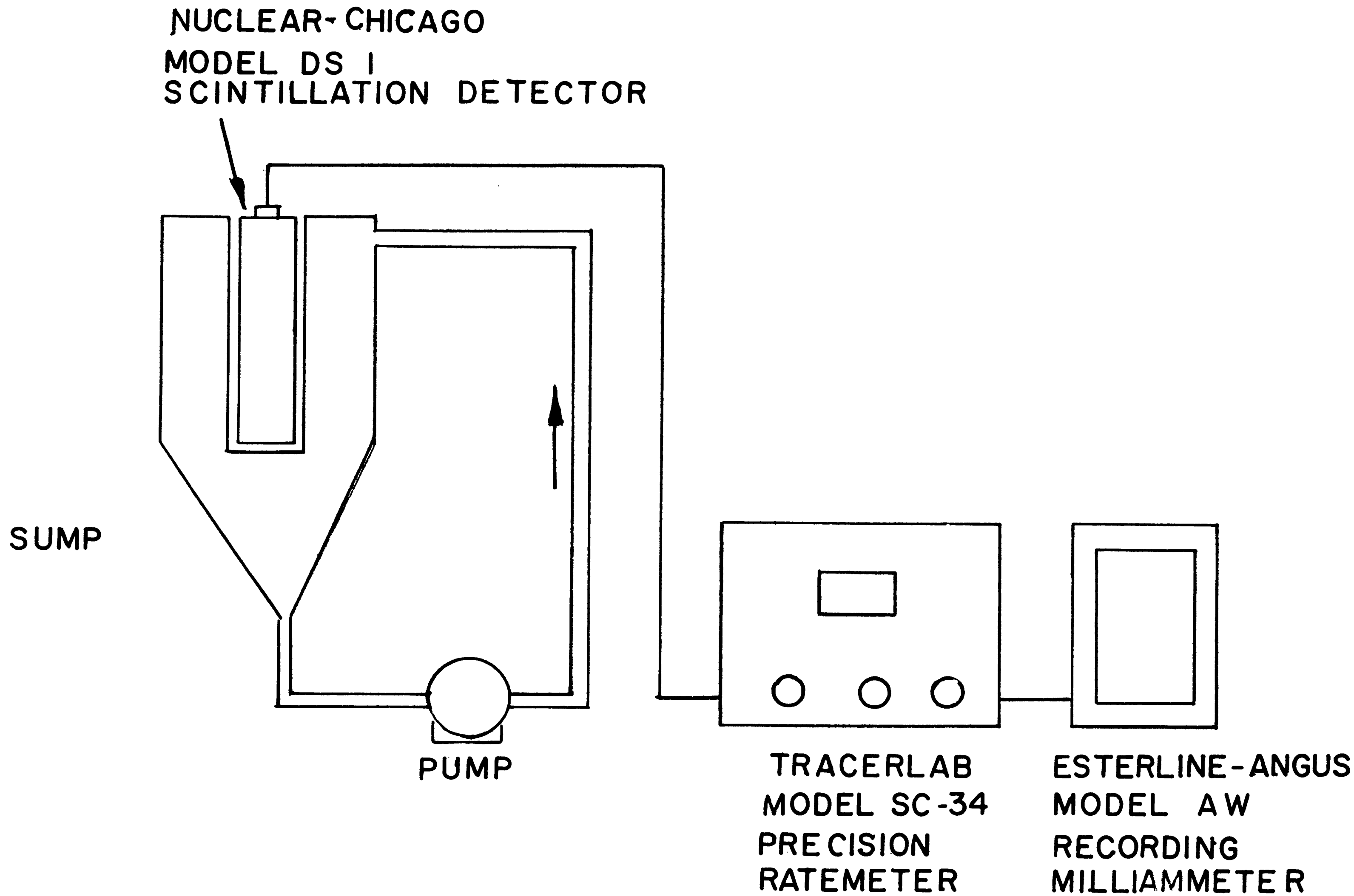

Fig. 5. Radioactive calibration system.

