FINAL REPORT

ON

BEARING-LUBRICANT ENDURANCE CHARACTERISTICS AT HIGH SPEEDS AND HIGH TEMPERATURES

PERIOD: September 1, 1962 through August 31, 1965

Contributors

C. J. Wachendorfer L. B. Sibley

Project Leader:

C. J. Wachendorfer

NASA Contract No. NASw-492
NASA Control No. HS-922
SKF Report AL65T068
SKF Code 6471 5289
SKF Project IX-1
SKF Reg. 451 5
423 2

Submitted to:

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LEWIS RESEARCH CENTER
21000 BROOKPARK ROAD
CLEVELAND, OHIO 44135

RESEARCH LABORATORY
SKF INDUSTRIES, INC.
ENGINEERING AND RESEARCH CENTER
KING OF PRUSSIA, PA.

FINAL REPORT

ON

BEARING-LUBRICANT ENDURANCE CHARACTERISTICS AT HIGH SPEEDS AND HIGH TEMPERATURES

PERIOD: September 1, 1962 through August 31, 1965

Contributors

C. J. Wachendorfer   L. B. Sibley

Project Leader: C J Wachendorfer

Approval: L B Sibley

Released: [illegible]

NASA Contract No.	NASw-492
NASA Control No.	HS-922
SKF Report	AL65T068
SKF Code	6471 5289
SKF Project	IX-1
SKF Reg.	451 5
	423 2

Submitted to:

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LEWIS RESEARCH CENTER
21000 BROOKPARK ROAD
CLEVELAND, OHIO 44135

RESEARCH LABORATORY
SKF INDUSTRIES, INC.
ENGINEERING AND RESEARCH CENTER
KING OF PRUSSIA, PA.

ACKNOWLEDGEMENT

The administration of this project by Messrs. N. Rekos and M. Comberiate of the National Aeronautics and Space Administration, Washington, D.C., is hereby acknowledged. Technical guidance from Messrs. W. J. Anderson and E. Zaretsky of the Fluid Systems Components Division, NASA Lewis Research Center, Cleveland, Ohio, is also acknowledged.

AL65T068

FINAL REPORT
ON
BEARING-LUBRICANT ENDURANCE CHARACTERISTICS AT HIGH SPEEDS AND HIGH TEMPERATURES

ABSTRACT

23670

This study was performed to determine operability, critical mode of failure and life of angular-contact ball bearings of advanced design at high temperatures in excess of 500°F using the best available fluid lubricants in a recirculating system. 205-size 25mm-bore angular-contact ball bearings made of vacuum-melted tool steels having high hot hardness were tested with a number of high-temperature fluid lubricants representing hydrocarbons, esters, and polyphenyl ethers. The test machine was designed to simulate a typical aerospace accessory drive system with nitrogen blanketing to minimize lubricant oxidation at high temperature. Bearing design parameters and cage materials were developed for successful operation under the test conditions. Results were obtained from over 300 bearings tested with 17 different lubricants at high load (300 to 500 lbs. thrust) and speeds from 20,000 to 45,000 rpm. These results indicate that satisfactory operation is possible at bearing temperatures at least up to 600°F. Ten bearings made of M-1 tool steel with silver-plated cages and tested with Socony Mobil XRM 177F lubricant, a synthetic paraffinic hydrocarbon with an anti-wear additive, ran at 600°F and 42,800 rpm without any sign of failure or lubrication distress to lives in excess of twice the AFBMA computed L_{10} life. Other hydrocarbons and presently available esters have failed to prevent glazing-type surface distress and/or smearing and bearing wear under these test conditions, which led to early failure of the bearings at temperatures below 600°F. A broad correlation was obtained between the occurrence of surface distress and low lubricant viscosity at the bearing operating temperature. This is believed to reflect an insufficient elastohydrodynamic lubricant film condition in the bearing. Insufficient boundary lubricant characteristics of some fluids were found to produce smearing (galling) type failures early in the tests.

TABLE OF CONTENTS

LIST OF ENCLOSURES

LIST OF ENCLOSURES (Con't)

AL65T068

FINAL REPORT ON BEARING-LUBRICANT ENDURANCE CHARACTERISTICS AT HIGH SPEEDS AND HIGH TEMPERATURES

FOREWORD

This is the Final Summary Report submitted in fulfillment of NASA Contract No. NASw-492, entitled "A Study of Bearing-Lubricant Endurance Characteristics at High Speeds and High Temperatures". It encompasses research conducted on all phases of this program from September 1, 1962 through August 31, 1965, and previously reported in Quarterly Progress Reports No. 1 to 11.

SUMMARY

Procedures for testing angular-contact ball bearings (25 mm bore) lubricated with the best available high-temperature lubricants in an inerted re-circulating lubrication system have been developed to the point where reliable long-life operation is obtained at high temperatures, speeds, and loads. Testing has been conducted in the 600-650°F temperature range at speeds of 42,800 rpm (DN 1.1×10^6) under high endurance-test thrust loads (459 lbs. on the 7205 bearings corresponding to AFBMA calculated L_{10} = 240 mill. revs. and a nominal maximum Hertz stress of 250,000 psi at the ball-race contacts). Initial problems of bearing alignment and thermal transients were solved by the use of appropriate clearances and a massive housing design in which temperature gradients could be kept to a minimum. Lubrication distress (glazing and smearing) of the ball paths and excessive cage wear occurred regularly in the initial testing and were eliminated, after considerable development work, by the use of better bearing and cage designs, better lubricants, refined test procedures and higher speed operation (initial testing was at 20,000 rpm).

In the screening phase of this program (Phase I), 134 bearings made of 2 high-temperature consumable-electrode vacuum melted (CVM) tool steels (CVM M-1 and CVM WB49) were tested with 17 candidate high-temperature lubricants at temperatures up to 700°F (with nitrogen blanketing) and speeds up to 45,000 rpm. A failure mode analysis for high-temperature high-speed bearings was made which indicates the importance of sufficient lubricant viscosity at the bearing operating temperature. Lubricants having insufficient viscosity produce a type of lubrication-caused surface distress (glazing) in the ball tracks. Continued operation under such poor lubrication conditions results in premature flaking failure of the bearing. On the basis of these screening tests, the maximum safe bearing operating temperature (for 7205 bearings under 365 lbs. thrust load and speeds of the order of 43,000 rpm) to avoid this glazing-type surface distress has been estimated for each of the lubricants tested, as listed in the following tabulation. (Nitrogen blanketing was used except where noted; oil flow rate was 100-700 cc/min., depending on viscosity and speed.)

LUBRICANT	ESTIMATED MAXIMUM BEARING OPERATING TEMPERATURE, °F	ESTIMATED MINIMUM SAFE VISCOSITY AT OPERATING TEMPERATURE, CS
HYDROCARBONS:		
Socony Mobil XRM 177F	> 610	< 2.10
Socony Mobil XRM 109F-1	590	2.45
Kendall Bright Stock 0846	580	1.51
Kendall Resin 0838 (Modified)	< 580	> 2.60
Esso FN-3157	< 500	> 1.07
Socony Mobil XRM 112	< 450	> 0.94
POLYPHENYL ETHERS:		
Monsanto OS-138	600	1.15
Monsanto MCS 353	< 585	> 0.70
Monsanto Skylube 600 (In air)	570	0.94
Monsanto MCS 365	< 570	> 0.94
Monsanto OS-124	< 550	> 1.05
Monsanto Skylube 600	< 540	> 1.08
Monsanto MCS 293	500	0.78
ESTERS:		
Sinclair Turbo S (Type 1048)	550	1.26
Esso Turbo Oil 35	515	1.36
Cellutherm 2505A	< 445	> 1.0
Socony Mobil RM 139A	440	1.08
Heyden Newport TP653B	400	1.03

It can be seen from this tabulation that the minimum safe viscosity corresponding to the maximum safe operating temperature for these lubricants varies only from about one to 2½ centistokes. For the operating conditions and test bearings used, this minimum safe viscosity corresponds to a calculated critical minimum ratio of the elastohydrodynamic oil-film thickness to the composite surface roughness at the bearing contacts equal to about 1.5 for the hydrocarbon base-stocks and about 0.9 for ester-base lubricants. There were no clearly successful polyphenyl ether tests so that the calculated critical ratio for this lubricant is probably considerably above 1.5. Due to known inaccuracies in the calculation of film thickness, the real value of the critical minimum ratio is probably close to 1.5 for all lubricants. In order to increase this ratio, there has been increased emphasis on surface finish for good performance of tool-steel bearings at high speeds and temperatures. Analysis techniques are available for computing lubricant film thickness, contact stress, and kinematic parameters in ball bearings for application of these results over a wide range of bearing size and operating conditions.

It was also found that avoidance of smearing (galling) failure requires adequate boundary lubrication and in this respect, esters and hydrocarbons with suitable additives were the lubricants found most successful. Control of ball-to-race sliding by assuring outerrace control of the ball spin through proper bearing design is also required to minimize smearing.

Other important results of this screening phase of the program have been the development of engineering criteria for the design of the cages for high-temperature high-speed tool steel bearings. Several candidate cage materials were evaluated in a simplified screening test and promising cage materials for wear resistance were obtained. Modifications of the cage design were tested in bearings. On the basis of these results, an optimum bearing design was selected and bearings manufactured of CVM WB49 and CVM M-1 steel, using wide-land silver-plated hardened M-1 steel cages, for endurance testing with selected lubricants in the endurance phase of this program (Phase II).

Phase II endurance testing at 42,800 rpm was completed on 6 groups comprising 14 to 30 bearings each and 6 smaller groups of 4 to 10 bearings each. The results of these endurance tests are summarized in the table below in which it is noted that only one test group failed predominantly by flaking (always with prior surface distress), whereas in the other short-lived groups, many bearings smeared.

Lubricant	Brg. Steel	Avg. Temp. °F	Thrust Load Lbs.	No. Brgs. Tested	No. Smearing and Flaking Failures	Brg. L_{10} Life Mill. Revs. Calculated AFBMA	Brg. L_{10} Life Mill. Revs. Est. From Smearing and Flaking Failures	Predominant Failure Mode
Hydrocarbons:								
Bright Stock	M-1	580	459	14	7	240	•	Smearing
Bright Stock with TCP	M-1	570	459	4	2	240	0.2	Smearing
XRM 109F-1	WB-49	585	459	8	4	240	3.1	Smearing
XRM 109F-1	M-1	580	459	18	6	240	1.3	Smearing
XRM 177F	WB-49	600	459	16	8	240	1.4	Smearing
XRM 177F	M-1	600	459	10	0	240	> 500.	No failure
Esters:								
Turbo Oil 35	WB-49	< 300	459	8	4	240	0.02	Smearing
Turbo Oil 35	M-1	500	459	30	10	240	58.8	Flaking
Turbo Oil 35	M-1	500	365	30	2	480	247.7	Smearing
Polyphenyl Ethers:								
Skylube 600	M-1	595	459	22	11	240	•	Smearing
Skylube 600**	M-1	600	459	8	4	240	•	Smearing
Skylube 600	M-1	481	365	6	3	480	•	Smearing

• No valid life estimates were obtained since most failures occurred by smearing shortly after start-up.

** These tests were run in an air environment, all others with N_2 blanketing.

These results show (XRM 177F oil with M-1 bearings) that spectacularly successful operation of tool steel bearings at 600°F is indeed possible. It is conjectured that the excellent performance of this particular bearing-lubricant combination resulted from (a) positive outer-race control of the ball spin by a sufficiently excessive ball spin torque on the outer ring compared to that on the inner ring, and (b) sufficiently effective boundary lubricating characteristics of the XRM 177F oil (containing an anti-wear additive) to prevent glazing surface distress. Based on the long life obtained with the Socony XRM 177F oil and M-1 bearings, there is no reason to suspect that any high-temperature derating of bearing fatigue life from the AFBMA standards used for more ordinary operating conditions is required. It is necessary, of course, to design and manufacture the bearings especially to provide the best possible lubrication conditions for the lubricants available and to find suitable lubricants. Bearings must have smooth ball-race contact surfaces, accurate contact surface geometry and be designed so as to minimize the ball-race sliding which is always inherent in high-speed bearings.

The results also show that about half the AFBMA computed L_{10} life of these bearings can be obtained at 500°F with Turbo Oil 35 under 365 lbs. load, but at 459 lbs. load, less than 1/4 of the computed life was realized. With this lubricant, surface distress was not completely prevented.

Numerous smearing failures shortly after start-up with Skylube 600 are attributed to the poor boundary lubricating properties of this fluid when used in nitrogen environment. Some bearings ran to over 3/4ths of the AFBMA computed L_{10} life with Skylube 600 in an air environment.

INTRODUCTION

In the development of accessory drives and power plant systems for advanced aircraft and missile applications, the need for reliable high-speed high-temperature bearings and lubricants has become ever more apparent. The low starting torque, simplicity of design and high reliability of rolling bearings make them ideally suited for the turbine-driven machinery in aircraft and missiles. However, a detailed working knowledge of the failure mechanisms and long-life endurance characteristics of high-speed bearing-lubricant systems at temperatures above 500°F is required for the development of new materials and design criteria for advanced systems. The purpose of this research project was to provide this knowledge and to evaluate the performance and reliability of the best available high-temperature bearing steels and the most recently developed high-temperature fluid lubricants in a bearing-lubricant system under conditions expected in advanced aerospace applications. In this manner, basic design and engineering data was obtained which allows more confident application of high-temperature materials and lubricants to rolling bearings.

