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DEVELOPMENT OF A GEAR AND SPLINE LUBRICANT TESTER

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FOREWORD

This report was prepared by the Western Gear Works, under USAF Contract No. AF 33(616)-496. The contract was initiated under Project No. 3044, Aviation Lubricants, (formerly RDO No. 613-14), and was administered under the direction of the Materials Laboratory, Directorate of Research, Wright Air Development Center, with Mr. R. C. Zurbrigg acting as project engineer.

ABSTRACT

Results of a compilation of lubricants, lubrication, and lubricant tester data include a selected bibliography with abstracts which has been arranged so that references are listed under the subject heading most applicable.

Design of a gear and spline lubricant tester which will accommodate, as test specimens, all the most popularly used types of gears; such as, spur, helical, worm, straight bevel, spiral bevel, and hypoid, as well as splines, is described. The lubricant tester will be capable of test gear speeds up to 30,000 RPM and tooth loads up to 6000 pounds per inch of face width.

The design of the tester was based on the analysis of information from sources listed in the bibliography, gear lubrication experience, and preliminary design studies of possible new simulation, as well as gear type testers.

It is concluded that a universal type, gear lubricant tester should prove advantageous in selecting lubricants for specific applications, as well as placing lubricants into a general classification of usefulness as gear lubricants.

The compilation of information, analysis, and final design of a gear and spline lubricant tester was conducted by Western Gear Works' Research Engineering Group.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

M. R. WHITMORE
Technical Director
Materials Laboratory
Directorate of Research

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INTRODUCTION

The investigation was primarily made to develope a suitable laboratory tester and test method which will closely simulate actual service conditions for evaluating existing and research spline and gear lubricants. Present day design trends are toward reduced unit envelope, higher speeds, extremes of operating temperatures, and greatly increased loading of splines and gears. These present day design trends are creating new lubrication problems and increasing the tendency for fret corrosion and other types of rapid wear with consequent reduced operating life.

The requirements for the test equipment are compact size, simplicity of operation, and ease of maintenance to facilitate functional and economical testing of small quantities of research lubricants. The equipment is required to be compact for the additional reason that it must be accommodated in a test chamber capable of being cooled and evacuated to simulate flight conditions.

The proposed test equipment is also completely suitable for testing and evaluating materials that future research may suggest as potential gear materials.

Phase I of the research program consisted of a literature survey and was concluded with a comparison of present testers on the basis of specific variables, performance, and cost. Recommendations for Phase II were that research effort be directed toward the design of a "gear type tester". Phase II consisted of design studies, refinements, and final layout and detailing of proposed gear and spline lubricant tester design.

SECTION I

BASIC PRELIMINARY INVESTIGATION

Literature Survey

To compile the available data on lubrication, libraries, airframe, and gear manufacturers, petroleum testing laboratories, and oil processing plants were contacted. Scientific and engineering journals, abstracts, indices, and reviews published during the past twenty years and more were carefully checked for pertinent information and additional references. Complete lists of the libraries searched and companies consulted are given in Appendix I and Appendix II, respectively. References are listed, under subject headings most applicable, in the bibliography.

Information gathered from the companies contacted was of importance to fully cover present lubrication problems and present test methods. This information was used in the analysis of testers and should be valuable in future research on this subject.

The procedure followed in each library, except the Western Gear Works' plant library, was the same. All of the library's index cards, related to lubricants, lubrication, and lubricant testing, were checked. Each article was examined and, if found to be worthy of reference, was included in the bibliography. The article was then studied more carefully and a brief digest prepared. The Western Gear Works' library was searched and all articles on lubrication were collected, read, included in the bibliography, and then filed in an indexed file for future easy reference.

As a result of this literature survey, a card index file was organized. Each card is filed under the subject heading most applicable and contains the usual bibliographical data and a digest or abstract covering the significant features of each publication.

On the basis of the information available, present lubricant testers were analyzed with reference to conditions occurring between the test surfaces and with regard to their performance and cost of operation. An analysis in chart form clarifies where present testers are different and serves as a basis of comparison for lubricant testers. The problems of a tester design could then be outlined and the direction of future effort planned.

Testers considered were those most generally used and those representative of types pertaining to gear lubricant testing. A brief description of these testers and test methods are as follows:

S.A.E. Lubricant Tester. In the conventional test method, two Timken test cups (T48651), of the type and quality standardized for lubricant testers, are rotated so their peripheral speeds relative to the area of closest approach are in the same direction. The speed of the upper cup is 14.6 times as high as the lower. The speed selected for the upper cup is usually 1000 rpm, although sometimes 500 rpm is used. As a preliminary to the test proper, there is a starting period in which the load is kept constant at 150 pounds (15 pounds scale reading) for 30 seconds. Thereafter, the load is increased at a uniform rate of 83.5 pounds per second until scoring (failure) occurs. The load at failure (scale reading) is the criterion adopted for tests with the S.A.E. machine.

The operating characteristics of the machine are such that automatic alignment is maintained between the two rotating cups so that the applied load is a true criterion of the actual pressure on the rubbing surfaces.

Timken Lubricant Tester. The basic test elements consist of a 1-7/8 inch cylindrical test cup rotating against a rectangular block held stationary in a holder. The test cup is rotated at 800 rpm. A steady load is applied through a lever arm system for a period of ten minutes. The standard load is obtained by applying a total of 33 pounds on the lever arm which produces approximately 20,000 pounds per square inch at the rubbing surfaces. Appearance of the blocks (scoring) after runs at various loads and weight loss of the block and cup after constant load run test is used to judge the quality of the lubricant.

<u>Four-Ball Lubricant Tester</u>. Four ½ inch SKF steel balls, or of metal under investigation, are arranged in the form of an equilateral tetrahedron. The basic elements are three lower balls held immovable in a clamp to form a cradle in which a fourth or upper ball is caused to rotate about a vertical axis under prescribed conditions of load and speed. An upward thrust given by a loading lever, acting on the ball holder, presses the three fixed lower balls against the upper rotating ball with a vertical force of 10 to 800 kilograms which results in a mean pressure on the rubbing surfaces of 15,000 to 60,000 kilograms per square centimeter. The ball holder is supported by a thrust ball bearing which permits horizontal displacement, insuring automatic centering of the underlying balls and making possible measurement of the frictional torque exerted on the lower balls. For specification purposes, three types of indices are in use; namely, the pressure-wear index, the mean Hertz load, and the seizure and weld points. All these indices require determination of the nature and size of the wear scar. The first and third indices, in addition, require determination of the load at which the scoring mechanism for the test balls changes.

Modified types consist of the Four-Ball Wear Tester, Four-Ball Top, and Four-Ball Wear Top. The Four-Ball Wear Tester has the same basic elements as the Four-Ball Extreme Pressure Tester. However, it was designed to operate in a much lower load range than the extreme pressure machine. The wear tester evaluates the anti-wear qualities of the base oils and the effects of additive materials in improving the anti-wear qualities of an oil. The size and appearance of the scar at a number of loads or the rate of wear per unit time under constant load may be used as measures of the anti-wear qualities of an oil.

The Four-Ball Top is an adaption of the Four-Ball bearing principle to a machine for measuring the coefficient of friction at low sliding velocities and high pressures. It consists of a weighted top which is arranged symmetrically around a single steel ball which in turn rotates in a cradle formed by the three lower clamped balls. Since the top is free-wheeling, only the frictional force between the upper and lower balls is involved in retarding the motion. Thus, to obtain coefficient of friction, it is only necessary to determine the time and number of revolutions until the top stops spinning, once it has been given a definite angular velocity by a falling weight. This machine is very useful for studying the fundamentals of lubrication.

The Four-Ball Wear Top is a motor-driven modification of the Four-Ball Top. It permits the measurement of friction by the deceleration method just described and also the measurement of wear by sustained operation for several minutes or hours. The rotor is motor-driven and can be brought up to the desired speed and then disengaged from the driving mechanism by means of a ratchet. Wear is determined by observing the wear spot and quantity of metal abraded, or by determining the time to wear through a thin plating placed on the top rotating ball.

Falex Lubricant Tester. The basic test elements consist of a $\frac{1}{4}$ inch steel pin rotating between a pair of V-blocks under a load applied by lever arms. The system is similar to a mechanically operated nutcracker. Since most of the wear occurs on the softer shaft, the area of contact and hence the specific pressure at the area of contact remains essentially constant as wear proceeds. Weight loss of the pin is used as a means of studying wear prevention quality and the continuous increasing load technique to measure extreme pressure properties of oils.

Almen Lubricant Tester. The tester was designed for testing lubricants for hypoid differential gears from the standpoint of film strength. A $\frac{1}{4}$ inch drill rod rotates at approximately 600 rpm in a $\frac{1}{2}$ inch split bushing made of SAE 2315 cold drawn steel. Pressure is applied to the bushing by means of a hydraulic and mechanical loading system. Friction torque developed is indicated through a second hydraulic system by a Bourdon gage. In conducting a test, the oil container is first filled with oil (25 ml). The machine is run for 30 seconds under no load. Load is applied at the rate of 2 pounds every 10 seconds until seizure (failure) occurs or 30 pounds have been applied. Each 2 pounds added produces 1000 pounds per square inch of projected area on the testing bearing.

PTR Lubricant Tester. The apparatus resembles a phonograph in that a steel pin 1 millimeter diameter rubs on a steel disc of 200 millimeter diameter. The machine operates at a temperature of 100°C. The wear of the pin is measured in relationship to running time. By producing a very thin film of test lubricant on the disc, the apparatus operates under conditions of pure boundary lubrication.

Durability Lubricant Tester. The test specimen consists of a 15 millimeter square stainless steel interferometer mirror in contact with a 1-3/4 inch diameter steel friction wheel. The test specimen is attached to a lever arm arrangement with a diamond on glass friction holder. When the friction between the wheel and the test specimen, due to failure of the oil film, is greater than the holding friction of the diamond on glass, the arm arrangement will move. Measurements are made during this movement by electrical methods and also the angle through which the movement is active.

Buckingham Lubricant Tester. Two rollers of approximately 3 inches in diameter and ½ inch thick are pressed against each other by means of a spring-loaded machine designed on the nutcracker principle. The rollers are driven either by eccentric phasing gears mounted on the same shafts, which cause different proportions of sliding and rolling to occur at different points on the peripheries of the rollers, or by phasing gears which produce a constant proportion of rolling and sliding on the peripheries. Oil is applied by a jet impinging on the outgoing side of the rollers. Scoring (failure) was determined on the rollers by visual inspection, magnetic pickup which indicated surface roughness, and a bearing block deflection measuring device employed to determine the coefficient of friction.

Thoma Lubricant Tester. The test specimens consist of two cylinders with crossed axis and in contact at the centers. One cylinder is loaded and the other is attached to a torsion meter. Cylinders are immersed in oil and both cylinders rotated in same direction. Torsion load at scoring is used to judge the quality of the lubricant.

Thornton Cam-Scuffing Rig. The cam-scuffing rig is a simulation-type test rig in which the cylindrical surfaces of two eccentrically mounted discs (cams) slide during part of their revolution against two flat surfaces (tappets), loaded by air pressure. The test oil is led, at controlled pressure and temperature, to two jets directed on to the contacting surfaces of the tappet and cam. Two sets of test pieces are used simultaneously, with the two cams mounted coaxially but out of phase by 180 degrees and separated by a distance piece, the plunger and tappet assemblies being suitably offset. The test oil is heated to 80°C. A run of 1 minute at a cam speed of 1250 rpm, load of 30 pounds per square inch is first performed. If scuffing has not occurred, the rig is restarted for the next run with the air pressure increased 5 pounds per square inch. This procedure, increase load 5 pounds per square inch every minute, is followed until both cams are scuffed (failure). The values for the scuffing load are the average air pressure at failure, in pounds per square inch, for four or more tests.

Navy Gear Wear Lubricant Tester. The tester consists essentially of a ½ inch diameter brass and steel helical gear combination mounted with axis at 90 degrees with each other. The power source delivers a sinusoidal reciprocating linear motion of 3.14 inches amplitude at 50 cycles per minute. The driving shaft is connected to the power source by a flexible cord working over a 1 inch diameter drum on the shaft. The driven shaft is fitted with a 1 inch diameter drum for applying the torque load to the gears. Load weights of 5 and 10 pounds are provided. Clean test gears are placed on the shafts of the tester with the brass gear on the driving shaft and the steel gear on the driven shaft. Weight losses per 1000 cycles are determined, after run-in, for 6000 cycles at 5 pounds load and 3000 cycles at 10 pounds load.

IAE Lubricant Tester. This machine is of the power-circulating type in which two pairs of gears are loaded torsionally against one another, the driving motor being required to provide only the frictional losses. The test gears are of coarse-pitch spur type, 3 ¼ inch centers, and made of case-hardened nickel-molybdenum steel. Loads are applied by means of a lever arm which in turn torques the system and loads the test gears. One gallon of lubricant is circulated through a heater and pressure control valve to the stationary gears at a feed temperature of 90°C. A lever load of 10 pounds is applied and the gears run for 5 minutes at 2000 rpm. The machine is then stopped for 5 minutes during which time the test gears are inspected, the load is increased by 5 pounds, and testing resumed. This procedure is repeated with further load increments of 5 pounds until failure occurs, the normal criterion of failure being the scuffing of the test gear teeth.

Thornton High-Speed Gear Rig. The apparatus uses spur gears as test pieces and was developed to measure the behavior of oils under high-speed conditions. The rig operates on the "four-square" power circulating system in which a pair of test gears is coupled by two shafts to a pair of "power-return" gears of much greater face width. The two sets of gears are loaded against each other by locking up a torque in the system by means of a torque coupling on one of the shafts. The gears are lubricated by an oil jet at 90°C., directed vertically downwards towards the meshing point of the gears. The standard test is to determine the load which causes scuffing during a 15 minute test run. The gears are given a run-in of 15 minutes with 15 pounds on the 2 foot lever arm. If the gears are undamaged, a sequence of increasing the load by 5 pounds and running for 15 minutes is followed until scuffing occurs. As a general rule, an oil is tested at 3000, 6000, 9000, and 12,000 rpm, using the four test faces of a single pair of gears with at least one set of repeat tests on a second pair of gears.

<u>Shell High-Speed Lubricant Tester</u>. * The test specimens consist of AGMA, Class IV or better, 6 DP spur gears, 3 inch center distance, 17 and 19 teeth, and ¼ inch face width. The apparatus is loaded by the "four-square" principle. Cluster gears of different ratio are contained in a movable housing. By moving this housing through a small angle, a torque is introduced and the test gears loaded. This loading is remote controlled and may be varied during operation. Speeds may be varied up to a maximum of 35,000 rpm. The procedure for testing will be to select a speed at which the test is to be run, pre-heat the lubricant to a specified temperature where it is controlled, apply a small initial load and run for a specified time, when it is stopped. Test gears are then inspected, if there is no sign of abrasion or scoring, the test is continued. The load is increased by increments and run for a specified time, stopped, and the test gears examined. This procedure is continued until abrasion or scoring of the test gear teeth. Another function of the tester is to determine the most favorable location for the application of the lubricant.

*Manufactured by Western Gear Works, Pacific Gear Plant, 1035 Folsom Street, San Francisco, California.

Ryder Gear and Lubricant Tester. Two parallel shafts are connected by two pairs of gears in the familiar "four-square" power circulating system. Small spur gears are the replaceable test specimens, while the large helical gears are permanent parts of the apparatus. The feature of this machine is the application of load to the test gears by axial movement of one helical gear relative to the other, accomplished by applying a known oil pressure to the piston-like hubs of the helical gears. From the helix angle and the area of the hubs, the tangential load may be calculated. The housing is divided into two compartments, one for the main gears and one for the test gears. The run-in procedure followed is: 200 pounds per inch of tooth face, 160°F. oil temperature, 5 minute run at 1100 rpm, 10 minutes at 1650 rpm, and 10 minutes at 2200 rpm. The loads are then successively increased until the inspections between load increments reveal failure by scuffing. The load at which 22 ½ per cent of the total tooth face area is scuffed is considered the failure load.

BREAK ALL SHARP CORNERS

GENERAL CLASSIFICATION OF TESTERS		SIMULATION TYPE										GEAR TYPE				
TESTERS		SAE (MODIFIED)	TIMKEN	4 BALL	FALEX	ALMEN	PTR	DURABILITY TESTER	BUCKINGHAM	THOMA	THORNTON CAM RIG	NAVY GEAR WEAR TESTER	IAE	THORNTON GEAR RIG	HIGH SPEED TESTER	RYDER
COMPARISON ON BASIS OF VARIABLES																
SURFACE CONDITIONS	MATERIAL				CONSTANT				VARIED	CONSTANT	VARIED			CONSTANT		
	HARDNESS					CONSTANT								CONSTANT		
	TYPE OF CONTACT	LINE		POINT	DISPERSED		POINT		LINE	POINT	LINE			LINE		
LOADING	STATIC					VARIED								VARIED		
	DYNAMIC			NONE				NONE						UNKNOWN		
	IMPACT					NONE PRESENT								UNKNOWN		
	VIBRATION						NEITHER EXTERNAL NOR FRICTIONAL WERE ACCOUNTED FOR							UNKNOWN		
MOTION	SLIDING			CONSTANT					VARIABLE					VARIABLE		
	ROLLING	CONSTANT				NONE PRESENT			VARIABLE	NONE	CONSTANT			VARIABLE		
	LENGTHWISE CROSSWISE	CROSSWISE				NONE			CROSSWISE	NONE	CROSSWISE			CROSSWISE		
	GEAR SIMULATION	GOOD			NONE				EXCELLENT	NONE	EXCELLENT	HELICAL GEARS		SPUR GEARS		
	SPLINE SIMULATION			NONE					NONE					NONE		
LUBRICATION	STRAIGHT MINERAL		POOR		GOOD									POOR		
	E.P. LUBRICANT		GOOD		POOR				GOOD					GOOD		
	BULK OIL TEMPERATURE	UNCONTROLLED	CONTROLLED		UNCONTROLLED				CONSTANT					CONTROLLED		
COMPARISON ON BASIS OF PERFORMANCE AND COST																
RIG PERFORMANCE	DIFFERENTIATION BETWEEN OILS	GOOD	GOOD	GOOD	FAIR				GOOD		GOOD			GOOD		
	REPEATIBILITY OF RESULTS	GOOD	GOOD	GOOD	FAIR				GOOD		GOOD	POOR		FAIR		
	CORRELATION WITH SERVICE	FAIR	POOR	FAIR	POOR				FAIR		FAIR	VERY POOR		FAIR		
EASE OF SPECIMEN PROCUREMENT	MATERIAL						ALMOST UNLIMITED							ALMOST UNLIMITED		
	SURFACE FINISH					FAIRLY EASE TO PRODUCE								NOT EASY TO PRODUCE		
	DIMENSIONAL CONTROL		EXCELLENT					EXCELLENT			FAIR			DIFFICULT - REQUIRES SPECIALISTS		
	SURFACE HARDNESS					EASE TO PRODUCE								NOT EASY TO PRODUCE		
	COST	FAIR		VERY CHEAP		CHEAP			FAIR	CHEAP	FAIR			RELATIVELY HIGH		
DEGREE OF SEGREGATION OF VARIABLES	HYDRODYNAMIC EFFECT	NONE	UNKNOWN	NONE	EXCELLENT POSSIBILITY		NONE	NONE	UNKNOWN	NONE	UNKNOWN			UNKNOWN		
	IMPACT LOADING				EXCLUDED BY DESIGN OF TESTER									UNKNOWN		
	MATERIAL					CONSTANT (CAN BE VARIED)			VARIED	CONSTANT	VARIED			CONSTANT (CAN BE VARIED)		
	CHEMICAL ACTIVITY					WAS NOT ACCOUNTED FOR								WAS NOT ACCOUNTED FOR		
	THERMAL EFFECT		HIGH		EXCESSIVE		HIGH	LOW		MEDIUM				MEDIUM		
	HARDNESS AND FINISH						COULD HAVE BEEN VARIED BUT WAS HELD CONSTANT							NOT EASY TO VARY		
OPERATION OF TESTER	TOTAL TIME OF TEST					30 MIN								6 TO 250 HOURS		
	EASE OF OPERATION	FAIRLY EASY		EASY			VERY EASY			FAIRLY EASY				NOT TO EASY - NO GREAT CONTROL OF VARIABLES		
	RECORDING RESULTS					MECHANICAL INSPECTION OF TEST PIECES FOR SCORING AND WEAR						LOSS OF WEIGHT		SCORING OF GEARS		

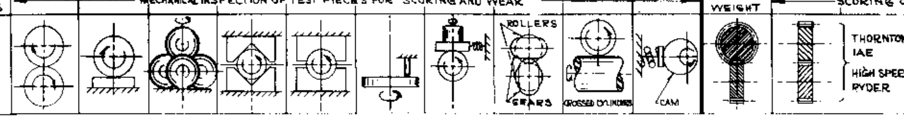


FIGURE I

WESTERN GEAR WORKS
LYNWOOD CALIF
LUBRICANT TESTER COMPARISON CHART
SCOTT
SAUTER
NONE
E 6501E5

SECTION II

ANALYSIS OF COMPILED INFORMATION

Testers studied in this research program have been separated into two groups. The basis of classification, being whether the tester contained actual gears as the test specimens, or used simple basic geometric test elements. Those using gears were classified as "Gear Type Testers", while all others were grouped under the heading of "Simulation Type Testers". The term, "Simulation Type Testers", does not necessarily mean that these testers simulated gear or spline type contact or motion. In some cases, such as the Four Ball Tester, it would be difficult to determine the exact machine element simulated.

