ENGINEERING RESEARCH INSTITUTE
THE UNIVERSITY OF MICHIGAN
ANN ARBOR

Final Report

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

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Project 2385

INGERSOLL KALAMAZOO DIVISION
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KALAMAZOO, MICHIGAN
SUBCONTRACT NO. 1615, U. S. NAVY CONTRACT NObs-3600

July 1956

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ABSTRACT

This report covers the testing and evaluation of final drives for the LVTP-5 type of amphibious vehicle.

The supply of lubricant furnished by the pump cluster, when operated at 2100 rpm, is adequate whether the oil is hot or cold.

The wear patterns indicate little or no shaft deflection.

The drives will operate without external cooling at 271°F, when loaded at 2200 ft-lb and 800 rpm.

Gear-tooth surface wear is low at maximum load.

The expected life of the drives, in their present design, is low at the maximum load.

OBJECTIVE

To test and evaluate final drives for the LVTP-5 type of amphibious vehicle.

INTRODUCTION

This report on Project 2385 covers the testing and evaluation of spur-gear final drives as designed and developed by the Ingersoll Kalamazoo Division of the Borg-Warner Corporation (hereinafter referred to as the Design Agent) for application to the LVTP-5 type of amphibious vehicle.

ORIGIN OF PROJECT 2385

Project 2385 originated through authorization of the Design Agent's Subcontract No. 1615, under the United States Navy Contract NObs-3600, to the Engineering Research Institute, The University of Michigan, Ann Arbor.

SCOPE OF PROJECT 2385

The scope of Project 2385 is limited to the intent of Part 2 of the Design Agent's Project 3600-10. Project 3600-10 covered three phases in the testing and evaluation of the spur-gear final drives, namely:

<u>Part 1</u>: Testing and evaluation of two drives in a vehicle at the Design Agent's testing facilities.

<u>Part 2</u>: Dynamometer testing of two drives at a suitable laboratory. (See Appendix I for Test Agenda.)

<u>Part 3</u>: Testing and evaluation of four drives in two vehicles at the U. S. Marine Corps Test and Experimental Unit, Camp Pendleton, California.

GENERAL DESCRIPTION OF EQUIPMENT AND TEST PROCEDURE

DESCRIPTION OF DRIVES TESTED

The general design of the two spur-gear final drives submitted by the Design Agent for test and evaluation combines the final drive and drop gear into a single unit, utilizing spur gears, and a hydraulic 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 inches below the input yoke. The arrangement of components in the new drive unit is shown in Fig. 3, Appendix III.

TEST EQUIPMENT

The final drives were mounted, with output shafts 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 would be at a high enough speed to be absorbed by the dynamometer; also, both drives may be tested at the same time. The port unit input was connected to a T-1200 transmission which was driven by an Allison V-1710 engine. The input shaft of the starboard unit was connected to a 2000-hp, eddy-current dynamometer, Fig. 1, Appendix III.

Lubrication and clutch-pressure oil was supplied by the standard LVT pump cluster, which was driven by an electric motor. During the gear-wear portion of the test, lubricating oil for the radioactive port unit was supplied by a separate gear pump, electrically driven. Radioactive oil was scavenged by a separate gear pump. See Fig. 2, Appendix III.

All instruments and controls were mounted remotely in the control room. Two Hewlett-Packard electronic counters were used to measure speed and revolutions of the drives. This gave an accurate determination of both input and output speeds and any possible clutch slippage. Dynamometer torque was measured with a Link Engineering Co. Unibeam, with the dial-type indicator located on the control panel. Thermocouples were installed at seven critical locations in each drive. The locations are shown in Fig. 3, Appendix III.

The intensity of radiation (hence the amount of metal) in the oil was detected by a Nuclear Chicago Scintillation pick-up head, indicated on a Tracerlab Rate Meter and recorded on an Esterline-Angus recording milliammeter.

TEST PROCEDURE

Tests of the spur-gear final drives were conducted in accordance with the "Agenda for Spur Gear Final Drive Test at University of Michigan at Willow Run dated 10 June 1954. A copy of this test agenda will be found in Appendix I. Upon reference to the test agenda, it will be noted that the procedure for testing is 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.

During the progress of the tests, a Load Schedule was prepared, showing loads on the drives in relation to drive speeds, duration of test runs, and accumulated time for the tests. This Load Schedule, covering the entire test program, will be found in Appendix II.

DISCUSSION OF TESTS AND RESULTS

In the following discussion of tests of the final drives and results obtained therefrom, a description is given of the method used in conducting tests under each phase of the test program, followed immediately by a discussion of test results.

I. OIL FLOW THROUGH FINAL-DRIVE ORIFICES

Description.—Each orifice was removed from the final drive and the time required for a measured quantity of oil to flow was recorded. The flow measurements were conducted under the following conditions:

Lubricant	Temperature	Pressure
SAE 30	85°F	20, 25, 30 psi
EP 80	130°F	18 psi
EP 80	240°F	12 psi

Test Results.—The oil flow rates through the final-drive orifices are shown in Table I below:

TABLE I

OIL FLOW MEASUREMENTS

Conditions			Flow in gpm, Pump Speed 2100 rpm		
			Gear-Mesh Spray Jets	Seals and Bearing Lube	Bearing Lube
SAE 30	85°F	20 psi	.76	.29	.15
SAE 30	85°F	25 psi	.94	.34	.19
SAE 30	85°F	30 psi	1.1	.39	.22
EP 80	130°F	18 psi	1.3	.43	.28
EP 80	240°F	12 psi	1.1	.38	.25

During the high-temperature measurements, no oil was discharged through the relief valve; therefore, the jets were receiving the entire capacity of the pump. Pump-cluster input speed during all tests was 2100 rpm.

II. FLUSH LUBRICATION SYSTEM

Description.—

A. The oil reservoir was filled to the proper level with SAE 30 oil.
B. The drives were operated at an input speed of 600 rpm with no load for two hours. The filters were checked each hour.
C. The system was drained and the filters cleaned.
D. Steps A, B, and C were repeated for a total of three times.
E. The oil reservoir was filled to the proper level with EP 80 oil.

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

III. BREAK-IN PHASE

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

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

Test Results.—The drives operated satisfactorily during the above loads.

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.

Test Results.—The wear pattern extended evenly across the tooth width, indicating little or no deflection of the shaft. All gears, bearings, and oil rings were in good condition.

V. WEAR PATTERN AT MAXIMUM LOAD

Description.—The drives were operated for 10 minutes at 800-rpm input and 5000-ft-lb torque. The gears and bearings were inspected after the above run.

Test Results.—All gears and bearings were in good condition and the wear pattern still extended evenly across the tooth width.

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

Description.—The oil coolers were removed from the lubrication system. The units were driven at an input speed of 800 rpm and the torque increased until a stabilized temperature of 271°F was reached in the oil sump. The final load was maintained for 30 minutes to insure that the stable state had been attained.

Test Results.—Before the drives were reassembled for the stabilized-temperature runs, it was noticed that one inner race on an intermediate shaft was displaced 1/8 inch toward the end of the shaft and was loose enough to be turned by hand. The inner race could not be turned when it was forced back into position on the shaft. After consultation with the Design Agent, it was decided to attempt the stabilized-temperature runs without reworking the inner race and shaft.

After a temperature of 250°F was reached in the final-drive sump, two inner races heated up rapidly, indicating that the races were slipping on the intermediate shaft. One of the slipping races was the race which had been loose before the test was started. The inner races were removed from both units and were chrome plated and ground to insure a .002-inch interference fit.

With an ambient temperature (measured at a point between the two final drives) of 100°F, the sump temperature stabilized at 271°F while the drives were loaded to 2200 ft-lb at 800 rpm.

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 loads and speed are varied and very low wear rates may be measured. Wear rates for many operating conditions may be determined in a short length of time and without visual inspection of the gears between load conditions.

An extra input pinion was obtained from Western Gear Works and was made from steel conforming to AISI 8620 specifications, which call for

C	.18 to .23%
Mn	.70 to .90
P	.040 max.
S	.040 max.
Si	.20 to .35
Ni	.40 to .70
Cr	.40 to .60
Mo	.15 to .25

After the final manufacturing process, the pinion was given a coating of Parker Lubrite.

A fine, high-speed grinding wheel was used to grind about 6 grams of metal from one side of the pinion for calibration purposes. This powder was collected by a magnet placed close to the off side of the grinding wheel. Samples of this powder were carefully weighed, wrapped in aluminum foil, and sent with the pinion to the reactor at Idaho Falls, Idaho, so that the pinion and the samples should receive the same neutron bombardment.

Examination of the sample packets, upon return of the pinion from the reactor, disclosed that some of the iron powder had been washed away during the underwater loading process used at the Idaho Falls reactor. It was decided to grind additional powder from the gear rather than attempt to use the remaining portions of the original samples.

The activated gear was chucked in a lathe, using long-handled tools, and approximately 1.5 grams of iron were removed, using a tool post grinder. The samples ground from the side of the gear had the same intensity of radiation as the gear-teeth surface. The activated powder was divided into three samples. The first sample weighed 102.4 mg, the second 350.6 mg, and the third 675.0 mg.

The calibration system, Fig. 4, Appendix III, was filled with 1.5 gal of oil and readings were taken to establish the background activity level. With the oil circulating, the smallest sample, containing 102.4 mg, was added to the system along with 1/2 gal of oil. The calibration system was checked periodically until the activity load became constant, indicating an even distribution of the powder in the oil. This reading was recorded and then the 350.6-mg sample was added with another 1/2 gal of oil. The system was again allowed to stabilize and the activity level recorded. Finally the 675.0-mg sample was added with another 1/2 gal of oil and a final stabilized reading taken. The preceding calibration procedure was followed for both chambers. From these data, the curve of Fig. 5, Appendix III, was made, showing activity vs milligrams of metal per gallon of oil. With this plot, any reading of activity in the identical counting chambers used in the lubrication oil system of the final drive can be converted to milligrams of metal per gallon of oil. If the tests cover a period of several days, it is necessary to apply corrections for the decay of radioactivity. This can be done by determining the calibration system's activity and drawing through that point and the origin a curve similar to the original curve of activity vs weight of metal.

The activated pinion was then installed in the port final drive with special long-handled tools. The lubricant used throughout the test was Texaco Universal Gear Lubricant EP 80. The lubrication system was filled with 5 gal of oil and the drives were run at 800 rpm. The drives were run at a constant load until a definite wear rate was shown on the Esterline-Angus chart recorder. The loads and duration of time at each load are shown below:

Load Schedule During Gear Capacity Runs

Load (ft-lb)	Time (minutes)
500	30
1000	30
1500	60
2000	60
2500	60
3000	120
3500	60
4000	60
4350	50
Bearing failure in starboard unit	
4500	30
4000	55
5000	20
4000 (650 rpm)	90
5000 (650 rpm)	45

Test Results.—The gear-tooth surface wear ratio and the total amount of metal worn off the tooth surface are shown in Fig. 6, Appendix III.