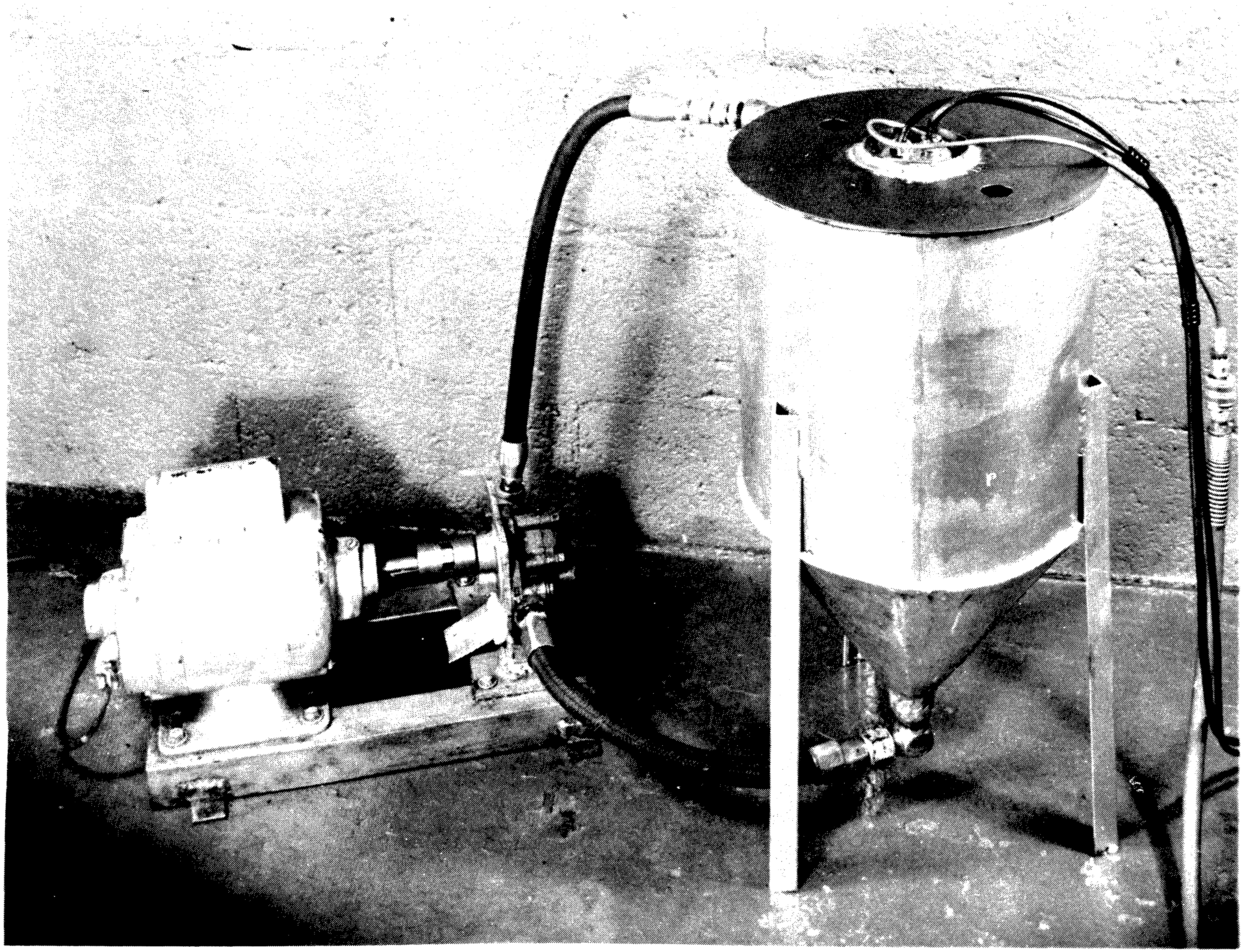

Fig. 6. Calibration system.

separation occurred. The calibration data are shown in Fig. 7. With this plot of mg/gal vs. activity, any reading of activity level in the identical counting chamber used in the final-drive lubrication system can be converted to milligrams of steel per gallon of oil (see Fig. 8). If the wear-rate studies cover a period of several days, it is necessary to apply corrections to account for the decay of radioactivity. This can be done by determining the calibrations system's activity at the later date and drawing a curve similar to the original curve through that point and the origin.

Pinion "A" was installed in the port final drive, using special long-handled tools. The drives were run until a definite wear rate was shown on the chart recorder.

Pinion "B" was irradiated at The University of Michigan's Ford Nuclear Reactor in Ann Arbor. This reactor is of the "swimming pool" type and the gear, sealed in an aluminum container, was suspended outside the reactor core face submerged under 22 ft of water. The following reactor operating conditions apply:

Approximate flux level (before gear was loaded into reactor)	3×10^{12}
Time in reactor	
Operating	418 hr
Not operating	2650 hr
Intensity of radiation at time of use (time of use was 18 days after removal from reactor)	2 roentgens

The calibration data for pinion "B" was obtained in the same manner as for pinion "A" and are shown in Fig. 9.

Pinion "B" was installed in the port final drive and the drives were run until a definite wear rate was shown on the chart recorder.

Results.—The wear rates at the various speeds and loads for pinions "A" and "B" are shown in the following table (page 20).

The wear rates and total wear for pinion "A" are also shown in Fig. 10.

The high initial wear rates may be attributed to surface irregularities. It was noted that the two extra input pinions supplied for the gear-wear studies had a much rougher surface finish than the gears which came installed in the gear box. Pinion "A" was also rougher than pinion "B."

The wear rates and total wear for pinion "B" are also shown in Fig. 11.

The irradiated pinion "B" was left installed in the starboard final drive during the gear and bearing life test. As may be seen, the wear rate dropped to zero once the machining marks had been polished from the tooth surface.