The research program was divided into two phases. In Phase I, candidate lubricants were screened for limiting load, speed, and temperature for reliable operation of tool-steel ball bearings, and design parameters of the bearings were developed for optimum performance. In Phase II, optimum bearing-lubricant combinations were endurance tested under predetermined test conditions in multiple bearing groups to establish design life and reliability parameters. Four test rigs developed by SKF Industries, Inc., to simulate a typical aerospace accessory drive were used on this program. Test procedures and data analysis were designed in such a manner as to facilitate the general applicability of the results to rolling bearings in all sizes.

As a supporting effort, testing of cage materials in an element test rig was also performed and has yielded screening information on cage material performance which was used in selecting the cages run in full-scale bearings.

BACKGROUND

The successful operation of rolling bearings at high speeds and high temperatures depends upon the appropriate selection of bearing materials, bearing designs, lubricants, and methods of lubrication. The following is a discussion of the technical background on which the selection of bearing and lubricant materials and designs for this program was based.

High-Temperature Bearing Steels

After a survey of elevated temperature properties, such as hot hardness, compressive yield strength, resistance to softening, and structural and dimensional stability of 29 candidate high-temperature bearing steels (1)*, the Crucible Steel Company of America, under contract to the Air Force, developed WB49 bearing steel, which maintains a hot hardness of Rc 57 after 500 hours at 1000°F (34). An extension of this work (2) resulted in the development of a corrosion-resistant high-temperature bearing alloy, WADC65, which has adequate temperature resistance (in the range 600-900°F), good dimensional stability and maintains a hot hardness of Rc 57 after 500 hours at 900°F. However, its application depends on the development of methods for overcoming the great difficulties encountered in forging this steel, which is not available in bearings for this reason.

Consumable-electrode vacuum-melted (CVM) M-50 tool steel has shown consistently better bearing fatigue characteristics than most analyses tested in the SKF Research Laboratory and in jet-engine service. M-50 has suitable hot hardness characteristics for use in bearings operating at temperatures up to 600°F, (Rc 57 after 1000 hours at 600°F). Another type tool steel, M-1, is a well-known dependable high-temperature bearing material which displays a hot hardness of Rc 57 after 1000 hours at 800°F. Other high-temperature tool steels exist such as M-2 and M-10 which have similar hot-hardness characteristics to M-1. These steels were repeatedly specified in earlier years for jet-engine bearings, but it has been the experience of SKF Industries, based on wide-ranging contact with jet-engine makers and on Laboratory evaluation,

*Numbers in parentheses refer to References listed at the end of this report.

that neither of these steels is competitive with the M-50, M-1, and WB49 group, in regard to endurance capabilities, melting and heat treatment capabilities or the ability to be reliably fabricated into acceptable bearings.

Hot hardness data taken after long-term exposure at high temperature on WB49 and M-1 tool steel, which are considered the most promising materials currently available for bearing operation at temperatures from 600°F to 1000°F, are given in the tabulation below:

Steel	Temperature, °F	Hot Hardness after Soaking at Indicated Temperature, Rc	Time at Indicated Temperature, Hours
WB49	600	65	1000
"	800	62	1000
"	1000	57	500
M-1	600	60	1000
"	800	57	1000
"	1000	52	1000

Endurance data are available on the above potential high-temperature bearing materials under a variety of temperatures and lubrication conditions. In the SKF Industries' Research Laboratory, a group of thirty 6309 bearings* with rings and balls made of consumable-electrode vacuum-melted (CVM) WB49 steel have been life tested at 210°F under a radial load of 4240 lbs., for which the AFBMA-computed L_{10} = 10 mill. revs. for the complete bearing (31) and the computed L_{10} = 10.8 mill. revs. for the inner rings only,

* 45-mm bore deep-groove ball bearings

according to the Lundberg-Palmgren theory (32). These tests were run at 9700 rpm with circulating Socony Mobil DTE Extra Heavy oil lubrication, and the endurance data obtained are summarized below.

Endurance of 6309 Bearings Made of High-Temperature Tool Steels

Steel	Speed rpm	No. of Failures: Inner Rings	Outer Rings	Balls	Combination	Est. L_{10} Life, Mill. Revs.: Complete Bearing	Inner Rings
CVM WB49	9700	9	6	12*	2	26	93
IVM M-1	1500	19	2	3	2	18	19

Also shown in the above tabulation are the results of an endurance test of thirty 6309 bearings made of induction vacuum melted (IVM) M-1 steel, which were run at 1500 rpm with drop-feed mineral oil lubrication (33). These M-1 bearing tests were run in 1957 at the same temperature and load as the more recent CVM WB49 tests. Since the L_{10} life of both the CVM WB49 and IVM M-1 bearings exceeds the AFBMA-computed L_{10} = 10 mill. revs. for these test conditions, these materials appear to be good candidates for high-temperature usage. In fact, the inherent endurance life of CVM WB49 steel, as reflected by the L_{10} life of the inner rings, may be several times the AFBMA L_{10} life, if balls having good endurance characteristics are used. It is the experience in the SKF Industries' Laboratory that 6309 bearings run at 9700 rpm have about twice the life of statistically similar bearings run at 1500 rpm due to lubrication effects on endurance (35-37) so that the life data at low speed given for IVM M-1 bearings in the above tabulation also reflects very good endurance characteristics for this steel.

---

*Many balls of this material exhibited micro-cracks in fabrication. Even though all cracks discernible in 100% magnetic particle inspection were eliminated from the test balls, it is suspected that the remaining balls had high residual stresses, possibly micro-cracks, that influenced life.

M-1 steel has a long history of both full-scale bearing and bench-type endurance tests. Room-ambient fatigue was studied by Bear and Butler (3) and Carter (4). The effect of various synthetic lubricants on the fatigue properties of M-1 tool steel at 100°F (5) and at elevated temperatures up to 450°F (6), were studies by Carter utilizing the NASA Fatigue Spin Test Rig. Anderson (7) ran a limited number of 110-mm-bore M-1 tool steel ball bearings at 14,000 rpm and 678°F, lubricated with a synthetic diester oil, concluded that fatigue life of M-1 bearings operating at this temperature might be much shorter than the AFBMA rating life for 52100 steel bearings lubricated with mineral oil at normal temperatures. The effect of lubricant viscosity on the fatigue life of M-1 tool steel has been studied (9) as has the effect of hardness on the fatigue resistance of both M-1 and WB49 steels (10) in the NASA spin rig element tester and the NASA 5-ball fatigue tester.

High-Temperature Lubricants

When the advent of gas-turbine power plants first pushed aircraft engine bearing operating temperatures beyond the capabilities of conventional petroleum oils, the development of synthetic lubricating fluids was undertaken. The guiding philosophy of most such lubricant development programs was to obtain high-temperature thermal and oxidative stability in fluids having relatively limited viscosity variation over wide temperature ranges. The most notable results of this early research were the diester-base lubricants used in the bulk of aircraft jet engines today (under the military specification MIL-L-7808). Di 2-ethylhexyl sebacate, the standard base-stock for MIL-L-7808 lubricants, very neatly combines the thermal stability of the ester grouping in a large molecule with the low variation of viscosity with temperature that is characteristic of the paraffin-type constituents in this fluid. The molecular-size, of course, was selected to combine pumpability at -65°F with adequate high-temperature viscosity for up to 350°F bulk engine oil temperature. Additive refinements and other ester-base oils (e.g. MIL-L-25336 and MIL-L-9236) have pushed the upper temperature limits to 450-500°F, and in other developments (MIL-L-23699 and a British jet-engine oil specification) ester-base lubricants with higher viscosity than MIL-L-7808 have been made available.

Striking recent advances in synthetic lubricant development, however, have been made with molecular types having much higher inherent thermal and oxidative stability than paraffinic straight-chain configurations. Such a lubricant is polyphenyl ether (11), which has considerable radiation stability as well, but, at the expense of the poor viscosity-temperature characteristics of aromatics. Thus the polyphenyl ethers usually have high pour points when compared with more conventional lubricants.

Another class of materials which has shown superior high-temperature lubricant characteristics are the synthetic hydro-carbons and highly-refined mineral oils (12) (13). Materials of this latter type have a useful liquid range on the order of -20 to 700°F, which, by deep de-waxing can be extended from -65 to 700°F. In addition, super refined mineral oils are susceptible to lubricity additives thereby offering the possibility of significantly improved performance through the utilization of additive packages.

Investigations into high-temperature high-speed ball-bearing lubrication in the temperature range of 400-1000°F have indicated that several types of "synthetic" fluids and mineral oils show promise of providing adequate lubrication over reasonable periods of time at various temperatures depending on the lubricant, the material it contacts, and the atmosphere in which the test operates. One of the most optimistic tests reported (concerned with the ability to operate without catastrophic failure) covers 20mm bore, M-10 tool-steel bearings operating at speeds to 60,000 rpm, loads to 200 lbs. and ambient temperatures up to 775°F, utilizing various available high-temperature lubricants, (14). These studies have involved mostly the supplying of the lubricant to the bearing in a "once-through" lubrication system. The effectiveness of various high-temperature additives, as well as base-stocks, was evaluated in terms of bearing operating feasibility with emphasis on lubricant-metal-environment chemistry as related to the lubrication processes involved. Recirculating oil tests also have been run using 35-millimeter-bore ball bearings under very light thrust loads and high speeds at temperatures up to 800°F. The only oil that would operate at all in a recirculating system at 800°F, however, (polyphenyl ether) did not protect the bearings from excessive wear of the raceways and cage. No fatigue flaking failures were obtained in these tests since the loads were not high enough. In almost every case with a recirculating system, the test was terminated because of excessive evaporation of the oil.

In related work done elsewhere (16) 150-millimeter-bore ball bearings made of CVM M-50 steel were tested at high thrust loads and speeds and found to have a fatigue life several times the computed value when the bearings were lubricated with a recirculating system using either polyphenyl ether or MIL-L-9236 type lubricants in air at a bearing temperature of 550°F and an oil-in temperature of 400°F.

High-Speed Bearing Kinematics and Contact Lubrication

Although the predominant motion at the ball-race contacts in ball bearings is a rolling motion, substantial sliding occurs at high speeds and under thrust load. It is this sliding motion that is thought to have important effects on lubrication and surface distress in rolling bearings. Palmgren (17) describes the following principal cases of partial or predominant sliding in the Hertzian contacts of ball bearings:

a. "Heathcote slip" due to the curved contact areas in the ball grooves.

b. Ball spin in ball bearings with non-zero contact angle, due to the fact that the condition of pure rolling is not satisfied in this configuration. (At least at one of the race contacts on each ball, there must be a spinning or twisting of the ball on its contact in addition to the rolling motion.)

c. Sliding of balls in bearings under combined load where circumferential load variations cause contact angle and deformed rolling diameter variations that result in variable rolling element speed against which cage forces militate (18).

In cases (a) and (c), the sliding velocities are given by the bearing kinematics. In case (b), sliding velocities depend on the "slip control" of the ring-ball contacts, i.e., whether there is sliding at both contacts or only one and, if so, which. Several investigators, e.g. Jones (19) and Shashaty (20), have analyzed the kinematics of ball spin, but in spite of substantial effort, the magnitude of spin cannot be accurately predicted because it depends on unknown friction parameters at the contacts.

Besides the above mentioned cases of "legitimate" or necessary sliding, there are many types of sliding in the contacts which are not necessary. Some of these are:

a. Sliding due to deviation from theoretical cage or rolling element speed (Palmgren (17) Macks, Nemeth, Anderson (21).

b. Sliding due to unavoidable inaccuracies of rolling element geometry (ball diameter differences, etc.).

The existence of these various instances of sliding is often obvious without direct experimental confirmation from kinematical evidence. Other cases, e.g., Heathcote slip, have been confirmed by observation of wear patterns in used bearings. Still others, e.g., ball spin, have been shown by special experiments (22).

Incidence of smearing on inadequately lubricated sliding surfaces is commonplace in rolling bearing practice. More subtle effects of a different type of surface distress and wear in the absence of gross sliding have been investigated by Lawrence and Schmidt (23) and Remorenko (24).

No satisfactory conditions for predicting the incidence of surface damage as a consequence of sliding have been found in the literature on rolling bearings. It is felt that this is due largely to the lack of depth in the analyses of micro-contact phenomena of welding and wear under the conditions encountered in rolling bearings.

In studies of the effect of lubrication on rolling contact fatigue, methods have been developed for demonstrating the separation of bearing surfaces by lubricant films formed hydrodynamically in the contact regions (25-28)*.

Rolling bearings form lubricant films at Hertzian contacts between bearing parts by a process called elastohydrodynamic lubrication. Lubricant pressures are generated hydrodynamically in the oil films adhering to the bearing surfaces as they are swept into the contact regions by the rolling motion. These pressures are balanced against the normal contact stresses required to deform the contact area elastically, hence, the term elasto-hydrodynamics.

---

*In (25) extensive further references are cited.