It is to be noted that the initial wear rate is high, even at low loads; but once break-in has occurred, the wear rate is almost negligible. After break-in occurred, the entire system was flushed to remove all metal particles, thus permitting the use of the most sensitive activity scales available. These scales indicated the final wear rates of 6.9 mg per hour at 4000 ft-lb and 13.9 mg per hour at 5000 ft-lb.

The high initial wear rates may be attributed to surface irregularities and to the fact that the extreme pressure compounds had not then formed on the gear-tooth surfaces.

Radiographs taken of the gear mating with the activated pinion showed that no metal had transferred from the pinion to the mating gear. The total absence of transferred metal also indicated that the gear teeth were meshing across the full width of the teeth with no deflection of the shafts.

Previous wear-rate investigations (Final Report No. 2138-3-F, Wear Rates of Final-Drive Sun Gear by Radioactive Method, The University of Michigan, Engineering Research Institute, June, 1954) discovered the sun-gear wear rate at 800 rpm and 5000 ft-lb torque, using EP 80 lubricant, to be 42 mg/hour or approximately three times the wear rate of the pinion gear investigated in this report. The average life of a sun gear in the field is estimated to be approximately 500 hours.

VIII. GEAR AND BEARING LIFE

Description.—The drives were to be operated at 800 rpm and 5000-ft-lb torque. It was found that the driving source could not be operated for extended periods of time at this load. The drives were operated at 650 rpm and 5000-ft-lb torque until failure.

Test Results.—The gears showed little or no wear at any stage of the program. The bearings of the intermediate shaft (gear, first reduction: pinion, second reduction) were a constant source of trouble throughout all phases of testing. Even though the inner races of these bearings had been pressed onto the shafts with an interference fit, slippage of the inner race on the shaft did occur. The first slippage was noticed after running a total of 14 hours and 40 minutes, under fairly light loads. At this time, all inner races were reworked to obtain a .002-inch interference fit. After 12 hours and 20 minutes under loads ranging from 500 to 4500 ft-lb, the bearings slipped again. Also at this time, one roller in the starboard unit bearing failed. Metal particles from this roller were impressed in the races of the bearing. New bearings were installed at this time and again the shafts and inner races were reworked to obtain a .002-inch interference fit.

All bearing slippage occurred after the temperature of the case adjacent to the bearing reached 180°F. Below this temperature, bearing slippage did not occur.

After 5-3/4 hours at loads ranging from 4000- to 5000-ft-lb input torque, the intermediate shaft failed on the outboard end where it necks down to receive the spacer behind the bearing inner races (see Fig. 14, Appendix III). This failure ended testing of the final drives in their present design.

Several times through the testing, difficulties were encountered due to the backing out of the helicoil inserts for the bolts holding the shaft housing to the outer housing. It is possible that the inserts were not driven deeply enough into the tapped holes, so the bolt ends caught the ends of inserts where the driving tong is removed.

CONCLUSIONS

The supply of lubricant furnished by the pump cluster, when operated at 2100 rpm, is adequate, whether the oil is hot or cold.

The wear patterns extend evenly across the gear-tooth width, indicating little or no deflection of the shafts and good alignment.

The drives will operate at a stable temperature of 271°F without oil coolers, when running 800 rpm and loaded at 2200-ft-lb torque.

Gear-tooth surface wear is very low (13.9 mg/hour) at the maximum load of 5000-ft-lb input torque and 650 rpm. The expected life of the final drives, in their present design, is very low at loads of 5000-ft-lb input and 650 rpm. Intermediate pinion shaft failure occurred after a total of 165 minutes at this load and speed. The total running time of the drives before failure may be distributed as follows:

25 hours at loads less than 3500-ft-lb input torque
8-1/2 hours at loads greater than 3500-ft-lb input torque.

RECOMMENDATIONS

From the results of the life test and previous bearing failures, it is apparent that a redesign of the shaft and bearings of the pinion, second reduction (Parts No. 29 and 75), is necessary.

An investigation should be made of the possibility of replacing or improving the helicoil inserts used in the drive housings.

APPENDICES

APPENDIX I

INGERSOLL PRODUCTS DIVISION
Borg-Warner Corporation
Kalamazoo, Michigan

Agenda for Spur Gear Final Drive Test
at
University of Michigan at Willow Run

June 10, 1954

Two (2) external meshing gear final drives are to be installed at Willow Run for stationary testing. These final drives will contain the steel output shafts and National face type outboard seals. The following agenda is to act as a guide in the testing of these drives and may require revisions determined during the course of testing:

I Oil flow through final drive orifices.

A. Fill reservoir with SAE 30 oil.
B. Remove each orifice from both units.
C. Measure the amount of oil flowing thru each orifice to determine the flow rate per minute at the following input speeds:

1. 400 RPM
2. 500 RPM
3. 600 RPM
4. 700 RPM
5. 800 RPM

II Flush lubrication system.

A. Fill reservoir to proper level with SAE 30 oil.
B. Drive input at 600 RPM for 2 hours without load, checking filters each hour.
C. Drain system and clean filters.
D. Repeat A, B, and C, for a total of three (3) times.
E. Fill reservoir to proper level with EP-80 oil.

III Break-in phase

A. Maintaining oil temperatures below 225°F, operate for:

1. One (1) hour at 400 RPM input and 1000 lbs. ft. load.
2. One (1) hour at 600 RPM input and 1500 lbs. ft. load.

IV Wear pattern due to deflection.

A. Install the following thermocouples.

1. Final drive cases at each accessible bearing outer race.
2. Both final drive sumps.
3. Oil reservoir.

B. Operate drives for thirty (30) minutes at 600 RPM input and 3000 lbs. ft. load recording above thermocouple temperatures.

C. Disassemble and inspect drives.

1. Inspect all bearings.
2. Inspect all gears noting location of contact or wear pattern.
3. Inspect all oil rings.

V Wear pattern at maximum load.

A. Operate drives for ten (10) minutes at 800 RPM input and 5000 lbs. ft. load.
B. Disassemble and inspect drives.

1. Inspect all bearings.
2. Inspect all gears noting wear pattern and extent of wear.

VI Torque required to stabilize drives at a temperature of 271°.

A. At 800 RPM input speed, increase torque until a stabilized temperature of 271° is reached in final drive oil sumps and record data.

VII Gear capacity before excessive surface breakdown.

A. Radio activate one 21 tooth input gear.
B. Drain lubrication system and replace with new EP-80 oil.
C. Determine the torque at which a marked increase of gear teeth surface wear takes place with the input speed held at 800 RPM and the oil temperature held below 225°F.

VIII Gear and bearing life.

A. Replace lubrication oil with new EP-80 oil.
B. Operate drives at 800 RPM input and 5000 lbs. ft. load and oil temperatures held below 225°.
C. Disassemble and inspect drives after first, third, seventh, twentieth hour, and at failure.

IX Submit complete report.

APPENDIX II

LOAD SCHEDULE

The load schedule for the entire program was:

Torque ft-lb	Speed rpm	Time min	Total Time hr-min	Remarks
0	600	360	6-00	
1000	400	60	7-00	
1500	600	60	8-00	
3000	600	30	8-30	
5000	800	10	8-40	
1000	800	20	9-00	
1500	800	20	9-20	
2000	800	15	9-35	
3000	800	165	12-20	
2800	800	140	14-40	All intermediate shaft bearings reworked to obtain .002-in. interference fit. Two of the four inner races had slipped.
3000	800	45	15-25	
3800	800	40	16-05	
2800	800	30	16-35	
2500	800	40	17-15	
2200	800	35	17-50	
0	800	45	18-35	
500	800	30	19-05	
1000	800	30	19-35	
1500	800	60	20-35	
2000	800	60	21-35	
2500	800	60	22-35	
3000	800	120	24-35	
3500	800	60	25-35	
4000	800	60	26-35	
4350	800	55	27-30	Starboard intermediate bearing failure. Both bearings replaced and reworked to obtain .002-in. interference fit.
4000	800	30	28-00	
4500	800	60	29-00	
4000	800	90	30-30	
5000	650	165	33-15	Port pinion—2nd reduction shaft failed.

APPENDIX III

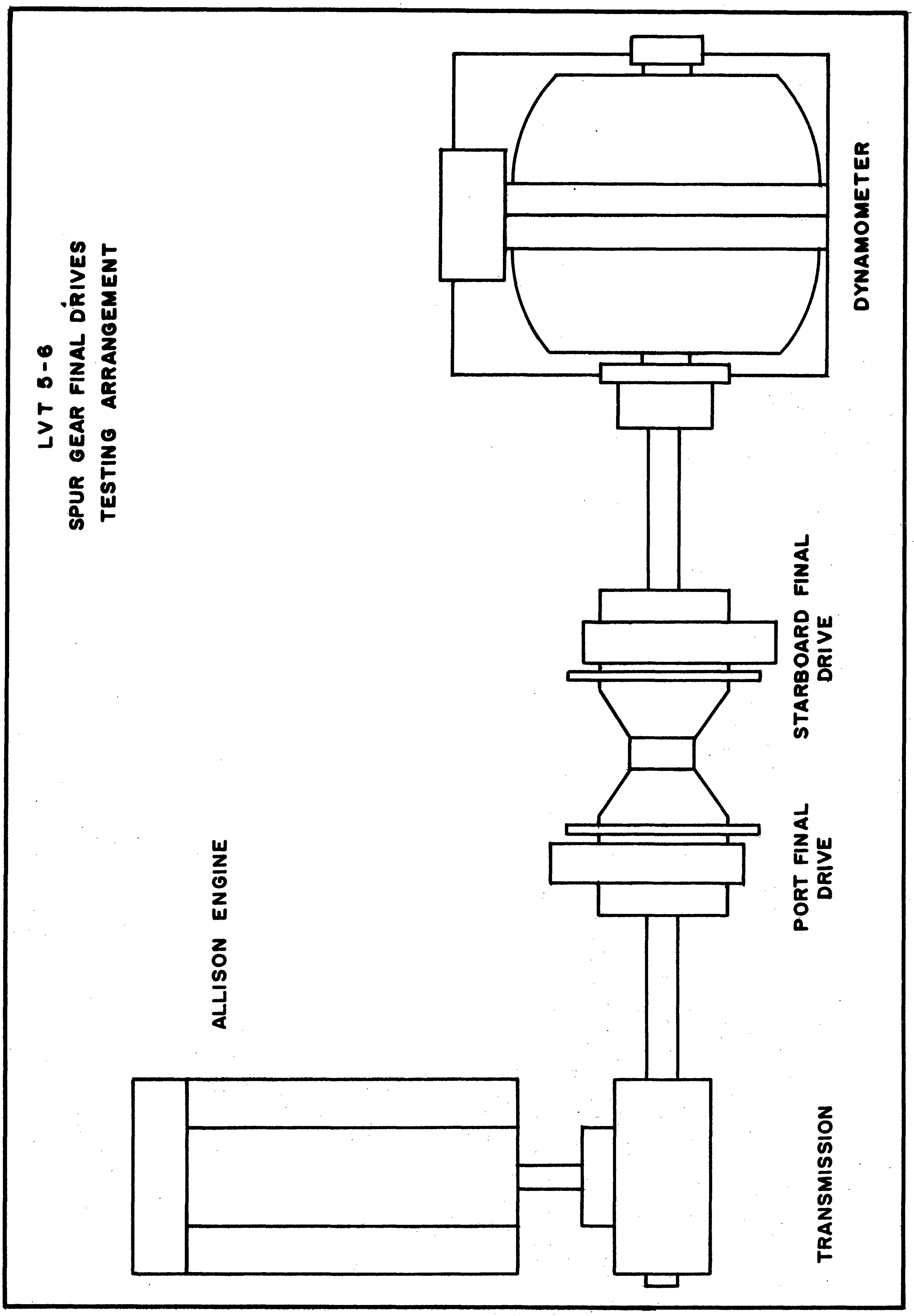
LVT 5-6
SPUR GEAR FINAL DRIVES
TESTING ARRANGEMENT
ALLISON ENGINE
TRANSMISSION
PORT FINAL DRIVE
STARBOARD FINAL DRIVE
DYNAMOMETER

Fig. 1.