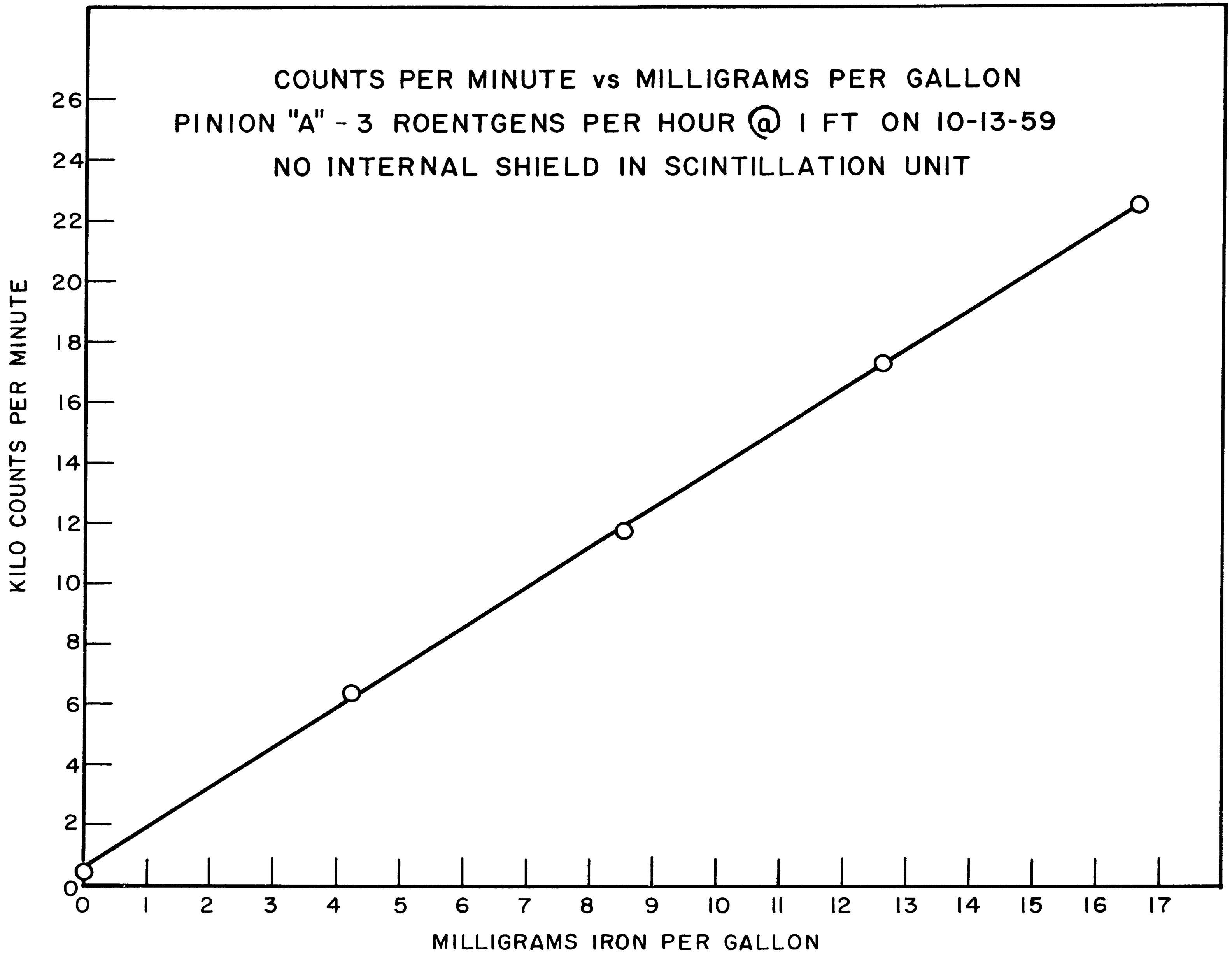

Fig. 7. Calibration data, pinion "A."