Conclusions drawn from each type are given in separate sections as follows:

Simulation Type Testers. Simulation testers, as a group, present a rather varied picture. Just about every type of configuration is represented. For instance, the type of contact varies from a point to an area contact of varying size and shape. Some have included only a few variable factors, (such as hydrodynamic effect), while others have employed more variable factors. For this reason, direct comparison of results is not obtained and these results do not correlate with service.

The picture they present appears confused, but does tell a story when analyzed as a group. Those that showed the greatest variation with service results were those that appeared to have the least number of variable factors usually occurring in gear meshes when considering all gear types. Those that attempted to simulate gear action, such as the Buckingham and Thornton cam rig testers, have restricted their action to that occurring in spur gears. When these testers tried to predict how a lubricant would work under service conditions with other type of gears, such as hypoid and worm, service results did not correlate.

Probably the most instructive points are not those that the testers as a group presented, but those that were not included. Lengthwise motion, such as that occurring in hypoid gears, was not simulated despite the now accepted fact that flash temperature becomes higher in this type of motion and the ability of a lubricant to perform seems very closely allied to temperature flash. Also, the time required for chemical action of the additives with the metal surfaces involved may determine whether a lubricant will operate satisfactorily or not, yet this has not always been fully considered. In all testers the time for chemical action was reduced as the speed increased. Some testers, such as the Timken and Four Ball, have one surface in contact continuously; thereby, making it practically impossible for chemical action to repair film rupture.

Some lubricants that have been rejected under a given load and speed might have proved very successful if more time had been allowed for chemical action to take place.

It is believed that these two variables, lengthwise contact and time for chemical activity, also have a decisive effect on the results a tester will give and should be adequately considered in a tester design. Attempts to correlate tester results with service indicate difficulty where these variables are ignored. More clearly defined results may possibly be obtained using these variables, thereby making the lubricant rating more closely correlate to service.

Simulation type testers have been found to have a poor correlation with gear service and have probably been designed and mainly used to determine, in the laboratory, the value of lubricants for specific or general application. The general opinion is that this type of tester has an important place in laboratory work and as a refinery control device, but is not a reliable indication as to the performance of a lubricant under gear service conditions.

Preliminary design studies have indicated that this type of tester may be improved and possibly, by careful design, could give results that correlate with service.

Gear Type Testers. Testers of this type show a marked similarity in all respects. The testers are of the "four square" power circulating type in which two pairs of gears are loaded torsionally against one another, the driving motor being required to provide only the frictional losses. The manner of accomplishing this action varied, but the principle was the same.

The present gear testers have also left out some gear contact interactions as only spur gears were used. Future designs should include a variety of gears such as hypoid and worm gears, as spur gears do not duplicate the type of contact in all gearing.

The time for chemical reaction of the lubricant and/or additives and the metal surfaces, decreases as the speed increases. It is felt that this variable should receive more attention and be carefully accounted for during the design of a tester.

The number of unknown variables; such as, hydrodynamic effect, external vibration, dynamic loading, and ambient conditions, all affect gear lubrication in aircraft and guided missiles. These must be considered when consistent results are required using testers to determine the value of a lubricant for gears. Control of these variables appears desirable, but may prove difficult.

Gear type testers seem to be similar to "go-no-go" gages, in that they give little specific detailed knowledge of the effects of the individual variables or their fundamental rating but accept or reject on an overall basis. This disadvantage is of small consequence as other testers are available for these specific variable analysis. A gear lubricant tester should incorporate the many variables occurring in a gear mesh and must be flexible so that service conditions can be simulated as closely as possible.

The general opinion of industry appears to be tending toward gear type testers as correlation with service has been better using gears as test specimens, and these testers are more readily aimed to service requirements.

SECTION III

DEVELOPMENT OF TESTER DESIGN

General Function and Design

From the results of the preliminary investigation described in Section I, the analysis of compiled information in Section II and preliminary design studies of possible new simulation, as well as gear type testers, it was determined that the research effort for the next phase of the program be directed toward the design of a "gear type tester". Because previous "gear testers" have been mainly restricted to "spur" type gears and have, therefore, had a narrow field of operation, the investigation of a gear type tester design, that will be much more universal in application, was proposed. This design would incorporate the ability to mount and test all of the most commonly used types of gears; such as, spur, helical, worm, straight bevel, spiral bevel, and hypoid, as well as splines, in conjunction with the lubricant variables so that conditions, more closely duplicating actual service requirements, may be accomplished.

Flexibility of the tester's range of variables; such as, load, speed, temperature, ambient conditions, and lubricant application, appears to be highly desirable. Test specimens should be actual gear elements that can be standarized as to form, size, and material for lubricant evaluation, but could be altered as to material and processing specifications for any specific evaluation desired for actual service requirements.

It was believed that previous testers have not correlated well to actual service conditions primarily because the testers were not designed nor set up to closely enough approximate the variables encountered in service. A universal type gear tester would prove advantageous in selecting lubricants for specific applications, as well as placing lubricants into a general classification of usefulness as gear and spline lubricants.

At this stage of development, it was desired that the testing device be designed as simply as possible with provisions for easy modification. It was felt that a preliminary study of the importance of each of the variables, listed in Figure I, was a definite prerequisite to the final design of a practical testing device, and it was expected that several modifications would be made or even different machines might be required depending on the results of the work. Fundamental requirements considered in the design were:

(1) Size and shape of gear and spline test specimens.

(2) Means for mounting specimens on rigid test spindles.

(3) Provision for introducing and measuring torque in a "four-square" torque system.

(4) Means of rotating or raising and lowering one test spindle relative to the other, providing for types of gears other than spur gears as test specimens.

(5) Method of driving test specimens at high speeds and loads.

(6) Confine test lubricant system to less than one quart capacity.

(7) Provision for the exposure of test specimens to simulated use conditions; such as, high temperature, low temperature, and high altitude.

The fundamental requirements for this device were so complex and incompatible that many design studies were considered before a design was found to fulfill the requirements.

Design Criteria

The Design Criteria Curve, Figure 2, is essentially a maximum expected test run on any one specific lubricant, designating gear specimen tooth loads versus speed at which scoring is anticipated. The curve of score loads was based on the following:

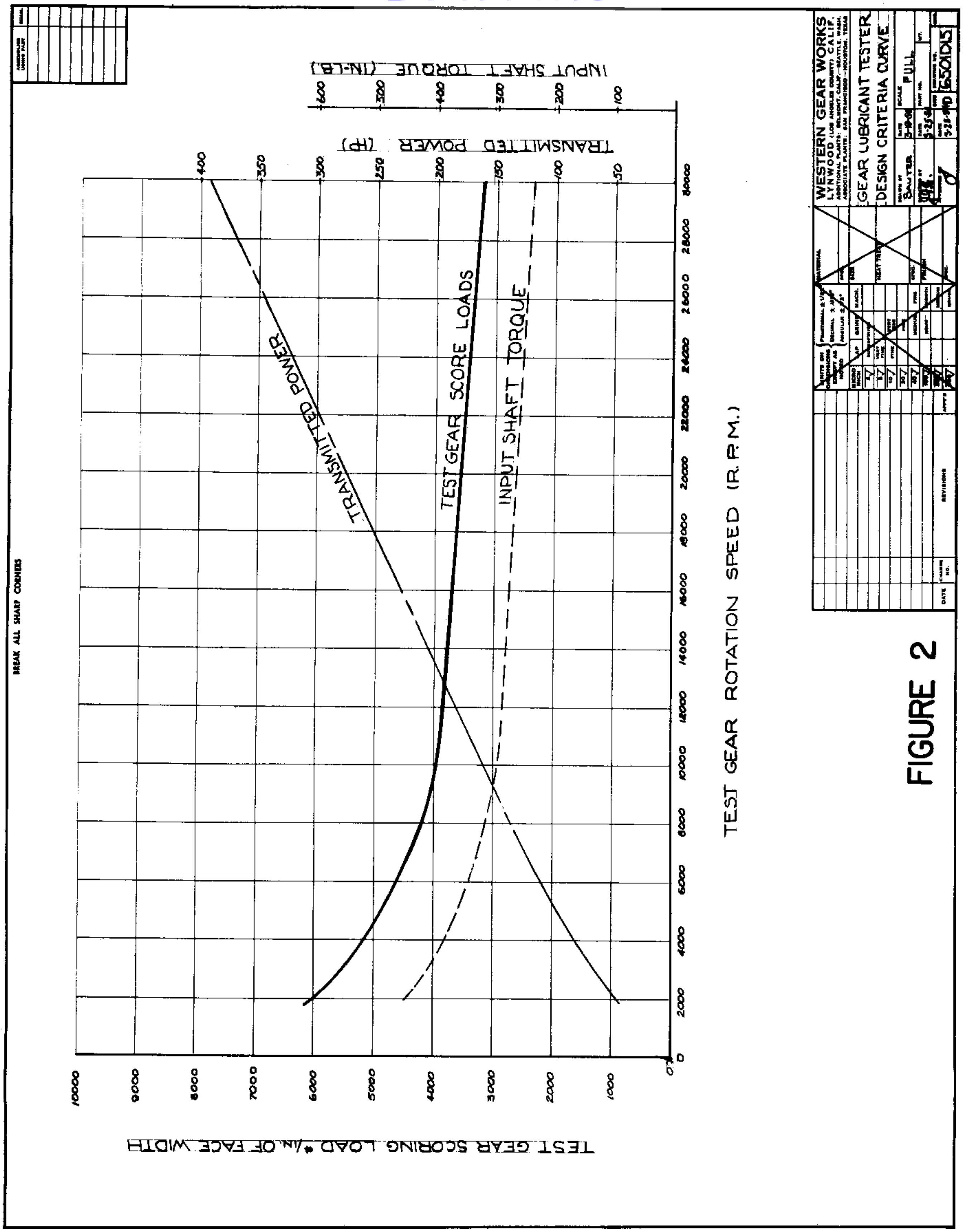

BREAK ALL SHARP CORNERS
INPUT SHAFT TORQUE (IN-LB)
600
500
400
300
200
100
TRANSMITTED POWER (HP)
400
350
300
250
200
150
100
50
TRANSMITTED POWER
TEST GEAR SCORE LOADS
INPUT SHAFT TORQUE
TEST GEAR SCORING LOAD #/IN. OF FACE WIDTH
10000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
TEST GEAR ROTATION SPEED (R.P.M.)
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000
22000
24000
26000
28000
30000
WESTERN GEAR WORKS
LYNWOOD (LOS ANGELES COUNTY) CALIF.
GEAR LUBRICANT TESTER
DESIGN CRITERIA CURVE
FULL
B5010I5

FIGURE 2

1. Results from tests performed on gear testers; such as, the I.A.E., Ryder, Thornton, and Shell Development testers.

2. Information from the literature survey on the load carrying capacity of lubricants.

3. Tooth loads encountered through the years of experience in the manufacture of gears by Western Gear Works.

To determine the stresses present in the test mechanism, the design had to be based on a criteria of tooth loads to produce scoring at the test gear specimens. Tooth loads previously encountered were found from the sources listed above.

The curve designates tooth loads much higher than are commonly used in gear practice and also higher than score loads determined on the previously mentioned testers. In the event of testing a new developed lubricant with a high load carrying capacity, the tester must be capable of transmitting these higher loads. Adequate allowances have been made in the determination of the design curve to allow for future lubricants of higher load carrying capacity.

The curve of transmitted horsepower indicates the horsepower being transmitted through the test mechanism and gear specimens due to the "four-square" power circulating principle, for the score loads shown at the same test gear speed. A value of 10% of the transmitted power was taken as the expected power loss. Therefore, the prime mover need only furnish this 10% loss, due to friction in the test mechanism.

The curve of input shaft torque indicates the torque on the input shaft for the corresponding score loads at the same test gear speeds.

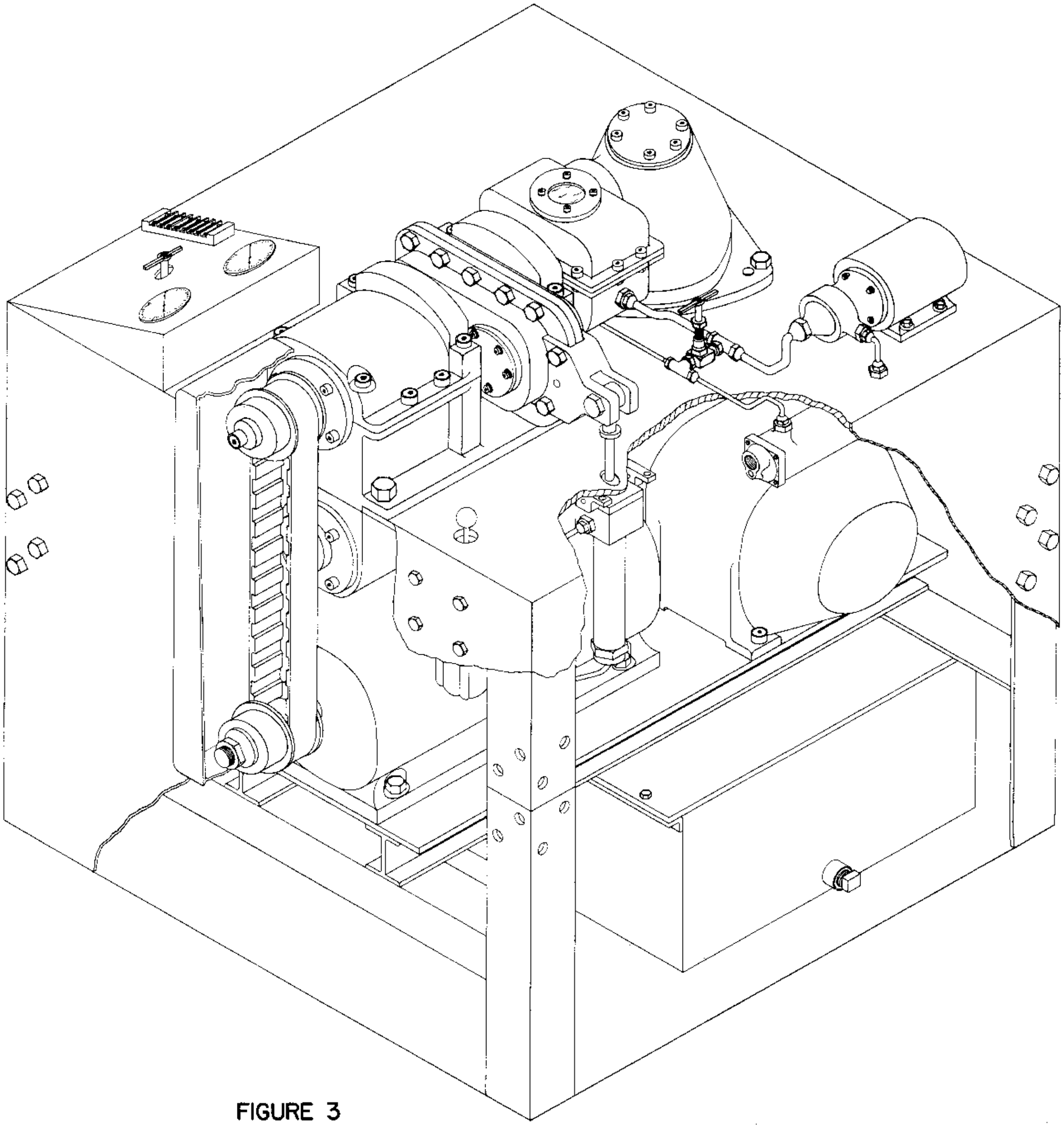
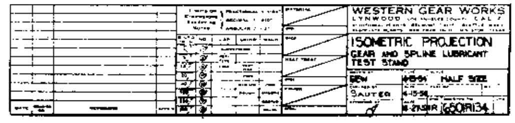


FIGURE 3


WESTERN GEAR WORKS
LYNWOOD
ISOMETRIC PROJECTION
GEAR AND SPLINE LUBRICANT
TEST STAND
HALF SIZE
SAUTER
4-15-54

SECTION IV

DESCRIPTION OF TESTER DESIGN

The gear and spline lubricant tester, as shown in Figure 3, is a self-contained unit. The test mechanism proper mounts on a test stand which houses the component parts of the tester. Component parts, Figure 4, consist of the following:

(1) Main oil lubricating and loading system.

(2) Drive motor with controls.

(3) Test lubricant circulating system.

The main oil lubricating and loading system and the test lubricant system are separate closed systems.

The main oil lubricating system consists of:

(1) Oil pump and electric motor.

(2) Pressure regulating valve.

(3) Pressure gages.

(4) Four-way selector valve.

(5) Two-way hydraulic cylinder.

(6) Manually controlled pressure relief valve.

(7) Lubricating oil reservoir.

Schematic layout of the main oil system is shown in Figure 4. A Gerotor type oil pump, mounted on a 2 horsepower electric motor, provides the oil pressure and capacity required to lubricate the drive and load mechanism, as well as pressurizing the hydraulic loading cylinder. A pressure regulating valve is used to obtain a constant pressure and oil flow for the lubrication of the test mechanism, regardless of the pressure change in the remainder of the system. A manually operated relief valve, mounted on the panel board, varies the oil pressure

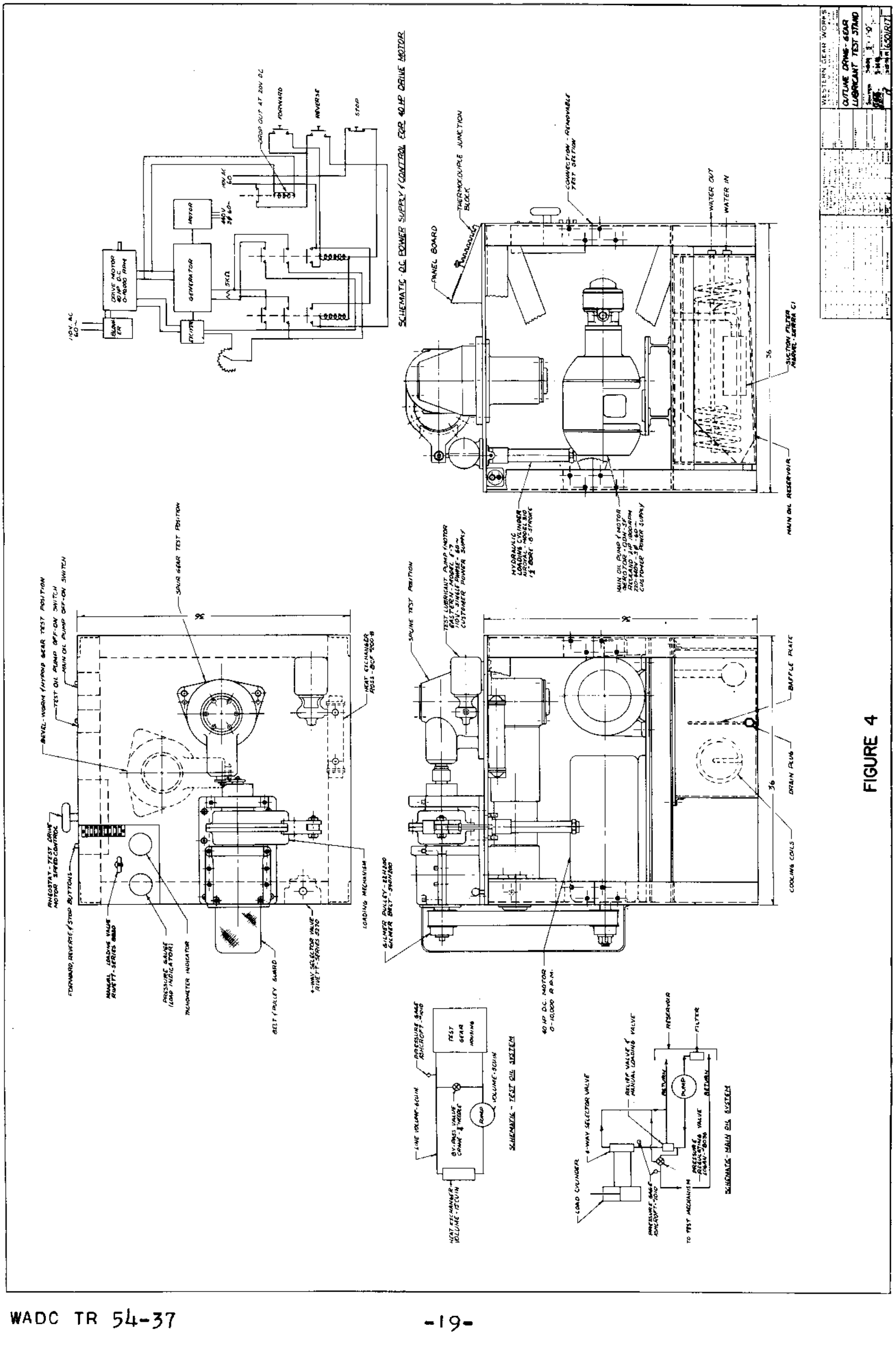

SCHEMATIC - D.C. POWER SUPPLY & CONTROL FOR 40 HP DRIVE MOTOR
SCHEMATIC - TEST OIL SYSTEM
SCHEMATIC - MAIN OIL SYSTEM
WESTERN GEAR WORKS
OUTLINE DRIVE-GEAR LUBRICANT TEST STAND

FIGURE 4

supplied to the hydraulic loading cylinder. The magnitude of pressure on the hydraulic cylinder depends on the load required at the test specimens. The tester will be calibrated to correlate oil pressure on the hydraulic cylinder to load in pounds per inch of face width on the test specimens. The direction of force applied by the hydraulic cylinder is controlled by a four-way selector valve. The two directions of applied force are necessary to load both faces of the teeth on the test gear specimens. Pressure gages indicate the pressure on the hydraulic cylinder and at the test mechanism. The pressure gage, indicating pressure on the hydraulic cylinder, is mounted on the panel board. The reservoir has a capacity of approximately 20 gallons and contains a suction filter for filtering the lubricating oil and water circulating coil for oil cooling purposes.