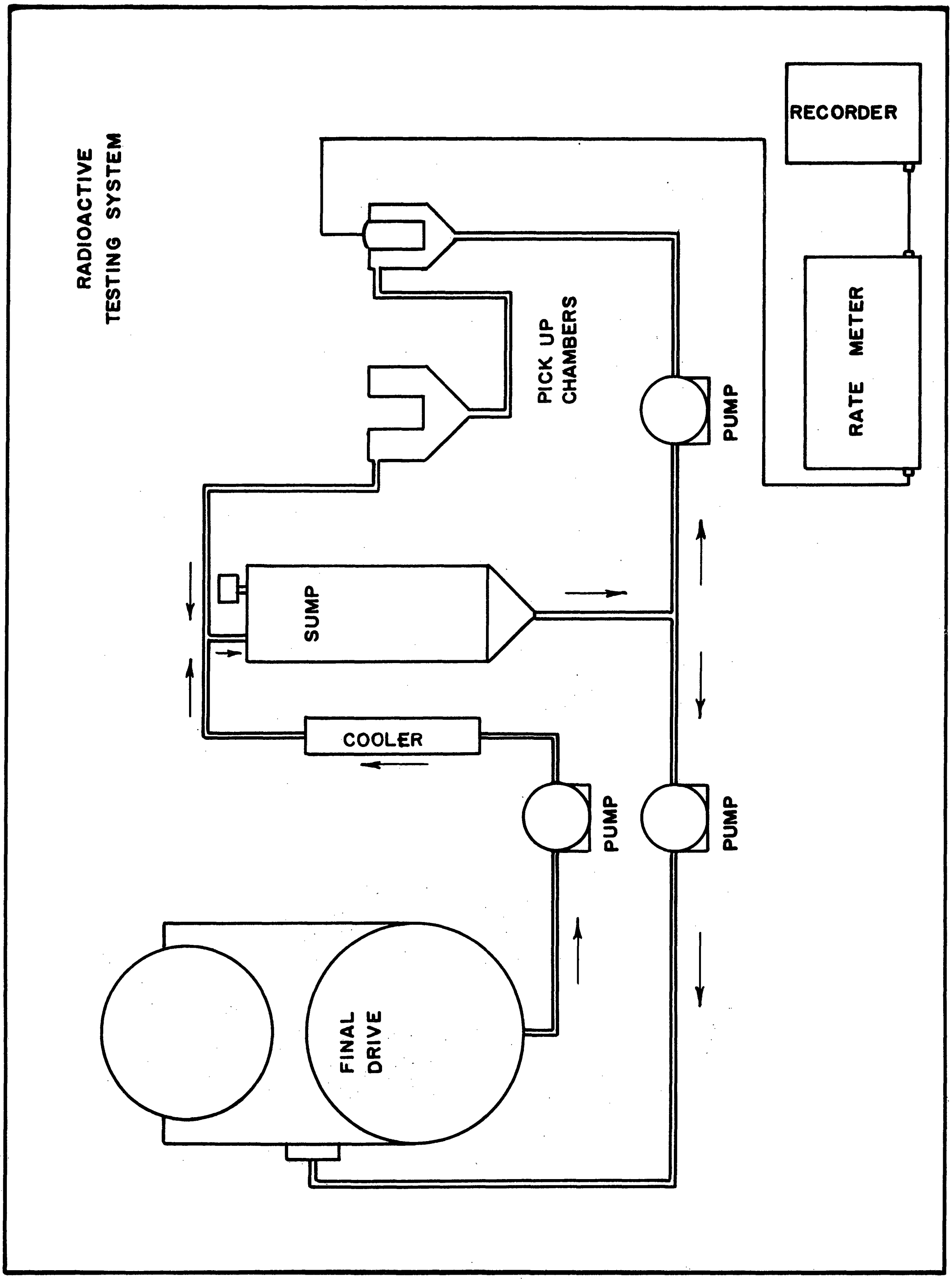
RADIOACTIVE
TESTING SYSTEM
RECORDER
RATE METER
PICK UP
CHAMBERS
PUMP
SUMP
COOLER
PUMP
PUMP
FINAL
DRIVE

Fig. 2.

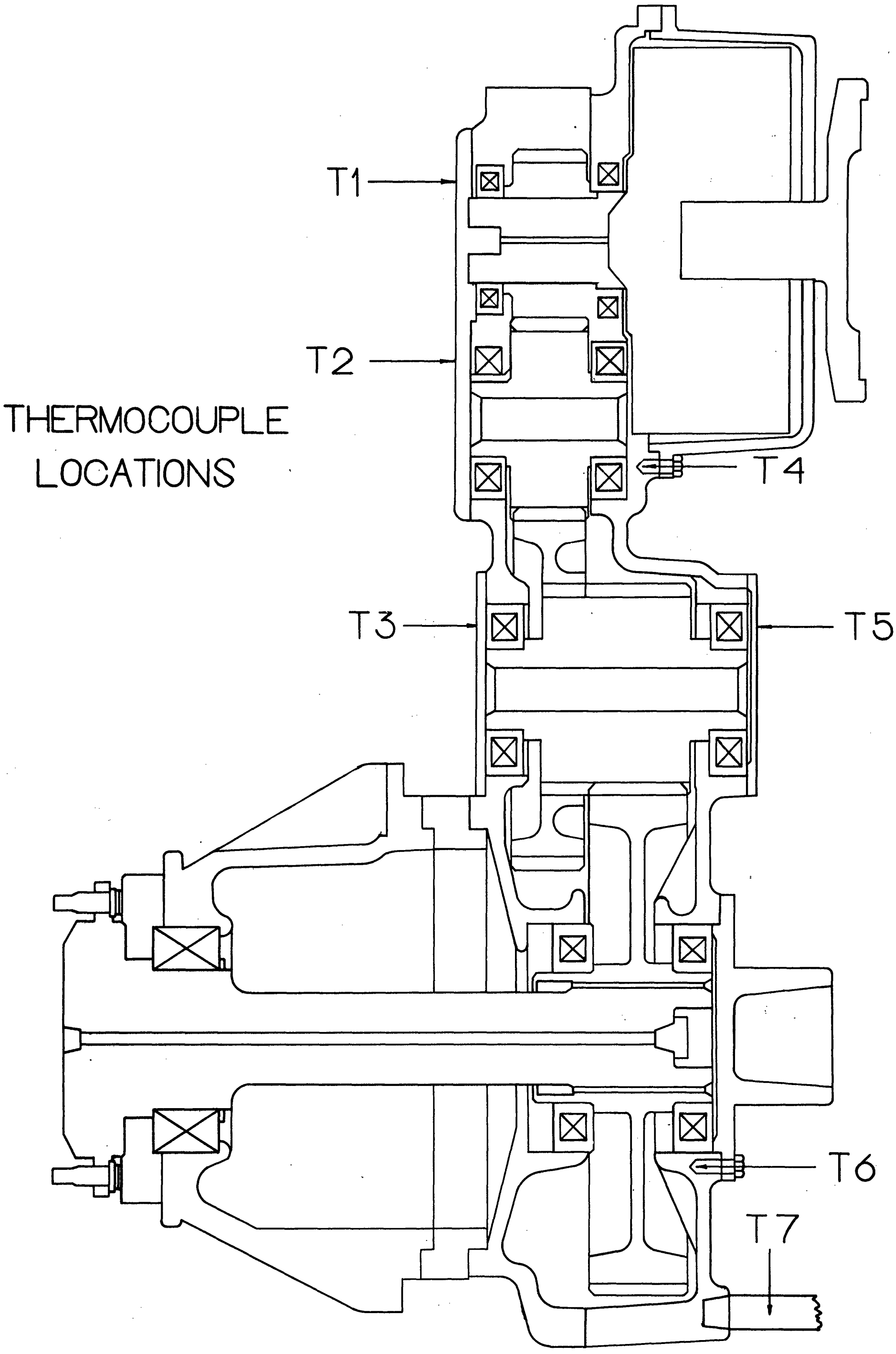

Fig. 3.