It has been shown in the SKF Industries' Research Laboratory (25), that under conditions of lubrication when significant contact of surface asperities occurs through the lubricant film, contacts occur at a rapid rate (several hundred micro-contacts per inch of track) and that these micro-contacts can be related to the micro-geometry (or roughness) of the contacting surfaces. Analysis of practical bearing surfaces has revealed that the surface micro-geometry can be represented by random functions. A description of the asperity distribution and spacing has been given (29).

Radioactive tracer wear tests have shown that an "adhesive type" wear occurs at an increasing rate as the hydrodynamic film becomes thinner (29, 30). The wear rate after run-in with several aerospace lubricants, as measured by a radiotracer technique (30), was found to be proportional to the total rolled-over area of asperity contact determined by the ball surface micro-geometry analysis already described (29). The proportionality constant is a characteristic of the lubricant, of the order of 1 microgram/in^3 for compounded ester-type lubricants, rating these as best, and 4 micrograms/in^3 for ester base-stock, 6 micrograms/in^3 for polyphenyl ether and 12 micrograms/in^3 for super-refined mineral oil base-stock without additives, rating these lubricants progressively poorer in the order given, regarding their microwear-preventing ability.

As the elastohydrodynamic oil film becomes progressively thinner, compared to the composite surface roughness at rolling bearing contacts, a type of surface distress known as "glazing" occurs in the rolling tracks and becomes more severe with thinner films. In severe cases this glazing results in micro-cracking of the surface which eventually leads to flaking failure, but at a greatly reduced fatigue life compared to that which can be obtained with the same bearings operated under more favorable lubrication conditions. The effect of the film-thickness-to-roughness ratio on endurance life of 52100 steel balls in rolling-contact 4-ball fatigue testing machines has been demonstrated (37).

TEST ELEMENTS

A. Test Bearings

Based on superior physical properties at high temperature, and on promising performance in bearings, discussed previously in this report, CVM M-1 and CVM WB49 tool steels were selected for test bearing rings and balls for this program. Composition of these steels is given below:

Element	Composition, % M-1	Composition, % WB49
C	0.75-0.85	1.00-1.10
Mn	0.15-0.40	0.20-0.40
Si	0.15-0.40	< 0.01
Cr	3.50-4.25	4.00-4.50
V	0.90-1.25	1.80-2.10
W	1.30-1.80	6.50-7.00
Mo	8.25-9.50	3.50-4.00
Co	None	5.00-5.50

Test bearings were manufactured to a 7205-size angular-contact design having counterbored outer races, inner-ring riding cages and ABEC 5 tolerances. Each of the bearing components (inner rings, outer rings, balls) of a given steel analysis was made from one heat of steel. As shown in Enclosures 1 and 2, bearings for initial testing were made according to the standard practice for high-speed aircraft quality bearings of this type which have a nominal contact angle of 17°, a radial looseness of 30-40 microns and narrow-width cages. As a result of early Phase I testing, however, the test bearings were modified until a final design evolved which was used in the Phase II endurance tests. This design, as shown in Enclosures 3, 4, and 5, calls for a nominal contact angle of 19°, a radial looseness of 40-50 microns and wide-land silver-plated bore-recessed hardened steel cages. In addition, the inner and outer rings of all final design bearings were black oxide coated to improve their resistance to lubrication-related surface distress, and the ring grooves were finished to a smoother surface finish (2-4 microinches, rms) than the standard bearings

(6-8 microinches, rms) except the CVM M-1 outer rings, for which the manufacturing method to produce the smoother finish on these counterbored tool-steel rings had not yet been developed at the time they were made.

The L_{10} life of the final design bearing (19° contact angle) was computed according to the AFBMA method given in Appendix I and is shown in Enclosure 6 as a function of thrust load applied to the bearing. A complete analysis of the dynamic contact angles, Hertz stresses, other kinematic parameters and estimated life and heat generation rates of both the 17° and 19° contact angle bearings, under all conditions of their operation in this program, is also described in Appendix I and the results are given in Enclosure 7. In most testing the maximum Hertz stress at the ball-race contacts was about 250,000 psi. The variation of Hertz stress with bearing design, speed and load is given in Enclosure 7.

B. Test Cages

Since early test results indicated that cage wear was a limiting factor in the operation of the test bearings under severe lubrication conditions, a separate test program was conducted to evaluate the performance of candidate cage materials with some of the promising lubricants in order to guide the development of improved cages for high-speed high-temperature tool-steel bearings.

Cage specimens manufactured from the following candidate high-temperature materials were run with several high-temperature lubricants in a modified element tester which is described in a later section of this report.

a. M-1 tool steel, annealed (Rb 96)
b. M-1 tool steel, hardened and tempered to Rc 40
c. M-1 tool steel, hardened to Rc 60
d. S-Monel, annealed (Rb 85)
e. S-Monel, hardened (Rc 33)
f. Haynes 25, age hardened (Rc 50)
g. Hidurel, annealed (Rb 80)
h. Polyimide, type SP-1
i. Polyimide, type SP-2
j. 440C (modified) stainless steel, hardened (Rc 52)
k. 440C (modified) stainless steel, hardened (Rc 57)

TEST LUBRICANTS

The lubricants selected for evaluation on this program can be divided into two general groups, those capable of temperatures higher than 500°F and those capable of temperatures up to 500°F.

A. Lubricants Reportedly Capable of Bulk Sump Temperatures Higher Than 500°F

1. Formulations based on super-refined mineral oils or hydrocarbons

a. Esso FN-3157, a super-refined naphthenic oil containing an additive package which enables especially good performance in a nitrogen-blanketed system.

b. Kendall Resin 0838 (modified), a naphthenic resin material having a rather high viscosity and better volatility properties than the polyphenyl ethers.

c. Socony Mobil XRM 112, a largely aromatic synthetic hydrocarbon reported to have good stability under beta radiation as well as at high temperature.

d. Kendall Bright Stock 0846, a highly viscous hydrocarbon petroleum product.

e. Socony Mobil XRM 109F, a paraffinic-type synthetic hydrocarbon. (XRM 109F-1 and XRM 109F-2 were different batches of the same basic XRM 109F formulation.)

f. Socony Mobil XRM 177F, the synthetic hydrocarbon XRM 109F containing a proprietary anti-wear additive.

2. Polyphenyl Ethers

a. Monsanto OS-124, a mixed isometric five-ring polyphenyl ether which possesses extraordinary resistance to degradation from heat, oxygen, radiation, hydrolysis and chemical attack.

b. Monsanto Skylube 600, the five-ring material OS-124 containing a proprietary antioxidant additive.

c. Monsanto MCS 365: OS-124 containing 2% (by weight) of tri-aryl phosphine sulfide, an anti-wear additive.

d. Monsanto OS-138, a higher viscosity 6-ring material which was furnished for this program by the Fuels and Lubricants Branch, Code ASRCNL, Wright-Patterson AFB, Ohio.

e. Monsanto MCS-293, a modified polyphenyl ether-type material which is characterized by outstanding thermal, oxidative and radiation stability in the presence of air at high temperature, and by a low pour point (-20°F).

f. Monsanto MCS-353, a more viscous version of MCS-293.

3. Fluorocarbon

a. DuPont PR-143, a fluorocarbon fluid having excellent thermal and oxidative stability, high temperature viscosity and low pour point (-20°F), furnished by Code ASRCNL, W-P AFB, Ohio.

B. Lubricants Reportedly Capable of Bulk Sump Temperatures Up to 500°F (Ester-Base Lubricants)

a. Celanese Cellutherm 2505A which qualifies under MIL-L-9236.

b. Heyden Newport Pentalube TP 653B which qualifies under MIL-L-9236 and has reportedly established a new standard of stability, lubricity and cleanliness for fluids at bulk-oil temperatures of 425°F.

c. Socony Mobil RM-139A, an ester-base fluid claimed to be thermally stable to 600°F.

d. Esso Turbo Oil 35, an ester-base fluid having higher viscosity than the MIL-L-7808 esters.

e. Sinclair Turbo S (type 1048 improved), a highly viscous ester-base oil.

The viscosity-temperature characteristics of these test lubricants are shown in Enclosure 8. Physical properties of each oil, with the exception of those listed as proprietary, are given in Appendix II.

TEST APPARATUS

A. Bearing Test Machines

In the studies that led to the establishment of the initial design conditions for a bearing tester, capable of running bearings at speeds of 40,000 to 50,000 rpm and with ambient temperature of both bearings and lubricants up to 1000°F, the following inherent design problems became apparent.

a) The design of a combined load machine, with the test bearing loads reacted through standard support or load bearings, would be difficult due to the need to keep the load bearings isolated from the severe test conditions.

b) A standard lubrication arrangement involving the pumping and delivery of lubricants at high temperatures from sources remote from the tester would present considerable problems of sealing, of keeping lubricant lines hot and clean, and would require a large volume of potentially costly lubricant. In addition, lubricant volume is a test variable influencing lubricant stability and it should be no larger than available in service.

c) The effects of temperature on the strength, dimensional stability and corrosion of construction materials are such that simple, sturdy shapes feasible from high-temperature steels are required.

d) Thermal expansions of the metals used in construction must be compatible with each other and with those of test bearing materials.

Design principles capable of solving the above problems were outlined as follows:

a) Use a two-bearing rig, for thrust loads, only.

b) Make the system essentially self-contained using a minimum of external piping for lubricants.

c) Construct critical parts of thermally stable stainless steels with suitable high-temperature properties and similar thermal expansion, compatible with that of the bearings.

On these premises, the test machine shown diagramatically in Enclosure 9 was developed. A photograph is shown in Enclosure 10. The test machine consists of two main sections, the test bearing housing and the lubricant reservoir.

As shown in Enclosure 11, the bearing housing assembly is essentially a housing block [101] *, in which are located the test bearing rotating assembly and elements for loading, heating, and lubricating the test bearings.

The housing block is machined from a solid forging of Allegheny Ludlum AM355 steel, a precipitation-hardened stainless steel which combines the corrosion resistance and formability characteristics of the austenitic stainless steels with the strength of the straight chromium, martensitic steels. This alloy retains useful strength from room temperature to 1000°F. (Yield strength at 1000°F is 70,000 psi 0.02% offset).

Located in the center of the housing block is a sub-assembly consisting of the shaft (including an integral screw pump), shaft liner, test bearings, balance washers, flingers and locknuts. This sub-assembly, shown in Enclosure 12, can be preassembled outside the test machine, a feature which facilitates mounting of the bearings and permits, if desired, balancing of the rotating parts as they would be run in the tester. The test bearings are mounted at room temperature with a line-to-line fit on the shaft and a press fit in the housing (obtained by heating the housing at assembly) so that the mounted bearing radial looseness is almost independent of operating temperature. This arrangement is required by the fact that the shaft and housing materials have greater thermal expansion than the bearings.

---

* Numbers in square brackets refer to item numbers in Enclosure 11.

The flingers (which act like small centrifugal pumps) are utilized to maintain a steady circulation of oil mist through the bearings.

The function of the pump liner is to produce the small radial clearances required by the screw pumps, the operation of which will be explained later.

When the shaft sub-assembly is mounted in the housing block, the outer ring of the drive end bearing fits into an accurate bore and seats against the shoulder formed by the housing cap [102] which is fastened to the end of the block. The pump liner slides into the same bore as the drive-end test bearing and is held in position by the oil tubes [103] which screw into the threaded holes on the bottom of the liner. The outer ring of the bearing opposite the drive end fits into the load plug [104] which slides into the large bore of the housing block. Thrust load is applied to the outer ring of this test bearing through the load plug by means of the dead weight and level arm system [105] which is mounted to the top of the housing block. The load is transmitted by the shaft to the drive-end bearing and from there to the housing cap. The load plug is equipped with an extension arm [106] by which it can be oscillated through a small arc in order to assure that it does not bind in the bore either initially or during operation of the tester.

In order to prevent the lubricants from seeping under and eventually coking around the load plug a shallow circumferential groove was cut into the load plug near the deeper drain groove (Enclosure 9). When nitrogen gas is fed into this shallow groove a positive gas flow toward the bearing prevents the lubricant from seeping under the load plug.

The shaft, pump liner, flinger, balancing washers, locknuts, housing cap, and load plug are made in most instances of Armco 17-4PH, a stainless steel similar to the AM355 steel from which the housing block is machined. Several spare parts were made from Vanadium Alloy Steel Company's hot formed No. 1 steel or modified 52100 steel, stress-relieved for 1000°F operation, which have compatible thermal properties to AM355.

The design and specifications of the above parts are such as to assure alignment of the bearing rings with respect to each other at all temperatures within 0.001 in. misalignment on a 3 in. diameter.

The housing block is maintained at the desired test temperature by means of cartridge heaters [H1] to [H7] which are imbedded in the block in a circular pattern around the shaft sub-assembly. Thermocouples [TC1] and [TC2] , which extend into the center portion of the block, monitor its temperature.

The temperatures of the test bearings and shaft liner are measured by thermocouples [TC3], [TC4], and [TC5] which contact their outer diameter.

The lubricant is supplied to the test bearings by means of screw pumps which are integral with the shaft sub-assembly. Essentially, these pumps consist of shallow square threads ground directly on the shaft with a major diameter that will allow only a small clearance with the inside diameter of the shaft liner. This small clearance forms an effective seal against lubricant "leakage" over the top of the threads.