A variable speed, 500 to 10,000 rpm direct current motor, is used as the driving motor. Schematic diagram of the DC power supply and control is shown in Figure 4. The motor incorporates a blower for cooling purposes plus an attached tachomometer generator. Compactness is a prerequisite so that the motor may be mounted within the test stand. The tachometer generator is connected to a tachometer indicator, mounted on the panel board, for speed indication. The speed control, with forward, reverse, and stop buttons, mounts on the side of the test stand for easy control. The equipment necessary for generating and regulating the required DC power is externally mounted. A high speed pulley and belt arrangement is used for power transmission from the drive motor to the test mechanism.

The test lubricant circulating system includes an oil pump and motor, heat exchanger, by-pass valve, pressure gage, and test gear housing. Schematic layout of test oil system is shown in Figure 4. The capacity of the system is restricted to not more than 1 quart with the test housing acting as a sump. The test lubricant is applied to the test gear specimens by a jet with pressure and flow controlled by the by-pass valve. A pressure gage is mounted in the oil inlet line near the test housing to indicate oil pressure. A heat exchanger is used to cool the test lubricant during a test run. The test gear housing incorporates an electric ring immersion heater for high temperature testing, as shown in Figure 5. Thermocouples are located within the housing to measure the air and/or the test lubricant temperatures. The thermocouple wires are connected to a junction block, located on the panel board, to which a temperature recorder and control may be attached for purposes of recording and controlling temperatures within the test housing. The housing contains a pyrex

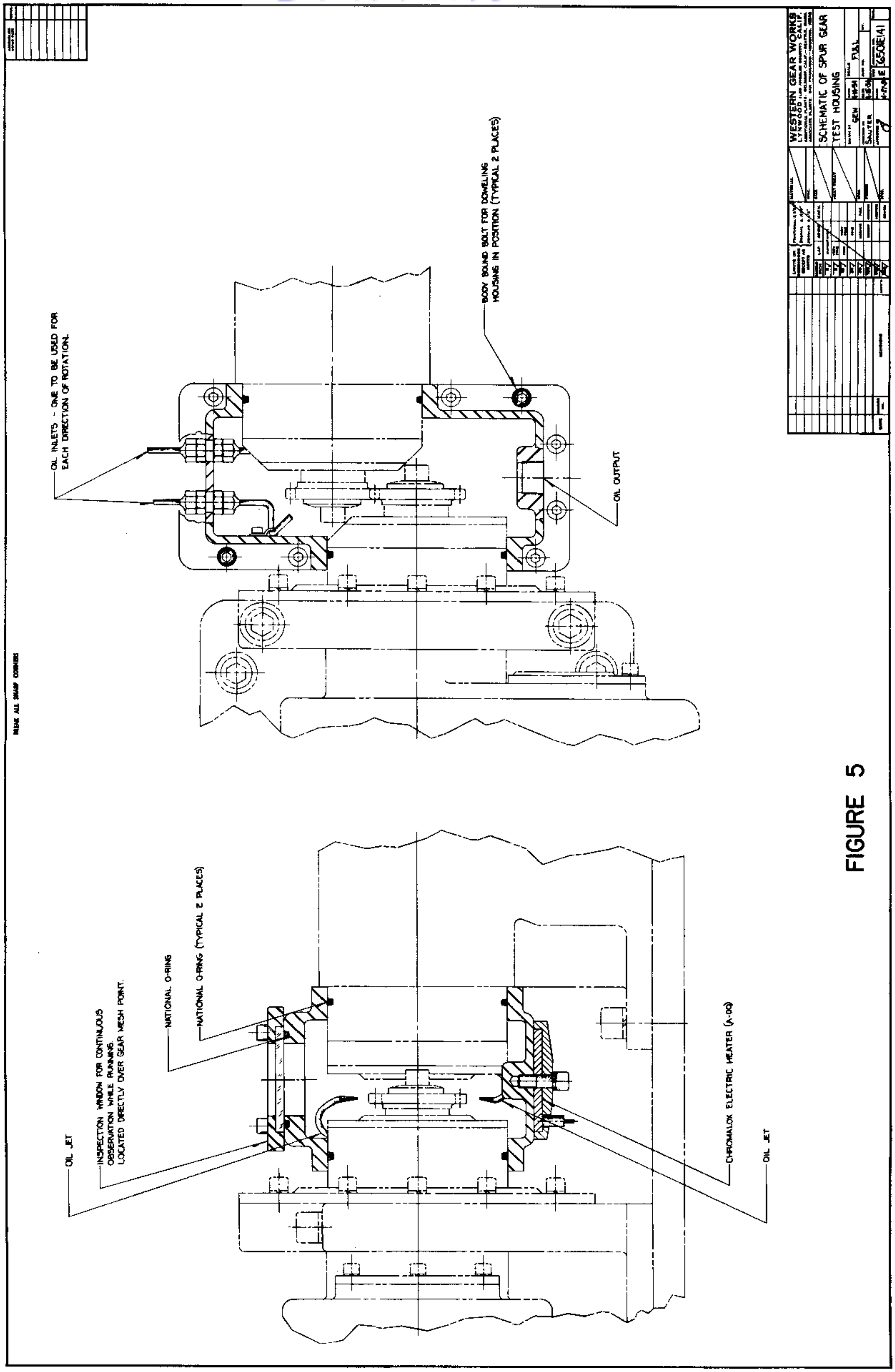

OIL INLETS - ONE TO BE USED FOR EACH DIRECTION OF ROTATION.
BODY BOUND BOLT FOR DOWELING HOUSINGS IN POSITION (TYPICAL 2 PLACES)
OIL OUTPUT
BREAK ALL SHARP CORNERS
OIL JET
INSPECTION WINDOW FOR CONTINUOUS OBSERVATION WHILE RUNNING LOCATED DIRECTLY OVER GEAR MESH POINT.
NATIONAL O-RING
NATIONAL O-RING (TYPICAL 2 PLACES)
CHROMALOX ELECTRIC HEATER
OIL JET
WESTERN GEAR WORKS
SCHEMATIC OF SPUR GEAR TEST HOUSING
SAUTER
FULL

FIGURE 5

inspection plate through which the test gear specimens are viewed during testing. If necessary, the top half of the housing can be removed for purpose of test specimen inspection without disturbing the test lubricant contained in the sump. The test housings are so designed that they can be easily and quickly removed for changing the test gear specimens.

The test mechanism proper is based on the "four-square" power-circulating system modified for the inclusion of the universal type tester principle. The modification consists of returning the power through a system of bevel gears rather than a single set of return gears. The use of bevel gears as power return gears permits greater freedom to the type of gears used as test specimens. The mechanism, as shown in Figures 6 and 7, consists of a speed increaser and torquing box, speed reducer, and a system of bevel gear sets and shafts for power return.

Figure 6 is an assembly drawing viewed from the top with the tester split on the top horizontal line. The movable head, (24), is in position for bevel gear test specimens.

Figure 7 is an assembly drawing showing the stationary and movable head in outline and the power return system split on the vertical axis. The movable head, (24), is in position for the use of splines as test specimens.

The speed increaser and torque box consists of two sets of spur gears, (99), (100), and (93) (94), with a step up ratio of 1:3. Torque is applied to the system by rotating one set of spur gears, (93) (94), about the other set, (99) (100). As gears, (93) (94), are rotated about gears (99) (100), due to the different number of teeth in the spur gears the test gear spindle, (99), is rotated through a greater angle than the drive shaft, (100). By introducing this difference of twist between two shafts within a locked system, torque is introduced within the system depending on the magnitude of rotation of the outer set of spur gears about the inner set. As shown in Figure 7, the power return gears and shafts are located beneath the test stand table top. After the test gear specimens are mounted in position, the system is closed and torque introduced at any point will be transmitted throughout the system. The introduction of torque will be controlled by applying pressure to the

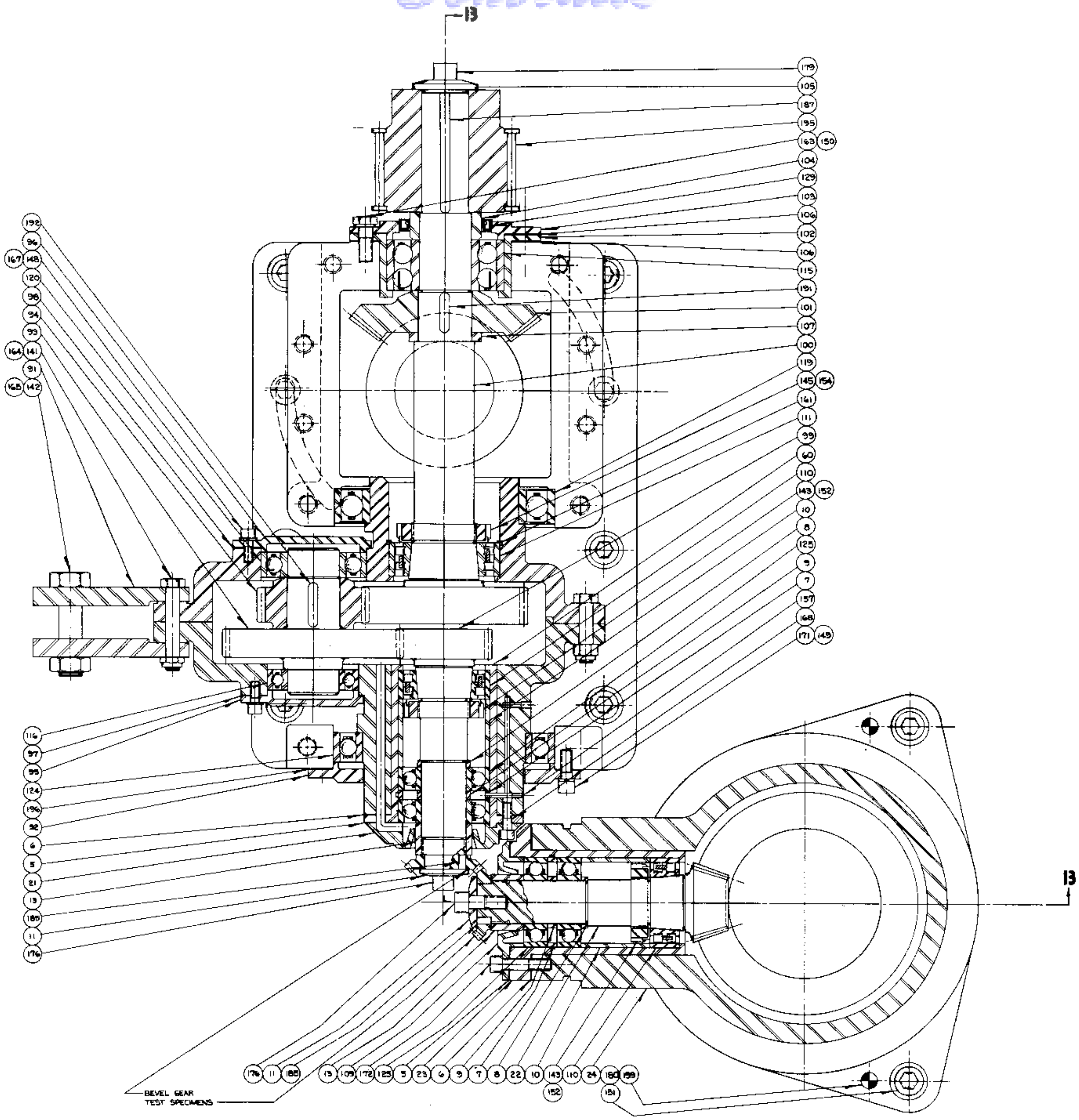
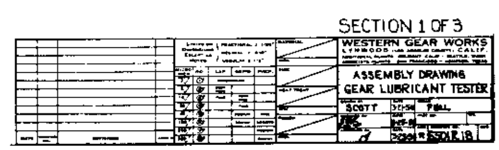


FIGURE 6. TOP VIEW-LUBRICANT
TESTER-BEVEL TEST
GEAR SETUP



hydraulic cylinder, which is connected to the torque arm (91). The torque arm is mounted on the housing, (90), which is free to rotate in bearings (119) and (124). As pressure is applied to the hydraulic cylinder, the housing will rotate through a larger angle thus rotating the cluster gears, (93) and (94), through a larger angle and introducing greater torque and higher loads on the test gears, the pressure is maintained so that there is a locked-up torque in the system. All gears within the system are loaded against each other, but as the loads are balanced, there is no resultant rotation. The gears may now be rotated and the horsepower transmitted by the test-gears is returned by the "power-return" gears. The external drive, therefore, has only to overcome the friction in the system; thus, this method economises in both capital and running costs over the absorption type of rig, in which large power-supplying and power absorbing equipment is required. The introduction of torque by this method permits the varying of tooth loads during operation, as well as providing a means of measuring magnitude of torque load during the test run.

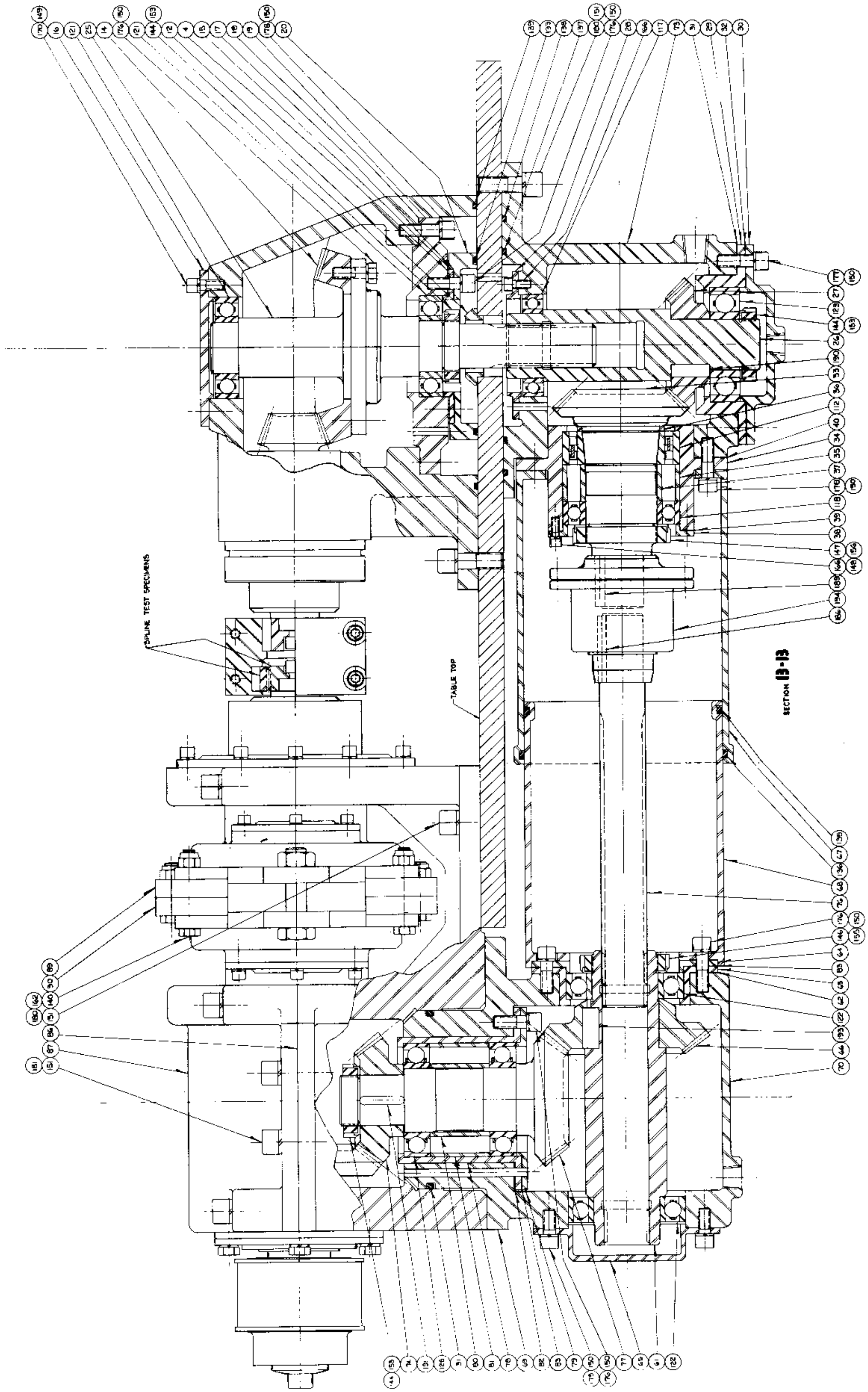
The test specimens are mounted on two high speed spindles, (99) and (22). One spindle, (22), can be moved through a 90 degree arc. The movable head, (24), and spindle, (22), permits the use of test specimens other than spur gears. The movable head, (24), contains a bevel gear set, (22) (14), with a speed reduction of 3:1. This reduction gear set makes it possible to return the power at a speed comparable with the driving speed. The male splined shaft, (25), transmits the power from the movable head, (24), to the return power system. This splined connection facilitates assembly at the different test positions.

The power return system, Figure 7, consists of the following:

(1) Movable return housing, (73), containing a female splined shaft, (26), and a bevel gear set, (27) and (33).

(2) Return shaft housing, (67) and (68), containing a steelfex coupling, (194), and a male splined shaft, (76), capable of extension or retraction.

(3) Return housing, (70), containing a female thru splined shaft, (61), and a bevel gear set, (66) and (77).



ASSEMBLY DRAWING

FIGURE 7. SIDE VIEW - LUBRICANT TESTER - SPLINE TEST SPECIMEN SETUP

The movable return housing, (73), can be moved similar to the movable head, (24), to facilitate mounting of the different types of test specimens. This movement is accomplished by allowing the two return housings, (73), and (70), to rotate about their vertical axis and the connecting splined shaft, (76), to extend or retract in accordance with the varying angle and distances between the two return housings. A telescoping housing, (67), and (68), encloses the splined shaft and coupling. The movable head, (24), and the return housing, (73), will be positioned with dowel pins and bolted to the test stand table at positions in accordance with the specimens to be used in testing.

The vertically mounted bevel gear, (77), in the return housing, (70), transmits power to a set of bevel gears, (74) and (101) in the stationary head, (87). Bevel gear, (101), is mounted on the drive shaft, (100), to which the external drive is supplying power through the pulley, (195), which is also keyed to the drive shaft. The drive shaft transmits the power and torque to the torque box, (90), and the circuit is completed.

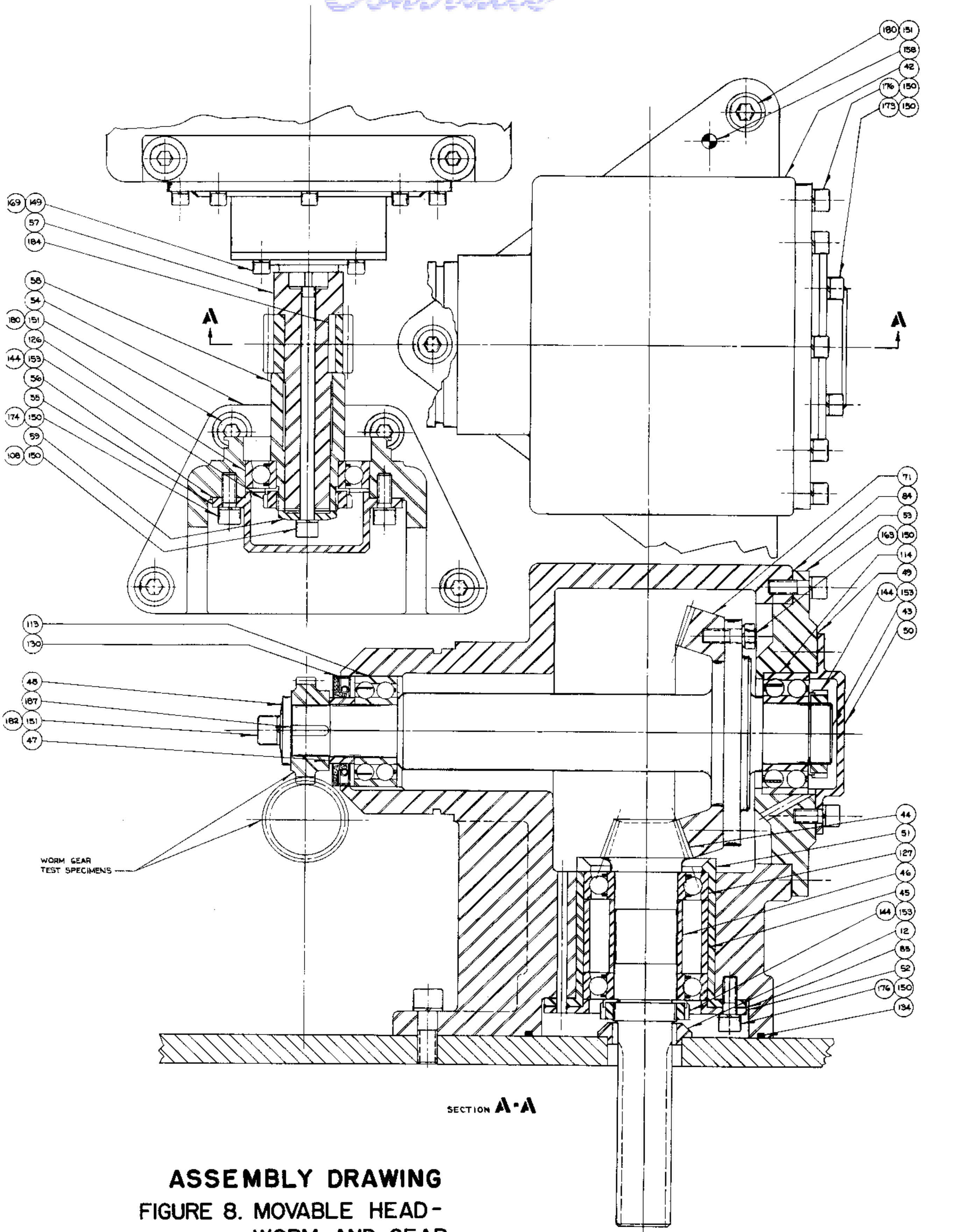
The direction of rotation of the test spindle, (22), in the movable head, (24), is determined by the type of gear test specimens being used; that is to say, the test spindle in the movable head turns one direction for spur gears and in an opposite direction for splines and bevel gears. To return the power to the drive shaft, (100), in the same rotational direction for all tests, it is necessary to have a rotation reversing gear mechanism incorporated in the system. This reversing of shaft rotation will be accomplished by rotating the return housing, (70), Figure 7, about its vertical axis which thereby reverses the direction of the vertical shaft, (77), returning power to the drive shaft. The female splined shaft, (61), in the return housing, (70), is designed to receive the male splined shaft, (76), from either side with a cap, (65), in place on the side not in use during a test. In tests using splines and bevel gears as test specimens, the splined shaft, (76), will be connected as shown in Figure 7. When spur gears are used, the return housing, (70), can be rotated 180 degrees about its vertical axis and the splined shaft, (76), inserted in the opposite end.