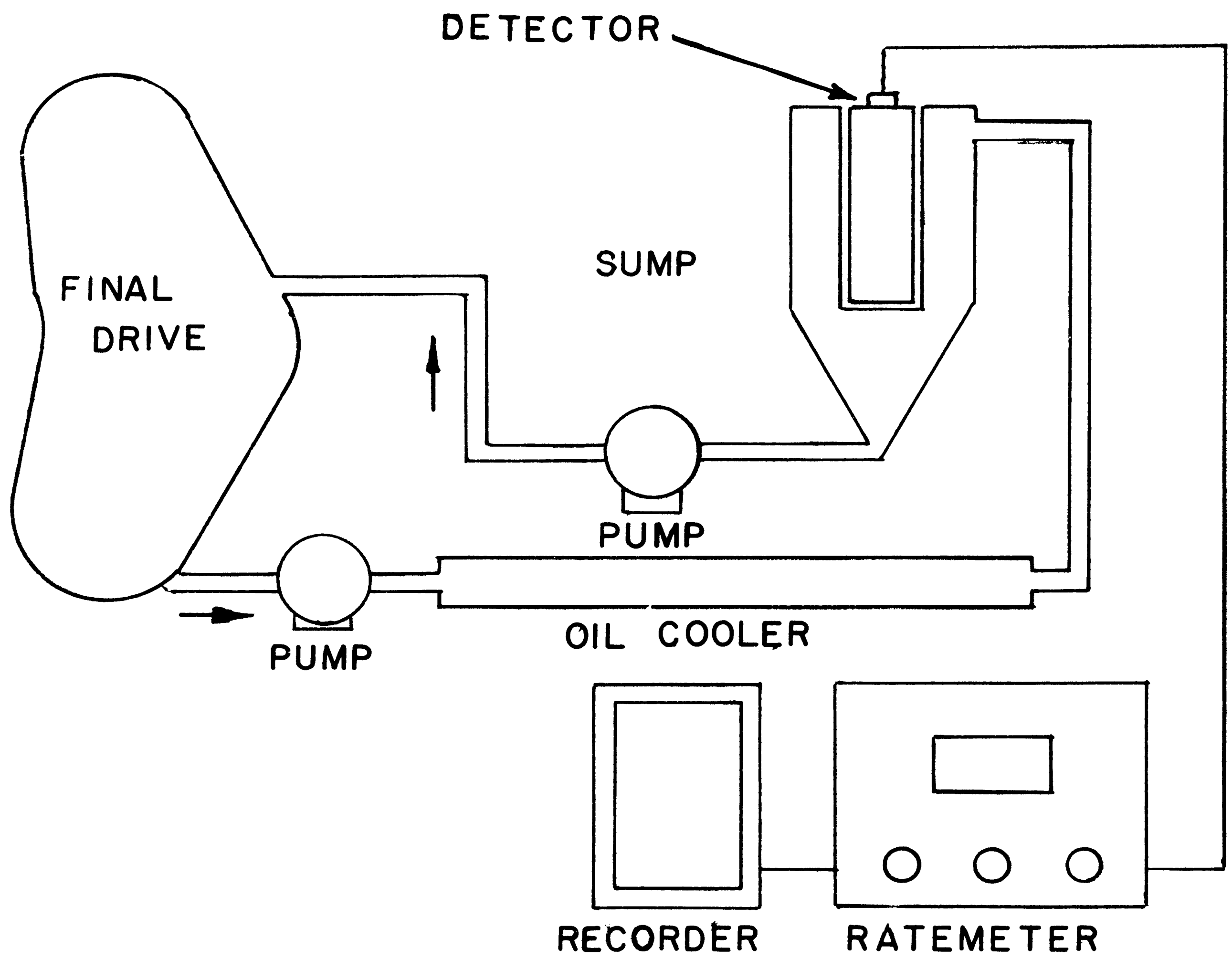

Fig. 8. Wear-measuring setup.

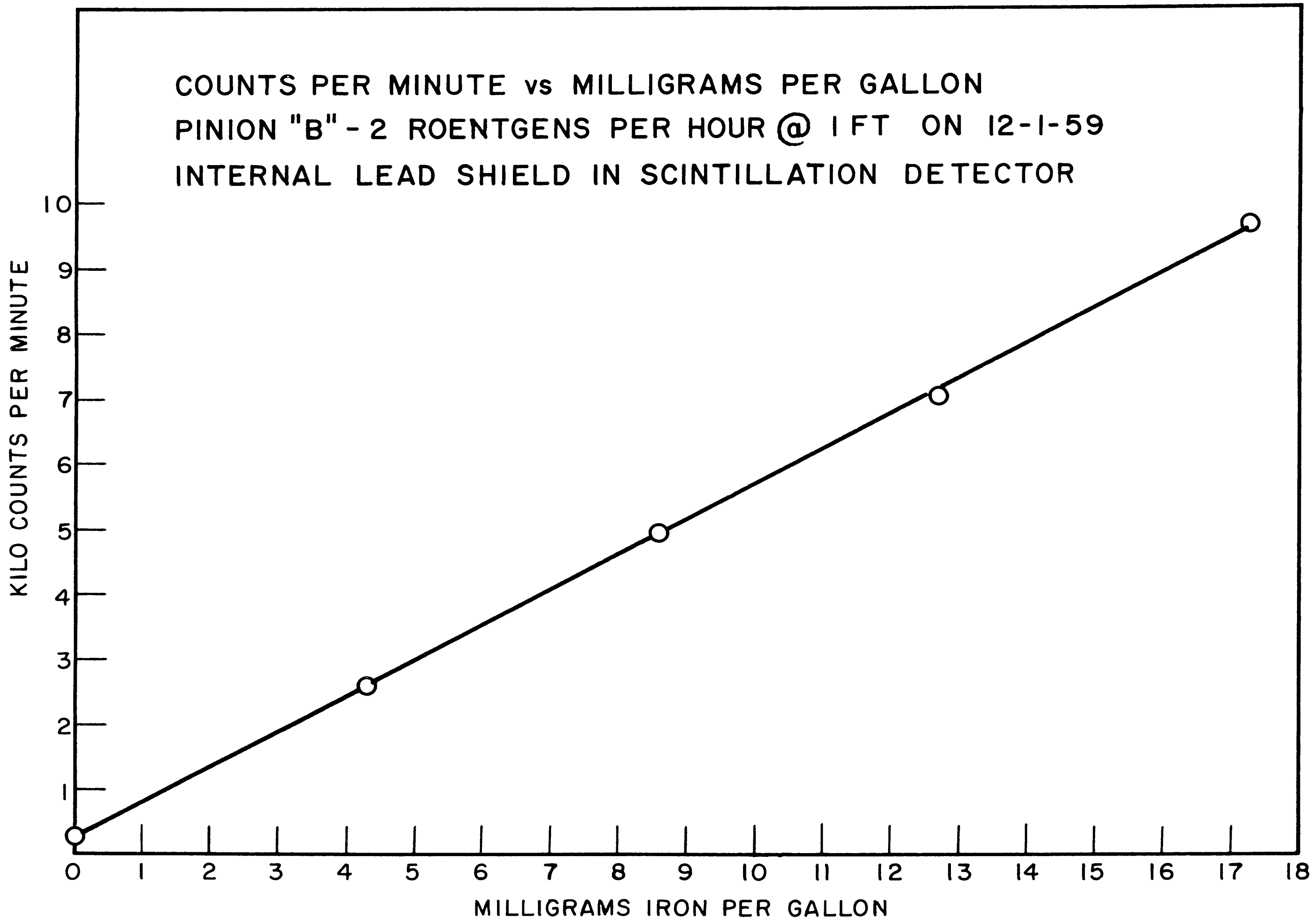

Fig. 9. Calibration data, pinion "B."