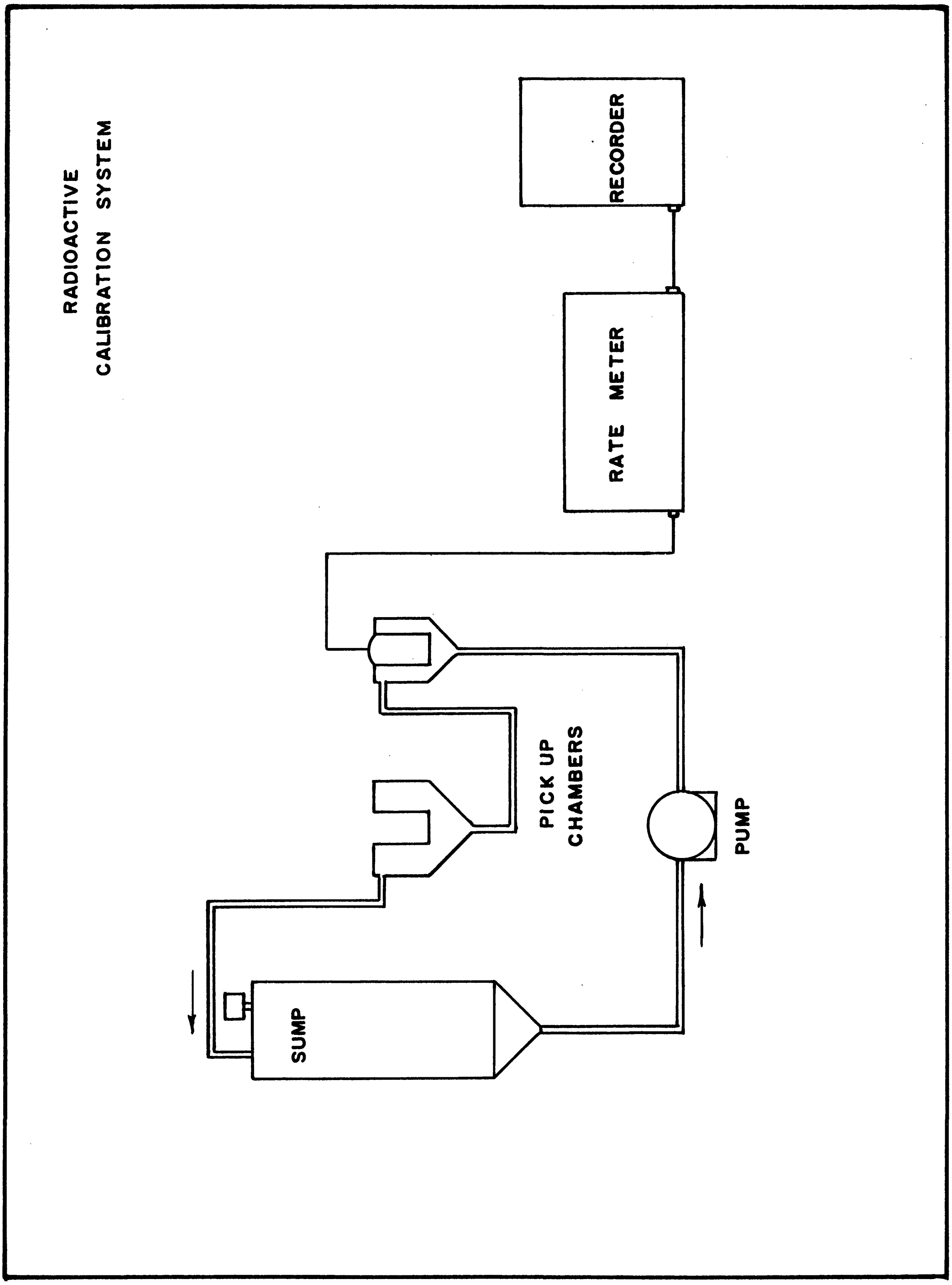
RADIOACTIVE
CALIBRATION SYSTEM
RECORDER
RATE METER
PICK UP
CHAMBERS
PUMP
SUMP

Fig. 4.

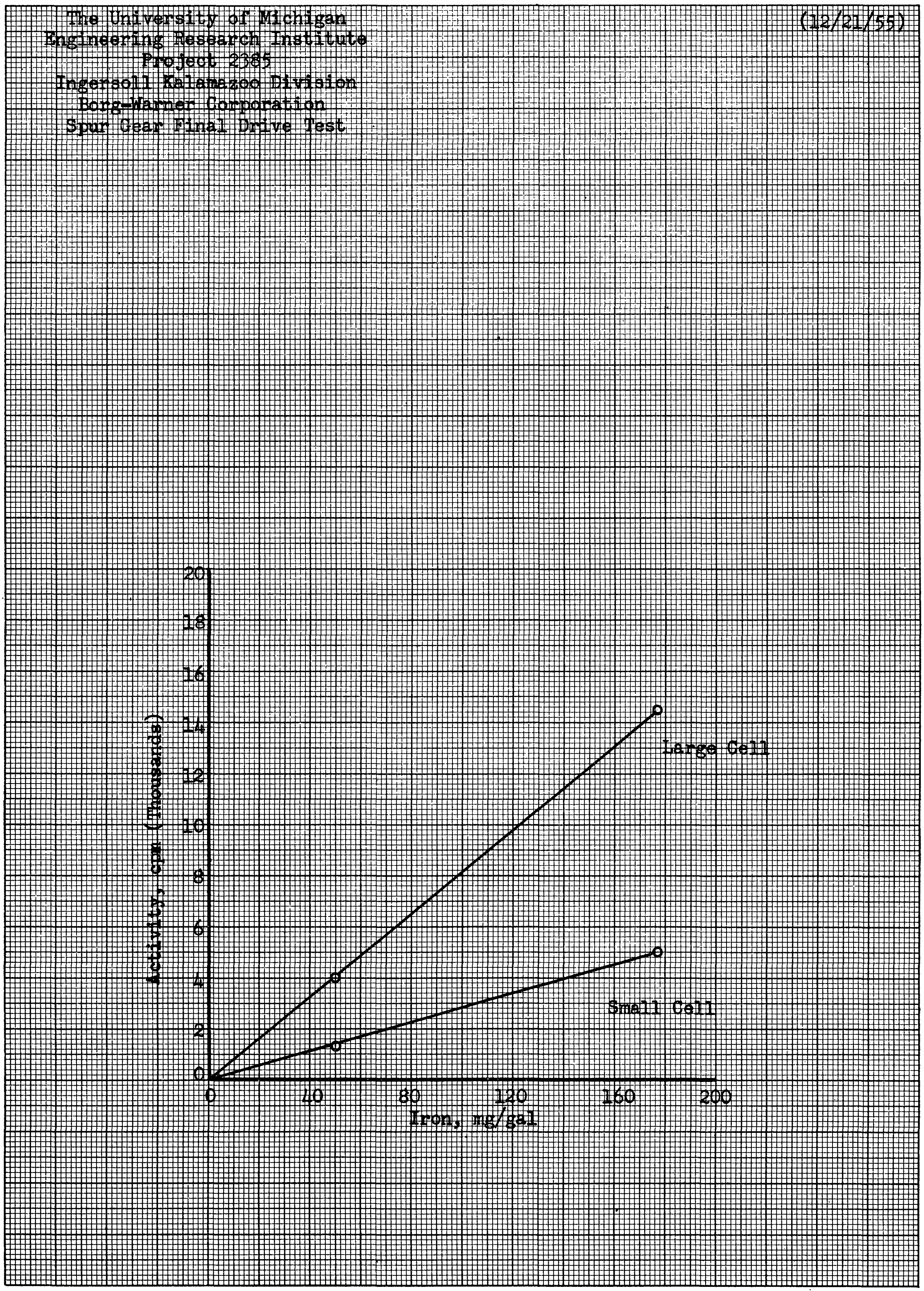
The University of Michigan
Engineering Research Institute
Project 2385
Ingersoll Kalamazoo Division
Borg-Warner Corporation
Spur Gear Final Drive Test
(12/21/55)
Activity, cpm (Thousands)
0
2
4
6
8
10
12
14
16
18
20
Iron, mg/gal
0
40
80
120
160
200
Large Cell
Small Cell

Fig. 5.

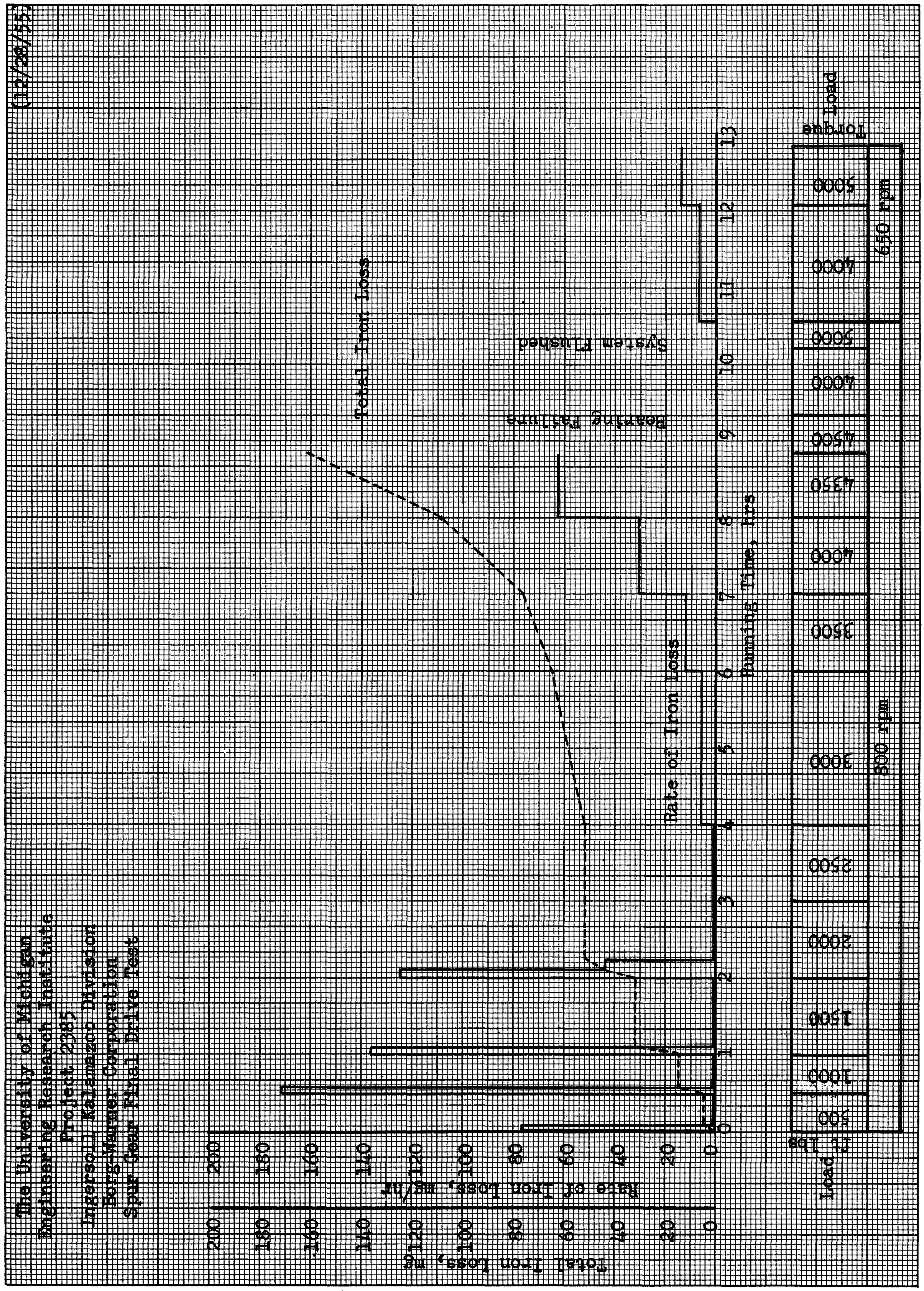
The University of Michigan
Engineering Research Institute
Project 2385
Ingersoll Kalamazoo Division
Borg-Warner Corporation
Spur Gear Final Drive Test
(12/28/55)
Total Iron Loss
System Flushed
Bearing Failure
Rate of Iron Loss
Running Time, hrs
Rate of Iron Loss, mg/hr
Total Iron Loss, mg
Torque Load
Load ft lbs
500 1000 1500 2000 2500 3000 3500 4000 4350 4500 4000 5000 4000 5000
800 rpm
650 rpm

Fig. 6.

DISCUSSION OF PHOTOGRAPHS

Figure 7 shows the Allison V-12 engine coupled to the T-1200 transmission. Some of the motors used for cooling fans and for driving oil pumps may be seen in the foreground.