Using spur gears as test specimens requires that the movable head, (24), be set in the position as shown in Figure 5. The test specimens listed above are keyed to the test spindles and held in position with an Allen head bolt, (176). The specimens are removed by disengaging the teeth of the test specimens, removing the bolt, and pulling the test specimens from the spindles.

In the testing of lubricants, using worm and gear as test specimens, it is necessary to have a set up similar to that shown in Figure 8. The test worm is mounted on an extended spindle shaft, (57), which is supported in a special bearing pedestal, (54). The test worm is removed by removing cap, (55), unscrewing bolt, (108), and then removing the extended spindle, (57), as a unit. Sleeve, (58), is then removed from spindle, (57), and the test worm is free to be removed from the spindle, (57). Due to the velocity ratio of the worm and gear set, it is necessary to have a special movable head, (42), for the testing of worm gears only. This movable head is shown in Figure 8, and incorporates a 1:3 speed increase bevel gear set, (71) and (44). The ratio of the test worm gear set is 9:1 so it is, therefore, necessary to increase the speed for return through the 1:1 power return linkage at a velocity comparable with the driving velocity of the drive motor. The gear of the test worm gear set is mounted on a spindle shaft, (43), similar to the other types of test gears. The vertical bevel gear shaft, (44), has a male spline on one end for transmitting the power to the movable return housing. The movable head is mounted similar to the movable head used for other test specimens.

The test gear specimens have a pitch diameter of 2 inches, face width of $\frac{1}{4}$ inch, diametral pitch may be varied, as well as material. All test gear sets have a 1:1 ratio except worm and gear set, which has a 9:1 ratio.

As previously described, the test gear housing incorporates an immersion heater for heating the test lubricant. Due to movability of one of the spindles for the use of different types of test specimens, it will be necessary to have different test gear housings for the different positions of the movable head and the varying types and sizes of gears to be used as test specimens. The test gear housings will be mounted as shown in Figure 5.



ASSEMBLY DRAWING
FIGURE 8. MOVABLE HEAD-
WORM AND GEAR
TEST SETUP

SECTION 3 OF 3
6501R18

The test stand table is of a height that permits easy accessability to the test section. Controls and gages are so positioned to afford simplicity of operation and control during a test.

As shown in Figure 4, it is possible to remove the top test section from the test stand in order to place the test section in an altitude chamber for testing lubricants under changing environmental conditions. The removable test section permits only the mechanism and test specimens to be subjected to these environmental changes. Hydraulic, electrical and driving power will be supplied through the wall of the chamber. Controls for varying speed and load will remain outside of the chamber.

In the event tests are to be conducted under vibratory conditions, provisions to be made for the application of external vibration are as follows:

(1) A solenoid operated valve with an electric timer will be placed in the hydraulic line previous to the loading cylinder. The solenoid valve will cause pulsations in the oil pressure applied to the hydraulic cylinder. These pulsations or fluctuations of load will be transmitted to the torque box and thereby vary the torque load applied to the test gear specimens. The frequency of the off-on pulsations may be varied by the electric timer. The rapid fluctuations of load at the test specimens will simulate torsional vibrations set up under actual in-use conditions.

(2) For the application of linear vibrations, provision is made for the mounting of a commercially available vibrator. These vibrators are essentially electromagnetic shakers with a wide range of frequency variation.

The introduction and control of externally applied vibration would present problems to the detriment of the test equipment. If the top section of the tester is vibrated, not only the test specimens, but also the driving mechanism is subjected to these vibrational forces. The application of external vibration tends to complicate the mechanism and limits the life of an expensive piece of equipment as the loading and driving gears would be subjected to all the abusive conditions imposed on the test specimens.

SECTION V

PROBABLE TEST PROCEDURE

The test procedure outlined herein is tentative and a more detailed test procedure will be determined after the tester is in operation.

(1) The test specimens are mounted on the test spindles. Movable head set in position depending on the type of specimens used in the test. Test gear housing is then assembled around the test specimens.

(2) Pour lubricant to be tested in test gear housing and start circulating pump. Adjust by-pass valve for required oil pressure and flow. Adjust heaters or flow of water through heat exchanger to obtain the desired test lubricant temperature.

(3) Start main oil lubricating pump and adjust pressure regulating valve for required pressure and flow to the test mechanism.

(4) Start prime mover and adjust speed to 1000 rpm. (Speeds mentioned herein refer to speed of test gear specimens). Load gear specimens to 1000 pounds per inch of face width by adjusting the manually controlled relief valve to increase the oil pressure to the hydraulic loading cylinder. (Tester will be calibrated to correlate oil pressure on the hydraulic cylinder to load in pounds per inch of face width on the test gear specimens). The test specimens are run for 15 minutes at this speed, 1000 rpm, and load, 1000 pounds per inch of face width.

(5) At the end of run-in time, the prime mover is stopped and test gear specimens are inspected for alignment and scoring. If the gears are in good alignment and are not scored, proceed with test.

(6) Increase speed to 2000 rpm and load test specimens to 2000 pounds per inch of face width. Run for 5 minutes, stop, and inspect gears for scoring. If gears are not scored, proceed with test.

(7) Subsequent tests run at 2000 rpm with load increments of 200 pounds per inch of face width for 5 minute periods until gears score. The normal criterion of failure being the scoring of the test gear teeth.

The foregoing test procedure can be run at any of the speeds within the range of the tester. Maximum test for one lubricant, upon which the design of the tester is based, would be to follow the preceding test procedure at speeds of 2000, 4000, 6000, 8000, 10,000, 15,000, 20,000, 25,000, and 30,000 rpm. Under these conditions, test time required for one lubricant would be approximately 7½ hours, based on the scoring loads shown on the Design Criteria Curve, Figure 2. Increments of loads and test specimen speeds will depend largely on the type of lubricant to be tested.

For testing grease lubricants, it would suffice to either butter the test gears or pack the housing around the test gears with the grease to be tested.

BIBLIOGRAPHY

I. BOUNDARY LUBRICATION

Barwell, F. T. The Effect of Lubrication and Nature of Superficial Layer After Prolonged Periods of Running. Symposium of Properties and Metallurgy of Surfaces. Institute of Metals. November 1952. pp. 101-122.

General review of current concepts of lubrication and wear of metallic surfaces. Mechanisms of continuous wear, scuffing, pitting, abrasion, and fretting are discussed. Bearing-surface modifications such as scuffing due to local high temperatures, plastic flow of surface crystals, and Beilby layer formation are described. Metallographic changes underneath the surfaces of ball bearings subjected to Hertzian-type stress distributions during rolling action are discussed. Photomicrographs are shown of sectioned steel balls having such metallographic changes, and of balls with incipient cracks and pits generated during test runs.

Surface coatings of various types such as oxide layers, phosphated and anodized surfaces, metallic soaps, inorganic salt extreme-pressure lubricants, and polymerized organic and silicone lacquers are described. Methods of formation, frictional and load-carrying characteristics, physical properties, and theories of the lubricating mechanisms of these coatings are discussed.

Barwell, F. T. Research on Friction and Wear. Engineering 172, 4478, 4479. November 1951. pp. 649-651, 697-699.

Paper discusses work in progress on friction and wear. In most cases, final results are not yet available, but the method of attack and the different problems being considered are presented.

The first part contains a discussion of wear on the boundary conditions in bearings. It is shown that phosphating the surface of mild steel produces a film, probably porous, which decreases seizure and scuffing. It is supposed that "extreme pressure additions" to lubricants also produce protecting films, although they may increase wear.

The latter part of the paper discusses various experiments under way to determine the features of hydrodynamic lubrication in different kinds of bearings. Author, in his conclusions, indicates that perhaps more will be learned by the design aspects of bearings than of the lubricants.

Beeck, O., Givens, J. W. and Smith, A. E. On the Mechanism of Boundary Lubrication. I. The Action of Long Chain Polar Compounds. Proceedings of the Royal Society. A177. 1940.

The effect of long-chain polar compounds on the coefficient of kinetic friction under boundary conditions has been studied using Boerlage four-ball friction apparatus in various modifications. With steel balls of the highest grade coefficients of friction, a great number of lubricants were measured as a function of the relative velocity of the rubbing surfaces.

The structure of thin films of these lubricants rubbed on polished mild steel surfaces was investigated by electron diffraction.

It was found that lubricants showing little or no surface orientation had a constant coefficient of friction of about 0.1 over the available velocity range from 0 to 1 cm/sec. With oils which showed high surface orientation imparted by addition of long-chain polar compounds, a sudden decrease of the coefficient of friction was observed at various velocities of the sliding surfaces, depending upon the compound used. By measuring the electrical resistance between the sliding surfaces it was found that the regions of sudden decrease of the coefficient of friction corresponding to a change from metallic contact to extremely high resistance.

The investigation shows that long-chain polar compounds act primarily by inducing the wedging effect and not by giving a direct protection to the surface.

Beeck, O., Givens, J. W. and Williams, E. C. On the Mechanism of Boundary Lubrication II. Wear Prevention by Addition Agents. Proceedings of the Royal Society. A177. 1940.

If two metal surfaces slide over each other in the presence of a lubricant and under high load, high pressure and temperatures prevail at those isolated spots which actually carry the load, leading to wear and possibly to breakdown.

The action of wear preventing agents under these conditions has been studied in detail and it has been found that such agents are effective through their chemical polishing action, by which the load becomes distributed over a larger surface and local pressures and temperatures are decreased. Especially effective are compounds containing phosphorus on other elements of Group V of the perodic system. These have been found to form a metal phosphide or homolog on the surface which is able to alloy with the metal surface, lowering its melting point markedly, and by this action aiding greatly in maintaining a polish.

A wear prevention agent reduces pressure and temperature through better distribution of the load over the apparent surface. Good wear preventing agents can never be good extreme pressure agents, and conversely, agents capable of preventing seizure under extreme conditions will generally not be able to reduce wear. These deductions are in good agreement with the observed facts.

Blok, H. Fundamental Mechanical Aspects of Boundary Lubrication. Presented to SAE, N. Y. May 24, 1949.

The physico-chemical advances cannot be fully exploited without a thorough knowledge of the mechanical aspects of boundary lubrication. Such terms as oiliness, film strength and load carrying capacity should be discarded because of vagueness. A clear definition of the mechanical terms such as real contact area, apparent area, flash temperature, bulk temperature, etc. are given. Types of contact such as concentrated (point and line contact) and dispersed contact (flat surfaces) were analyzed in detail. On the bases of real mechanical conditions existing in boundary lubrication, it was proposed that the following four classifications be recognized.

1. Mild boundary lubrication (low temperature and low press).
2. High temperature boundary lubrication (high temperature and low press).
3. High press boundary lubrication (low temperature and high press).
4. Extreme boundary lubrication (high temperature and high press).

In extreme boundary lubrication the effects of temperature, and not pressure, influences seizure tendency. The role of the lubricant in the above classifications was extensively discussed.

Frictional vibration under conditions of boundary lubrication was discussed. Critical speeds were found at which vibration occured with a slip-stick seizure resulting. The frictional vibration was thought to cause the seizures and not the seizures causing the vibration. Theory of vibration connected with boundary lubrication and a discussion of current literature on the subject were given.

Bowden, F. P. The Influence of Surface Films on the Friction and Deformation of Surfaces. Symposium of Properties and Metallurgy of Surfaces. Institute of Metals. November 1952. pp. 197-212.

Paper is an exposition of the facts of a large variety of surface films on sliding of both metals and nonmetals. It bears out the statement that friction and damage are influenced more by these films than by the materials themselves.

Degassed metals seize when slid and indicate coefficient of friction over 100. Small amounts of gases absorbed on surface greatly reduce friction and damage. Similar effect is observed with nonmetals, but of smaller magnitude. Other surface conditions reduce wear but not necessarily friction. Relative hardness of oxide and base metal determines effectiveness of this film. Hard oxide on soft base such as tin and aluminum breaks under light load. Hardened Beilby layer of polished surface provides similar protection. Pronounced reduction in wear with boundary lubricants is shown to be result of solid metallic soap formed on surface. These are not effective beyond the melting point of the soap. Chlorine and sulfur are not effective below temperatures which will cause attack of the surface. Low shear-strength films such as indium, molybdenum, disulfide, and polytetrafluorethylene may also be interposed to prevent damage to sliding surfaces.

Reviewer believes such fundamental understanding of the mechanism of prevention of damage to be especially valuable when considering problems involving sliding surfaces.

Bowden, F. P. The Mechanism of Boundary Lubrication. Proceedings of the Third World Petroleum Congress. The Hague. Section 7. 1951. pp. 328-348.

To date, boundary lubrication has been studied much less intensively than hydrodynamic lubrication. This is due in part to the fact that the mechanisms and parameters involved when bearing surfaces are separated by irregular lubricant films of only molecular dimensions have not been thoroughly understood.

Starting with description of metallic friction and wear mechanisms, author proceeds to describe influence of chemical attack and temperature on boundary lubrication. Basic principle is that sliding friction is severely increased by localized adhesion at tiny contact points. Oxide films, chemical reactions, or thin films or low shear-strength materials on hard substrates reduce friction by inhibiting welding or permitting easy surface smearing. All postulations are demonstrated by well-designed experiments using electron diffraction and radioactive tracer techniques.

Bowden, F. P. The Influence of Surface Films on the Friction, Adhesion, and Surface Damage of Solids. Annals N. Y. Academy of Science 53. June 1951. pp. 805-823.

A review and extension of work by Bowden and associates on the role of surface films. Friction of outgassed materials and surfaces exposed to contamination by a variety of substances is discussed. The performance of some lubricant additives is explained by these observations. Work on friction and adhesion between steel and indium lends further support to the "welded-junction" theory of friction.

Bowden, F. P. and Tabor, D. The Friction and Lubrication of Solids. Oxford. Clarendon Press. 1950. p. 337.

Monograph presents the major findings of the authors and their associates in the Laboratory on the Physics and Chemistry of Solids at Cambridge University, England, and covers a period of research of approximately fifteen years. The text is extremely well written and illustrated.

Subject matter treated is primarily for surfaces in full or partial metallic contact, little space being devoted to the hydrodynamical aspects of lubrication; consequently, the text is essentially nonmathematical. Many beautifully designed experiments of unusual simplicity are used to establish key concepts associated with such topics as: the real area of contact between solids, the surface temperature of sliding surfaces, the nature of polishing, the physical nature of metallic and nonmetallic friction, the role of the contaminating film, and the boundary friction of lubricated surfaces.

Topics are included that might not be expected to be found in a work of this sort. These include a treatment of the nature of contact between colliding solids, and a discussion of chemical reactions produced by friction and impact. The manner in which explosives decompose is considered as part of this latter material.

Being essentially a monograph, the book follows closely the researches of the authors. As a result, the coverage is not well balanced on the basis of modern textbook standards. The space allotted to dry friction and its causes far outweights that assigned to wear phenomena and the chemical aspects of boundary lubrication. However, the monograph approach makes it possible to gain the reader's confidence and provides very stimulating reading. This book will be invaluable to those doing research in the field of boundary friction and related topics, and will exert a major influence in the thinking in these areas for many years to come.

Bristow, J. R. The Measurement of Kinetic Boundary Friction or the Experimental Investigation of "Oiliness". Proceedings of the Institution of Mechanical Engineers. 160. December 1949. pp. 384-402.

Apparatus for determining variation of coefficient of boundary friction with sliding velocity at very low velocities is described; and a collection of test results obtained with different lubricants using a variety of temperatures, metals, and surface finishes is presented. The dynamic aspects of test apparatus for boundary lubricant is discussed and significance of stick-slip phenomenon reviewed. Author contends that results presented support theses originally put forth by Sir William Hardy.

Brummage, K. G. An Electron-Diffraction Study of the Heating of Straight-Chain Organic Films and Its Application to Lubrication. Proceedings of the Royal Society. Series A. Vol. 191. November 11, 1947. pp. 243-252.

Thin films of normal paraffins, fatty acids, and esters upon surfaces of aluminum, cadmium, copper, nickel, silver, mild steel and stainless steel, were examined by electron diffraction during heating. In all cases the diffraction patterns showed the films to change from an orderly crystalline arrangement of molecules to an expanded state of individual molecules at a rather definite temperature. This disorientation was found to be reversible, the orderly arrangement returning when the specimen was cooled.

Films deposited by either the Langmuir-Blodgett technique, or from isohexane, gave the same disorientation temperatures within an experimental precision of 2 to 5 C. The disorientation temperature was found to increase with film thickness and to depend also upon the substate and film materials. While no explanation of the influence of these variables upon the disorientation temperature is offered, a correlation was observed between the bulk melting point of the film material and the disorientation temperature for members of a homologous series.

The disorientation temperature was found to correspond to the temperature at which the smooth friction of a lightly loaded, slowly moving slider abruptly increased with increase in temperature. The temperature marking the onset of "stick-slip" sliding was generally found to lie above the disorientation temperature, which indicates that even an expanded film is absorbed on the metal surface. It is suggested that the few cases in which the smooth friction transition point is observed to lie above the disorientation temperature are due to the formation of a chemical compound between the substrate and the film.

Charron, F. Viscosity Under Rapidly Changing Pressure. (Viscosite sous Pression Rapidement Variable) C. R. Academy Science, Paris. Vol. 225. November 17, 1947. pp. 919-921.

The author has investigated the question whether the change in viscosity with pressure of lubricants is an instantaneous change, or whether the viscosity change is a function of the time. While the problem is complicated by temperature changes due to the pressure changes, his experiments with a "ballistic viscosimeter" (C. R. Acad. Sci., Paris, 1947, vol. 224, pp. 1472-1474) have indicated that the viscosity does not reach its steady-state value instantaneously, but that there is a certain hysteresis or time lag in the viscosity increase.

Further experiments are planned to prove this point and to investigate it further. The effect is important for those applications of lubricants where shock or impact in the bearings may cause very high pressures of short time duration.

Clayton, D. An Introduction to Boundary and Extreme Pressure Lubrication. British Journal of Applied Physics Supplement. No. 1. January 1951. pp. 25-32.

This comprehensive review on boundary and extreme pressure lubrication includes explanations of boundary friction, in particular, the adhesion theory. Dealing with the influence of the main parameters on friction values, velocity is considered first, together with explanation of friction-induced vibrations in terms of the main type of friction vs. velocity curve. A simple general statement on the effects of temperature and surface roughness on friction cannot yet be given. Brief references are made to the high temperatures reached at sliding contacts, and to electrical effects. In terms of these fundamental concepts, a sketch is given of the mechanism of running in, of wear (including wear-testing and anti wear additives), of the behavior of surfaces under severe conditions leading to scuffing (including the function of extreme pressure lubricants). Finally, the value of solid lubricants is mentioned, and also the benefits that can be obtained by chemical pretreatment of solid surfaces.

Johnson, R. L., Godfrey, D. and Bisson, E. E. Friction of Solid Films on Steel at High Sliding Velocities. NACA TN 1578. April 1948. pp. 1-65.

This research is a continuation of a study reported by Johnson, Swikert, and Bisson (Nat. adv. Comm. Aero. tech. Note, no. 1442, Oct. 1947). The authors report the results of friction and wear studies made with surfaces coated with various inorganic films which are of importance in run-in, in extreme-pressure lubrication, and in supplementary or dry lubrication. Additional studies were made using standard physical, chemical, and metallurgical techniques, including electron diffraction. The relation of the findings to friction and lubrication theory and practice is discussed.

The apparatus used for the friction and wear tests consisted basically of an elastically restrained spherical steel rider sliding on a rotating steel disk. The range of sliding speeds employed was from 50 to 8000 fpm with loads from 169 to 1543 grams (108,000 to 225,000 psi, initial Hertz surface stress). The inorganic films studied were Fe_2O_3, Fe_3O_4, $FeCl_2$ FeS, MoS_2, and graphite. These were applied to the disk surface only. Of these Fe_2O_3, $FeCl_2$, MoS_2, and graphite were found to be the most beneficial as regards friction and wear.

Johnson, R. L., Godfrey, D. and Bisson E. E. Friction of Surface Films Formed by Decomposition of Common Lubricants of Several Types. NACA TN 2076. April 1950. p. 28.

The effect on friction of decomposition films formed by heating various petroleum and synthetic lubricants in contact with a steel surface is studied, using an apparatus employing an elastically restrained spherical rider moving in a spiral path on a rotating disk. The decomposition films cause significant reductions in friction. A thin film of the decomposition product of a silicone polymer in conjunction with the undecomposed liquid was particularly effective in lowering friction. Authors believe these low values are due to low shear strength of the film rather than hydrodynamic action of a very viscous film.