As the shaft rotates, a velocity gradient is induced in the lubricant filling the threads between the inner wall of the rotating "channel" formed by the threads and the stationary outer wall formed by the shaft liner. Thus, there results in the threads a flow of lubricant which, depending on the back pressure developed, will deliver up to half the volume generated by the thread cross-sectional area moving axially in any given period of time.

On leaving the pump the lubricant enters an annular groove which is cut into the shaft. There are two paths through which it can leave this area. One is the annular capillary formed by the shaft and the liner and the other is a by-pass drain which leads to the sump. The flow to the sump can be restricted by a needle valve [107], so that by regulating the quantity of the by-pass flow, the flow through the annular capillary can be controlled.

The oil which flows through the capillary splashes against the test bearing and provides lubrication for the bearing. Drain lines on both sides of the bearing return the oil to the sump. (A valve in the inboard drain line is used to control the proportion of the oil flow which passes through the bearing). The oil does not flow directly into the sump, but in each case it passes through a sight flow indicator [108] made of 300-series stainless steel which is mounted to the side of the tester.

Needle valves are provided on the return side of each sight flow indicator. By closing one valve and timing the rise in level for a given volume the oil flow rate can be determined. Also with this valve wide open a rough approximation of the oil level in the corresponding sump can be obtained.

Since coking in the annular capillary or in the oil tubes would result in a respective rise or fall in output pressure of the pump, this pressure is continually monitored by means of a pressure gage [112] , which in some cases is a Nak-filled membrane-type. The oil pressure gages are equipped with high and low pressure electric alarm and automatic shut-off mechanisms. In this manner, surface failures due to lubricant starvation during operation are prevented.

The lubricant for each test bearing is drawn from a separate sump through an 80 or 450 mesh stainless steel screen [111] (depending upon the degree of coking taking place) which is submerged in the sump. The individual sumps, which are interconnected to maintain equal oil level, are contained in the lubricant reservoir [201] which is machined from a solid forging of AM355 stainless steel and mounts directly to the bottom of the test bearing housing assembly.

The desired ambient temperature of the lubricant is maintained by cartridge heaters [H8] to [H14] which encircle the sumps. The temperatures of the sump walls are monitored by thermocouples [TC6] and [TC7] which extend into the reservoir housing to a point 1/8" from the sump wall. The lubricant temperature is also monitored by thermocouples [TC8] and [TC9] which pass through the walls and into the lubricant.

Inert-gas blanketing of all lubricant-filled regions in the machine is achieved by slowly feeding nitrogen through ports at the top of each sump cavity and into each bearing chamber. Air and excess nitrogen is purged through sump vents and a labyrinth seal on one end of the shaft. Priming of the screw pumps is accomplished by closing all return lines and vents to the sumps and increasing the nitrogen pressure.

Early testing with a single nitrogen blanketing line (supplying both sumps but not the bearing cavity) revealed that lubricant would flow from one sump to the other, eventually starving one bearing, even if the pump supply pressures to the two bearings were identical. This apparently was the result of slight differences in the pressure-flow characteristics of the pumps or in the distribution of the nitrogen pressure through the machine. Lubricant flow between the sumps was monitored through a connecting line and controlled by pressure balancing at the pump inlets with the separate nitrogen supply lines to the sumps. In addition, a separate supply of nitrogen was fed directly into the test bearing areas, rather than being supplied into these regions from the sumps in order to further reduce any possible disturbing pressure differentials in the machine. These methods have permitted stable operation.

Of the four test machines utilized in this program, one is designed for variable-speed operation and used primarily in exploratory Phase I testing. Since endurance testing was to be conducted on multiple-bearing groups at preset test conditions, the remaining three test machines were designed for constant-speed operation.

The variable speed test machine is driven by a Westinghouse 50-horsepower DC motor and Ward-Leonard Control system. Speed control of this drive unit ranges from a few hundred to 3500 rpm. The constant speed test machines are driven by 60 horsepower. 3560 rpm Louis Allis part-winding start motors. Test shaft speeds up to 50,000 rpm are possible through V-Belt drives and Snow-Nabstedt speed-increaser gear boxes.

The instrumentation cabinet for the variable-speed test machine is shown in Enclosure 13.

In view of the extreme conditions of speed and temperature it was considered advantageous to have the instrumentation, in addition to controlling and monitoring test parameters, provide for the complete shutdown of the apparatus upon any malfunction affecting proper operation of the machines.

For this reason, the indicators for the screw pump output pressures are, as mentioned previously, equipped with electric contacts which through a system of relays, shut off both the motor and the heaters when the pump pressure exceeds or falls below preset values.

Excessive pressure would be a result of blockage in the outlet paths of the pump while low pressure would result from cavitation, supply line blockage or clogged filters. In both cases the test bearings would receive inadequate lubrication, a condition which would not be tolerated for any length of time under the severe test conditions.

Excessive vibration stemming from bearing failures, misalignment or any other reason also initiates complete shutdown of the test apparatus by means of a vibration switch mounted to the load arm.

Protection against over-heating of the housing due to breaks in thermocouple lines, is provided by a special thermocouple burn-out circuit in the instrumentation used in controlling the temperature of the housings.

Separate temperature control of the bearing housing and the lubricant sump is accomplished by separately regulating the power to each set of heaters by means of two separate electrical systems consisting of a proportioning-type controller.

Proportioning control of the heaters in the housing block is based on the deviation in temperature between that set on the controller and that measured by the thermocouple [TC1] located above the tester shaft. Similar control of the lubricant reservoir heaters is based on the deviation between the preset temperature and that measured by thermocouple [TC7] which monitors the sump wall temperature.

Additional control of heaters [H6] and [H7] and [H13] and [H14] is provided separately by means of variable transformers, which are capable of cutting down the voltage input to these sets of heaters. This manual reduction of supply voltage to those heaters is necessary since, due to the separate control of each housing, a thermal transition zone seems desirable to prevent controller hunting when the bearing housing and lubrication reservoir are at different temperatures. A thermocouple embedded in the housing block between heaters [H6] and [H7] continually monitors the temperature in this area so that adjustments can be made until equilibrium is maintained.

The temperatures of the lubricant in each sump, of the oil out of each test bearing, of the shaft liner and housing block are continuously recorded on multiple-point recycling-type temperature recorders. The bearing outer-race temperatures are monitored on either the IBM 1710 Logger Scanner system currently in use at the SKF Engineering and Research Center or a dual-channel continuous-point strip-chart recorder. Testing experience has shown that, after all temperatures reach equilibrium, the oil sump temperature can be controlled within ± 13°F, but at no more than 60°F lower than the test bearing temperature at 42,800 rpm test speed, and that each test bearing temperature can be controlled within ±10°F of its mean temperature. The heater control thermocouple is in the housing, so that the mean temperatures of the two test bearings are not controlled separately, but are usually observed to differ up to 17°F, although differences up to 80°F were observed in some early testing.

The testers are located in flash proof, heavy walled individual test cells for protection against operational hazards.

B. Cage Compatibility Tester

A modified rolling element fatigue tester which has a test head configuration as shown in Enclosure 14 was used to screen potential cage materials for wear characteristics under simulated high-temperature cage operation when lubricated with selected test lubricants. A photograph of this equipment is shown in Enclosure 15.

Three 1/2" balls are positioned by a cage specimen between two flat washer-shaped bearing rings. A load mechanism at the top of the test machine applies a vertical load on the balls through the drive shaft, on which the upper washer is mounted. The test cage rests on the lower stationary flat washer and is center about a sleeved guide post having about 0.015" eccentricity with respect to the axis of the rotating washer. Thus, the balls are loaded against the cage pockets by the variation in linear speed of the balls around the track and the tendency of the balls to creep radially outward. Tests are not run at high enough speed for the centrifugal force on the balls to be an appreciable fraction of the normal ball-to-washer load. The operating conditions selected for these tests were 1000 lbs. load 1200 rpm and temperatures of either 500 or 700°F.

A drawing of the test cage is shown in Enclosure 16. The pockets are tapered in order to develop a small downward component of the ball pocket load which will keep the cage firmly seated on the stationary flat washer. The cages have six pockets but only three (equally spaced) are used at a time for testing. In this way two tests can be run with each test cage.

The flat washers, balls, and guide sleeve are made of M-1 tool steel, so that all wear surfaces on the cage are in contact with materials likely to be encountered by cages in high-temperature bearings.

The entire test head is maintained at the desired ambient temperature (up to 800°F) by means of two 500-watt heaters which are controlled by an off-on type controller with a time proportioning device for minimizing variation at the set point. A thermocouple which contacts the side of the stationary flat washer (Enclosure 14) measures the control temperature for the heaters which is also temperature recorded in the data.

A nitrogen atmosphere is maintained in the oil reservoir and around the test washer, and a once-through bath lubrication system is used with the flow rate to the test specimen maintained by an oil drip cup so that the oil level in the test chamber was kept above the ball contacts on the lower washer. In most tests a gravity feed system was employed; however, a nitrogen-pressurized forced-feed system was used when lubricating with high-viscosity oils.

In the event of a ball or washer flaking failure, or other major malfunction, a vibration-sensitive switch mounted at the top of the test configuration automatically shuts the machine down.

RESULTS AND DISCUSSION

A. Cage Compatibility Tests

In order to evaluate the performance of candidate cage materials for resistance to wear under operating conditions simulating those in high-temperature ball bearings, the following test method was used with the cage compatibility tester.

The size of the elliptical wear scar area produced in the test cage pockets, due to contact with the ball under the action of the ball-to-cage forces, is used as the criterion for evaluating the performance of cage material-lubricant combinations. This is a realistic test because the wear depends on the wear-resistance of the ball-cage contact and on the magnitude of the ball-cage forces, and these forces are produced by the lubrication-dependent traction forces at the rolling ball-flat contacts. The wear scar area in the cage pockets can be accurately measured, whereas coking and metal transfer on the cage specimen makes weight measurements doubtful.

During each test, the tester was disassembled and the major and minor axis of the wear scar in each test cage pocket was measured at half-hour intervals (for calculation of the elliptical wear scar area), and the tests were terminated when stabilization of the wear scar (due to formation of a hydrodynamic film) or smearing in the cage pocket occurred. This result was usually produced within 2 hours' test time under the selected operating conditions of 1000 lbs. load and 1200 rpm of the flat washer set-up in the cage compatibility tester. Boundary lubrication conditions were most clearly evident in the early part of the test as metal-to-metal contact allowed the wear scar to increase progressively in size. Therefore, a good comparison of the cage materials tested under boundary lubrication conditions could usually be made at the end of one-half hour of testing.

Fifty different combinations of cage and lubricant materials were evaluated, and the tabulation in Enclosure 17 lists the average wear scar area measured with each combination at each test temperature after one-half hour running time. Tests were conducted at two temperature levels, 500°F and 700°F, to bracket the range of interest for this program. All 11 candidate cage materials were tested with Esso

FN-3157 highly refined mineral oil at 500°F and with Monsanto OS-124 polyphenyl ether at 700°F, and some materials were also tested with other oils, as tabulated in Enclosure 17.

M-1 and S-Monel cage materials were evaluated with a wide variety of selected lubricants at 500°F and 700°F, and the results of these tests are plotted in Enclosures 18 and 19, respectively, which show the change in wear scar area with running time.

Discussion of Cage Compatibility Results

As shown in Enclosure 17, the cage materials having the lowest wear rate at both 500°F and 700°F with FN-3157 oil and Monsanto OS-124 oil respectively were aged S-Monel (Rc 33) and hardened M-1 (Rc 60), based on the wear scar area after one-half hour running time. Fairly good wear performance was also obtained at both temperature levels with 440C modified stainless steel (with molybdenum for improved hot hardness) hardened to Rc 52 or 57. The polyimide plastic also had surprisingly low wear at 500°F making this material a promising candidate for bearing cages at least up to 500°F. At 700°F test temperature, however, which is the limiting temperature for this material quoted by the manufacturer DuPont, the polyimide cage wore quite heavily and cracked during the test.

In selected test combinations with other oils shown in Enclosure 17, the hardened M-1 (Rc 60) and the aged S-Monel (Rc 33) cage specimen showed the least wear of all cage materials tested with every oil. There was considerable variation in the wear performance of these materials in the different oils, however, as illustrated in Enclosure 18 and 19. At 500°F (Enclosure 18) the hydrocarbon oils, which generally have higher viscosity than the ester oils, reached an equilibrium wear scar area which was smaller than the equilibrium wear attained by the ester oils. The cage specimens wore at a smaller rate with the ester oils than with the hydrocarbons, however, (in some cases the wear scar area had not reached equilibrium at the end of the two hour test period), which reflects the superior boundary lubricating characteristics of the ester lubricants tested. In all tests conducted with the Kendall resin 0838 lubricant a heavy coating of hardened coke residue was observed on the test specimen. Since it was suspected that this coke residue was caused by insufficient protection of the oil from oxidation, several tests were repeated with the flow of nitrogen blanket gas increased approximately 10 to 1 (to greater than 18 SCFH), with no apparent effect on the amount of coke formation during the test. It is concluded, therefore, that this coking reflects the inherent thermal instability of the oil.