Pinion "A"			
Speed, rpm	Torque, ft-lb	Wear Rate, mg per min	Duration, min
800	0	9.13	9
		2.17	11
		0.76	20
		0	7
			47
800	500	2.54	10
		0.95	20
			30
800	1000	2.25	10
		0.85	20
			30
800	1500	1.09	15
		0.76	15
			30
800	2000	1.17	30
800	2500	2.08	30
800	3000	2.17	15
800	3500	4.68	21
650	4000	3.04	14
650	4500	4.48	15
650	5000	2.72	10
650	2000	0	15
650	3000	0.28	20
800	3500	3.38	8
650	2000	0	43
650	3000	0.29	30
650	4000	0.91	20
650	5000	2.27	15

Pinion "B"			
Speed, rpm	Torque, ft-lb	Wear Rate, mg per min	Duration, min
800	0	6.31	4
		0	12
			16
800	500	2.58	10
		0	5
			15
800	1000	2.94	5
		0	10
			15
800	1500	1.47	5
		0	10
			15
800	2000	0.98	15
800	2500	2.94	10
		0	5
			15
800	3000	1.96	15
800	3500	1.56	15
650	4000	1.63	15
650	4500	2.35	15
650	5000	2.72	20
650	4500	0.29	20
650	4750	0.74	27
650	4900	0.91	20
800	3500	0.96	15
		0.07	31
			46
800	3000	0	20
1000	2500	0	30
650	5000	0.93	18
650	5000	0.23	27
650	5000	0.11	30
650	5000	0.69	19
650	5000	0.04	93
650	5000	0.36	29
800	0		
650 to	4500	0.22	75
650	5000	0	120
650	5000	0	180
650	5000	0	180
650	5000	0	120
650	5000	0	270
650	5000	0	120

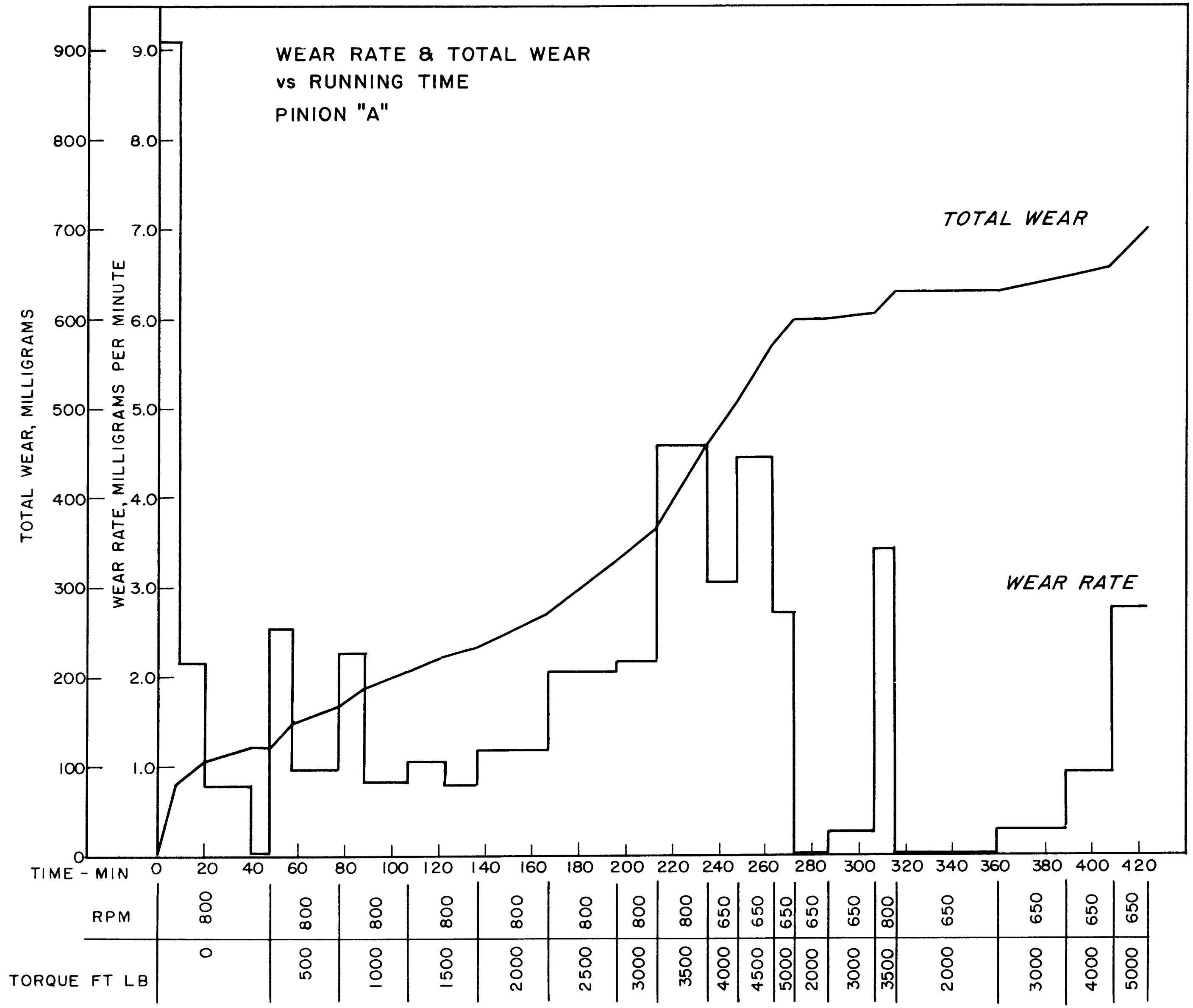

Fig. 10. Wear data, pinion "A."