Johnson, R. L., Peterson, M. B., and Swikert, M. A. Friction at High Sliding Velocities of Oxide Films on Steel Surfaces Boundary Lubricated with Stearic Acid Solutions. NACA TN 2366. May 1951.

Friction experiments indicated that lubrication with stearic acid as an additive in cetane was effective at sliding velocities up to 3000 ft/min. both for clean steel surfaces and for surfaces coated with Fe_2O_3 (1000 A° thick). A prepared film of Fe_3O_4 (1000 A° thick) prevented lubrication failure with 0.5 percent stearic acid in cetane at sliding velocities higher than 7000 feet per minute. It is suggested that the lubrication failures found at a sliding velocity of 3000 ft. per minute are caused by the melting of the metallic soap formed at the liquid metal interface. The type of oxide and the film thickness are important in determining the effectiveness of stearic acid as an additive in lubrication at high sliding velocities.

Johnson, R. L., Swikert, M. A., and Bisson, E. E. Friction and High Sliding Velocities. NACA TN No-1442. October 1947.

Reports experiments conducted with kinetic-friction apparatus with steel specimens over ranges of speeds between 50 and 6600 ft./min. and loads from 169 to 2232 grams.

Kinetic friction decreases with sliding speed for dry and for some boundary-lubricated surfaces at high sliding speeds. Amonton's law was verified at all speeds with dry and boundary-lubricated surfaces. Changes in physical characteristics of material resulting from high surface stressing and occurrence of ferrous oxide on the sliding surface are possible causes for reduction in kinetic friction with high sliding velocities.

Tabor, D. Collisions through Liquid Layers. Engineering. Vol. 167. February 18, 1949. pp. 145-147.

The author presents a study of collisions between metal spheres and between spheres and flat metal surfaces. Experiments with contact surfaces dry and again with films of lubricating oils between the surfaces are discussed. It is found from measured electrical conductance across the area of contact that the viscosity of the oil influences the condition under which metal to metal contact occurs. There are cases in which metal to metal contact does not occur but nevertheless plastic deformation of the metal does occur.

Calculations based on the theory of viscous flow of fluids indicate that fluid pressure in excess of the yield stress for the metal and fluid temperature as high as 3000 C can be developed. Such pressure and temperature conditions have an adverse effect on the lubricant.

The results of the investigation are said to be of importance in the selection of lubricants for such machine elements as gears and bearings.

Tabor, D. Mechanism of Boundary Lubrication. Proceedings of the Royal Society. Ser. A. Vol. 212. May 22, 1952. pp. 498-505; Disc. p. 516.

How the theory of metallic friction is modified in the presence of boundary films. Minute metallic junctions contribute to the frictional resistance and play a major part in wear of lubricated surfaces. Effect of temperature shows that the most effective lubrication is provided by a solid boundary film which possesses a close-packed, strongly oriented structure. A recent study of metal transference from one surface to the other as sliding takes place, using radioactive metals, shows that, at the melting point of the lubricant film, a marked increase in pickup and friction occurs. At still higher temperature, a second deterioration in lubricating properties occurs, corresponding to desorption of the lubricant film.

I.-A. PHYSICAL ASPECTS

Almen, J. O. Dimensional Value of Lubricants in Gear Design. SAE Journal. September 1942. pp. 373-378.

Explaining his title, Mr. Almen points out that, as a result of the modern demand for more intensive use of structural materials, many design details are dependent upon specially compounded lubricants, and these lubricants have thus become inseparable parts of the design. Mr. Almen establishes the hypothesis that lubricant failure in gears and other highly loaded surfaces results in welding of small areas of the mating surface for the following reasons:

1. The rubbing action removes weakly adhering films permitting contact between clean metal surfaces.
2. The temperature of a thin surface layer is very high from the friction of sliding under high load.
3. Welding of two surfaces can occur at temperatures considerable below the melting points of a metal if the pressure is great.
4. Welding will not occur if a sufficiently tenacious contaminating film is formed on the rubbing surfaces.

Considerable data, including three dimensional charts, are presented to show the effect on the scoring limit of the lubricant, the pressure, the temperature, the sliding velocity, and the hardness.

Burwell, J. T. and Strang, C. D. On the Empirical Law of Adhesive Wear. Journal Applied Physics. January 1952. pp. 18-28.

Authors start from the premises that wear increases in a general way with distance of travel or time of running, but generally not in a linear fashion, and that wear generally decreases with increasing hardness of the rubbing surfaces.

A simple fly wheel apparatus is described with a rubbing pin of several shapes, cylindrical, conical, and spherical. For a number of different low loads, the volume of abraded material is linear with distance of travel, provided that a steady condition of operation was reached before any data were taken. The highest load corresponded to about 2/3 the yield strength of the rider material. Experiments in a higher stress region, where the rate of wear increases very rapidly with stress to relatively high values, indicate that the sharp increase in slope of the h/LP vs. pressure curve (h is average depth of material removed, L distance of travel, P average normal stress over the nominal contact area) cannot be a result of breakdown of the liquid lubricant film at some critical stress. The bearing of the results on the running-in of machine parts is discussed.

Bisson, E. E., Swikert, M. A., and Johnson, R. L. Effect of Chemical Reactivity of Lubricant Additives on Friction and Surface Welding at High Sliding Velocities. NACA TN 2144. August 1950.

Kinetic friction experiments were conducted on steel specimens lubricated with cetane containing sulfur and chlorine compounds of different chemical reacturties. Data were obtained at sliding velocities from 75 to 7000 feet per minute with loads from 269 to 1017 grams (126,000 to 194,000/in.^{2} initial Hertz surface stress). Greater chemical reactivity of additives increased the critical sliding velocity at which lubrication failure occurred, thereby substantiating a limiting rate hypothesis. Activity of the active atoms was more important than the number of active atoms available for reaction. Increasing the load decreased the critical sliding velocity as a linear function of initial Hertz surface stress.

Burwell, J. T. and Strang, C. D. Metallic Wear. Proceedings of the Royal Society. Serial A. Vol. 121. May 22, 1952. pp. 470-477.

Experiments on wear between a cylindrical metal pin and a hardened steel disk. Under steady-state conditions at light loads, it was found that volume of material worn away is proportional to load and to length of path traversed. Results are discussed in relation to the problem of running-in newly assembled machine parts.

Johnson, R. L. and Swikert, M. A. Friction at High Sliding Velocities of Surfaces Lubricated with Sulfur as an Additive. NACA TN 1720. October 1948.

Kinetic friction experiments were conducted on steel specimens lubricated with sulfur-type E.P. lubricants over ranges of sliding velocities between 50 and 800 ft. per minute and loads from 269 to 1543 grams (126,000 to 225,000 lb.-in.^{2} initial Hertz surface stress). Supporting evidence was found for a theory that rate of chemical reaction is a limiting factor in lubrication by E.P. additives of surfaces operating at high sliding velocities. Variations in load and sulfur concentration had no appreciable effect on the sliding velocity at which lubrication failure occurred.

Schnurmann, R. Thermoelectric Experiments with Extreme-Pressure Lubricants. Journal of Applied Physics. Vol. 20. April 1949. pp. 376-383.

Extreme-pressure lubricants (EPL) are designed to reduce friction at high pressures, and prevent abrasion of the surfaces of the rubbing elements. The teeth of hypoid gears are subjected to pressures up to 1.5 to 3 X 10^4 kg per cm

For testing EPL the author employs a device in which a rotating steel ball is pressed against the flat surface of a metal block (Timken test block, flat surface of a steel roller, etc.). The thermal electromotive force generated at this contact is taken as a measure of friction. If lubrication is insufficient, abrasion takes place and the galvanometer reading increases at a greater rate than the normal load. With pure mineral oil at 18 rps, for instance, the electromotive force was proportional to the normal load only up to a load of 80 kg. The addition of 2 per cent by volume of chlorinated paraffin wax extended this region of proportionality to a load of more than 128 kg. The final wear impression had a diameter in the first case of 1.61 mm, in the latter of only 0.79 mm.

The author employs EPL of various kinds, with different concentrations and several metals and metallic films. The results suggest the existence of an optimum concentration of additive. The author makes an attempt to explain this fact physically.

.-C. MECHANICAL WEAR

Barwell, F. T. Some Aspects of Research on Friction and Wear. Transactions of the Institution of Engineers and Shipbuilders in Scotland. 1951-1952. 95. Part 2. pp. 64-91.

Paper is an excellent discussion of problems inherent in rubbing surfaces and current research toward understanding and solution of these frictional problems. Topics discussed include nature of surfaces, effect of surface oxides on lubrication, possible mechanisms of wear, fretting corrosion, and extreme pressure lubricants. Turning to hydrodynamic lubrication, author touches upon dynamic loading, fluid pressures between parallel surfaces, and experimental studies of high-speed bearings, including vortex flow in eccentric bearings. Lubrication of ball bearings is treated briefly and a possible explanation is put forth for the wide variation in the lives of similar ball bearings.

Blok, H. Gear Lubricant-A Constructional Gear Material. Ingenieur 63, 39, 0.53-0.64. September 1951.

The power-transmitting capacity of highly loaded tooth gears is influenced by their lubrication. Hydrodynamic lubrication (full fluid film) is seldom realized in tooth gears. The lubrication is generally a boundary lubrication (the load borne by direct contact tooth to tooth), and the scoring of the tooth faces may be a critical phenomenon; this often can be avoided by using special oils, the so-called extreme pressure (E.P.) oils, which contain chemically active additives.

Blok, H. War on Wear. Engineering 173, 4502, 4503. May 1952. pp. 594, 625-626.

Wear is the undesirable migration of the material of a solid surface due to overstressing caused by mechanical forces through continuous or repetitive motion relative to a fluid or solid in contact. Wear may be single-sided, as in the cavitation erosion by a fluid, or double-sided for two mating solid surfaces. In the latter case, besides the temperature effects, the surface contact may be dispersed or concentrated, the surface finish may present summits to plastic flow, and the surface pressure may exist with or without sliding. Pitting of gear teeth is a surface fatigue which is often arrested by strain-hardening, a self-restoring factor. Abrasive wear may be caused by an interference or cutting action, or by the interlocking of the surface irregularities. The wear particles cause further erosive abrasion. Adhesive wear occurs between materials having high adhesive qualities--junctions may be formed requiring the forced separation by tearing and shearing.

Protective measures to reduce wear are discussed under three classifications: (1) The principle of the protective layer includes protective plating and chemical coatings, and contact inhibitors such as hydrodynamic fluid films, boundary layers, and the absorption properties of extreme pressure lubricants; (2) the principle of conversion considers the use of lubricants, surface materials, and proper design to convert destructive to permissive wear; and (3) the principle of diversion of wear from one surface to another.

Wear is controlled by the choice of lubricants, materials, and design. Further research is suggested particularly for quantitative analysis in the problem of wear.

Bowden, F. P. and Ridler, K. E. W. The Temperature of Lubricated Surfaces. Proceedings of the Royal Society. Series A. Vol. 154. May 1, 1936. p. 640.

A method is described for measuring the surface temperature of sliding metals. The temperature depends upon the load, speed, coefficient of friction and thermal conductivity of the metals, and is in good agreement with theory. With readily fusible metals the surface temperature reached corresponds to the melting point of the metal. With less fusible metals the local surface temperature may exceed 1800°F.

Even with lubricated surfaces the temperature (under boundary lubrication conditions) is high and may exceed 1100°F. This high surface temperature will cause a local volatilization and decomposition of the lubricant and is a cause of the breakdown of the boundary film.

Bowden, F. P. and Tabor, D. The Seizure of Metals. Institute of Mechanical Engineering Journal. Vol. 160. No. 3. 1949.

When metal surfaces are placed together, the area over which they touch is usually very small. As a result the localized pressures at the points of real contact are sufficiently high to produce plastic flow of the metal. Even under static conditions, these high pressures may produce "cold welding" between the surfaces, and the metallic junctions so formed may be very strong. These junctions are sheared during sliding, and if they are stronger than the parent metals heavy damage may result which is not limited to the interface at which intimate contact occurs. The growth and extension of this localized damage constitutes seizure.

Bowden, F. P. and Others. A Discussion on Friction. Proceedings Royal Society. Series A. 212, 1111. May 1952. Pages 439-520.

Fundamental and significant work is reported on three phases of friction: Friction of metals, friction of nonmetals, and boundary and extreme pressure lubrication. In the first category, results reported indicate that, with most metals, natural oxide layer is sufficient to prevent metallic contact at small loads, and that surface roughness, thickness of oxide film, and relative hardness of oxide and metal substrate affect degree of protection afforded bare metal. Coefficient of friction is found to decrease linearly with hardness. Metallic surfaces sliding together become coated with an amorphous or Beilby layer formed when asperities become molten and smear out. Coefficient of friction between clean surfaces is found to be on the order of 25 to 100. Metallic wear is found to be proportional to load and to distance traversed with load pressures below 1/3 the hardness; with loads above this value, wear rate is greatly accelerated.

In the second category, frictional behavior of several plastics and effect of outgassing on frictional properties of diamond, graphite, and carbon are reported. Friction between single textile fibers is reported to decrease with increasing load.

In the third category, desirable characteristics of boundary lubricants and effect of surface condition are discussed together with an interferometric technique for measuring thickness of surface films. Action of extreme pressure lubricants is explained.

Bowden, F. P. and Moore, A. J. W. Internal Stresses Produced by the Sliding of Metals. Institute of Metals Monogram Report Serial No. 5. 1948. pp. 131-137.

When a hemispherical steel surface slides over a copper surface, a clearly defined track is formed. Direct examination shows little apparent difference between the track produced under clean and under lubricated conditions. However, for clean sliding, sections reveal much more severe distortion of the copper beneath the surface of the track. Sections were made at 90 deg to the direction of sliding and 5 deg to the surface of the steel. Study of the clean sliding of copper on steel shows that adhesion and a plucking out of the steel occurs at the interface. When the surfaces are lubricated, a similar process takes place on a reduced scale. When a copper contact slides on a copper surface, the damage is much more severe.

It is concluded that the internal stresses produced below the surface of the copper are formed by the formation and shearing of metallic junctions during sliding. Continued sliding would give a distorted layer extending far beneath the surface.

Buckinham, E. Qualitative Analysis of Wear. Mechanical Engineer. August 1937. pp. 576-578.

Names six types of wear, and identifies, with a reasonable degree of accuracy, both their causes and their effects. (a) Pitting, appears to be the result of compressive fatigue of the material caused by the repeated high compressive stresses set up by the contact of curved surfaces under load. When these stresses exceed the compressive endurance limit of the material, particles or flakes are sheared out of the surface. (b) Abrasion, is caused by the presence of foreign matter such as grit or metallic particles between the contacting surfaces. On gear teeth, it results in scratches or fine grooves running more or less at random in direction of sliding between surfaces. (c) Shearing, or cutting, takes place when because of other types of wear, rough surface finish, misalignment of the parts, or other inperfections, sharp corners or edges are present which cut through the oil film and score the mating surfaces. (d) Spalling occurs when the shear stresses set up by the movement of the elastic and plastic wave ahead of the contact area between the curved surfaces exceed the shear strength of the material. In some respects is similar to pitting but does not appear to be a phenomenon that is caused by fatigue. (e) Galling or scuffing, results from a momentary failure of the oil film, causing high local temperatures and also a plastic flow of the skin surface of the material. (f) Seizing, is an extreme case of galling, local temperatures are so high during momentary failures of the oil film that particles of material are actually welded or brazed onto the contacting surfaces. Of the six foregoing types of wear, failure of lubrication is responsible for only the last two, galling and seizing.

Burwell, J. T. Mechanical Wear. American Society for Metals. January 1950.

The papers in this volume were all presented in connection with a special summer conference on Mechanical Wear, which was held at the Massachusetts Institute of Technology, Cambridge, Massachusetts in June 1948.

The papers cover laboratory and service experience on wear in internal combustion engines, steam turbines, brake materials, journal bearings, gears, etc.

The papers also cover the effect of various factors on wear in any or all of the above types of service.

The confused nonmenclature in the field, and lack of clear definition of such common words as "scuffing", "scoring", "galling", and "erosion", testify to the state of undigested knowledge. On the other hand, mechanical wear and means of curing it are becoming daily more important in modern industry.

For these reasons, the M.I.T. conference was held and the purpose of this volume is to make available to a wider audience the valuable papers and principal discussions.

Chalmers, B., Forrester, P. G., and Phelps, E. F. Kinetic Friction in or Near the Boundary Region. Proceedings of the Royal Society. Series A. Vol. 187. December 13, 1946. p. 439.

An investigation has been made of the causes underlying change of friction with sliding velocity. The method adopted was to measure the friction of several different combinations of materials under three different conditions: dry, with excess of various lubricants, and with thin films of various lubricants applied by two different methods. The experiments made show that changes in friction with velocity may be derived from at least three sources. First, such a change may be derived from the properties of the clean metal surfaces. Certain combinations of materials show, when unlubricated, a decrease in friction with increasing velocity, and this tendency may remain in a modified form even when a lubricant is added. Various hypotheses to explain this tendency are discussed. Secondly, with a boundary film of lubricant present, friction may increase with increasing velocity, and the evidence suggests that this change is due to partial destruction of the boundary film, the rate of which destruction rises with increasing sliding speed. Thirdly, when there is excess of lubricant, fluid film effects may occur even with restricted areas of contact and sliding speeds as low as 0.5 cm/sec.; such fluid film effects bring about a decrease in friction with increasing velocity. This tendency is most marked when one component is a soft metal, and may well explain the good antifriction properties of the Babbitt and similar alloys. A brief investigation was also made of the effects of surface finish and of continual sliding or "running-in", and the results obtained are discussed in the light of information gained from the experiments on sliding velocity.

Clayton, D. and Jenkins, C. H. M. Physical Changes in Rubbing Surfaces on Scuffing. British Journal of Applied Physics. Suppl. No. 69-77. January 1951.

Cast-iron surfaces rubbing against a steel ball have been found to develop a thin 10^{-4} or 10^{-5} inch layer of a "white constituent" when scuffing occurs or is approached. This material has not been fully identified, but the results point to it having a basis of two phases, viz., cementite and a quenched high-carbon ferritic phase developed from austenite, resulting from the high temperature developed by rubbing. In laboratory work, using a 4-ball apparatus modified to take piston-ring segments, this white material was also obtained in some degree with mineral oil under milder conditions than those causing scuffing, but not with the castor oil which proved better in preventing scuffing in running-in an aero-engine. Very excellent microphotographs show a smooth surface of this white material interrupted by cracks where the material has torn out. Paper points out a very interesting effect which should be investigated further.

Evans, L. S. and Tourret, R. The Wear and Pitting of Bronze Disks Operated Under Simulated Worm-Gear Conditions. Journal of the Institute of Petroleum. 38, 344. August 1952. pp. 652-668.

Authors study wear and pitting occurring during rolling and sliding contact of peripherally loaded disks. Problem is applicable to worm-gear wear, as in automotive transmissions. Bronze disk with lands is operated against steel disk of smaller diameter. Wear is determined by measuring the decrease in depth of calibrated scratch. Pit formation, and surface and subsurface cracking are determined by photomicrography of cross sections.

Authors suggest that wear is caused by surface becoming hardened and brittle so that small particles become detached. Pitting is initiated by either surface or internal cracks, or both. Mechanisms of formation of surface and internal cracks are discussed, and crack propagation by hydraulic forces in lubricant is postulated.

Feng, I. M. Metal Transfer and Wear. Journal of Applied Physics. 23, 9. September 1952. pp. 1011-1019.

The interface between each pair of opposed high spots of two solid surfaces pressed together is shown to be roughened to the form of a saw-tooth profile by the plastic deformation that occurs. This new experimental finding is the basis of author's hypothesis that the perfect matching and the consequent mechanical interlocking so achieved at the interface are the primary cause of the shear component of friction and of the two types of wear characterized by metal transfer and by the formation of loose wear particles, respectively.

In contrast to Bowden and others, author considers welding as the consequence and not as the cause of friction. More precisely, the shearing induced by sliding is supposed to take place not at the interface but, because of strain-hardening, at a section at some distance from and more or less parallel to the interface. A temperature flash is thought to be induced at the interface by the heat of shearing, and thus the sheared-off peak may become welded to its opposed high spot, provided that diffusion at the interface is rapid enough and the consequent adhesion is strong enough. In cases where this condition is not fulfilled, the sheared-off peaks would form loose wear particles.

Author claims that his hypothesis solves several explanatory difficulties inherent in the older, the welding hypothesis. One such difficulty lies in explaining the formation of loose wear particles and the preponderance, in certain cases, of this kind of wear as compared with metal transfer. Another such difficulty relates to the high order of magnitude (100 and more) of the coefficient of friction between two clean metal surfaces. Reviewer finds the new hypothesis exceedingly thought-provoking, but feels that its correctness can be judged fully only after the additional evidence announced by the author has been published.

Godfrey, D. Investigation of Fretting Corrosion by Microscopic Observation. NACA TN 2039. February 1950.

An experimental investigation using microscopic observation of the action was conducted to determine the cause of fretting corrosion. Glass and other noncorrosive materials, as well as metals, were used as specimens. Convex surfaces at frequencies of 60 cycles or less than 1 cycle per second, an amplitude of .001 in. and a load of 0.2 pound.

Fretting corrosion was concluded to be caused by the removal of finely divided and apparently virgin material due to inherent forces and that its primary reaction is independent of vibratory motion or high sliding speeds. Fretting corrosion occurred to clean non metals and metals readily and glass microscope slides and steel balls provided an excellent method for visual studies.