In the tests at 700°F with different oils (Enclosure 19), again the more viscous hydrocarbons seem to reach an equilibrium wear scar area in the two-hour test period, whereas the less viscous (at this temperature) polyphenyl ether had not reached an equilibrium wear scar area at the end of the two-hour period, although the cage wear rate in some cases was quite low (with the aged S-Monel cage). An attempt was made to improve the lubricating ability of the hydrocarbon oils FN-3157 and 0846 Bright Stock at 700°F by the addition of molybdenum disulphide powder. No reduction in the wear scar area was detected (in fact the wear increased with FN-3157). However, the MoS_2 did not stay in suspension in these preliminary tests and therefore these results (Enclosure 17) may not be a true evaluation of this principle of lubrication.

Another quite promising lubricant tested at 700°F was the fluorocarbon, DuPont PR-143. As shown in Enclosure 19, the wear of the hardened M-1 (Rc 60) cage specimen with this oil was among the lowest tested. A similar test with S-Monel exhibited high wear (see Enclosure 17), which may be the result of some corrosive attack on the silicon in this cage material (39). The extreme stability of this lubricant, even in an air environment, together with its good high-temperature viscosity (1.3 cs at 700°F) and low pour point (-20°F) make it a promising candidate for full-scale bearing testing. No bearing tests with DuPont PR-143 were conducted on this program, however, because of its corrosivity to the structural materials in the tester over the extended bearing test periods (39, 40).

As a result of the cage compatibility testing discussed here, hardened M-1 tool steel and S-Monel cages were made for bearing testing and were found to be more resistant to cage wear than the softer steel cages used in initial bearing testing. Extensive experience in jet engine bearing cages indicates that a light silver plate (0.001 - 0.002" thick) on steel cages operates very satisfactorily and therefore, as a further improvement, silver-plated hardened M-1 cages were made and tested in the 7205 bearings on this program and were found to have improved resistance to cage wear over the unplated cages.

B. Bearing Tests

1. Phase I Testing

Screening tests were conducted on 21 different bearing-lubricant combinations which included 134 bearings of CVM M-1 and CVM WB49 tool steel and 17 of the 18 high-temperature lubricants listed previously (DuPont PR-143 fluorocarbon was tested only in cage compatibility tests).

In all Phase I tests N_2 blanketing was utilized, but four Phase II tests were conducted with a polyphenyl ether lubricant in an air environment. Maximum oil flow to each bearing was maintained, which was in the range of 90 to 390 cc/min. at 20,000 rpm depending upon the viscosity of the oil at the test temperature. All the tests were started with the 1000cc sump supplying each bearing filled with test oil and make-up oil was added periodically throughout each test in order to replace that lost through decomposition, evaporation, and seal leakage.

In early stages of this program, testing of the bearings was conducted at 20,000 rpm and suspended after 30 hours (36×10^6 revs. at 20,000 rpm) in order to make a preliminary selection of the lubricants for endurance testing. It was the intent to determine whether the available lubricants would permit bearing operation at temperatures greater than 500°F and to develop the means of obtaining long enough bearing lives to permit life rating of bearing-lubricant combinations. Such ratings were to be obtained in Phase II of this program. As the number of lubricants under consideration narrowed down, the speed was increased to 42,800 rpm (selected as the most favorable in the range of 40,000 - 50,000 rpm from the standpoint of critical frequencies in the tester and drive system) and the time-up life was extended to 90 hours (231.1×10^6 revs. at 42,800 rpm) in preparation for Phase II testing. Cooling fans were employed during each test to dissipate the heat generated in the housing block at the high test speeds and loads and to provide satisfactory control of the bearing operating temperatures.

The thrust load applied to the bearings was 150 lbs. in the initial stages of testing and later increased to various values up to 918 lbs. Based upon a preliminary selection of lubricants, determination of proper position of the ball track in the ring grooves and desired radial looseness, a thrust load of 365 lbs., for which the AFBMA computed L_{10} = 480 mill. revs, was subsequently selected for most Phase I testing and initial Phase II tests. Later, the test load was increased again to 459 lbs. thrust (AFBMA L_{10} = 240 mill. revs.) when it was found in early Phase II testing that no primary fatigue flaking failures (not lubrication induced) were obtained at the lower load.

Low Speed Phase I Tests

The results obtained with CVM M-1 and CVM WB49 steel bearings at 20,000 rpm test speed are shown on composite plots of temperature vs. test life in Enclosures 20 and 21, respectively. In these enclosures, failed bearings are indicated by the dotted-line symbols and unfailed bearings by solid-line symbols, the shape of the symbol indicating the load and the number in each symbol representing the lubricant used, as listed in the legend on each enclosure. Bearings were designated as failed if they had undergone fatigue flaking, surface distress, or smearing to the extent that they were inoperable. Unfailed bearings were those in which little or no surface distress had taken place.

The bearings plotted in the region between 0-1 hour test life are those where the test was terminated very shortly after startup due to thermal instability or smearing of the bearings.

From Enclosures 20 and 21, the most promising high-temperature oil candidates, based on the bearing condition and life attained at the highest possible temperature, in the ester-base, hydrocarbon and polyphenyl ether categories respectively, were: Esso Turbo Oil 35, Socony Mobil XRM 109F and Monsanto OS-124 (the only polyphenyl ether candidate tested at this time). A close second choice in the ester-base class was Socony Mobil XRM 139A. The Kendall Bright Stock 0846 was also considered as a prime hydrocarbon candidate.

High Speed Phase I Tests

The test speed was later increased to 45,000 rpm and it was observed that the maximum temperature capabilities of the test lubricant candidates seemed to be somewhat higher at the higher speeds, so that the limiting temperature for each oil at high speed was established as 515°F for Esso Turbo Oil 35, 590°F for Socony Mobil XRM 109F and 500°F for Monsanto OS-124. Plots of test temperature versus test time at these higher test speeds for the M-1 and WB49 bearings are given in Enclosures 22 and 23, respectively, on which all Phase II tests are also plotted for comparison. As seen from these high-speed test plots, the number of failures that occurred very early in the tests was greater than at the lower test speed. These early failures were largely of a smearing type rather than a glazing or flaking-type failure that occurred after longer test times. The tendency of the high speed was to induce a larger number of early smearing failures, but once a bearing could be operated without smearing failure for a few hours, it would run to very long lives, so that there is a region of intermediate length of life in Enclosures 22 and 23 where there were very few bearing failures.

Various modifications of polyphenyl ether lubricants were tested, such as Monsanto Skylube 600, which exhibited the same maximum temperature capability of 500°F as OS-124 when tested at the same speed and load conditions and appeared to have satisfactory resistance to sludging which the unmodified OS-124 did not unless metallic copper sheets and screening were immersed in the oil as an inhibitor.

As a result of the above Phase I tests, conditions of initial Phase II endurance tests were set as 365 lbs. thrust load and 42,800 rpm. The following high-temperature oils were selected for Phase II testing with N_2 blanketing at their respective selected bearing operating temperatures.

a) Esso Turbo Oil 35 at 500°F
b) Socony Mobil XRM 109F at 600°F
c) Monsanto Skylube 600 at 600°F

The temperatures selected for Turbo Oil 35 and XRM 109F were based on the maximum safe bearing operating temperatures found in Phase I testing. It was decided to test Skylube 600 at 600°F, in the absence of even successful tests at this temperature, since this lubricant is not considered a promising candidate for any lower temperatures for other reasons (e.g. cost, pore point).

Further Phase I testing was conducted later with the newer polyphenyl ether lubricants, Monsanto MCS-293, MCS-353, and OS-138. The modified polyphenyl-type lubricant, MCS 293 and the more viscous version MCS 353 (both of which have lower pour points than Skylube 600), gave somewhat improved performance at temperatures of 500 to 600°F in a nitrogen atmosphere over the more conventional 5-ring polyphenyl ether, Skylube 600. In another series of tests (Phase II) conducted with Skylube 600 at 600°F in an air environment instead of a N_2 blanket as previously tested, 50% of the bearings survived for appreciable lives (76 to 189 mill. revs., compared to all bearings tested in N_2 at 600°F having run no more than 1.4 mill. revs. because of smearing failures) which reflected the improved lubrication conditions provided by an oxidizing atmosphere with this lubricant. The modified polyphenyl ethers MCS 293 and MCS 353, tested in an N_2 atmosphere, have about comparable performance to Skylube 600 tested in air. All polyphenyl ethers tested, however, caused lubrication failures at lives less than the calculated L_{10}.

In another late test, a new candidate ester-base lubricant, Sinclair Turbo S, was tested at high speed and found to have about comparable performance to the previously selected ester for endurance tests, Esso Turbo Oil 35.

Discussion of Phase I Results

During the course of the Phase I testing on this program, several different modes of bearing failure were recognized, as follows:

1) Surface distress in the ball tracks, which gave the tracks a glazed appearance shown in Enclosure 24b (compared to the normal condition of bearing tracks after running, Enclosure 24a), was observed with all lubricants tested. This glazing seemed to become more pronounced as the temperature was increased, developing under severe conditions into a superficial pitting of the surface in the glazed regions, shown in Enclosure 24c. Continued operation under glazing conditions resulted in flaking failure of the bearing, shown in Enclosure 24d, with the flakes having a depth comparable to the depth of the maximum Hertz shear stresses at the ball-race contacts (32), whereas the earlier surface pitting was much shallower (Enclosure 24c). This glazing phenomenon can be related to the elastohydrodynamic oil-film thickness at the ball-race contacts, as will be described.

2) Smearing was observed in the ball tracks involving gross metal transfer or galling of the ball-race surfaces as shown in Enclosure 25a. This is a distinctly different failure from the glazing described in (1). Smearing occurs when gross sliding exists at the ball-race contacts, as a result of unfavorable kinematic conditions in the bearing, to be described.

3) Cage wear shown in Enclosure 25d, compared to a tested cage showing no wear in Enclosure 25c, occurs when there is an unfavorable combination of lubricant, materials and geometry at the sliding contacts on the cage (on the inner-ring guide surfaces, as in Enclosure 25, or in the ball pocket).

4) Fatigue flaking in the bearing tracks, shown on the WB49 ball in Enclosure 25d, occurs at the end of the inherent fatigue life of the steel from which the bearing is made, but is a function of the lubrication conditions under which the bearing is operated (35-37).

The lubricants tested in Phase I of this program are listed in Enclosure 26 with the test speed and maximum temperature at which each lubricant can run with the 7205 test bearings under 365 lbs. thrust load. Where operation without surface distress per (1) above was achieved, the correspondent temperature is listed. Where all tests showed surface distress, the tests with the least degree of such damage are quoted. The condition of the ball-race tracks in bearings run under the listed conditions is noted for each lubricant, and on this basis, an estimated minimum viscosity for satisfactory bearing operation was determined for each lubricant. Using this "minimum safe viscosity", the pressure-viscosity characteristics of lubricants given in the literature (41), and the measured surface roughness of the inner ring grooves and balls in the test bearings, the estimated minimum safe ratio (h/σ) of the isothermal elastohydrodynamic (EHD) film thickness to the composite surface roughness for each lubricant was computed for the inner-ring-ball contacts in the test bearings as described in Appendix I and is also listed in Enclosure 26.

The calculated critical value of h/σ for tests in which the bearing remained serviceable without serious surface distress ranges as follows (Enclosure 26).

Critical h/σ		
hydrocarbons	$1.5 \sim 2$	
polyphenyl ethers	$\sim 0.75 \sim 1$	(uncertain, few good results)
esters	$0.5 \sim 1$	

It was shown in (30) that calculated film thickness for hydrocarbons tends to fall close to experimental values, but it comes out too low requiring a correction factor of $1.5 \sim 2$ for esters and too high requiring a correction factor of about 0.5 for the polyphenyl ether OS-124. No experimental values for other polyphenyl ethers are available. Thus, the real limiting value of h/σ for the test conditions is probably close to 1.5 for hydrocarbons and esters.

For OS-124 and Skylube 600 polyphenyl ethers, the real critical value of h/σ may well be of the order 1.5, but such a high value was never achieved in tests, which may explain why these fluids did not perform well. For the MCS 293 and 353 fluids, no experimentally supported critical value of h/σ can be given and the best estimate, from calculated film thickness only, is of the order of 1.

2. Phase II Tests

In all, 12 groups comprising 4 to 30 bearings each, of CVM M-1 and CVM WB49 steel bearings, including a total 192 bearings, were endurance tested in Phase II on the program. It was originally intended that 30-bearing groups would be tested at all times in order to obtain reliable bearing-life estimates, for rating lubricant-bearing material combinations. However, because of poor lives experienced with some of the bearing-lubricant combinations and a desire to test alternative combinations, only two of the groups were fully completed.

The bearings were endurance tested until failure or their time-up life of 90 hours (231.1×10^6 revs.) was reached, whichever occurred first. One group, in which no failures occurred within the 90-hour test period, was tested for periods up to 280 hours. Rules guiding the interpretation of failures were set-up and are given in Appendix III. Overall results of the endurance tests are tabulated on the next page and are discussed in detail in the following sections.