Figure 8 shows the two spur-gear final drives connected together and mounted in their frames on the bed plate. Several thermocouples may be seen in bolts securing the bearing cover plates.

Figure 9 is an overall view of the test setup. The 2000-hp eddy-current dynamometer may be seen in the foreground. The standard LVT pump cluster, driven by an electric motor, appears at the right in the picture.

Figure 10 shows the control console and instrumentation used in the test. The precision ratemeter and recording milliammeter appear in the left foreground. The dynamometer controls, the Link torque meter, Hewlett-Packard electronic counters, engine controls and instruments, and the Brown potentiometer for temperature determination appear in that order, left to right, across the console.

Figure 11 shows the special pumps, oil coolers, and sump required for the radioactive wear-rate portion of the test.

Figure 12 shows the counting cells through which a sample of the lubricating oil was pumped. During actual use, these cells were surrounded with approximately five inches of lead to eliminate any stray radiation.

Figure 13 shows the calibration counting cells which were identical in construction with the counting cells used in the test. Again, in use, these cells were shielded with lead.

Figure 14 shows the shaft failure of the pinion, second reduction. Note that the failure occurred at the necked down portion of the shaft, where the spacer ring is located.

Figure 15 shows the damaged gear teeth due to shaft displacement after the shaft failure.

Figure 16 shows damage done to idler gear when the intermediate shaft was displaced.

Figures 17 and 18 show case damage due to shaft failure.

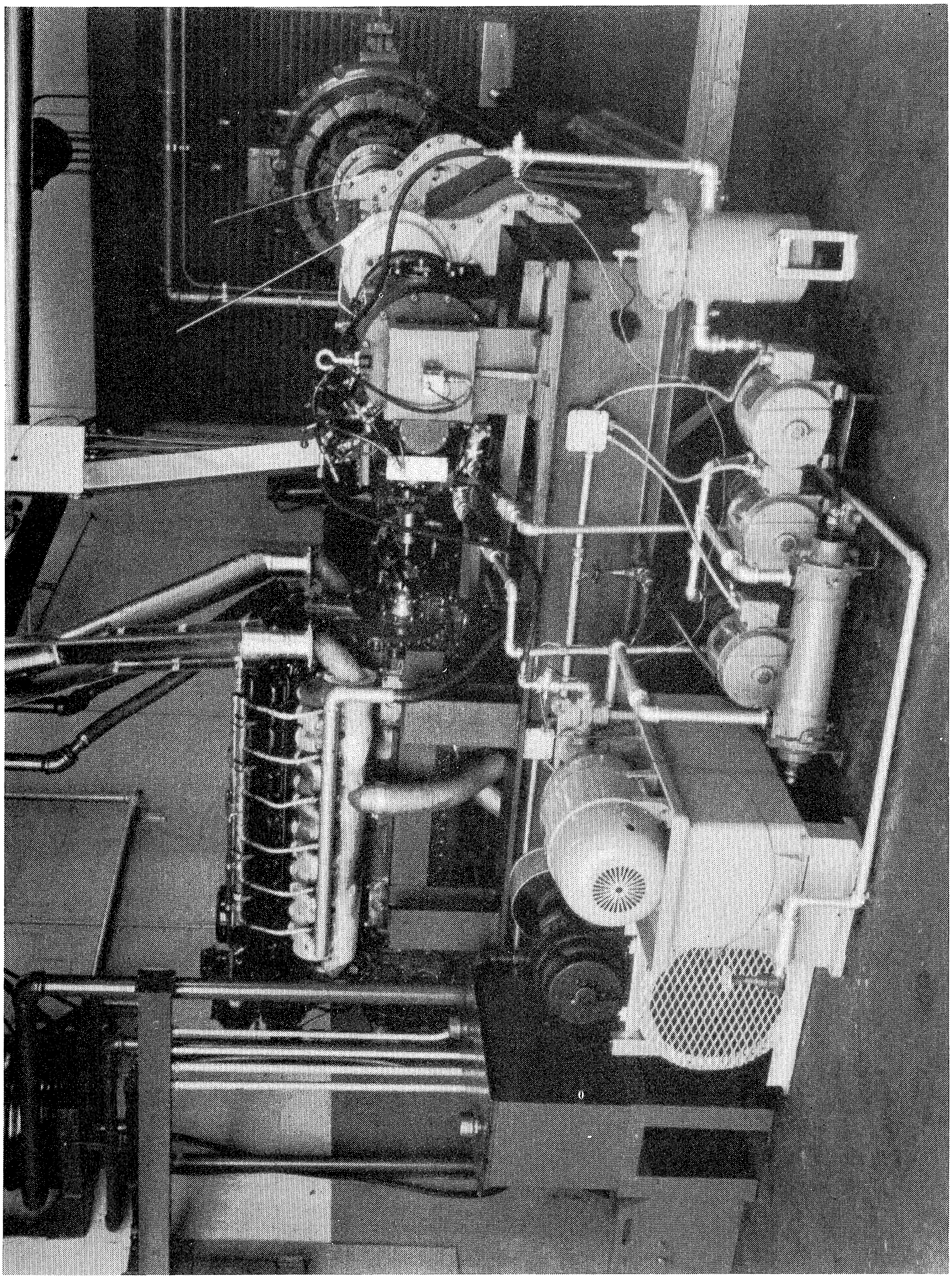

Fig. 7.

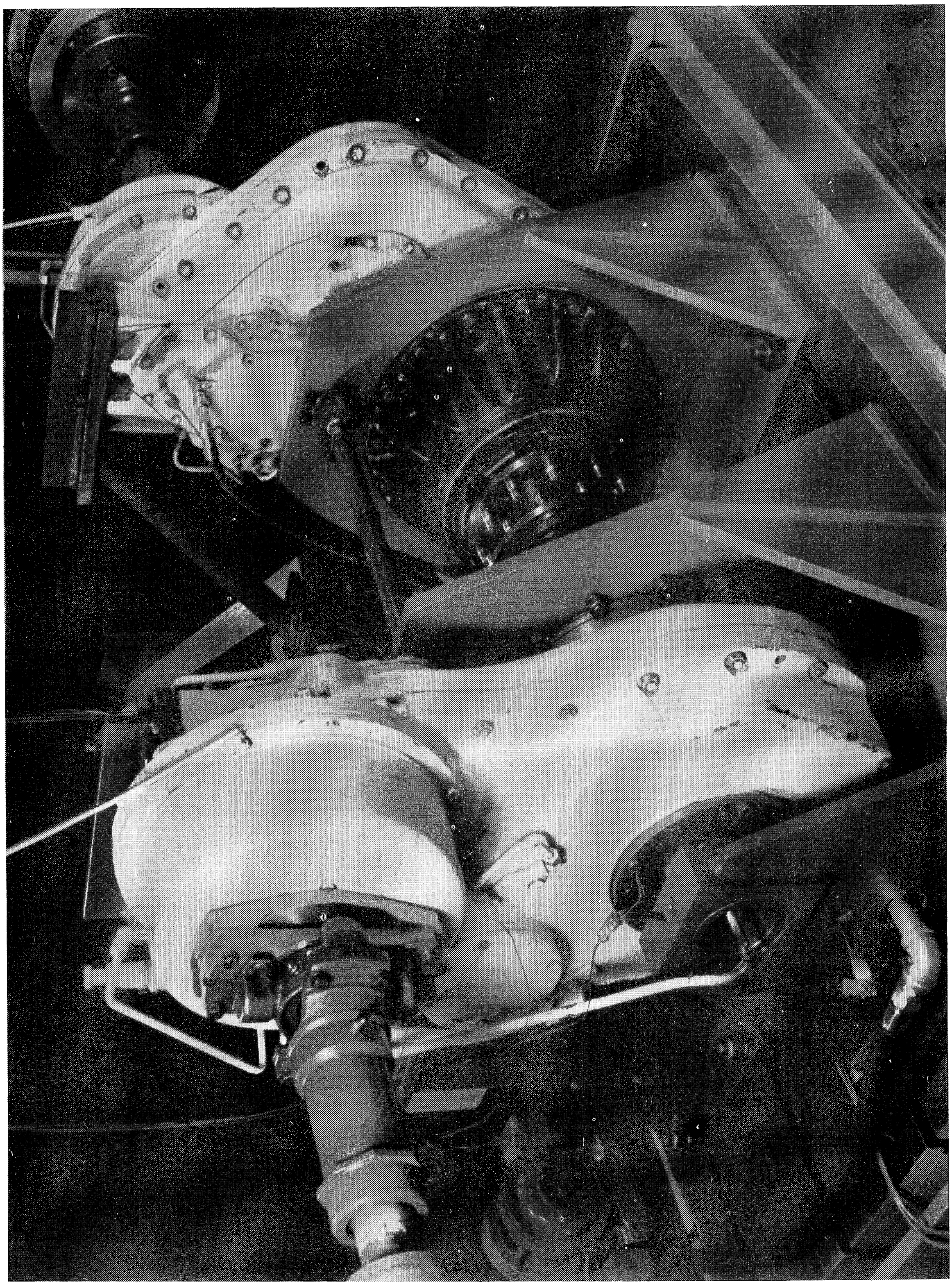

Fig. 8.