Godfrey, D. and Bisson, E. E. Effectiveness of Molybdenum Disulfide as a Fretting-Corrosion Inhibitor. NACA TN 2180. September 1950.

Six methods of applying molybdenum disulfide MOS_2 to steel as a fretting-corrosion inhibitor were evaluated. Inhibitor action between steel balls vibrating in contact with glass flats was microscopically observed; number of cycles required to produce incipient fretting corrosion was noted. Experiments with steel flats vibrating in contact with steel flats were also conducted. A bonded MoS_2 coating delayed fretting corrosion for 28,000,000 cycles for steel against glass and about 10,000,000 cycles for steel against steel. Dry MoS_2 and mixtures with water, oil, or grease appreciably delayed fretting corrosion, without an inhibitor, specimens fretted immediately. Effectiveness of any inhibitor is dependent on its ability to prevent metallic adhesion continuously.

Lane, T. B. and Hughes, J. R. A Practical Application of the Flash-Temperature Hypothesis to Gear Lubrication. Proceedings of the 3rd World Petroleum Congress. The Hague. Section 7. 1951. pp. 320-327.

The postulate of Blok, that seizure will occur in gears when the temperature in the contact reaches a value characteristic of the combination of sliding system and lubricant, is investigated to explain the behavior of oils in a gear rig at different speeds. An empirical formula is given confirming Blok's hypothesis, but this formula is only applicable to the special experimental arrangement used for the investigations.

Lloy, J. M. Literature Survey of the Effect of Sub-Zero Temperatures on Fuels and Lubricants. NRC MP. July 1952.

The purpose of this literature survey was to provide in a readily accessible form a compilation of abstracts dealing with the sub-zero operation of aircraft and vehicles with particular emphasis on the part played by petroleum products in such operation.

The main difficulty in dealing with the problem, which is quite evident in the literature, is the large number of variables involved. Many of the authors of the papers reviewed here have endeavored to list these, but considerable disagreement still exists as to which variables dominate in a given case.

Until the relative importance of the variables is established, it will not be possible to define "limits" for satisfactory operation at sub-zero conditions.

Mansion, H. D. Some Factors Affecting Gear Scuffing. Journal of the Institute of Petroleum. 38, 344. August 1952. pp. 633-645.

The tested gear was driven by another similar gear with pressure loads rising from about 10 to about 70 lb., usually at a speed of about 2000 rpm. The load at the beginning of scuffing (in 5 minutes) measures the quality of the lubricant. Main results: Scuffing load rises with rising kinematic viscosity of the lubricant at a given temperature; the influence of the temperature is small in the range of 40 to 110 C; suitable additions to the lubricant may increase the scuffing load by a factor of up to 3.

Ovenhoff, R. F., Hand, J. W., and McGinniss, J. H. Bimonthly Report No. 10 on Full-Scale Engine Performance on Aviation Oils. Report No. RL-6M-52 (38) ASTIA July 1, 1952.

Progress is reported in the investigation of full-scale engine performance of aviation oils. Test No. 7 was carried out to check the reproducibility of the SOD full-scale test operation by evaluating a straight mineral oil that was almost identical with the reference lubricant used in Test No. 1. The lubricant used in Test No. 7 (grade 1100 aviation oil) was blended from solvent-extracted mid-Continent base stocks and contained no additive except for a small amount of an approved pour depressant. The reference oil used in test No. 1 (grade 1120 aviation oil) was blended from the same components in slightly different proportions. Results obtained in test No. 7 show that the engine test is satisfactorily reproducible and that the performance of the test oil agrees very well with that of the reference oil (test No. 1) in all respects.

Pomey, J. Friction and Wear. NACA TM 1318. March 1952.

Report covers general survey of field of friction and wear, including discussions of hydrodynamic lubrication, boundary lubrication, dry friction, and seizure. Definitions are derived and fundamentals discussed in considerable detail. Effects of surface films (absorbed lubricants or chemically reacted compounds) are studied. Some experimental results are presented.

Reviewer believes this is an excellent compendium of basic information. Extensive foreign bibliography is included.

Rabinowicz, E. Metal Transfer During Static Loading and Impact. Proceedings Physical Society. (B) 65, Part 8, 392. August 1952. pp. 630-640.

Experiments are described which investigate the metal transfer taking place when a radioactive hemispherically ended slider is pressed normally into a flat surface of the same or another metal. It is found that, under very varied conditions, metal fragments are transferred from one surface to the other, and this shows in a direct way that strong junctions are formed between metal surfaces in contact. The amount of metal transferred is small. In a typical case (copper on steel, clean, with load of 4 kg), the pickup is 2 x 10^{-11}g, corresponding to a uniform layer of thickness 0.1 angstrom over the area of the identation. Experiments in which the load is varied suggest that, at higher loads, the oxide layer on the metal surfaces is broken up to a larger extent than at lower loads, and a more than proportional increase in metallic interaction and transfer takes place. An analogous effect is observed in the presence of boundary lubricants.

When the surfaces are impacted together, very similar results are obtained. Somewhat less pickup is observed than for static loading, and the difference is probably due to the fact that it takes time for strong junctions to be formed. Impact experiments with surfaces covered by lubricants show that a lubricant layer may be trapped between the surfaces, and this produces a large reduction in pickup without greatly reducing the amount of plastic deformation.

Smooth Surfaces Don't Always Give Least Wear. SAE Journal. Vol. 60. June 1952. pp. 57-60.

Research studies show that smooth surfaces wear more slowly than rough ones under ideal laboratory conditions. However, in applying the results to production machinery, it is necessary to exercise considerable caution. In many cases, fits of wearing surfaces are seldom those envisaged by designers, and best results are obtained by allowing some initial roughness. Effects of secondary influences.

II. LUBRICATION PROBLEMS

Cunningham, R. Closed-Circuit Lubrication System Applied to a Turbo-Jet Aircraft Engine-Final Report. Pennsylvania State College. (ATI No. 189102). September 1952.

The report describes in detail the procedure followed in modifying a current production turbo-jet engine to the closed-circuit system. The resultant oil system was then subjected to simulated flight conditions in the laboratory, involving operation at altitudes to 70,000 ft., climb and dive, inverted flight starting procedures at ambient temperatures down to -65°F. The development and testing programs in the laboratory were followed with installation on full-scale engines for ground and flight testing at Wright Air Development Center.

Hutt, E. T. Lubrication and the Load-Carrying Capacity of Gears. Lubrication Engineering. Vol. 8. August 1952. pp. 180-182, 201-202.

The functions of oil supply in relation to recognized types of gear failure. The oil is shown to be much more than a mere lubricant in the exact sense of the word. Ways in which operating conditions and choice of lubricating oil can influence scoring and pitting of gears in service. Some research findings concerning influence of oil properties and other factors.

Tourret, R. and White, N. Aeration and Foaming in Lubricating Oil Systems. Aeronautical Engineering Review. 24, 279. 137. May 1952. pp. 122-130.

Paper gives a broad picture of the circumstances governing the formation of foams in lubricating systems, with particular reference to aircraft engines. The basic physics of the foam-formation process is discussed first, and the effects of solubility, surface tension, and buoyancy are considered briefly. The various stages in the formation and decay of an oil form are described qualitatively, and the effects of viscosity and temperature on the rate of formation are discussed. The second part of paper relates to the influence on foam formation of design factors in the lubricating system itself. The effect of tank size and of pipe design is considered, and various means of improving the de-aeration characteristic of a given system are suggested. Results of tests on the effect of various detergent additives in increasing the initial foaming volume are also quoted, together with some discussion of the effectiveness and limitations of chemical anti-foaming agents.

Paper is, in the main, a survey of the literature of this somewhat neglected subject, and 38 references to original contributions are given.

Unconfirmed Minutes of the Meeting of the Group and Panel Leaders of the Aircraft Reciprocating Engine Lubricant Performance Group. Coordinating Research Council, Inc. ASTIA (ATI 159044). April 5, 1951.

The general scope of the project was to develop research technique for the evaluation of lubricants for reciprocating aircraft engines which will permit a better understanding of the requirements and performance of petroleum products and engine equipment. Laboratory test techniques were to be developed which would give results comparable to those secured in flight tests for six special compounded oils as compared with straight mineral oils.

Separate projects were set up as, full-scale multi-cylinder engine tests, single-cylinder engine oil performance tests, single-cylinder engine rocker box coking tests, single-cylinder engine preignition tests, single-cylinder engine spark plug fouling tests, and bench tests.

No results or conclusions were presented as the individual groups were just being organized.

II.-A. GEARS AND SPLINES

Albert, E. V. Fretting Corrosion--How to Eliminate It. Steel. April 5, 1948. p. 72.

Fretting corrosion is mechanical rather than chemical in nature and is a function of load, motion, and oxygen. It was found that fretting corrosion increased with highly finished surfaces and was more severe with steel surfaces of the same hardness than when surfaces of different hardness are used. Anti-fretting corrosion lubricants does not appear too likely in view of the influence of load and motion upon the phenomena involved. To reduce the tendency of fretting corrosion, it was suggested that the following be done: Elimination of oxygen, adequate coating to reduce friction; treating or plating surface to increase friction and reduce slipping, installation of a gasket to absorb motion and prevent metal transfer, increase hardness of one or both surfaces, induce residual compression stresses, increase motion in order to maintain a protective film, and reduce viscosity of lubricant and increase its tenacity. The hardness of the surfaces must not be the same.

Kelley, B. W. A New Look at the Scoring Phenomena of Gears. SAE 804. September 1952.

This paper points out the need of more accurate formulas to cope with the lesser ability of light oils to provide scoring protection. The most important reason why the scoring phenomena have not been adequately expressed in a formula is the lack of specific knowledge of lubrication. The author goes into detail in applying surface temperature formula to gears. This critical temperature would limit the use of mineral oils on gears. In conclusion, purely empirical formulas, although necessary as a temporary design tool, are rarely expandable and cannot be trusted for advanced designs. The approach which attempts to deal with more fundamental concepts is more desirable because of its probable expandability.

II.-B. TESTING METHODS

Casey, R. J. Report on Lubricating Oil Requirements of J-34 Turbo-Jet Engines Under Low Temperature Conditions - Project TED No. NAM-PP509. Naval Air Material Center. Aeronautical Engine Laboratory. Philadelphia. ASTIA (ATI 72093). January 1, 1950.

Tests were conducted to determine the lubricating oil requirements of the J-34 turbo-jet engine under low temperature, high-altitude starting conditions. All lubrication data were obtained from runs on an X24G-4B engine incorporating AN X 24C-4D lubrication system. It was found that performance of the oil pressure and scavenge pumps is sufficient to maintain oil flows over an altitude range of sea level to 37,000 feet with oils of 20,000 centistokes viscosity. The bearing temperatures remain within limit under these conditions. The oil jet holes at No. 2 and No. 3 bearing are susceptible to clogging by foreign matter. This causes an increase in bearing temperature owing to the reduced oil flow and thereby may lead to bearing failure.

III. HYDRODYNAMIC LUBRICATION

Cope, W. F. The Hydrodynamical Theory of Film Lubrication. Proceedings of the Royal Society. Series A. Vol. 197. June 1949. pp. 201-217.

This paper is a re-examination of the theory of hydrodynamic lubrication between two surfaces in steady relative motion. Consideration of the relative magnitudes of the various terms in the equations for typical cases allows considerable simplification. Thus the inertia terms in the momentum equation, and the dilation and conductivity terms in the energy equations are neglected. It is noted that the exact solution of the equations requires an equation of state which is simple only for a gas. It is shown that, to obtain a suitable bearing-pressure distribution with a gas as lubricant, the geometric slope will have to be considerable to over-come the effect of density variation, while with a liquid the density variation with pressure is small compared to its decrease with temperature, and a parallel-face bearing is possible. Thus a gas requires a "geometric wedge" bearing in which the distance between the two surfaces decreases in the direction of motion, while a liquid will work in a parallel-face bearing due to a "thermal wedge" in which the density decreases in the direction of motion.

With the aid of simplifying assumptions (side leakage neglected, analytically simple equations of state and viscosity laws) several cases have been solved, and their results are summarized. The load-carrying capacities of the two wedges are shown to be about the same when the viscosity varies greatly with temperature, but the thermal wedge is shown to be superior if the viscosity variation is small. The thermal wedge is mechanically simpler, but the required clearances are small so that efficient fluid filtration is required.

The author concludes by noting that successful film lubrication requires a wedging action of some kind, either geometric or thermal. The first type is important in many applications, but the second may altogether outclass it if the conditions of small viscosity variation with temperature, large coefficient of cubical expansion, and small clearance are obtained.

Parker, R. C., Farnworth, W., and Milne R. The Variation of the Coefficient of Static Friction with the Rate of Application of the Tangential Force. Proceedings of the Institution of Mechanical Engineers, Applied Mechanics. (W.E.P. No. 59). 1950. pp. 179-184.

This investigation was undertaken because of its possible relation to the occasional failure of safety clutches to slip at high over-loads. Coefficient of static friction was measured for a number of different materials under impact conditions. An analysis of the conditions of impact, derived from collision between solid bodies and from a suitable hydraulic system, showed the latter to be the more convenient and equally effective method of applying tangential force. Method adopted consisted of feeding oil at a constant rate into an oil cylinder, the piston of which constituted the striking head. The friction members were so arranged that the struck member was in the form of a bar that was clamped between two massive plates. The normal force was applied hydraulically and the tangential force was measured by wire-resistance strain gages.

Static friction was measured for rates of application of the axial force up to 10^3 tons per sec. Under dry conditions, friction increased with rate of applied force, an effect more pronounced between a nonmetal and a metal than between metals. Lubrication with oil appeared to diminish the variation of friction with rate of application of force for all combinations of materials tested. Magnitude of this effect was sufficient to contribute significantly to the type of failure mentioned above.

Thomsen, T. C. The Practice of Lubrication. McGraw-Hill. New York. 1951.

A general descriptive text on the applications of lubricants to diesel engines, steam engines, and other industrial engines. The Lohmeyer and Thurston oil tester were briefly mentioned. On oil testing, the author says, "The only reliable way of testing lubricants is to test them under actual working condition by applying them to the machinery they are to be used in and watch the results". This is a good text for the operating engineer.

Wannier, G. H. A Contribution to the Hydrodynamics of Lubrication. Quarterly Applied Mathematics. April 1950. pp. 1-32.

Paper is composed of three sections. Section 1 deals with a new derivation of the well-known Reynolds equation. This arises from the Stokes equations in first approximation if all quantities entering latter are expanded in powers of the film thickness. Simple formulas are expanded, giving such unknowns as the transverse pressure gradient from the Reynolds solution. The standard assumption that latter is zero is superfluous. In Section 2, pressure distribution, resultant load, torque, and coefficient of friction are calculated for the closed infinite cylindrical bearing (no side leakage). Starting from Stokes equations, author applies the known electrostatic solution of the Laplace equation to solving the problem of flow between two eccentric circles. A comparison with Reynolds limit of very thin oil film shows that the differences are not very marked. Section 3 deals with the closed spherical bearing with side leakage. Solution of the exact mathematical problem being too difficult, pressure, resultant load, torque, and coefficient of friction are calculated in the Reynolds limit; this also has not been done up to this time. Analysis shows that side leakage does not play a very essential role in the theory of lubrication. All the main features are the same in the two cases: pressure maximum near the point of least clearance, region of negative pressure of the exit side, antisymmetry of the pattern. A factor of the order 1/2 indicates the effect of side leakage on the pressure.

Lubrication of Industrial and Marine Machinery. J. Wiley and Sons, Inc. New York. Chapman and Hall, Ltd. London. 1945.

The book gives an elementary description of all phases of lubrication without really giving working knowledge of the subject. It is good for definitions and to obtain a general grasp of the subject.

III.-A. GEAR TEETH

Cameron, A. Hydrodynamic Theory in Gear Lubrication. Journal of the Institute of Petroleum. 38, 344. August 1952. pp. 614-622.

Paper attempts to explain the operation of gears on the basis of hydrodynamic forces involved. Analysis and experiment are based on simulation of gear operating conditions by sliding and rolling of edge of a disk on the face of another with axes crossed. Test equipment permits measurement of speed, friction, and normal load. Experimental results indicate coefficient of friction is sensibly independent of speed until scuffing occurs at some critical speed.

Theoretical analysis first considers rigid disks and constant oil viscosity. Discrepancies between theory and test are considered, and effects of pressure viscosity, film temperature variation, and elastic surface deformations are noted. These considerations provide improved qualitative agreement with experiment.

Gatcombe, E. K. The Nonsteady-State Load-Supporting Capacity of Fluid Wedge-Shaped Films. Transactions of the American Society of Mechanical Engineers. 73, 8. November 1951. pp. 1065-1075.

Thickness and load-supporting capacity are considered of fluid wedge-shaped films between the peripheral surfaces of circular rotating disks on parallel shafts, which have their center-to-center distance continuously changing, thus causing the film thickness to alternate at moderately high frequencies.

Exact analytical solution of steady-state problems (i.e., disks revolving at constant angular velocities, and lower disk and assembly not vibrated) is accomplished by transformation of viscous flow equation from Cartesian to bipolar coordinates. Earlier approximate solution gives results close to the exact answers. These are, for disks 3.0000 in. in diameter, angular velocity (-183) rad per second, minimum thickness of wedge 0.00002 in., and oil viscosity 5.5×10^{-6} lb. second per sq. in.: Maximum pressure, 26,700 psi; load-carrying capacity, 225 lb. per in. of face width; width of band of contact, 0.034 in.

The nonsteady-state solution (i.e., when the disks are simultaneously rotated and the lower one, with its assembly, is rotated) is obtained experimentally. Apparatus and its calibration are described. D-c-excited coil is attached to lower shaft and pick-up coil to upper shaft (though neither coil rotating). Signal produced in pick-up coil by vibration is fed to galvanometer-type oscillograph fitted with automatic camera. Results show that wedge can carry 2.22 times as much load as steady-state solution indicates. Load-supporting action of film can be six times as large as that resulting from rotational effects of disks. Author's conclusion is that oil-film wedge in engineering bearings and gears supports the loads known from practical experience because they operate under nonsteady-state conditions.

Lane, T. B. and Hughes, B. A Study of the Oil-Film Formation in Gears by Electrical Resistance Measurements. British Journal of Applied Physics. October 1952. pp. 315-318.

In pursuit of information concerning the nature of lubrication in gears, measurements of the electrical resistance between the meshing teeth of rotating gears have been made, using a cathode-ray oscillograph. Photographs of the trace on the screen of the cathode-ray oscillograph are shown indicating values of resistance ranging from infinity to zero. The variation of resistance depends on the load, though the resistance is always lowest where the sliding speed is highest and has its maximum value in the neighborhood of the pitch-line where there is no sliding motion. Absolute values of film thickness could not be determined.

M'Ewen, E. The Effect of Variation of Viscosity with Pressure on the Load-Carrying Capacity of the Oil Film Between Gear-Teeth. Journal of the Institute of Petroleum. Vol. 38. No. 344. August 1952. pp. 646-650.

Author thinks that hydrodynamic lubrication of gear teeth is possible, and he attempts to calculate the minimum film thickness. He uses Reynolds differential equation for the pressure in an oil film and assumes that there is no side leakage. Next, assumption is made with an expression for viscosity-pressure dependence containing two arbitrary constants. Author concludes that "for surfaces of sufficiently high finish" the maximum of the hydrodynamic zone increases as the two-thirds power of relative radius of curvature.

III.-B. BEARINGS

Cameron, A. Hydrodynamic Lubrication of Rotating Disks in Pure Sliding. Journal of the Institute of Petroleum. 37, 332, August 1951. pp. 471-486.

Current lubrication theory predicts that a fluid film cannot exist between the peripheries of two disks which are loaded against each other and which have surface speeds of equal magnitude but opposite direction, resulting in pure sliding. Author postulates that solution of the Reynolds equation with variable viscosity term might result in a final load equation which would predict same load capacity for the pure sliding case. Differential equations including effects of viscosity variation along and across the film, surface temperatures due to frictional heating, elastic deformation of the surfaces, and adiabatic heating of the oil are set up in the appendix. A simplified analysis is made, based on two large assumptions, and predicts the existence of a fluid film. Simple experiments are conducted with rotating disks, and a fair correlation with analytical results is obtained.

Hersy, M. D. Theory of Lubrication. J. Wiley and Sons, Inc. New York. 1936.

Main portion of the book is concerned with hydrodynamic lubrication. A historical introduction is given and then the classical hydrodynamic theory is developed in detail, temperature rise in bearings and oiliness occupy the latter part of the book.

Michell, A. Lubrication: Its Principles and Practices. Blackie. London. 1950.

Early part of the book deals with the physical properties of lubricants and the numerical constants of lubricating fluids. Slider bearings, journal bearings and rolling bearings are given in detail. A short chapter on boundary lubrication was given, but the book mainly dealt with fluid lubrication and the distribution and treatment of lubricants in service.

Needs, S. J. Boundary Film Investigations. Transactions of American Society of Mechanical Engineers. Vol. 62. 1940.

The paper reports some of the results of a four-year study of thin films of vegetable and mineral oils conducted in Dr. Kingsbury's laboratory under his direction. Aside from the problems of filtration, selection of the proper metal for the test surfaces and the preparation of the surfaces, the experimental work divided itself into two distinct parts. Part I describes attempts to discover indications of increasing viscosity in thin films of oil formed by two plane disks approaching each other under loads of the order of 0.20 psi with no tangential motion. Part II describes experiments with plane surfaces approaching under loads up to 800 psi with continuous relative rotation of the surfaces. It also describes a method of measuring the minimum film thickness reached under these conditions. Experiments show that steel surfaces have a profound influence on the behavior of lubrication films up to thickness greater than 60 millionths of an inch. These films are capable of supporting loads at least as great as 800 psi. Under such loads, the film thickness reaches a constant minimum which remains unchanged indefinitely even with continuous relative rotation of the surfaces.