In this discussion, life estimates are given based on a combination of several failure modes. Where failures are all or in part, due to smearing, the estimated life has meaning only as an indication of the success or failure of that particular test series, but not of the inherent life of the bearing-lubricant combination. Of course, even in the absence of smearing, the life is that of the bearing-lubricant combination run at the test temperature, because surface distress often has reduced life. Only the few tests without surface distress reflect the inherent fatigue life of the bearing material.

Lubricant	Brg. Steel	Avg. Temp. °F	Thrust Load Lbs.	No. Brgs. Tested	No. Smearing and Flaking Failures	Brg. L_{10} Life, Mill. Revs. Calculated AFBMA	Brg. L_{10} Life, Mill. Revs. Est. From Smearing and Flaking Failures	Predominant Failure Mode
Hydrocarbons:								
Bright Stock	M-1	580	459	14	7	240	•	Smearing
Bright Stock with TCP	M-1	570	459	4	2	240	0.2	Smearing
XRM 109F-1	WB-49	585	459	8	4	240	3.1	Smearing
XRM 109F-1	M-1	580	459	18	6	240	1.3	Smearing
XRM 177F	WB-49	600	459	16	8	240	1.4	Smearing
XRM 177F	M-1	600	459	10	0	240	>500.	No Failure
Esters:								
Turbo Oil 35	WB-49	<300	459	8	4	240	0.02	Smearing
Turbo Oil 35	M-1	500	459	30	10	240	58.8	Flaking
Turbo Oil 35	M-1	500	365	30	2	480	247.7	Smearing
Polyphenyl Ethers:								
Skylube 600	M-1	595	459	22	11	240	•	Smearing
Skylube 600**	M-1	600	459	8	4	240	•	Smearing
Skylube 600	M-1	481	365	6	3	480	•	Smearing

• No valid life estimates were obtained since most failures occurred by smearing shortly after start-up.

** These tests were run in an air environment, all others with N_2 blanketing.

Test Results with Hydrocarbon Lubricants

CVM M-1 Bearings at 580°F with Kendall Bright Stock 0846 under 459 Lbs. Load (C/P = 6.2)

A group of 14 black-oxide coated CVM M-1 bearings (with improved groove surface finish on the inner rings only) were endurance tested using one of the most promising hydrocarbon candidates; Kendall Bright Stock 0846. Six bearings smeared at lives ranging from 0.13 to 54.9 mill. revs. and one bearing glazed and flaked at 201.8 mill. revs. Of the smearing failures, four occurred almost immediately after startup. No valid life estimates were obtained due to the wide variation in lives.

CVM M-1 Bearings at 570°F with Kendall Bright Stock 0846 (Containing TCP)

A group of four bearings (with improved groove finish on the inner ring only) was tested with the bright stock hydrocarbon lubricant containing 0.5% by weight of tricresylphosphate (TCP). Two bearings smeared at lives of 0.13 and 8.37 mill. revs., and one bearing was slightly glazed at 8.37 mill. revs. The resultant maximum likelihood L_{10} estimate was 0.2 mill. revs., indicating no significant improvement by the addition of TCP to the lubricant.

CVM M-1 Bearings at 580°F with Socony Mobil XRM 109F-1 under 459 Lbs. Load (C/P = 6.2)

Of the eighteen bearings tested, (improved finish only on inner rings), three suffered smearing failures at lives of 2.2 to 3.3 mill. revs., three were glazed and flaked at lives of 2.3 to 17.5 mill. revs., two reached their time-up life without failure and the remainder were treated as suspended tests at lives up to 17.5 mill. revs. The resultant maximum likelihood L_{10} estimate was 1.3 mill. revs.

CVM WB49 Bearings at 585°F with Socony Mobil XRM 109F-1 under 459 Lbs. Load (C/P = 6.2)

Since it was suspected that the very early smearing failures experienced in these high-speed tests may be related to the increased ball slip which might occur on the inner rings having improved finish, a group of CVM WB49 bearings having the standard aircraft quality finish on both rings were endurance tested for comparison. Of the eight bearings tested, three smeared after lives ranging from 0.13 to 106.5 mill. revs., two (one of which was a companion bearing to a smeared bearing) were glazed and/or pitted at lives of 27.5 to 70.9 mill. revs., and the remaining bearings were undamaged companion bearings treated as suspended tests. The maximum likelihood L_{10} estimate for these results was 3.1 mill. revs., which is not significantly different from the lives obtained in previous groups.

CVM M-1 Bearings at 600°F with Socony Mobil XRM 177F under 459 Lbs. Load (C/P = 6.2)

Socony Mobil XRM 177F lubricant, which is XRM 109F-2 containing a proprietary lubricity additive, was used to endurance test ten CVM M-1 bearings (with improved finish on inner-ring only). All ten bearings ran out to their time-up life of 90 hours (231.1 mill. revs.) without failure. Inasmuch as this was the best performance of any of the bearing-lubricant combinations tested thus far, all ten bearings were remounted and tested further under the same conditions. Again all ten bearings ran out until their accumulated time-up lives ranged from 437.0 to 726.5 mill. revs. with no signs of surface distress or failure. Enclosure 27 shows the excellent appearance of the longest-lived bearings of this group, which ran without failure to over 3 times the AFBMA computed L_{10} life under these test conditions. Coking of the XRM 177F lubricant occurred on the outside of the rig, but all internal nitrogen-blanketed parts were exceptionally clean and the filter screens (80 mesh) from the oil sumps, through which all the lubricant supplied to the test bearings was circulated, were remarkably free of decomposition products and other debris.

CVM WB49 Bearings at 600°F with Socony Mobil XRM 177F Under 459 Lbs. Load (C/P = 6.2)

Since the performance of the CVM M-1 bearings and XRM 177F was so favorable, a group of 16 CVM WB49 bearings (with improved finish on both races) were tested with the same lubricant under similar conditions. Eight bearings in eight tests smeared at lives ranging from 1.02 to 190.7 mill. revs. resulting in a maximum likelihood L_{10} estimate of only 1.4 mill. revs.

Test Results with Ester-Base Lubricants

CVM WB49 Bearings at 500°F (intended) with Turbo Oil 35 under 459 Lbs. Load (C/P = 6.2)

A group of eight CVM WB49 bearings (improved finish on both races) were tested with Esso Turbo Oil 35 lubricant at an intended temperature of 500°F. One bearing in each of the four tests smeared immediately after startup before reaching the desired test temperature.

CVM M-1 Bearings at 500°F with Turbo Oil 35 under 459 Lbs. Load (C/P = 6.2)

Ten of thirty bearings tested (improved finish on inner-ring only) developed smearing and/or flaking failures (8 glazed and/or flaked at lives ranging from 26.5 to 230.8 mill. revs., and 2 smeared at 49.6 and 77.8 mill. revs.), five of the remaining 20 bearings reached their time-up life without failure and 15 were treated as suspended tests (one flaked companion bearing to a flaking failure, one smearing failure attributed to coke deposits which accumulated over a weekend shutdown, one glazed companion bearing to a flaking failure, 7 slightly glazed bearings, and 5 bearings in good condition). There results give an L_{10} estimate of 58.8 mill. revs., which is 25% of the computed AFBMA L_{10} = 240 mill. revs. Considering all failure modes in these tests, there were a total of 13 failures, resulting in an L_{10} estimate of 47.7 mill. revs. Hard coke deposits were experienced in these tests which were cooled down and restarted over a weekend during the course of testing.

CVM M-1 Bearings at 500°F with Turbo Oil 35 Under 365 Lbs. Load (C/P = 6.2)

A full complement of 30 bearings (improved finish inner rings) were tested and only two of these resulted in smearing and flaking failures. Considering all types of failures, i.e., those induced by cage wear as well as hard coke deposits, there were a total of eight failures. Of the remaining bearings, 14 ran until their time-up life and 14 were treated as suspended tests in accordance with the failure evaluation rules. Hard coke deposits were noted in all test bearings especially in those that were stopped, cooled down, and restarted over a weekend during the course of testing. It is believed that these deposits may have precipitated some failures. The maximum likelihood L_{10} estimate based on smearing and flaking failures was 247.7 mill. revs. which is 51.6% of the computed AFBMA L_{10} life of 480 mill. revs.. With all types of failures considered, the maximum likelihood L_{10} estimate is 116.9 mill. revs., which is 24.4% of the computed AFBMA L_{10} life.

Inasmuch as surface distress and cage wear in these bearings were predominant over fatigue failures, it was concluded that the 42,800 rpm, 365 lbs. load and 500°F test condition is on the edge of the safe operation range for this bearing-lubricant material combination.

Polyphenyl Ether Lubricants

CVM M-1 Bearings at 595°F with Skylube 600 Under 459 Lbs. Load (C/P = 6.2)

Endurance testing of a group of 22 bearings (improved inner-ring finish only) conducted with the five-ring polyphenyl ether lubricant in the standard N_2 atmosphere indicated that lubrication-related surface distress occurred within several minutes after startup. Of the eleven smearing and flaking failures, ten (five tests) occurred within 5.4 mill. revs. (6 bearings smeared at 0.15 to 0.26 mill. revs. and 5 glazed and flaked at 1.03 to 98.6 mill. revs.), making it impossible to obtain a valid L_{10} life estimate.

CVM M-1 Bearings at 600°F with Skylube 600 Under 459 Lbs. Load (In Air)

Since N_2 blanketing of the tester for reduction of oxidative decomposition of the lubricants is not required with Skylube 600, which has inherent high-temperature oxidative stability, and since oxide films formed from an oxidizing atmosphere may improve the boundary lubricating properties of an oil, four additional tests (8 bearings having improved finish on inner rings only) were conducted under the same conditions as the previous group, except that air replaced the N_2 atmosphere. Of these eight bearings, two smeared at lives of 0.8 and 0.5 mill. revs., one smeared at a life of 189.3 mill. revs., one glazed and flaked at 76.3 mill. revs., while their companion bearings were in good condition. These results indicate a significant improvement over tests run with the same bearing-lubricant combination with N_2 blanketing.

CVM M-1 Bearings at 481°F with Skylube 600 Under 365 Lbs. Load (C/P = 7.8)

Because of the short lives experienced with this lubricant, three tests (6 bearings) were conducted in an N_2 atmosphere with reduced thrust load. Here again extremely short lives were obtained when three bearings smeared at lives of 0.03 to 1.36 mill. revs. and one companion bearing was slightly glazed at 1.36 mill. revs.

Discussion of Phase II Test Results

The smearing performance of the test bearings at 600°F depends on the surface geometry characteristics of the bearings. XRM 177F oil worked well with M-1 bearings having improved finish on the inner rings only, but poorly with WB49 bearings, which have higher hardness and should resist smearing better than M-1 steel, but had improved finish on both the inner and outer rings. Since the chromium content of these two bearing steels is similar, one would not expect a significant difference in oxide film formation on the surfaces affecting boundary lubricating properties.

In order to explore surface roughness effects more thoroughly, an analysis of the kinematic conditions existing at the ball-race contacts was performed for these bearings under the imposed test conditions, using a high-speed digital computer. The method is described in Appendix I. Under these operating conditions with the bearing design used in Phase II testing (and illustrated in Enclosures 3 and 4), and assuming a Coulomb-type coefficient of friction of 0.06 at the ball-race contacts, (the spinning torque and the spinning heat generation are proportional to the coefficient of friction assumed), the operating parameters given in Enclosure 7 were computed. In addition, the ball spinning torques at both inner and outer races were computed as follows, for 459 lbs. thrust load at 42,800 rpm shaft speed.

	Present 19° Contact Angle Bearing	Modified Design
Ball spin torque on inner, in-lbs.	0.081	0.076
Outer ball spin torque, projected on inner, in-lb..	0.097	0.115

Also given in the above tabulation and in Enclosure 7 are the computed operating parameters for a modified 7205 bearing with all design and operating conditions identical to those given above except that the ball-race conformity on the inner ring (the inner-ring groove radius expressed in percent of the ball diameter) is changed from 52.2% to 53% and the conformity on the outer ring is changed from 53.2% to 52.3%. It is seen that the inner-ring spin torque decreases and the outer-ring spin torque increases by this modification.

In order for ball control to exist on the outer race, i.e. for "pure" rolling to occur at the outer-race-ball contacts and all spinning to occur at the inner-race contacts, the ball spinning torque on the inner ring must be less than that at the outer ring contact. If it is assumed that the coefficient of sliding friction is the same at both race contacts, then this criterion for outer-race ball control is satisfied for the standard bearing design under the above operating conditions. However, one might expect that the sliding friction at the ball-race contacts will depend on the surface roughness, the lubrication conditions, and the amount of sliding taking place, with its attendant temperature rise in the contact area. According to the above analysis, with the present design bearing, if the coefficient of sliding friction on the inner ring is more than 17% greater than on the outer ring, then the criterion for outer-ring control is no longer satisfied and gross sliding of the ball would occur. Also the outer-race contact must absorb the traction forces generated by the gyroscopic moment on the ball which would tend to make it even less able to prevent gross ball sliding.