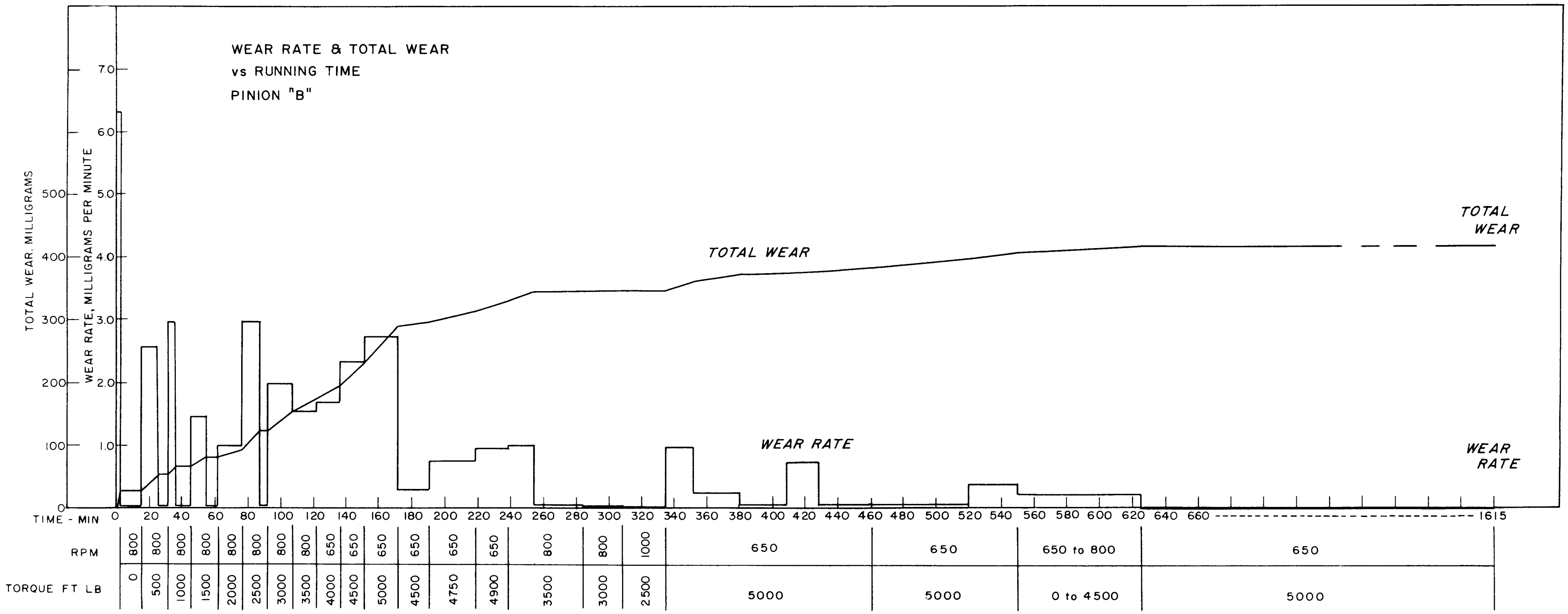

Fig. 11. Wear data, pinion "B."

VIII. GEAR AND BEARING LIFE

Description.—The drives were operated at 650 rpm and 5000 ft-lb torque. As mentioned in the previous section, the irradiated pinion "B" was left in the starboard final drive to observe its wear rate with respect to time. Because of the radiation hazards to personnel, the starboard final drive was not disassembled for inspection during the life test.

Results.—The drives had operated for a total of 271 minutes at 650 rpm and 5000 ft-lb torque when the input pinion of the port unit failed. Sections of the outer case of the gear tooth material had started to flake off which greatly accelerated the failure (see Fig. 12).

A tooth from the failed input pinion was ground, and the metal hardness was measured on the end surface of this tooth and at .010-in. increments beneath the surface. The hardness was measured on the Rockwell Superficial Hardness 15-N scale and then converted to Rockwell-C scale. The results of this investigation are shown in Fig. 13. The surface hardness was only Rockwell C-53 while Drawing No. 1200618 calls for a minimum of Rockwell C-60. The carburized hardness falls off at .040 in. beneath the surface, while the drawing calls for a minimum of .055 in.

A new final drive was obtained and installed on the port side of the test unit. The drives were operated for two hours at the life test loads and then inspected. This inspection revealed a slight scuffing of the tips of the input pinion teeth on their outboard ends. This scuffing caused a slight sharp edge to be formed on the tip of the wearing surface of each tooth. The bearings were slightly burnished. The drives were operated for three additional hours at the life-test loads and then inspected. No significant changes were noted at this time. The drives were operated for an additional five hours and again no significant changes were noted.

The drives had operated for an additional 6-1/2 hours at the life-test loads when a major failure occurred in the T-1200 transmission. After consultation with the Ingersoll Kalamazoo Division, it was decided to terminate the life test at this point.

The drives were disassembled and inspected. The following visual observations were made at that time.

Final Drive No. 14. 16.5 hours at the life-test load of 5000 ft-lb and 650 rpm.

The piston ring seal for the clutch oil supply was in good condition. The diametral clearance between the cover projection and the shaft hole was 0.020 in.

A slight amount of wear was found at the tips of the input pinion teeth on the outboard side of the pinion. This wear was not any greater than that noted after the first two-hour inspection.