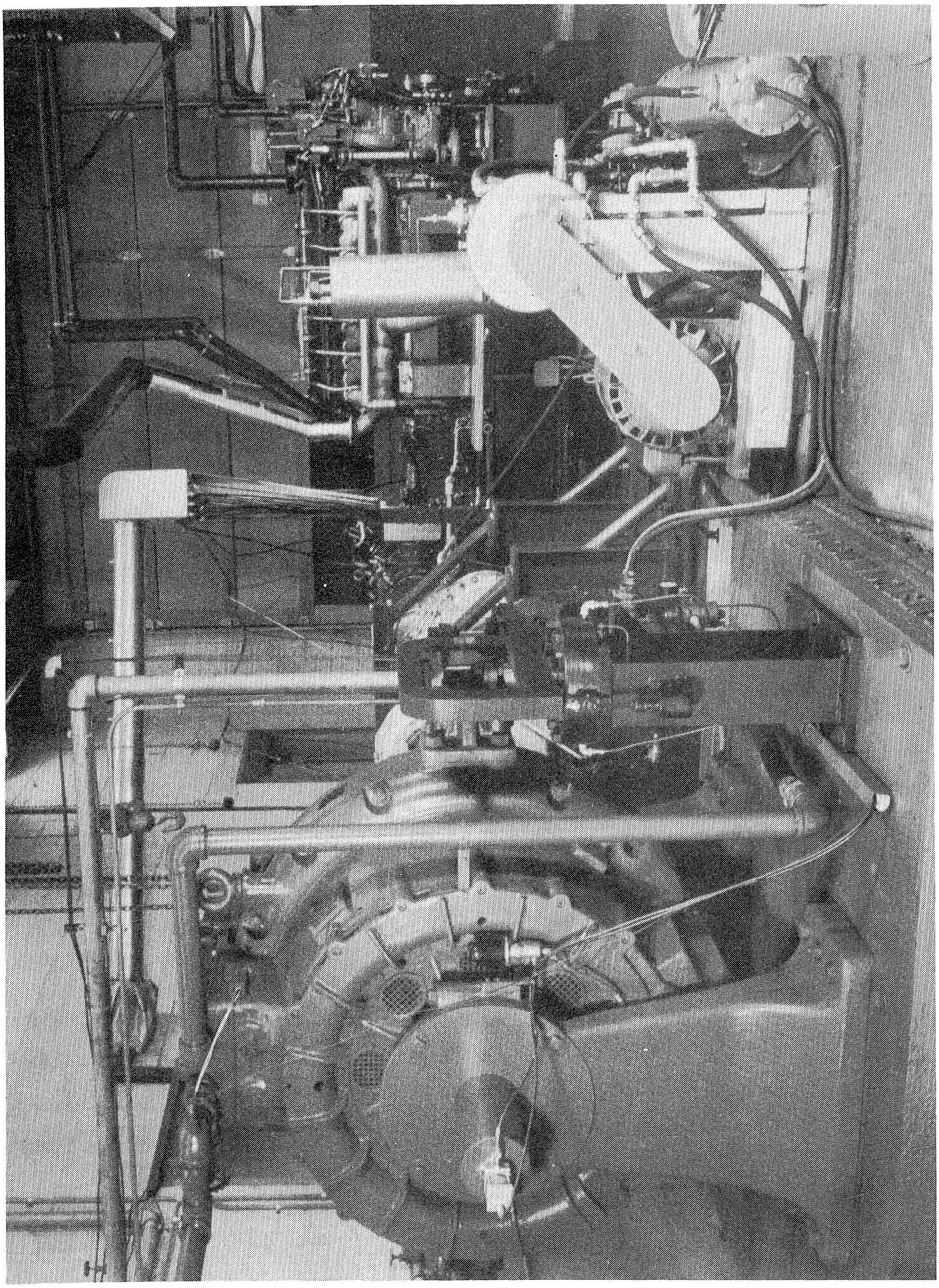

Fig. 9.

Fig. 10.

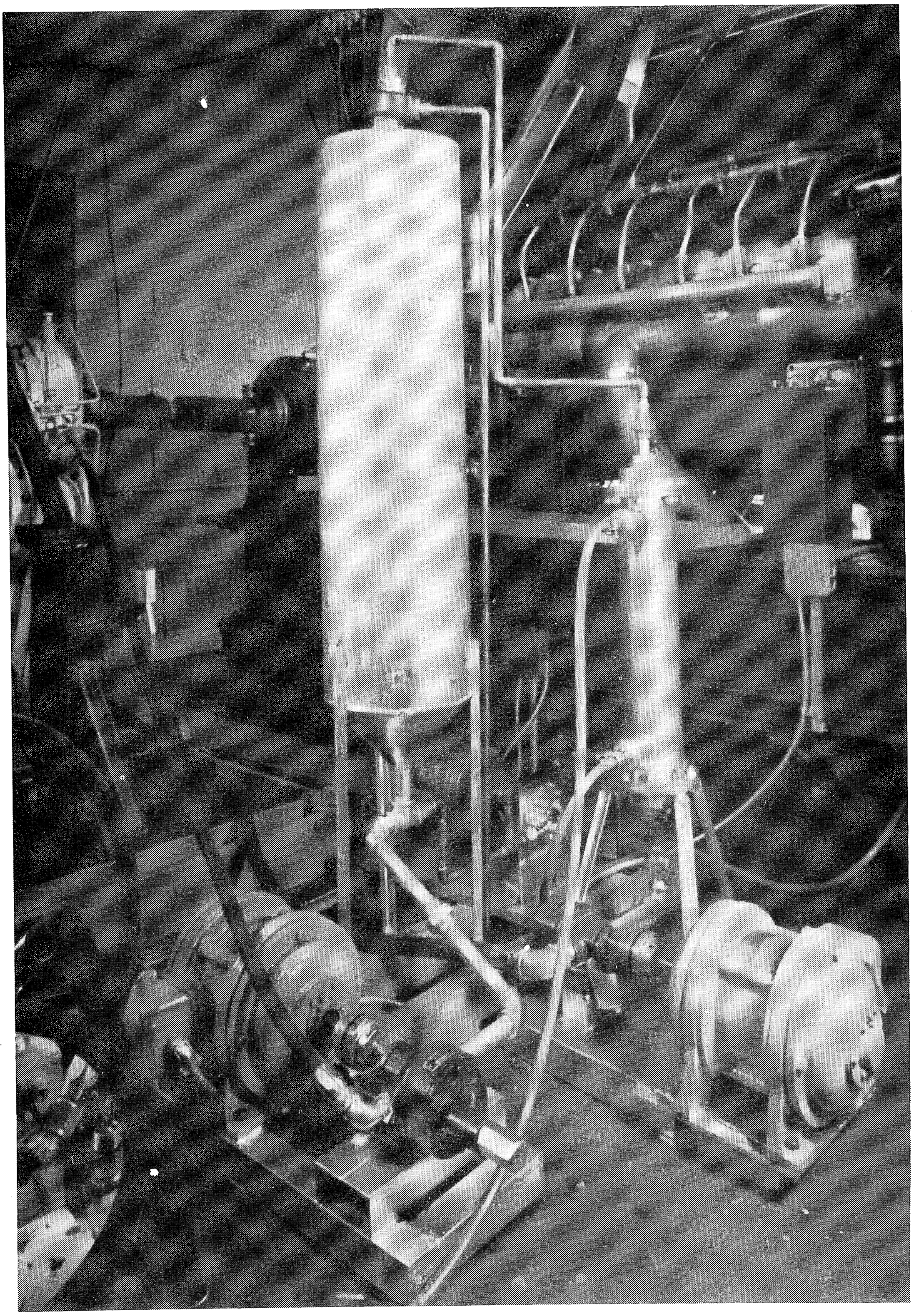

Fig. 11.

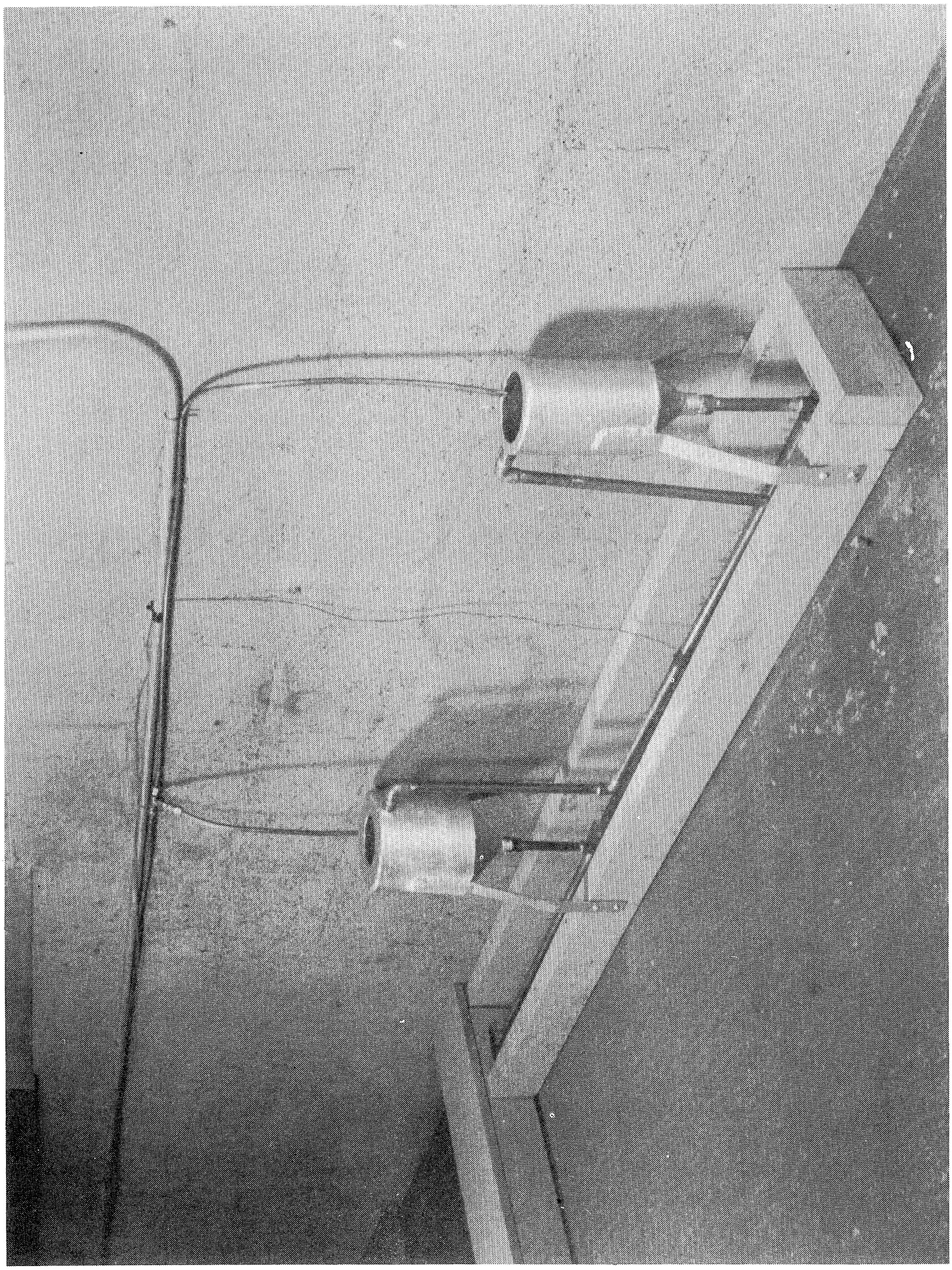

Fig. 12.