The success of the M-1 bearings tested at 600°F with XRM 177F oil is probably attributable to the rougher surface finish on the outer races than on the inner races, which would improve outer-race control. The WB49 bearings having the smoother finish on both races probably do not generate sufficient spin friction torque on the outer ring to insure outer-race control so the balls spin or slide excessively on both rings (probably stabilized by gyroscopic forces instead of by race contact forces), thus making it more difficult for the lubricant to prevent surface distress and smearing. It is undesirable to design bearings having different surface roughness on the inner and outer rings, since the roughness in the ball tracks on both rings probably tend to equalize with extended running due to run-in phenomena. The modified design described above should provide a greater margin of outer-race control by an adjustment of the ball-race conformities on both the inner and outer rings and is expected therefore to perform better when both races are of the same roughness.

CONCLUSIONS

1. 7205-size ball bearings made from M-1 tool steel of Rc 62 minimum hardness can be operated satisfactorily at 600°F, 42,800 rpm, and a nominal maximum Hertz stress of 250,000 psi with Socony Mobil XRM 177F as the lubricant under an inert blanket. All of 10 bearings run in this manner attained twice the AFBMA L_{10} catalog life or more.

2. For successful high-speed high-temperature operation under thrust load, ball bearings must be designed with low surface roughness and accurate groove geometry to prevent surface distress (glazing) with available lubricants. It is conjectured that ball-groove conformities must be selected to produce substantially greater ball spin torque on the outer ring than on the inner ring during operation. This geometry will assure outer-race control and prevent gross skidding and smearing of the ball tracks or balls.

3. Ester-base lubricants, such as Esso Turbo Oil 35 and Sinclair Turbo S, lubricate quite satisfactorily at temperatures up to 500°F in a nitrogen-blanketed system using 7205-size bearings at 42,800 rpm speed and a nominal maximum Hertz stress of 250,000 psi.

4. All polyphenyl ethers tested in the nitrogen atmosphere caused lubrication failures at 600°F or lower, at lives less than the calculated L_{10} in these tests. A maximum bearing temperature of 550°F may be feasible for successful lubrication with two modified polyphenyl ether lubricants, Monsanto MCS 293 and MCS 353, as well as with the 5-ring polyphenyl ether, Monsanto Skylube 600, (the latter in an air environment).

5. Short-term cage screening tests with DuPont PR-143 fluorocarbon at 700°F in nitrogen indicate promising high-temperature performance of this lubricant.

6. Cage material screening tests showed that hardened M-1 tool steel and aged S-Monel gave the lowest wear of the (unplated) materials tested at 500°F and 700°F with several of the candidate high-temperature lubricants. Silver-plated hardened tool steel cages having wider than standard guide lands were found best and operated with negligible wear in all bearing tests in which no lubrication distress occurred at the ball-race contacts.

7. Three modes of lubrication-type failures that can occasionally occur prior to "classical" fatigue flaking were identified in this bearing testing as follows:

a. Surface distress (glazing) can occur in the ball tracks (and can result in subsequent early flaking failure) if there is insufficient elastohydrodynamic lubricant film thickness at the ball-race contacts (film thickness/roughness ratio less than about 1.5). This failure mode is eliminated by: (1) increasing the viscosity of the lubricant at the bearing operating temperature, or (2) improving the surface finish at the critical ball-race contacts, or, (3) increasing speed without increasing bearing temperature.

b. Gross wear of the cage occurs if the cage sliding contacts are inadequately lubricated. This in turn can cause cage failure or an acceleration of ball-race glazing. This failure mode can be avoided in cases of marginal lubrication by changes in the cage material and design.

c. Smearing or gross metal transfer, distinct from the glazing described in (a), can occur at ball-race contacts, caused by severe sliding which may arise in thrust-loaded ball bearings due to unfavorable kinematic conditions. The smearing can be eliminated in cases of marginal lubrication either by improving the boundary lubricating properties of the lubricant or by reducing the amount of sliding which occurs at the ball-race contacts in the bearing by appropriate modifications in the bearing design.

REFERENCES

1. G. K. Bhat, A. E. Nehrenberg, "A Study of the Metallurgical Properties that are Necessary for Satisfactory Bearing Performance and the Development of Improved Bearing Alloys for Service up to 1000°F", WADC TR 57-343, Astia Document No. 142117, November, 1957.

2. G. Steven, T. V. Philip, "Development of a Corrosion-Resistant Bearing Steel for Service in Aircraft at Temperatures up to 1000°F", WADC TR 59-390.

3. H. R. Bear, R. H. Butler, "Preliminary Metallographic Studies of Ball Fatigue Under Rolling-Contact Conditions", NACA TN 3925, March, 1957, 27 p.

4. T. L. Carter, "Preliminary Studies of Rolling-Contact Fatigue Life of High Temperature Bearing Materials", NACA RM E57K12, April, 1958, 27 p.

5. T. L. Carter, "Effect of Lubricant Base Stock on Rolling-Contact Fatigue Life", NACA TN 4161, February, 1958, 28 p.

6. T. L. Carter, "Effect of Lubricant Base Stock on Rolling-Contact Fatigue Life With Liquid and Dry Powder Lubricants", NACA TN 4163, January 1958, 40 p.

7. W. J. Anderson, "Performance of 110-Millimeter-Bore M-1 Tool Steel Ball Bearings at High Speeds, Load, and Temperatures", NACA TN 3892, January, 1957, 38 p.

8. T. W. Morrison, H. O. Walp, R. P. Remorenko, "Materials in Rolling Element Bearings for Normal and Elevated (450°F) Temperature", published in ASLE Transactions, Vol. 2, No. 1, 1959.

9. T. L. Carter, "Effect of Lubricant Viscosity on Rolling-Contact Fatigue Life", NACA TN 4101, October 1957, 25 p.

10. T. L. Carter, E. V. Zaretsky, W. J. Anderson, "Effect of Hardness and Other Mechanical Properties on Rolling-Contact Fatigue Life of Four High Temperature Bearing Steels", NASA TR D-270, March 1960, 51 p.

11. C. L. Mahoney, "Nuclear Radiation Resistant High Temperature Lubricants", WADC TR 59-173, July, 1959, 123 p.

12. E. E. Klaus, M. R. Fenske, "High Temperature Lubricant Studies", WADC TR 56-224, Astia Document No. AS97291, September, 1956.

13. E. E. Klaus, M. R. Fenske, "Fluids, Lubricants, Fuels and Related Materials", WADD TR 60-989, March, 1965.

14. J. B. Accinelli, S. J. Beaubien, "Fundamentals of High Temperature Bearing Lubrication", Contracts No. NOas 58-6141-C, NOw-60-0270-C, and NOw-62-0239-C.

15. J. B. Accinelli, and S. J. Beaubien, "Bearing Lubrication Under Severe Conditions", S-13918, Shell Development Company (1964).

16. J. H. Gustafson, "High Temperature Oil Evaluation in Full-Scale Bearing Fatigue Tests", ASLE Paper No. 63AM2A-3 (1963).

17. A. Palmgren, Ball and Roller Bearing Engineering, SKF Industries, Inc., Philadelphia, Pa. (1959).

18. L. Haworth, "Ball Bearing Cage Bore Wear", Machinery LLoyd (1951).

19. A. B. Jones, "Ball Motions and Sliding Friction in Ball Bearings", Technical Report to Navy Department, Bureau of Ships, Nobs 72203 (circa 1954). See Also "A General Theory for Elastically Constrained Ball and Radial Roller Bearings under Arbitrary Load and Speed Conditions", ASME Jrl. of Basic Engrg., pgs. 309-320 (1960).

20. A. J. Shashati, "A Theoretical Investigation of the Angular Contact Ball Bearing", ScD Thesis, Mechanical Engineering, MIT (1958).

21. F. Macks, Z. Nemeth, and W. J. Anderson, "Preliminary Investigation of Molybdenum Disulfide-Air-Mist Lubrication for Roller Bearings Operating to DN Values of 1 x 10^6 and Ball Bearings Operating to Temperatures of 1000°F", NASA RME 51G31 (1951).

22. R. Shevchenko, and P. Bolan, "A Visual Study of Ball Motion in a High-Speed Thrust Bearing", SAE Paper (January 1957).

23. J. C. Lawrence, and W. E. Schmidt, "Radioisotope Techniques Applied to the Investigation of Surface Contact in Rolling Element Bearings", ASLE Paper No. 62AM 6B-1 (1962).

24. R. Remorenko, "Informal Discussion of Radiotracer Wear Studies of Ball Bearings", presented at ASLE-ASME Lubrication Conference (1962).

25. N. E. Sindlinger, J. A. Martin, D. F. Huttenlocher, "Progress Report Nos. 1-11 and First Summary Report on Influence of Lubrication on Rolling Contacts", U. S. Navy Contract No. NOw-61-0716-C.

26. E. V. Zaretsky, L. B. Sibley, and W. J. Anderson, "The Role of Elastohydrodynamic Lubrication in Rolling-Contact Fatigue, Prepared for ASME-ASLE Lubrication Conference, October, 1962.

27. A. W. Crook, "Some Physical Aspects of Lubrication and Wear", Contemp. Physics, 257-271 (1962).

28. L. B. Sibley, and F. K. Orcutt, "Elastohydrodynamic Lubrication of Rolling Contact Surfaces", ASLE Trans. 4, (2), November, 1961.

29. T. Tallian, Y. P. Chiu, et al, "Lubricant Films in Rolling Contact of Rough Surfaces", ASLE Trans., 7, (2) November, 1961.

30. T. Tallian, E. F. Brady, J. I. McCool, and L. B. Sibley, "Lubricant Film Thickness and Wear in Rolling Point Contact" ASLE Preprint No. 65AM 4A4 (1965). See also "Partial Elastohydrodynamic Lubrication in Rolling Contact", Paper No. 14, Symposium on Elastohydrodynamic Lubrication, Inst. Mech. Engrs., London (1965).

31. AFBMA Standards, "Method of Evaluating Load Ratings for Ball Bearings", Section No. 9, the Anti-Friction Bearing Manufacturers Association, Inc., 60 East 42nd Street, New York 17, N. J.

32. G. Lundberg, and A. Palmgren, "Dynamic Capacity of Rolling Bearings", Acta Polytechnica, Mech. Engr., 7 (1947); See also "Dynamic Capacity of Roller Bearings", Acta Polytechnica, Mech. Engr., 2 (1952).

33. H. O. Walp, R. P. Remorenko, and J. V. Porter, "Endurance Tests of Rolling Contact Bearings of Conventional and High Temperature Steels Under Conditions Simulating Aircraft Gas Turbine Applications", WADC Technical Report 58-392, July, 1959.

34. T. V. Philip, A. E. Nehrenburg, and G. Steven, "A Study of Metallurgical Properties that are Necessary for Satisfactory Bearing Performance and the Development of Improved Bearing Alloys for Service up to 1000°F", WADC TR-373, Part II, October, 1958.

35. T. W. Morrison, T. Tallian, H. O. Walp, G. H. Baile, "The Effect of Material Variables on the Fatigue Life of AISI 52100 Steel Ball Bearings", ASLE Trans., 5 (2), 1962.

36. T. W. Morrison, T. Tallian, H. O. Walp, G. H. Baile, "The Effect of Trace-Element Content and Multiple Vacuum Melting on the Fatigue Life of Ball Bearings made from AISI 52100 Steel" Inst. of Mech. Engrs., Session 3, Paper 7.

37. R. R. Valori, T. Tallian, and L. B. Sibley, "Elastohydrodynamic Film Effects on the Load-Life Behavior of Rolling Contacts", (to be published).

38. A. B. Jones, Mechanical Design and Systems Handbook, Section 13, McGraw-Hill, New York.

39. W. H. Gumprecht, A New Class of High Temperature Fluids - PR-143, Research and Development Division Contribution No. 381 Organic Chemicals Department - Jackson Laboratory, E. I. du Pont de Nemours and Company, Wilmington, Delaware.

40. AFBMA Standards, "Method of Evaluating Load Ratings for Ball Bearings", Section No. 9, The Anti-Friction Bearing Manufacturers Association, Inc., 60 East 42nd St., New York 17, N.Y.

41. The American Society of Mechanical Engineers Pressure-Viscosity Report, Volumes I and II.

42. Lieblein, J., Zelen, M., "Statistical Investigation of the Fatigue Life of Deep Groove Ball Bearings," Journal of Research, National Bureau of Standards, Vol. 57, No. 5 (1956).

43. Tallian, T., "Weibull Distribution of Rolling Contact Fatigue Life and Deviations Therefrom", ASLE Transactions, Vol. 5, 1, April (1962).

44. McCool, J., Tallian, T., "Estimating Weibull Parameters From Censored Data by Maximum Likelihood" - To be published.

APPENDIX I

TEST BEARING ANALYSES

The test bearings shown in Enclosures 1, 3, and 4, and the proposed modified design described in Enclosure 7, have the following mean design parameters and typical operating conditions.