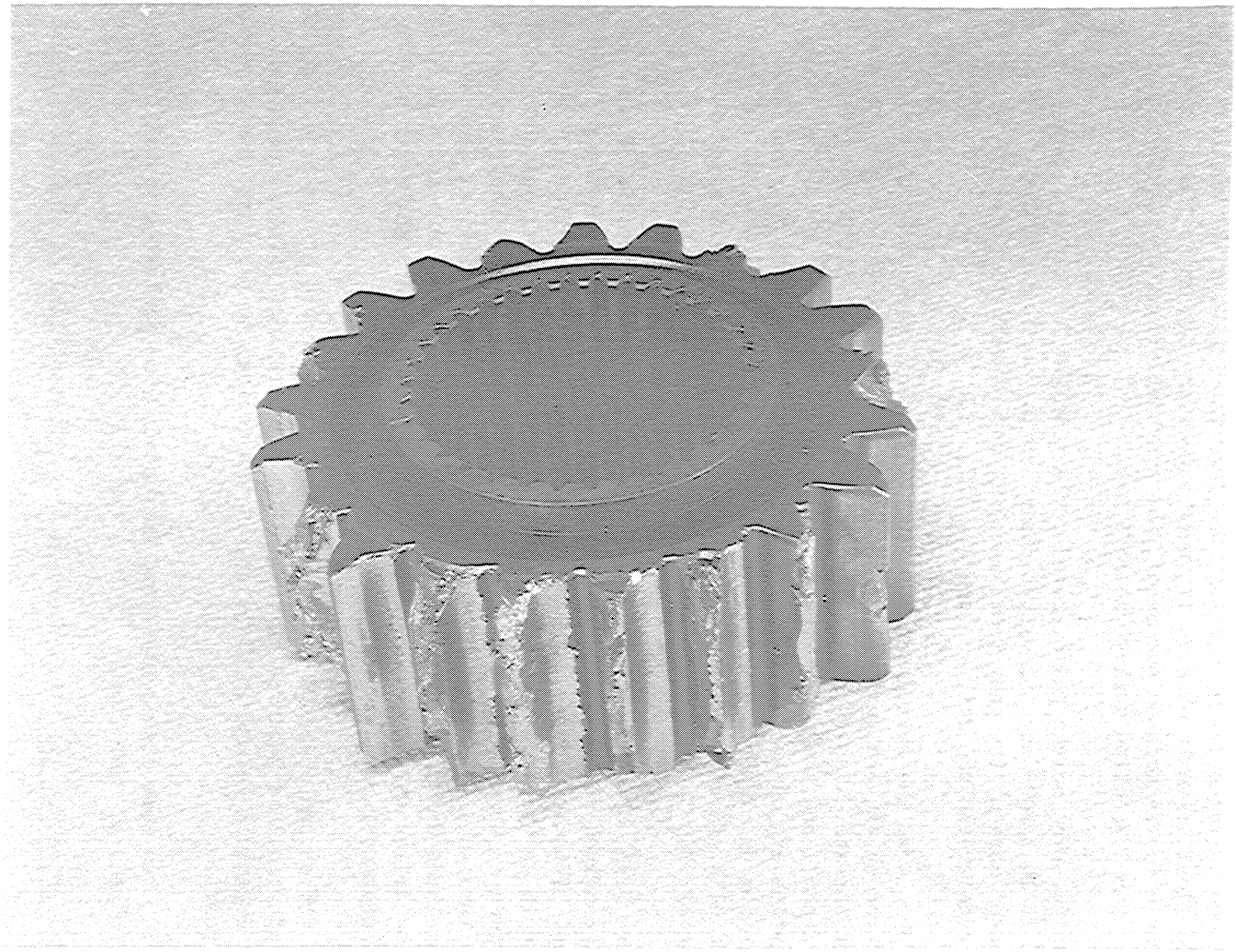

Fig. 12. Failed input pinion.