IV. LUBRICANTS

Agster, R. T. Performance of RPM Aviation Oil 860 in Pratt and Whitney R-2000 Engines Pan American World Airways Pacific Division. Standard Oil Co. of California, Product Acceptance Dept. San Francisco, California. ASTIA (ATI 48184). January 1948.

To demonstrate the service performance of RPM aviation oil 860, the oil was installed in the Pratt and Whitney 4-2000 engines of a fleet of Douglas DC-4's. The inspections of the first four engines, which had completed a full overhaul period, showed RPM 860 performs satisfactorily in P & W 2000 engines, both from the standpoints of low wear and light engine deposits. During the period of operation with RPM 860, oil screen plugging and collapse was reduced to a point where it was no longer a problem. It is believed that this should be credited to the oil, though minor changes in operating conditions and changes in screen cleaning frequency also made important contributions.

Bouman, C. A. Properties of Lubricating Oil and Engine Deposits. MacMillian. London. 1950.

Discusses the classification, manufacture, physical, and chemical properties of lubricating oil before starting in on the lubrication phenomena occurring within the engine. Engine deposits are analyzed in great detail. One chapter deals with the maintenance of the lubricant in the engine. Practical testing of lubricants is confined to physical properties of the oil.

Eckardt, H. and Krienke, C. F. Friction Coefficients of Lubricants. Air Material Command, Wright-Patterson Air Force Base, Dayton, Ohio. ASTIA (ATI 27066). February 1947.

A device used for the measurement of the friction coefficient of lubricants is described. The oil to be tested is spread on a level plate and a steel pin slides over the thin layer of oil. The plate is rotated and the force the oil exerts on the pin serves as a criterion for the measurement of the friction coefficients. Results indicate that the friction coefficient of a great number of oils of equal viscosity but of different composition differs only slightly. Additions of graphite tricresyl phoshate and calciumchlorostearate did not improve the friction coefficient. Fatty oils showed lower values.

Glass, E. M. Lubrication Engineering: The U. S. Air Force's Approach. Society of Automotive Engineers. November 1948.

Portion of the paper attempted to describe the general characteristics, limitations and proposed developments with regard to currently used aircraft lubricants.

The need for a lubrication program was outlined to standardize lubricants so that the least number need be procured, supplied and used.

The pertinent points of the program of lubrication engineering, as is covered by specification AN-L-32, were discussed.

Benefits of the lubrication program, after three years of effectiveness, has been the reduction of the number of lubricants in stock from the 400 special lubricants to approximately 20.

The most important benefit of the program as practiced by the USAF is the welding together a partnership between the design engineer and the lubrication engineer with the result that aircraft equipment in addition to being mechanically engineered is lubrication engineered in the mutual creation of an air force second to none.

Godfrey, D. and Nelson, E. C. Oxidation Characteristics of MoS_2 and Effect of Such Oxidation on its Role as a Solid-Film Lubricant. National Academy of Sciences. TN 1882. May 1949.

The oxidation characteristics of molybdenum disulfide MoS_2 and the effect of such oxidation on its role as a solid-film lubricant were determined. An analysis was made by x-ray diffraction and special equipment was used in the coefficient-of-friction determinations. A coating of MoS_2 serving as a high-temperature solid-film lubricant maintained low coefficient of friction values during its oxidation as long as an effective subfilm of MoS_2 remained. The oxidation product MoO_3 alone produced very high friction. The MoS_2 began to oxidize at a very low rate when heated in air at 750°F. the rate steadily increased and became high at 1050°F. The MoS_2 maintained its original hexagonal structure when heated to 1000°F. in vacuum.

Weber, W. Viscosity Measurements of Fluids Under High Pressures. Part 2. Angew. Chem. B. Vol. 20. April 1948. pp. 89-96.

This paper describes two series of experiments, conducted on 20 different oils, in order to investigate the relation between viscosity and pressure at various temperatures. The viscosity measurement was made using a small sphere falling along an inclined tube. A first series covered pressures up to 1000 atm and temperatures up to 60°F.; a second series on which fuller details are given, covered pressures up to 200 atm only, but temperatures up to 120°F.

The analysis of the results is very careful. For each oil the pressure coefficient of the logarithmic relation between viscosity and pressure was determined; this relation has been confirmed. The dependence of the pressure coefficient on the temperature has been studied but none of the pertinent empirical relations proposed by previous authors has been satisfactorily confirmed. Some efforts are made to connect the physical characteristics involved with the chemical nature of the oils.

Woods, W. W. and Robinson, J. V. Comparative Foaming Characteristics of Aeronautical Lubricating Oils. NACA TN 2031. February 1950.

Comparative data are presented on the volume of foam and the stability of foams of aeronautical lubricating oils (new and used) produced at 100°C. (212°F.) by the air-bubbling method.

An empirical expression has been developed by which the rate of foam rise may be predicted from a known rate of aeration, by making use of two parameters characteristic of the oil.

Physical Chemistry of Lubricating Oils. New York Book Division. Reinhold Publishing Corp. 1951.

A very complete and advanced treatment of the physical and chemical properties of lubricating oils including synthetics. There was no mention of oil testing or test machines. This would be a good reference for specific questions concerning the property of a lubricant.

IV.-A. GREASES

Compondu, D. H. Application of Bentone Lubricants. Iron and Steel Engineering. September 1952. p. 165.

Bentone lubricants cannot truly be called greases in that the oils are not thickened by a soap, but are gelled by a chemical compound obtained by reacting a purified bentonite and an organic base. The qualities of these lubricants are summarized as follows: they have no melting or dropping point; contain no metallic soaps, or fatty acids; are water resistant and are insoluble in the face of extreme water conditions; have greater adhesive properties than any other type lubricant because of their greater affinity for metals; shear structure is lower than other lubricants; and never lose their extraordinary adhesive and lubricity characteristics. Their practical limit of lubrication seems to be 500°F. although 1500°F. has been successfully reached.

Burger, R. J., Rubin, B., and Glass, E. M. Study of Rust Preventive Properties of Greases. Lubrication Engineer. Vol. 8. No. 1. February 1952. pp. 2-3, 26-27.

Investigation of wheel bearing greases conducted in laboratory and in service aircraft on petroleum, Diester, Ucon oil, and silicon greases. Specification AN-G-15 greases showed lack of adequate corrosion protection while AN-G-25 greases exhibited adequate corrosion protection properties under conditions of high humidity and temperatures.

Glass, E. M. and Salzberg, L. F. Grease Lubrication of Military Aircraft and Components. Material Laboratory Engineering Division, Air Material Command. ASTIA, ATI, Technical Data Digest. December 15, 1947. p. 7.

Knowledge of the ingredients or compository of a grease gave no indication of the performance of such a grease in equipment under actual operating conditions. In addition to performance tests, it has also become necessary to establish a more fundamental physical-chemical approach, in order to have a more complete understanding of greases.

The most important job for the future in connection with grease development is the necessity of infinitely greater correlation between service-performance and laboratory-performance tests together with refinement of the laboratory methods for greater reproducibility. As far as the immediate future is concerned, grease lubrication of aircraft will be limited to approximately four greases, AN-G-5A, AN-G-10 (improved). AN-G-15 and AN-G-25. It is felt that these specifications overlap sufficiently within their performance ranges to take care of all present and possible future requirements.

Klemgard, E. N. Lubricating Greases: Their Manufacture and Use. Reinhold Publishing Corp. N.Y. 1937.

A book of practical and scientific data on the manufacture and use of lubricating greases, including the chemical analysis of many commercial lubricants abstract of important patents, formulas, and practical processes for the manufacture of greases. The operation and construction of the timken oil tester is briefly given.

Pixley, F. E. Gear Lubrication Test Gun Door Actuator. Report No. NA-47-230. ASTIA (ATI No. 18408). March 17, 1947.

The purpose of the test was to determine the operating life of gear door actuator mechanism worms and gears under a specific load condition with various lubricants. The five lubricants tested were AN-G-25, AN-G-15, AAF-3607, AN-G-5, AN-G-5 plus 5% (wt.) AN-G-24 and one set run without lubrication.

Conditions of test were: operate worm at 120 rpm (worm gear at 2 rpm); 50 in. lb. torque load to worm gear; air temperatures $225 \pm 22^{o}F.$; operate until failure due to lack of lubrication.

Results were as follows: AN-G-25 at 13,500 revolutions, gear showed minimum of wear; AN-G-15 at 13,500 rev. failure was impending; AAF-3607 at 13,500 rev. failure was impending; AN-G-5 at 13,500 rev. gear failed; AN-G-5 plus 5% (wt.) AN-G-24 at 13,500 rev. only slight wear; no lubrication, gear failed at 2100 rev.

There was no noticeable wear of any of the worms.

Swakon, Edward A. and Brannen, Cecil G. Development of High-Temperature Grease. Standard Oil Co. ASTIA (ATI No. 170965). November 1951.

Eighteen experimental and commercial greases have been tested to date in the high-speed, high-temperature bearing tester (ABEC-NLGI). Of the greases evaluated at 450°F., copper phthalocyamine grease has given the best performance (about 250 hours) followed by Dow Corning 44A (175 hours).

Emphasis has recently been placed on determining which of the physical properties of grease are the principle causes for bearing failure in the high-speed high-temperature bearing tester. Leakage and evaporation tests run on a number of silicone greases showed a wide variation in results but sufficient data have not been obtained, as yet, to allow conclusions to be drawn. Further work on the four-ball machine will be discontinued because a complete lack of correlation was shown between performance in the bearing tester and wear in the four-ball machine.

Continued work with new compositions for use as thickners was carried out for the dual purpose of assessing their value as thickeners and for obtaining data on the physical properties of greases made from these materials and silicone oil. Greases made from silicone oil and thickeners containing lanthanum oxide, which had been precipitated in the presence of (a) colloidal silica and (b) carbon black, showed the undesirable properties of the silica greases in one case and of the carbon black greases in the other.

Symposium on Functional Tests for Ball Bearing Greases. Presented at 51st Annual Meeting A.S.T.M. Detroit, Michigan. June 23, 1948, Philadelphia, 1949.

The purpose of the symposium was to guide future developement of functional testing of ball bearing greases. The following articles were presented at the symposium:

1. "Developement of Functional Grease Test Methods for the Aircraft Industry", by D. H. Moreton.

2. "Grease-An Oil Storehouse for Bearings", by D. F. Wilcock and Marshall Anderson.

3. "Laboratory Performance Tests for Antifriction Bearing Greases", by M. Herbst.

4. "Service Experience with Grease", by W. T. Everitt.

5. "Factors Affecting Simulated Service Tests of Greases", by S. M. Collegeman.

IV.-B.-2. <u>OILS</u>

Mangolin, E. L. <u>Properties of Specification MIL-L-7808 Oil (Synthetic Oil)</u>. Tech. Note: WCNE-52-532-487. ASTIA (ATI No. 164226). August 11, 1952.

The purpose of the paper was to present a compilation of data on the properties of specification MIL-L-7808 oil. This specification is a performance specification for oil suitable for turbo-prop engines and such new turbo-jets as the J-57, J-65, and J-71. It is not believed that petroleum oils can qualify due to the requirements. For comparison typical grade 1010 jet engine oil was included in data given.

Esso Turbo Oil 15, the only presently qualified MIL-L-7808 oil, swells rubber five times as much as grade 1010 oil.

MIL-L-7808 oil is similar to grade 1010 oil at room temperature but has better viscosity characteristics at extremely low and high temperatures.

Synthetic oils range from 10% to 20% heavier than comparable petroleum jet oils.

MIL-L-7808 oil is less volatile, has a lower value of specific heat, greater thermal conductivity, greater load carrying ability, and superior foaming characteristics than petroleum jet oils.

Mosteller, J. C. <u>Fluids, Lubricants, Fuels, and Related Materials</u>. Petroleum Refining Laboratory, Pennsylvania State College, School of Chemistry and Physics. ASTIA (ATI No. 158975). December 1951.

First quarterly report summarizing work being done during the period of October to December 1951. Synthetic studies with PRL 3161 gear lubricant are continuing using full scale tests on turbo-prop and turbo-jet engines. Lubricant is proving successful so far. Physical properties are being evaluated. Improved shear stability over mineral base oils was determined. A specification procedure and limits for PRL thin film oxidation and corrosion test for application to synthetic gear lubes was prepared. Study of various base stocks for synthetic lubricants is essentially complete. Composition and properties, wear, and lubrication characteristics, as well as oxidation and corrosion have been determined. Oxidation and deterioration of an AN-09 oil, that was in storage for approximately three years, was investigated and found that it did not meet oxidation stability requirements but did meet corrosion test specifications. An attempt to determine the reason for reduced oxidation stability will be made.

Oxidation and corrosion stability tests at 500°F. with phenothialine inhibited di-2-ethylhexyl sebcate (PRL 3207) and PRL 3161 are being made with the complete results not yet compiled.

A program to study the inhibitor concentration change during an oxidation and corrosion test is in progress with partial result obtained.

Wear characteristics of Di-sec-amyl sebacte with tricresyl phosphate anti-wear additive was made on a four-ball tester. The wear remains high at the 1 kilogram load even though wear reduction over that of the ester without an anti-wear additive is noted for the 10 kilogram load.

Murray, S. F., Johnson, R. L., and Bisson, E. E. <u>Effect of High Bulk Temperatures on Boundary Lubrication of Steel Surfaces by Synthetic Fluids</u>. NACA TN 2940. May 1953.

An experimental study was made of the effect of high lubricant bulk temperatures on the boundary lubricating effectiveness with steel surfaces of various types of synthetic fluids considered as lubricants for turbine engines.

Synthetic fluids were generally effective lubricants at higher temperatures than comparable petroleums.

Lubrication with esters and possibly other fluids is believed to result from formation of a chemisorbed soap film by free acids (contaminants) in the fluid. Failure results from deterioration of metal soap and oxide films which is influenced by temperature effects on the bulk fluid. Thermal stability can be associated with viscosity grade within a given class; however, no correlation could be made between lubrication failure temperature and the viscosity at failure temperatures for the various fluids. At temperatures up to its decomposition point, a silicate ester showed more promise than the other lubricants studied.

Murray, S. F. and Johnson, R. L. Effects of Solvents in Improving Boundary Lubrication of Steel by Silicones. NACA TN 2788. September 1952.

Silicones best satisfy the viscometric requirements for lubricants for turbine engines because of the known synthetic fluids. A study was conducted to establish the effect of solvents on boundary lubrication by silicones. Boundary-lubrication data were obtained which are considered substantiating evidence for a hypothesis that in solutions of solvent blended with silicones, the silicones form a closely packed and oriented absorbed film on ferrous surface. The solutions reduced friction and prevented surface failure even when the solvent, as well as the silicone, was an extremely poor lubricant. These data indicate that satisfactory lubrication is the result of solvation effect rather than a lubrication additive effect of the solvent because 30 to 50 percent of solvent was necessary for good results. The best results were obtained with solvents having dipole moments. Solutions of diesters in silicones may be practical lubricants.

Rubin, B. and Glass, E. M. The Air Force Looks at Synthetic Lubricants. SAE Quarterly Transactions. Vol. 4, No. 2. April 1950.

Synthetic lubricants (materials that are wholly or largely nonpetroleum) are being looked at with favor by the Air Force, the authors indicate, because of limitations in the performance of petroleum lubricants at extremely high and extremely low temperatures.

The types of materials investigated as synthetic lubricants included: dibasic acid ester, polyalkylene glycols, organic silicon compounds, halogenated hydrocarbon, and inorganic radical-organic base ester.

Extensive studies and experiments with the above synthetic lubricants were performed to determine their advantages and shortcomings.

The ever extending of the temperature ranges provides a problem in the rise of synthetic oils. The objective is to produce an item capable of giving reliable performance over the range -75 F. to 400 F. in a wide variety of applications. If such a material can be made for both rolling and sliding friction applications, the benefits in terms of simplification of the logistics and maintenance problems would be incalculable. Such a lubricant would require not only a stable liquid component, but also new thickness agents and additives suitable for high temperatures.

Woods, W. W. and Robinson, J. V. Aeration of Aircraft Lubricating Oils over a Range of Temperature. NACA TN 1846. April 1949.

An investigation was conducted to demonstrate the existence of air emulsions in aeronautical lubricating oil at 100°C. and the amount of air in the emulsions has been measured. The three mechanical systems that were investigated to produce air emulsions were an electric kitchen mixer, the colloid mill, and the gear-pump circulating system.

The greatest measured air content of any emulsion produced was about 15% by volume with oil containing lubricating additives, where as unmodified oil formed emulsified aeration, was reduced to a minimum of 2% to 4% by use of either Dow Corning fluid type 200 or a mixture of glycerol with aerosol OT.

These two mixtures completely eliminated surface froth.

Zuedema, H. The Performance of Lubricating Oils. Reinhold. New York. 1952.

A well written book that discusses the performance characteristics of lubricating oils; such as, rheology, oxidation, bearing corrosion, sludge, and lacquer deposition, emulsification, foaming and wear. A short section is given on manufacturing methods. Several of the well known testers (Almen, Timken, SAE, Four Ball) were very briefly described on a general level.

Effective Lubrication Range for Steel Surfaces Boundary Lubricated at High Sliding Velocities by Various Classes of Synthetic Fluids. NACA TN 2846. December 1952.

It was the endeavor of this program to find new types of lubricants to replace current petroleum oils for the turbine type aircraft engines of the immediate future. The present lubricants (specification MIL-0-6081A, grades 1010 and 1005) are not completely satisfactory because they are either too viscous for adequate pumpability at low temperatures (-65°F.) or because they have marginal lubricating ability and excessive volatility at present bearing operating temperature (under 350°F.).

Experiments were performed on classes of diesters, polyethers, silicones, silicone diester blend, silicate ester, phosphonate esters, halocarbon and petroleum MIL-0-6081 (grade 1010) as a comparison only. A kinetic-friction apparatus, consisting of a steel ball sliding on a lubricated flat, steel disk, was used. Sliding velocities from 75 to 18,000 feet per minute and a load of 269 grams with very thin films of synthetic fluids were used in gathering data.

Sliding-friction data and surface-failure properties show that a number of the synthetics are more effective boundary lubricants at high sliding velocities than petroleum oils of comparable viscosity. A blend of silicones and a diester, an alkyl silicate ester, and a compounded diester (containing lubrication additives) were more effective boundary lubricants at high sliding velocities than were comparable diesters from which most widely accepted synthetic lubricants are made. A diester failed to lubricate nonreactive surfaces which indicates that the lubrication mechanism for diesters may involve chemical reaction with the lubricated surfaces.

IV.-C. MILITARY SPECIFICATIONS

Lubrication. U. S. Dept. of the Army. Government Printing Office. (Technical Manual, 9-2835) 1949.

This manual explains the fundamental principles of lubrication, lists the lubricants, lubrication publications, and lubricating equipment available to personnel and analyzes the basic types of surfaces on bearings which must be lubricated. Lubrication of types of material also is discussed. No attempt has been made to give definite lubrication procedure for specific items. Main purpose of the manual is to give the reader a general understanding of the subject of lubrication as applied to military material.

V. TESTERS - GENERAL

Backoff, W. J. Performance Testing of Gear Lubricants. SAE 398. November 1949.

Describes a performance testing program to determine whether 2-105B lubricants would or would not give satisfactory performance when subjected to various testing procedures. Findings were that the 2-105B qualification tests do not cover the full range of requirements for all types of service and from the several test procedures investigated, the Chrysler Mt. Truck, the General Motors shock and the Pure Oil Co.. High speed tests should be added to the L-19 and L-20 tests in order to cover the most severe types of service.

In summarizing the gear test work, it is believed that there are four possible courses for the marketer of gear lubricants which allows each to choose how many applications one lubricant can economically cover and that road testing evaluation of gear lubricants will clearly define the class into which each lubricant falls.

Clower, James Ira. Lubricants and Lubrication. 1st Edition., New York and London, McGraw-Hill Book Co., Inc. 1939.

Standard treatment of properties of lubricants and lubrication practices. The text is essentially practical with advanced theories left to other books that treat them in detail. Essential purpose of the book is to give a working knowledge of the subject. The Almen, Timken, SAE, and the Navy Work-Factor Machine for Testing Lubricants are described very briefly.

Downs, D. and Pigneguy, J. H. Lubricating Oil Tests. Automotive Engineering. Vol. 39. February 1949. pp. 51-58.

A series of accelerated engine tests has been evolved for rating oils, especially lighter duty oils containing anti-oxidants, detergents, and dispersants. The tests are made on water-cooled gasoline engines. Piston cleanliness tests last 50 hr, the temperature of the piston being 200 C and that of the oil 95°C; ratings are given depending on the extent of carbon deposit on the skirt. Cold-sludging tests last 200 hr, the temperature of the jacket and the oil being lower than in the previous test. Big-end bearing corrosion tests last 90 hr; copper-lead bearings are used and the oil temperature is maintained at 100°C. Cylinder and ring wear is tested in a special crosshead engine providing separate lubricating oil for the cylinder and the crankcase assembly. Ring sticking tests are carried out for standard periods of 5 hr, the cylinder temperature increments between successive tests being 5°C. until a quarter of the top ring is right in the groove.

Diagrams and photographs of pistons show that oil performance can be essentially improved by some additives.

Klaus, E. E. Wear and Lubricating Characteristics of Some Mineral Oil and Synthetic Lubricants. Petroleum Refining Laboratory Pennsylvania State College, Report No. P.R.L. 4-52, June 1952.

The purpose of the study was to evaluate the relative lubricity of various types or classes of organic compounds as lubricants and lubrication additives.

A total of sixty-four laboratory wear and lubrication testers were reviewed. These testers were described briefly with no test results given.