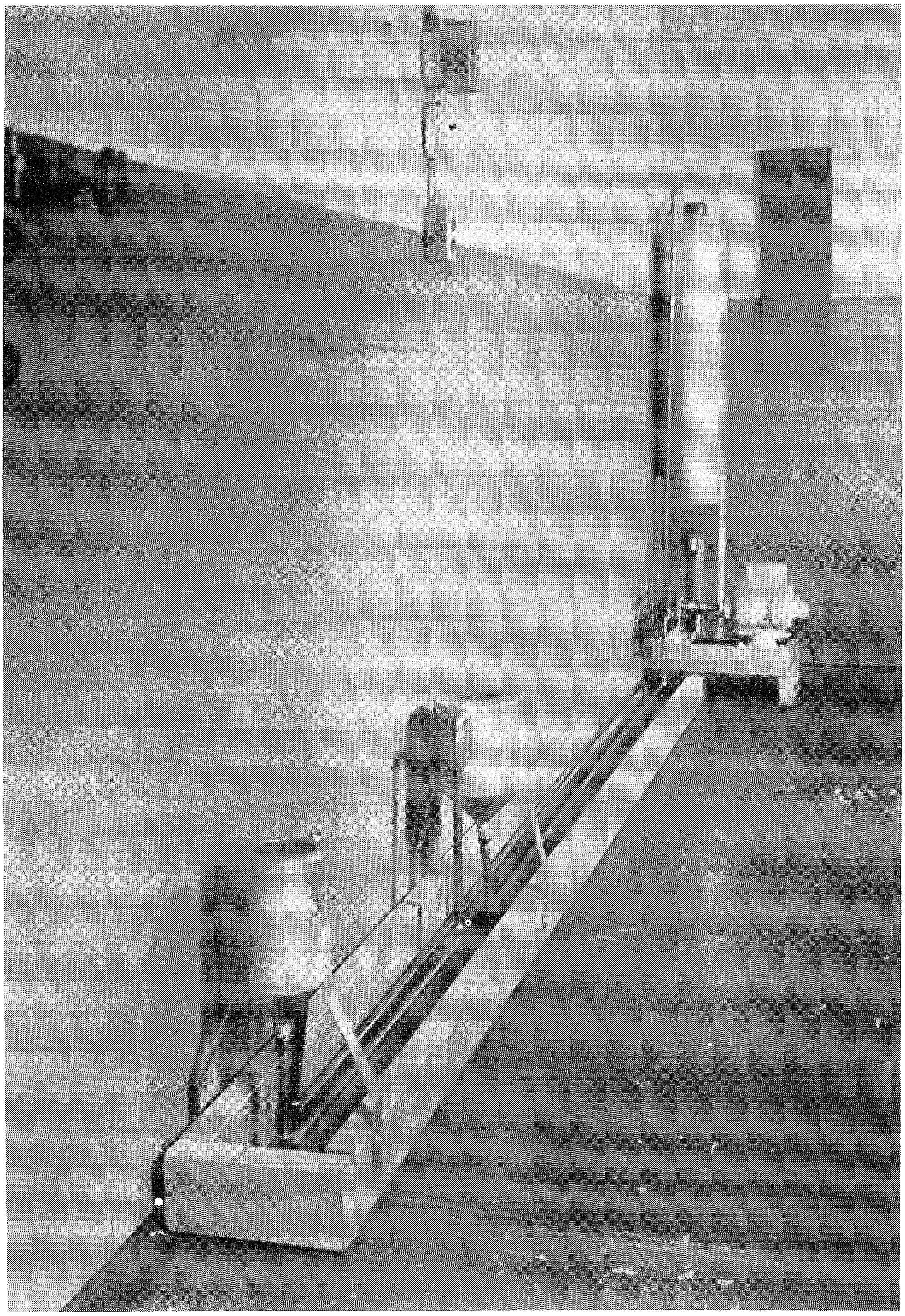

Fig. 13.

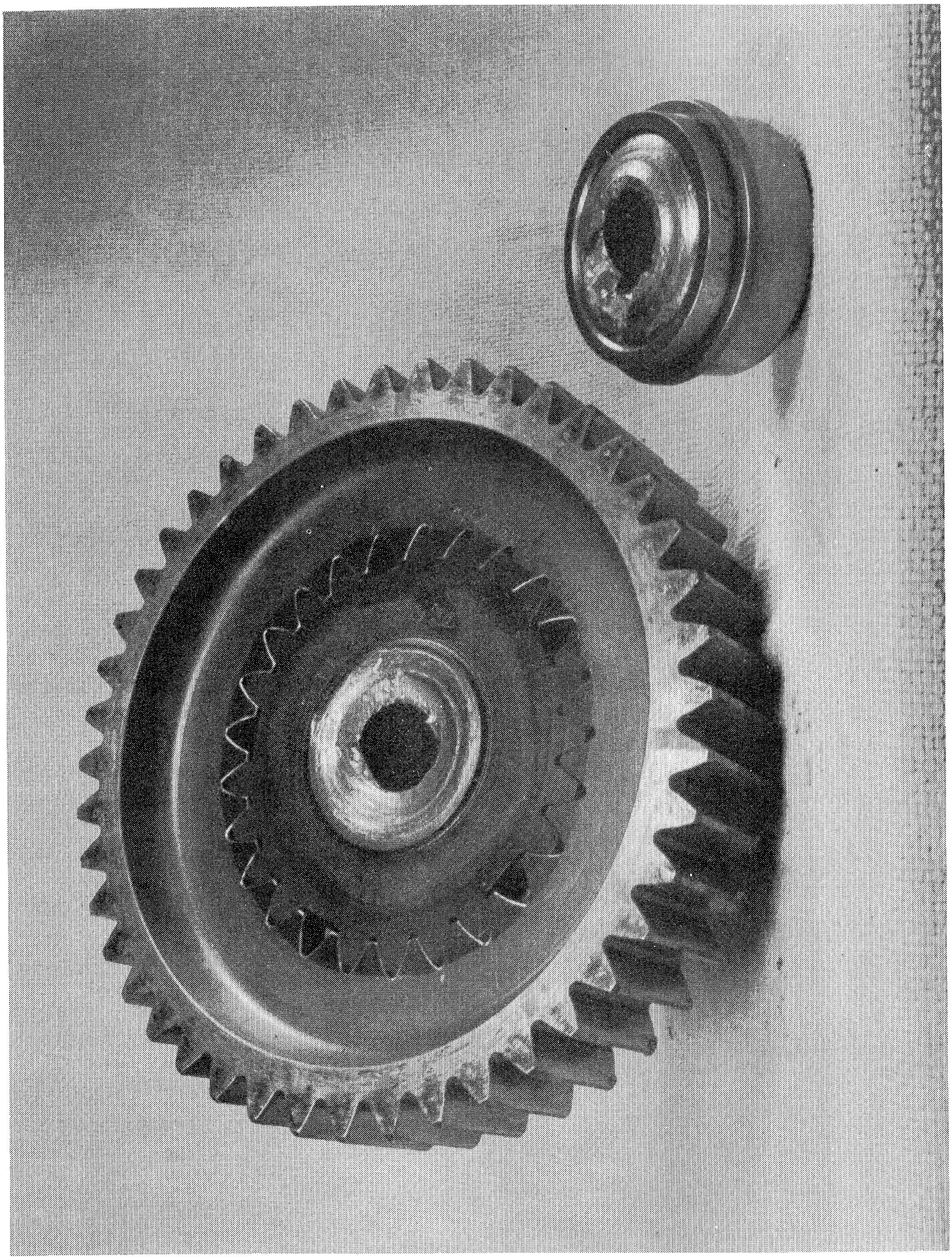

Fig. 14.

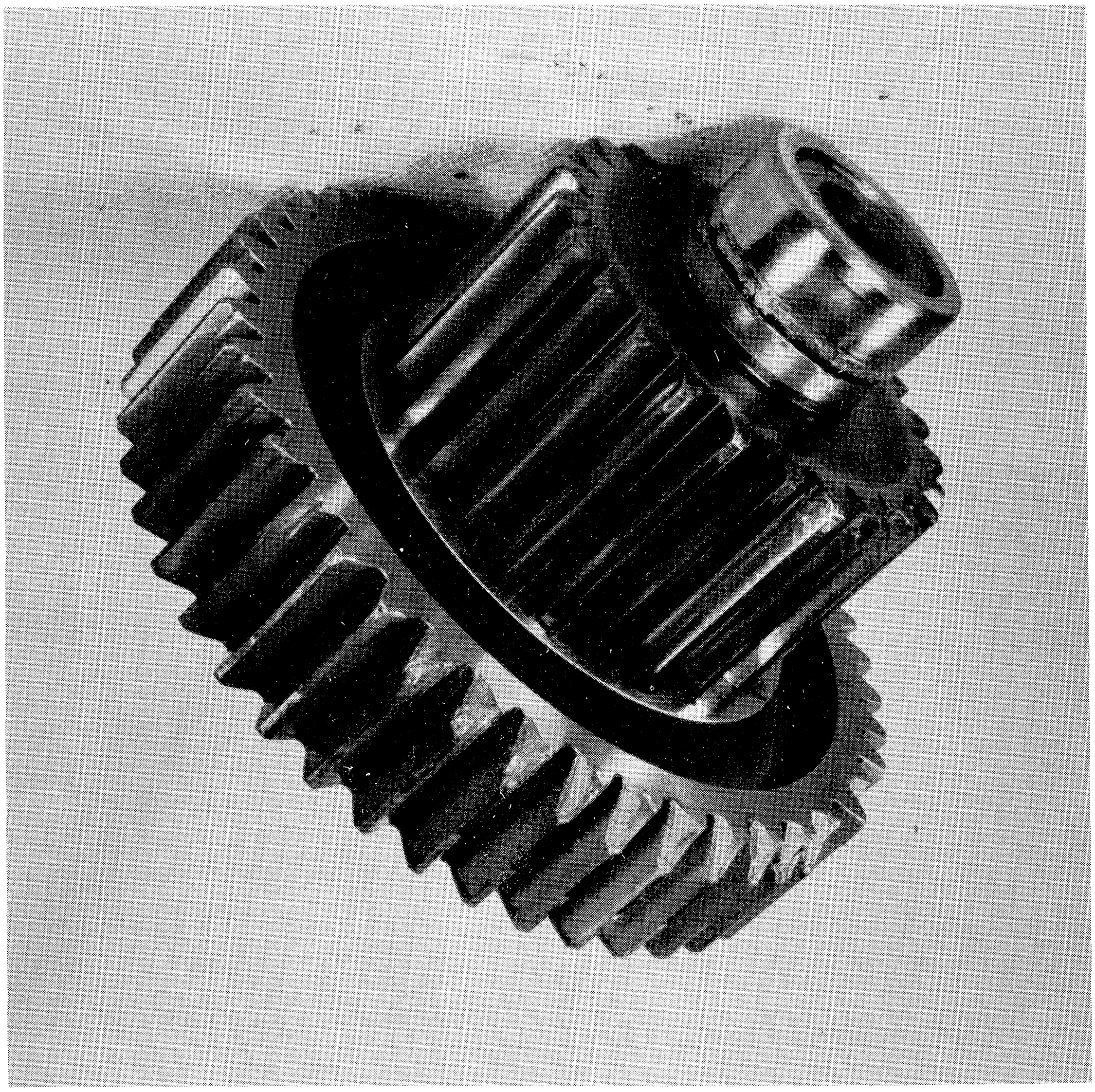

Fig. 15.