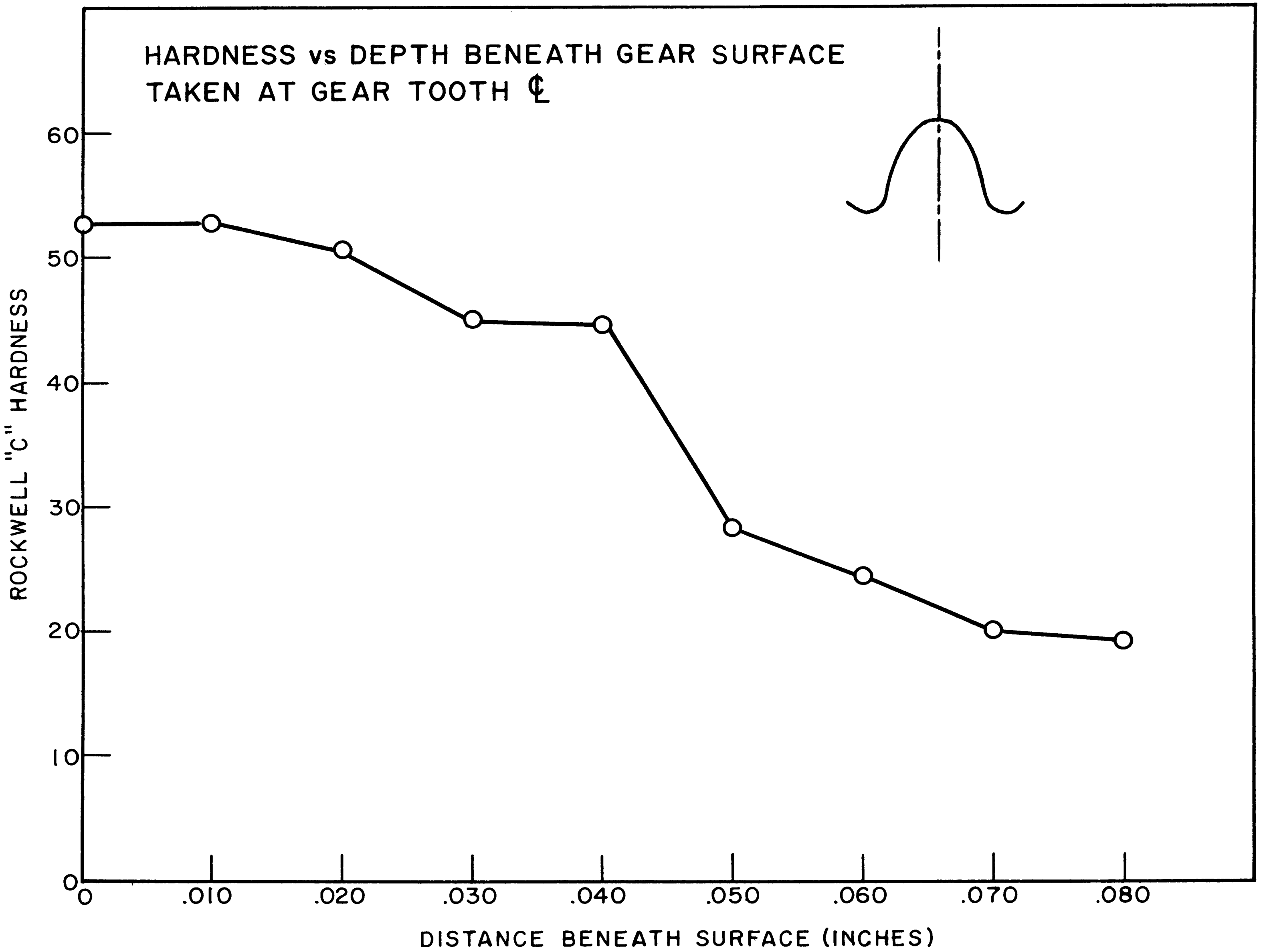

Fig. 13. Gear-hardness data.

Nonuniform contact pattern between the idler and gear, first reduction. Contact is more pronounced on the inboard side of the idler.

Nonuniform contact pattern between the pinion and gear, second reduction. The pattern is more pronounced on the inboard side.

All bearings were burnished but in good condition.

Final Drive No. 5. 20.1 hours at the life-test load of 5000 ft-lb and 650 rpm.

A slight amount of wear was found at the tips of the input pinion teeth on the outboard side of the pinion.

Fairly uniform contact patterns between all gears.

All bearings burnished but in good condition.

The following load schedule, with pertinent remarks, applies to the entire project.

Speed, rpm	Torque, ft-lb	Power, hp	Duration, min	Remarks
600	0	0	120	(A)
600	0	0	120	
600	0	0	120	
400	1000	76	60	
600	1500	171	60	
600	3000	343	30	
650	5000	618	10	
800	0	0	47	(B)
800	500	76	30	
800	1000	152	30	
800	1500	228	30	
800	2000	305	30	
800	2500	381	30	
800	3000	457	15	
800	3500	533	21	
650	4000	495	14	
650	4500	557	15	
650	5000	618	10	
650	2000	247	15	
650	3000	371	20	
800	3500	533	8	
650	2000	247	43	
650	3000	371	30	
650	4000	495	20	
650	5000	618	15	
800	2000	305	60	(C)
800	3000	457	20	
800	3000	457	143	
800	3500	533	127	
800	0	0	16	(D)
800	500	76	15	
800	1000	152	15	

(A) Final drives Nos. 5 and 6 installed.
(B) Input pinion in No. 5 replaced with irradiated pinion "A."
(C) Pinion "A" replaced with original pinion.

Speed, rpm	Torque, ft-lb	Power, hp	Duration, min	Remarks
800	1500	228	15	
800	2000	305	15	
800	2500	381	15	
800	3000	457	15	
800	3500	533	15	
650	4000	495	15	
650	4500	557	15	
650	5000	618	20	
650	4500	557	20	
650	4750	588	27	
650	4900	606	20	
800	3500	533	46	
800	3000	457	20	
1000	2500	476	30	
650	5000	618	75	
650	5000	618	141	(E)
800	0	0	15	(F)
800	500	76	5	
800	1000	152	5	
800	1500	228	5	
800	2000	305	5	
800	2500	381	5	
800	3000	457	5	
800	3500	533	10	
650	4000	495	10	
650	4500	557	10	
650	5000	618	120	
650	5000	618	180	
650	5000	618	180	
650	5000	618	120	
650	5000	618	270	
650	5000	618	120	(G)

(D) Pinion in No. 5 replaced with irradiated pinion "B."
(E) Input pinion in No. 6 failed.
(F) Final drive No. 14 installed.
(G) T-1200 transmission failure.

CONCLUSIONS

The supply of lubrication oil furnished by the pump cluster, when operated at 2100 rpm, is adequate.

The wear patterns indicate nonuniform contacts between the pinions and gears.

The drives will operate without external cooling at 265°F, when loaded at 3500 ft-lb and at 800 rpm.

The input pinion wear rate is zero at 5000 ft-lb and 650 rpm once break-in has occurred.

If the gear specifications are maintained, the expected life of the final drives, at the maximum load of 5000 ft-lb and 650 rpm, is better than 20 hours.

An input pinion that did not meet the metallurgical specifications failed after operating only 4.5 hours at 5000 ft-lb and 650 rpm.

RECOMMENDATIONS

The diametral clearance between the cover projection and the input shaft pilot hole should be increased to at least 0.020 in. to prevent seizing and damage to the oil seal-ring.

Metallurgical and dimensional specifications must be maintained if long life is expected.