Mac Coull, N., Ryder, E. A., and Scholp, A. C. An Oil Corrosion Tester. Presented to SAE. January 1942.

A bench corrosion tester was built by the Texas Company & Pratt and Whitney Aircraft Co. Results correlated fairly well with practice, but it was emphasized that the tester does not eliminate the need for a full scale test. The tester mainly eliminates the very poor oils. Operation, construction, etc. of the tester is given in detail. It is a sleeve bearing running in an oil bath that is thermally controlled. This tester does not reproduce the temperature flash as in boundary lubrication and its use in connection with a gear tester is questioned.

Mayer - Bugstroem. Lubricity Tests on Oils with the Oil Testing Machines of Almen, Wieland, and Thoma. ASTIA (ATI No. 18556). March 1938.

A series of lubricating oils, especially aircraft engine oils of various manufacture and composition, were tested in the oil-testing machines of Almen, Wieland, and Thoma. No satisfactory correlation of the results obtained by the various machines could be determined. Only in the case of a very different composition of the oils (fatty oils compared to mineral oils) the evaluation of the lubricity was the same in all three machines. Obviously, between the mineral oils and the slightly fat-modified oils, the difference in lubricity are too insignificant to be noticed within the recording errors of the machines.

Consequently, a lubricity determination by one of these machines does not agree with the measurements of any other machine designed for the same purpose. That means, an application of these data in large-scale production will be impossible. Further tests are, therefore, scheduled to test several oils under static or dynamic load as to their behavior in an aircraft-engine bearing. These oils can then serve as reference materials.

Schnurmann, R. Mechanical Methods of Testing Lubricants. Transactions of the Instrument Measuring Conference. Stockholm. 1949. pp. 154-164.

This is a review covering some work of the author and other workers in the fields of non-Newtonian fluid flow and the friction of solids under conditions of boundary lubrication.

Temporary and permanent reductions in viscosity of fluids under high rates of shear are discussed and a technique for capillary-type measurements of viscosity is presented which permits high shearing stress with laminar flow and negligible viscous heating of the fluid.

It is emphasized that in boundary friction the "coefficient of friction" is not fixed by a choice of rubbing materials and lubricant but is a function of a number of operating conditions such as load, sliding velocity, and temperature. These factors and the relationship between wear and normal load are discussed at length, the arguments being supported by data obtained on a four-ball apparatus. It is concluded that a number of boundary lubricants can be rated by measuring the friction force as a function of normal load, or when Amonton's law is known to be valid for a given load range, by observing wear rate as a function of normal load.

Schoekel, H. Tests with the Oil Testing Machine Developed at the DVL. AFTS 937. May 1947.

In this paper, an oil testing machine developed at the DVL, and tests with this machine, are described. First, five lubricating oils of various viscosities were examined. The evaluation of the tests in the friction-viscosity graph gave a varying valuation of the oils. The tests were repeated with the airplane engine oils, three of which were modified fats. The modified fatty oils had, at the start of the loaded machine, a lower friction number than a corresponding mineral oil. A checking of the experiments indicated errors, which were due to the lack of accuracy in the measurements at the determination of the friction number. In order to eliminate the errors, the use of a modified oil-testing machine with friction balance, would be necessary. An attempt to eliminate the differences in the condition of the machine by comparison with a reference oil was without success.

Wenzel, H. Development of An Oil Testing Machine. AAF TS929. February 1947.

Describes the development of an oil testing machine for the definite determination of the point at which the oil film tears due to the influence of stress and temperature.

A vertical shaft, supported by a radial ball bearing, runs with its lower end in the test bearing. The test bearing is of strong steel pipe lined with white metal and split so that a load can be applied with a spiral spring. The main advantage is the fact that the lubricating power of the oils can be observed when various bearing metals are used. The oil was supplied to the test bearing by a drip oiler.

For the determination of the measuring method, several preliminary tests were made with total lubrication, which showed definite differences between the individual oils in their friction coefficients and in the bearing temperature. No test results were made available.

V.-A. GEAR TYPE

Hughes, J. R. and Tourret, R. Mechanical Testing of Gear Lubricants. Engineering Vol. 175, No. 4542. February 13, 1952.

The paper describes a gear tester developed at the Thorton Research Center of the Shell Petroleum Company. The gear tester is known as the Thorton high-speed gear rig. The rig operates on the "four-square" power circulating system in which a pair of test gears is coupled by two shafts to a pair of "power-return" gears of much greater face width. One of the shafts has a torque coupling for applying the load. The test-gear system is designed to provide control of both rate of flow and oil temperature. Each gear is divided into two parallel sets of teeth by a groove cut in the periphery. By using the two sets and the reverse faces of each set, four tests may be carried out with each pair of gears.

The standard test carried out is to determine the load which causes scuffing during a 15 minute test run. An oil is tested at four speeds, 3000, 6000, 9000, and 12,000 rpm, using the four test faces of a single pair of gears with at least one set of repeat tests on a second pair of gears.

The work carried out on the tester show that smaller lower-speed machines rate lubricants satisfactorily for high-speed conditions, but much is still unknown about the behavior of oils at high speeds.

It was found that, approximately, load carrying capacity is proportional to N to the minus 2/3 power, and power is proportional to N to the plus 1/3 power. The paper presents a mathematical analysis of these equations and shows that the experimental results compare to some degree.

V.-A.-1. NAVY GEAR

Tingle, E. D. and Calderwood, G. F. N. A Critical Examination of a Proposed Specification Gear Wear Test. Royal Aircraft Establishment. Technical Note No. Chem. 1149.

Experiments with the tester indicated a need for greater stability in the gear rig, and gears of high quality subjected to strict inspection. Most wear occurred during the running in phase, indicating this may not be a valid phase to test in. The tester conformed to specification AN-G-25 and its essential parts were a pair of small, fine pitch spiral gears, meshing at a shaft angle of 90°. One gear being leaded brass, the other stainless steel, loading was unidirectional but rotation was reversed every revolution. Correlation of test results with service experience was considered unlikely with the tester in its present form.

V.-A.-2. RYDER

Ryder, E. A. A Gear and Lubricant Tester-Measures Tooth Strength on Surface Effects. A.S.T.M. Bulletin No. 148. October 1947.

A simple machine has been constructed for the bench testing of gear materials and gear lubricants at Pratt & Whitney Aircraft. Intended for aircraft engine research, the machine uses as test specimens a pair of gears which are like those used in an actual engine but as simple as possible to manufacture. The various factors affecting the load capacity of gearing are studied separately. The load on the test can be changed without stopping the machine. Lubrication of the test gears is controlled from oiling of the remainder of the rig. Some remarks on the philosophy of bench testing are included. The machine described here was developed in 1941 for the investigation of aircraft engine gears. However, it is suitable for the study of other spur gearing by changing the test gears, speed, etc. Before describing the test rig, the reasons for doing any kind of bench testing are discussed; such as, the effect of speed on fatigue strength, and on scuffing resistance.

Ryder, E. A. A Test for Aircraft Gear Lubricants. A.S.T.M. Bulletin 184. September 1952.

Pratt & Whitney aircraft bench machine for testing gear materials and lubricants has been put to use by several laboratories and a great deal of experience is now available. Machine offers satisfactory method of assessing antiscuffing properties of gear lubricants, and its use is being called for in oil specifications; current procedure for this purpose is described.

V.-A.-3. GENERAL ELECTRIC

Shipley, E. E. and Dudley, D. W. Evaluation and Comparison of a Silicone Oil Using Small Case Hardened Test Gears. General Electric Apparatus Department. Technical Information Series. ASTIA (ATI No. 114250). July 1, 1951.

A General Electric gear test stand (1098566-100) was used for both oils tested. Oil temperature of both tests being 300-311°F. with a 1600 K-factor load. From the wear point of view, silicone oil was far inferior to 1005 oil, and quite unsatisfactory for lubrication of spur gears under load. Silicone oil caused threaded parts to stick and fixed splines became quite difficult to disassemble. Considerable silicone fumes were present in the test area with silicone oil up to speed and temperature of 300°F., however, the oil did not foam, indicating the vapor was due to elevated temperature.

Shipley, E. E. Evaluation and Comparison of Plexol #244 Oil Using Small Case Hardened Test Gears. General Electric Apparatus Department. Technical Information Series. ASTIA (ATI No. 144459). January 1, 1952.

Plexol #244 oil will lubricate spur gears satisfactorily under 1600 K-factor load and at a temperature of 300°F. However, the duration or life of a set of gears using this oil is limited. The performance of plexol #244 at 300°F., as compared to 1005 oil at 120-130°F., would be rated as slightly inferior. From the wear point of view, the plexol oil is about the same or slightly inferior. The clinging quality of plexol oil is about the same or slightly inferior. The clinging quality of plexol oil was better than 1005 oil, but showed a tendency to form a sticky, greasy, substance which seemed to be caused from breakdown of the oil. A General Electric gear test stand (1098566-100) was used for both oils tested.

V.-A.-4. TIMKEN

Kolarik, I. S., Zeiler, C. A., and Kipp, E. M. Effect of Variations in Viscosity of Lubricants Upon Timken OK and PSI Values. ASME. July 1952.

The Timken lubricant tester is often used in the laboratory in establishing Timken OK loads and PSI valves at OK loads for lubricants of the extreme pressure type, particularly for those used as gear lubricants. The present investigation was conducted to determine the magnitude of hydrodynamic - lubrication effects associated with viscosity in determining OK loads. The effect of rubbing speeds was also studied.

Results were that viscosity does effect the tests results and any tests which are run must be compared on a like viscosity basis. Similarly, speed is a factor in the load carrying ability of the test oils and must be considered in the application of a lubricant for a specific purpose. Also, test results are affected by the surface finish of the test members used on the machine.

V.-A.-5. S.A.E.

McKee, S. A., Bitner, F. G., and McKee, T. R. Apparatus for Determining Load-Carrying Capacity of Extreme-Pressure Lubricants. SAE Transaction 193. pp. 402-408.

Describes an attempt to develope an apparatus that would, as far as possible, simulate the action of gears and also maintain a constant ratio between applied load and actual pressure throughout the test run.

The tests with this machine, together with previous test with other machines, tend to indicate that in a given mechanism, operating under extreme-pressure lubricant, the factors involved are such that the relative load-carrying capacities for various lubricants under specified conditions of operations do not necessarily apply to all operating conditions. They indicate, also, that among the factors of significance are: speed of rubbing, lubricant temperature, and rate of loading.

The operating characteristics of the machine used are such that automatic alignment is maintained between the two rotating test rolls so that the applied load is a true criterion of the actual pressure on the rubbing surfaces. The point at which lubrication fails is quite definite and may be readily determined while the machine is in operation.

Over the operating range covered, the test results obtained with this machine appears to rate the lubricants in reasonable agreement with ratings based upon service performance in automotive gears.

White, H. S. and McKee, S. A. Progress Report on Turbo-Prop Lubricants Project for April to June, 1952. National Bureau of Standards Report No. 1709. ASTIA (ATI No. 159611). June 27, 1952.

Tests were made with 19 oils during this quarter. One mineral oil and four synthetic oils were tested on the McKee E. P. lubricants testing machine at various temperatures at 600 rpm.with a shaft-speed ratio of 47:15, and with operating periods of 18 hours at loads of 400, 800, and 1200 pounds. With the modified SAE E.P. lubricants testing machine operating with a shaft-speed ratio of 3.4:1, an oil-flow rate of 300 gm/min, an oil inlet temperature of 210°F. and with periodic load increases, three synthetic oils were tested at 600 rpm and 19 oils were tested at 765 rpm.

Tabulated results were given.

White, H. S. and McKee, S. A. Summary Report on Turbo-Prop Lubricants for Fiscal Year Ending June 30, 1952. NVS Report 1745. ASTIA (ATI No. 159470). June 30, 1952.

A summary is given of the work done during the fiscal year 1952 on turbo-prop lubricants. An investigation was made of the possibilities of the modified SAE E.P. lubricants testing machine and the McKee E.P. lubricants testing Machine for determining the antiwear and the load-carrying properties of turbo-prop lubricants. Twenty-seven lubricants were tested. Various techniques of operation were used. Operation of the two machines were studied. Data are presented also for oils tested for stability at high temperatures on the McKee oil stability apparatus.

At the present time, the data available from gear tests (in other laboratories) on these lubricants, are not sufficient for a comparative analysis. However, the methods outlined show some promise for controlling the anti-wear and the load-carrying properties of turbo-prop lubricants.

Laboratory Wear Tests with Automotive Gear Lubricants. SAE 400, November 1949.

This paper describes the use of the S.A.E. machine for the determination of the wear with gear lubricants under conditions simulating high torque and low speed.

S.A.E. machine modified so that upper cup was sprayed with oil from an external oil reservoir-circulating system was used.

Lubricants used included a Navy contract 1080 mineral oil, five lubricants conforming to U.S. Army specification 2-105B, five lubricants conforming to Federal specification VV-L-761 and one lead-soap active-sulphur lubricant. All were S.A.E. 90 grade.

Data indicated that under the conditions present, there was a run-in period of high rate of wear for a few hours, after which the wear settled down to a fairly constant rate. Also noted that at the higher loads, the rate of wear with some of the lubricants showed a marked increase, whereas with others there was little effect.

The wear with mineral oil was low up to the point where it would not carry the load, whereas with the lead-soap active-sulphur lubricant (which will carry a high shock load) the wear was relatively high at all but the lightest load.

The data are of interest in that they show significant differences in the performance of lubricants meeting the requirements of the same specification.

The evidence of a run-in wear under some conditions is of considerable importance and should be taken into account in the evaluation of lubricants with respect to wear. One advantage of this type of test is that the rate of wear after the run-in period is not materially affected by the original roughness of the test cups as is the case with the usual load-carrying capacity tests with this machine.

The lubricants containing the more chemically-active additives, for withstanding higher shock load, tend to show the greater wear. This is in agreement with the known service performance of some of these lubricants; particularly, the active-sulphur lubricant and the non-additive mineral oil.

V.-A.-6. SHELL 4-BALL

Blok, H. Seizure-Delay Method for Determining the Seizure Protection of EP Lubricants. SAE Transactions. Vol. 45. 1939.

The paper points out that it is not recognized fully that it is the local temperature at the surface of contact and not the local specific pressure that chiefly determines the occurrance of seizure under extreme pressure lubrication conditions. It appears typical for extreme-pressure-lubrication conditions, as met in gear practice, that the temperature flash is much higher than the bulk temperature.

With existing conventional test methods for the determination of the protection against seizure afforded by E.P. lubricants, a considerable rise of the bulk temperature mostly occurs as it cannot be controlled sufficiently; thus, leaving an unknown margin for the temperature flash, it renders impossible a reliable determination. Shows how, in the development of the test method for determining the protection against seizure on the four-ball apparatus, the so-called seizure-delay method. Due consideration has been given to the preceding views and to what extent its results correlate better with practice than the conventional methods, (SAE & Timken).

Howlett, J. Film Lubrication Between Spherical Surfaces: With an Application to the Theory of the Four-Ball Tester. Journal of Applied Physics. Vol. 17. 1946. pp. 137-149.

An analytical investigation into the ability of a ball to support a load under hydrodynamical conditions. Results were found to indicate that the ball could support a load of only a few hundred grams and it, therefore, must be concluded that boundary lubrication is the mode of lubrication.

Larsen, R. G. Study of Lubrication Using the Four-Ball-Type Machine. Shell Development Co. ASTIA (ATI No. 14945). May 7, 1945.

The report constitutes a review and description of the use of several machines employing four-ball-type bearing for research on lubrication processes and evaluation of oils. Description of these machines, uses for which they are intended, and experimental techniques used in studying lubrication phenomena are presented. The methods of measurement of wear, friction, and seizure are given, and lubrication testing apparatus is described.

In the discussion of the various four-ball machines, no detailed experimental techniques are given. However, great care should be taken in cleaning the balls and ball holders.

Once the conditions of operation of the four-ball apparatus, which correspond to a given application, have been established, it provides a very useful means for studying not only the lubricant, but other factors as well.

V.-B. MISCELLANEOUS BENCH-TYPE

Bassett, W. B. Performance Characteristics of Automotive Gear Lubricants. SAE. August 1948.

Describes a program for the development of laboratory axle test procedures which could be used in evaluating automotive gear lubricants. The basic requirements of hypoid lubricants are classified as: (1) Load carrying ability; (2) Stability; (3) Desirable miscellaneous properties. Points out the uncorrelated results of bench tests; such as, the Almen, Timken, S.A.E., and Falex testers when used on hypoid lubricants. The laboratory is devoting considerable effort toward the development of a laboratory test procedure which would give better correlation with service.

Ducus, E. N., Coleman, F. F., and Roess, L. C. A New Experimental Approach to the Study of Boundary Lubrication. Journal of Applied Physics. Vol. 15. December 1944.

A description is given of apparatus for measuring the relative ability of rubbed down monolayers of polar lubricants to maintain low friction under test conditions which do not permit replacement of the lubricant. This quality of a lubricant is called its "Durability". The clean polished rim of a slowly rotating steel wheel rubs on the monolayer deposited on a polished steel flat. An account is given of the preparation of the surfaces and films. It was not found possible to prepare reproducible test surface-monolayer combinations; hence, it was necessary to compare each lubricant with a standard deposited on a separate area of the same surface. Significant differences are found in the relative durabilities of a number of polar compounds. Values are found to increase with the number of carbon atoms per molecule. Reduction of durability and molecular orientation results when the film is flushed with a fine stream of benzene. On chromium, also, the orientation is destroyed by the solvent, but the durability is not changed.

Hughes, J. R. and Tourret, R. Mechanical Testing of Gear Lubricants. Engineering. February 27, 1953.

The Thornton cam-scuffing rig is a simulation-type test rig in which the cylindrical surfaces of two eccentrically-mounted discs slide during part of their revolution against two flat surfaces loaded by air pressure. The conditions encountered in gear meshing, of a combination of rolling and sliding, are, therefore, simulated. Repeatable results were obtained on lubricants that were known to vary widely in their wear ability. This type of test has the advantage over gear testers in that a wider choice of material can be made and a greater ease of production is obtained. The exact conditions in gear meshing is not simulated, as the sliding contact does not vary and the rolling contact cannot be varied except by dimensional changes of the disks. As the knowledge on the subject of gear lubrication increases, it may be possible to perform an increasing portion of test work on the simpler simulation rigs, but this would always be in conjunction with works on gear rigs.

Tingle, E. D. An Instrument for Investigating Boundary Lubrication Properties. Royal Aircraft Establishment. Farnborough. ASTIA (ATI-107251) March 1951.

A rapid method of comparing the boundary lubricating properties of materials and the durability of solid lubricant films is described. A modified Wells instrument provides a record of the friction between lubricated metal surfaces at a range of sliding speeds from 0.26 to 5.2 cm/sec. A standard preparation technique for comparative tests is described.

Windish, W. W. German Methods for Determining Lubricity of Oils and Greases. AAF Translations. May 1947.

The report presents a view of the German progress, and a description of the different apparatus used in an attempt to develop an oil-test apparatus which will give: (1) a numerical value for the oiliness of a lubricant; (2) a value which is easily reproduced by various laboratories using this same apparatus; (3) and a value which correlates accurately with the actual service use of the lubricant.

The apparatus described were the Four Ball, Almen-Wieland, PTR, Thoma, Bearing and Gearwheel Apparatus of Prof. Heidebroek, Siebel, Oil-Test Apparatus of Z.F., DVL Oil-Test, Falex, Chain, Wire, Wear and Grinding.

Conclusions as to the Four Ball apparatus were that it operated with excessively high loading and they suggested that it would be possible that the thermal and mechanical requirements of the lubricating spot could be decreased by changing the speed, and possibly the load. It was considered very good for evaluating extreme pressure lubricants with additives, in comparison with lubricants without additives, but not good for lubricants without additives alone, as the accuracy of the machine was not good enough for small differences.

It was suggested that possibly the chain apparatus, using the wire adaptation was the better for evaluating oils alone.

Further research will be necessary to prove the advantages of the chain apparatus.

The Siebel apparatus was also considered very good for evaluating additives to lubricants; however, its results were similar to the Four-Ball for lubricants without additives.

Symposium on Industrial Gear Lubricants. Presented at a meeting of Technical Committee D-2 on Petroleum Products and Lubricants. Detroit, Michigan. A.S.T.M. June 22, 1948.

Purpose of papers was to present studies of various field problems encountered in the lubrication of industrial gears. It was suggested that perhaps the average of the results of several testers would be a better evaluation of an oil than by the use of just one. The use of a cathode ray oscilloscope to determine the vibration in a gear box was explained. Claims were made that it could detect such things as smooth action, deformed teeth, out of round, low loads, initial failure, and severe failure. The instrument may prove to be a useful tool in boundary lubrication studies.

APPENDIX I

<u>Libraries Searched</u>:

1. Wright Field Technical Library
   Wright-Patterson Air Force Base, Ohio

2. Armed Service Technical Information Agency
   Dayton, Ohio

3. California Institute of Technology
   Aeronautics Library
   Pasadena, California

4. Pacific Aeronautical Library
   Los Angeles, California

5. University of California at Los Angeles
   Engineering Library
   Los Angeles, California

6. Los Angeles Public Library
   Los Angeles, California

7. Western Gear Works Library
   Lynwood, California

APPENDIX II

Facilities Interviewed:

1. Shell Development Company
   Emeryville, California

2. California Research Corporation
   Richmond, California

3. Monsanto Chemical Corporation
   St. Louis, Missouri

4. Douglas Aircraft Company
   Santa Monica, California

5. Union Oil Company
   Research Laboratories
   Brea, California

6. Boeing Aircraft Company
   Seattle, Washington

7. California Research Corporation
   El Segundo, California