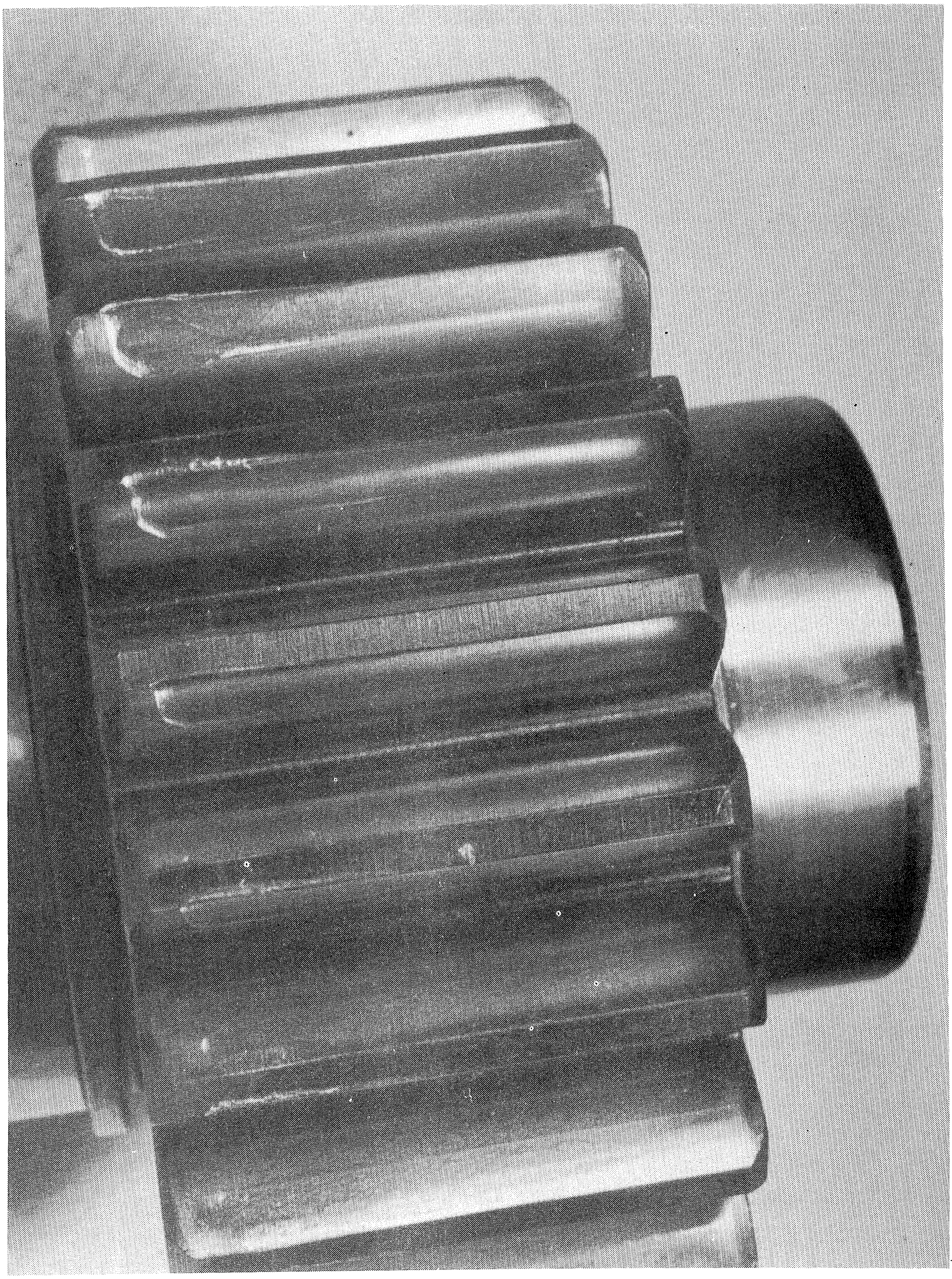

Fig. 16.

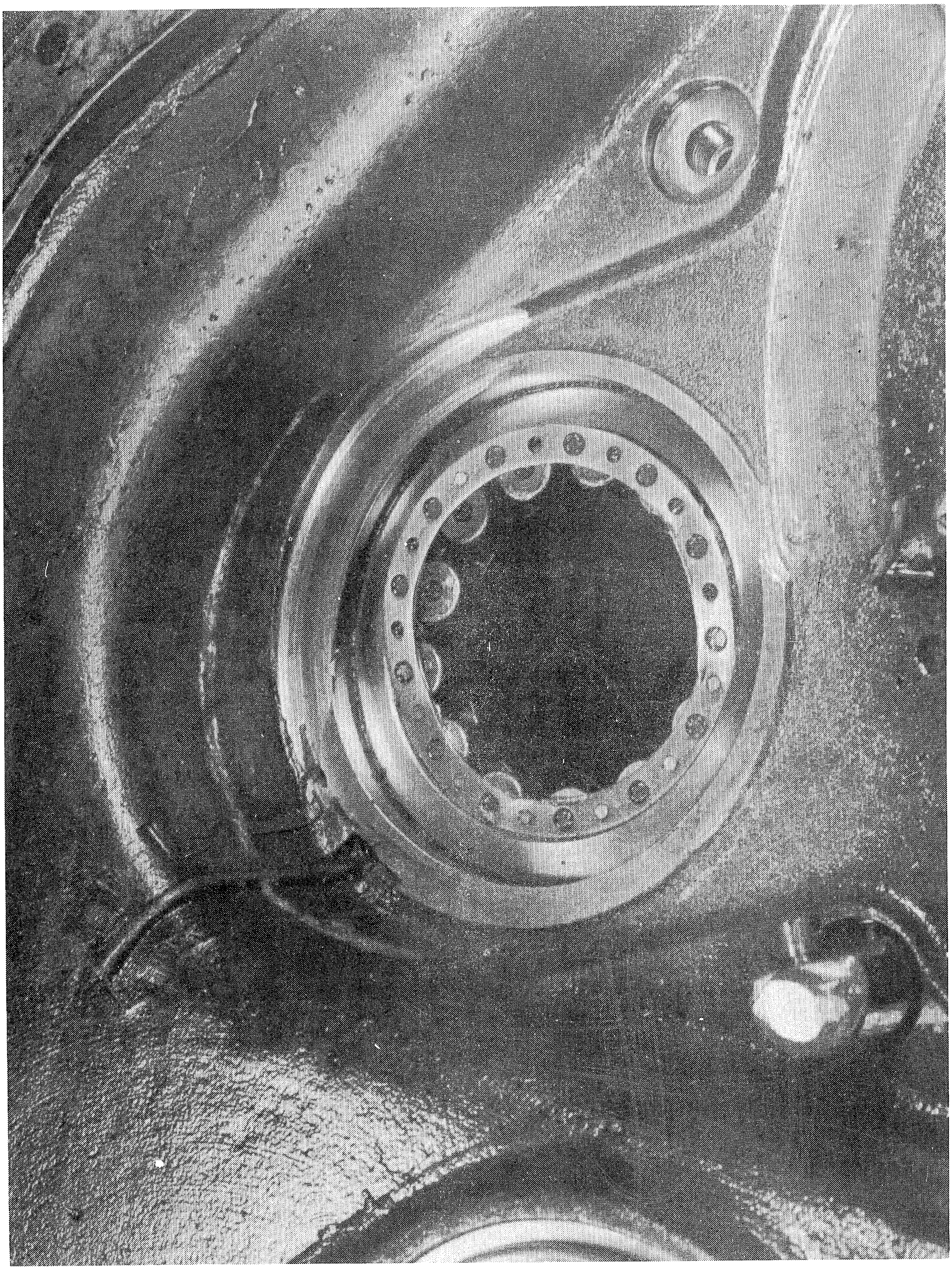

Fig. 17.

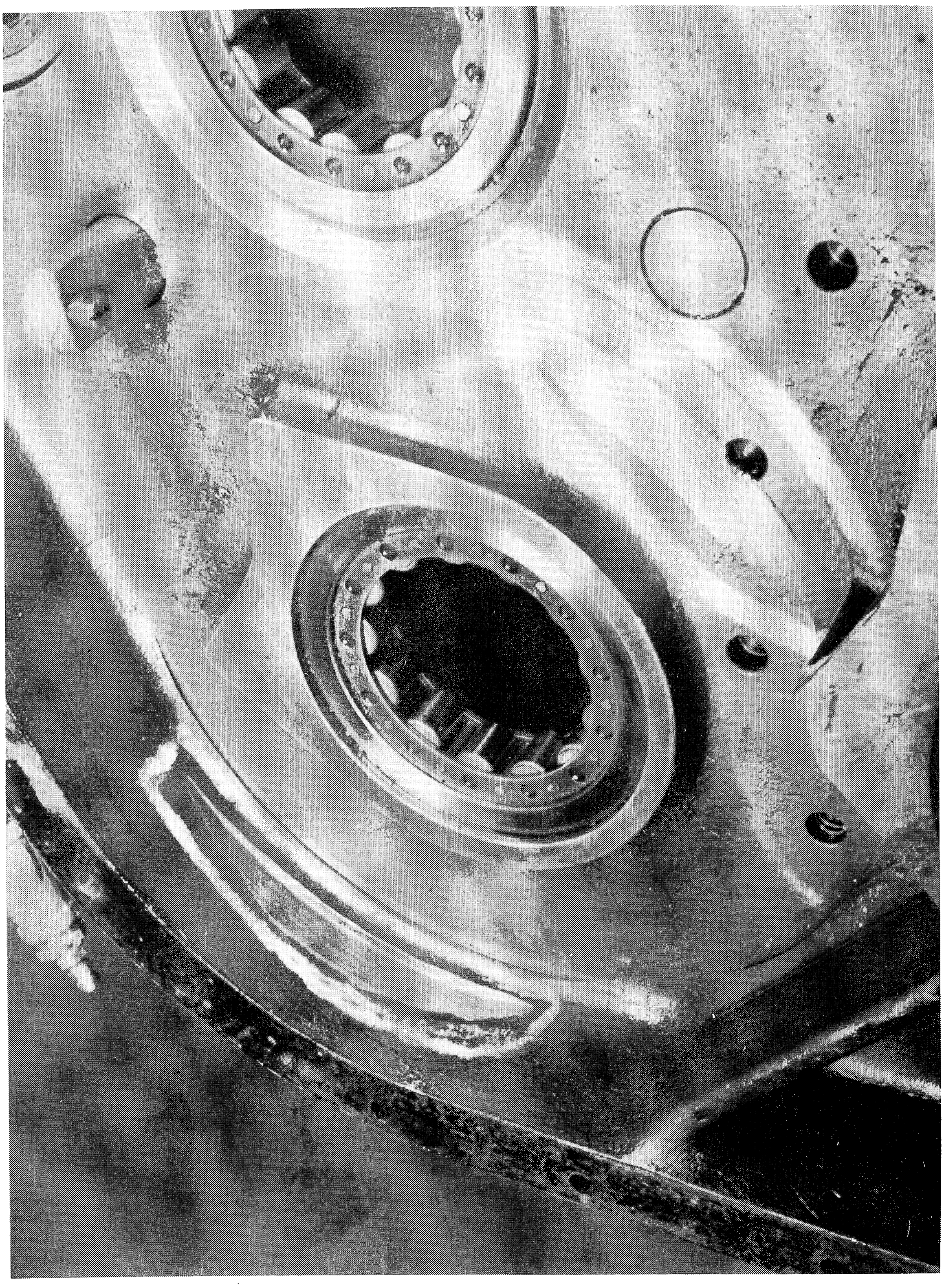

Fig. 18.