		Encl. 1	Encl. 3 & 4	Proposed Modified Design
Outer ring groove conformity	f_o	= 0.532	0.532	0.523
Inner ring groove conformity	f_i	= 0.522	0.522	0.530
Ball diameter, in.	D_a	= 0.3125	0.3125	0.3175
No. of balls	Z	= 12	12	12
Radial looseness (mounted), in.	P_d	= 0.0007	0.0011	0.0011
Pitch diameter, in.	d_m	= 1.537	1.537	1.537
Inner ring speed, rpm	n_i	= 20,000	20,000-42,800	42,800
Thrust load, lbs.	F_a	= 365	365-459	459

1. AFBMA Computed Life

According to standard AFBMA load rating practices (31), the basic dynamic capacity **C** of a ball bearing, defined as the radial load under which 90% of all bearings made of standard 52100 steel will survive one million revolutions of the inner ring without fatigue flaking of the rolling surfaces (32), is given by the following formula:

$$C = f_c \left(i \cos \alpha \right)^{0.7} Z^{\frac{2}{3}} D_a^{1.8} \qquad [1-1]$$

where i = No. of rows of balls (in this case, one)
f_c = a material and geometry factor which is derived in (32) and tabulated in Table 11 of (31) as a function of

$$\gamma = \frac{D_a \cos \alpha}{d_m} \qquad [1-2]$$

APPENDIX I (Cont.)

for which $f_c/f = 0.611$ in the present case (19° bearing shown in Enclosure 3 and 4).

where f = a material factor found from tests to be 7450 when inch and pound units are used.

Therefore $C = (0.611)(7450)(\cos 19°)^{0.7}(12)^{\frac{2}{3}}(0.3125)^{1.8}$

$= 2855$ lbs.

Table 1.2 of (31) gives the fraction Y of the thrust load F_a equal to the equivalent radial load P

$$Y = 1.0$$

so that the ratio of the capacity to the equivalent load for 459 lbs. thrust load

$$\frac{C}{P} = \frac{2855}{459} = 6.22$$

Using the inverse cube load - life law from (32) and (31), the AFBMA computed L_{10} life for which 90% of all bearings in the present case are expected to endure under the specified operating conditions is

$$L_{10} = \left(\frac{C}{P}\right)^3 = (6.22)^3 = 240 \text{ million revolutions } [1\text{-}3]$$

$$= \frac{(240)\,10^6}{60\,(42{,}800)} = 93.5 \text{ hours at } 42{,}800 \text{ rpm.}$$

2. Ball Forces and Motions in the Test Bearing During High - Speed Operation

As the speed of an angular-contact ball bearing is increased, the centrifugal force on the balls increases until it becomes

APPENDIX I (Cont.)

a significant portion of the ball loads from external bearing loads, at which time the foregoing calculation methods must be altered in order to reflect adequately the actual operating characteristics of the bearing. Further increases in speed result in the generation of significant gyroscopic moments on the balls which then must also be considered.

Cage Speed

The ball centrifugal force is dependent upon the cage speed. By assuming that the balls do not slip on the races, and that the ball-race contact angle on both rings is equal to the nominal operating design contact angle α, the following formula can be derived (17) as a first approximation of the cage speed.

$$n_m = \frac{1}{2} n_i (1 - \gamma) \qquad [1\text{-}4]$$

where $$\gamma = \frac{D_a \cos \alpha}{d_m}$$

The Hertzian deflections at the ball-race contacts when the thrust load is applied to the bearing cause an increase in the contact angle. Also, the centrifugal force on the balls causes the contact angle on the outer race to be smaller than that on the inner race to maintain an equilibrium of forces on the balls. Later in the analysis, these contact angles will be computed and a corrected cage speed will then be calculated by assuming that ball control exists at the outer raceway contacts (i.e., rolling with no spinning exists at the outer race contacts), using the following relationship taken from (19).

$$n_m = n_i \left[\frac{1 - \frac{D_a}{d_m} \cos \alpha_i}{1 + \cos(\alpha_i - \alpha_o)} \right] \qquad [1\text{-}5]$$

APPENDIX I (Cont.)

Contact Angle Divergence From Ball Centrifugal Force

When a ball bearing operates under thrust load at high speed, the contact angle α_i between the inner ring and ball and α_o between the outer ring and ball differ due to the increase of the ball load on the outer ring caused by the centrifugal force of the ball.

The governing equations for determining the two unknown contact angles α_o and α_i were derived from the conditions of compatibility and force equilibrium (38). They are:

$$\left[(f_i - \tfrac{1}{2})D_a + \delta_i\right]\cos\alpha_i + \left[(f_o - \tfrac{1}{2})D_a + \delta_o\right]\cos\alpha_o = (f_i + f_o - 1)D_a\cos\alpha \quad [1-6]$$

$$\frac{F_a}{Z}\left[\cot\alpha_o - \cot\alpha_i\right] = F_c \quad [1-7]$$

where the centrifugal force F_c and Hertzian deformations δ_o, δ_i are given as follows:

$$F_c = \frac{W}{g}\frac{d_m}{2}\left(\frac{2\pi}{60}\right)^2\left(1 - \frac{D_a}{d_m}\cos\alpha_i\right)^2\left[1 + \cos(\alpha_i - \alpha_o)\right]^{-2} n_i^2 \quad [1-8]$$

$$\delta_i = C_i\left(\frac{F_a}{Z\sin\alpha_i}\right)^{\frac{2}{3}} \quad [1-9]$$

$$\delta_o = C_o\left(\frac{F_a}{Z\sin\alpha_o}\right)^{\frac{2}{3}} \quad [1-10]$$

(C_i and C_o are Hertzian constants which also depend on the contact angle, W is the weight of one steel ball in the bearing and g is the acceleration of gravity) and the gyroscopic moment on the ball is derived later in this Appendix.

APPENDIX I (Cont.)

Equations [1-6] and [1-7] are nonlinear algebraic equations which cannot be solved in closed form. Using the Newton-Raphson iteration method with a high-speed digital computer, solutions for the inner and outer race contact angles α_i and α_o are obtained quickly for any specific case.

The corresponding ball forces Q_i and Q_o can be calculated by the following relations:

$$Q_i = \frac{F_a}{Z \sin \alpha_i} \qquad [1\text{-}11]$$

$$Q_o = \frac{F_a}{Z \sin \alpha_o} \qquad [1\text{-}12]$$

Ball Spin Control

The balls in an angular-contact bearing must spin as well as roll on one or the other of the races. Spinning is defined as the component of the ball motion represented by rotation about an axis drawn from the ball center to the center of the race contact. Ball spin will occur on that raceway which has the smaller frictional resistance to the spinning moment. In most high-speed ball bearings, ball spin occurs at the inner-race contact, due to the high centrifugal loading on the outer race. This condition is known as "outer-race control", and it will occur for operating conditions which conform to the following inequality (38).

$$Q_o\, a_o\, \mathcal{E}_o \cos(\alpha_i - \alpha_o) > Q_i\, a_i\, \mathcal{E}_i \qquad [1\text{-}13]$$

where a_o, a_i = the major semi-axes of the outer and inner race contact areas, respectively.

$\mathcal{E}_o, \mathcal{E}_i$ = complete elliptical integrals of the second kind with a modulus of the ratio between the major and minor axes of the contact areas on the outer and inner rings, respectively.

The spinning torque at the ball-inner-race contact M_s is

$M_s = 3/8\ \mu_s\, Q_i\, a_i\, \mathcal{E}_i$

where μ_s = the coefficient of sliding friction at the contact

APPENDIX I (Cont.)

The spin-to-roll ratio on the inner race is then computed according to the following formula derived from geometric considerations (38).

$$\left(\frac{\omega_s}{\omega_R}\right)_i = \left(1 - \frac{D_a}{d_m}\cos\alpha_i\right)\tan(\alpha_i - \beta) + \frac{D_a}{d_m}\sin\alpha_i \quad [1\text{-}14]$$

where $$\tan\beta = \frac{\sin\alpha_o}{\cos\alpha_o + \frac{D_a}{d_m}} \quad [1\text{-}15]$$

3. Bearing Life Estimate at High Speed

From (32), the basic dynamic capacity Q_c of the inner-ring or outer-ring contacts in a ball bearing, taken separately and assuming standard 52100 bearing steel and normal lubrication conditions, is:

$$Q_{c_{i,o}} = 7450\left[\frac{2f_{i,o}}{2f_{i,o}-1}\right]^{0.41}\frac{(1 \mp \gamma_{i,o})^{1.39}}{(1 \pm \gamma_{i,o})^{\frac{1}{3}}}\left(\frac{D_a}{d_m}\right)^{0.3} D_a^{1.8} Z^{-\frac{1}{3}} \quad [1\text{-}16]$$

where the upper signs refer to the inner-race contacts and the lower signs refer to the outer race. For the inner-race contacts, since considerable sliding occurs there due to the ball spin, the capacity is modified by a factor λ_p which accounts for the reduction in fatigue life due to this spin (32, 38).

$$\lambda_p = 1 - \frac{1}{3}\sin\alpha_i \quad [1\text{-}17]$$

From Eq. [1-3]

$$L_{10_o} = \left(\frac{Q_{c_o}}{Q_o}\right)^3$$

$$L_{10_i} = \left(\frac{Q_{c_i}}{Q_i}\right)^3$$

and the life of the bearing, from (32)

$$L_{10} = \frac{1}{\left(\frac{1}{L_{10_i}^{1.11}} + \frac{1}{L_{10_o}^{1.11}}\right)^{0.9}}$$

4. Gyroscopic Forces on the Balls

As the speed of an angular-contact ball bearing increases, the balls rotate around their own axes at very high speed. Since the direction of each ball axis is forced to change as the ball orbits around the bearing, there is a gyroscope moment generated as a result of this moment applied to the ball angular velocity vector.

Assuming outer-race control and steel balls rotating about their own axes at speed n_a, and about the bearing axis at speed n_m, the gyroscopic moment M_G can be derived as follows (38):

$$M_G = I \, n_a n_m \sin\beta \qquad [1\text{-}18]$$

where $I = \dfrac{\pi \rho D_a^5}{60\, g}$ and ρ is the density of steel

$$n_a = n_i \Big/ \left[\left(\frac{\cos\alpha_o + \tan\beta \sin\alpha_o}{1 + \frac{D_a}{d_m}\cos\alpha_o} + \frac{\cos\alpha_i + \tan\beta \sin\alpha_i}{1 - \frac{D_a}{d_m}\cos\alpha_i} \right) \frac{D_a \cos\beta}{d_m} \right] \qquad [1\text{-}19]$$

and $$\tan\beta = \frac{\sin\alpha_o}{\cos\alpha_o + \frac{D_a}{d_m}} \qquad [1\text{-}15]$$

and n_m is given by Eq. [1-5].

This moment is resisted by the tangential tractive forces at the ball-race contacts, which require a coefficient of sliding friction μ_c at the ball-race contacts to prevent skidding of the balls

$$\mu_c = \frac{2 M_G}{D_a Q_o} \qquad [1\text{-}20]$$

5. Elastohydrodynamic Oil Film Thickness

From the isothermal theory of elastohydrodynamic (EHD) lubrication reviewed in (25), the following formula for the minimum oil-film thickness h at the critical inner-race contacts in a ball bearing can be derived.

$$h = G \left(\mu_o \alpha \, n_i \right)^{0.727} Q_{max}^{-0.06} \qquad [1\text{-}21]$$

APPENDIX I (Cont.)

where μ_o = dynamic viscosity of the oil at the bearing operating temperature

α = pressure coefficient of viscosity

Q_{max} = maximum ball load

Here G = a bearing geometry parameter computed for steel bearings by the formula below

$$G = 8.92 \left(R_i \frac{D_a}{d_m}\right)^{1.09} \left(\frac{2 R_o}{D_a}\right)^{0.727} e_a^{\,0.09} \left(\Sigma\rho\right)^{-0.03} \qquad [1-22]$$

where $R_i = \frac{d_m}{2} - \frac{D_a}{2} \cos\alpha$

$R_o = \frac{d_m}{2} + \frac{D_a}{2} \cos\alpha$

$\Sigma\rho$ = curvature sum at the inner-race contacts

e_a = a Hertzian variable given in (17).

The composite surface roughness σ_h of the inner-race contacts is

$$\sigma_h = \sqrt{\sigma_i^2 + \sigma_b^2} \qquad [1-23]$$

where σ_i = rms roughness of the inner-ring groove

σ_b = rms roughness of the balls

6. Heat Generation Calculations

The general expression for the heat loss H_l due to load and hysteresis phenomena in ball bearings (17) may be defined as follows:

APPENDIX I (Cont.)

$$H_l = 0.04026\, f_1\, F_\beta\, d_m\, n_i \qquad [1\text{-}24]$$

In equation [1-24] the quantity f_1 is given by the following relationship:

$$f_1 = 0.001 \left[\frac{F_o}{C_o}\right]^{0.33} \qquad [1\text{-}25]$$

where

$$F_o = Y_s\, F_a \qquad [1\text{-}26]$$

The magnitude of Y_s as a function of the contact angle is given in (17).

F_β in equation [1-24] is defined as follows:

$$F_\beta = 0.9\, F_a \cot\alpha \qquad [1\text{-}27]$$

where α in equation [1-27] is the nominal value.

The general expression for the loss H_V due to viscous considerations is defined as follows: (17)

$$H_V = 0.57376\, f_o\, \nu_o^{\frac{2}{3}}\, n_i^{\frac{5}{3}}\, d_m^3 \qquad [1\text{-}28]$$

Where f_o is a correction factor depending on the type of lubrication; see reference (17).

The spinning heat generation H_S at the inner raceway may be defined by the following relationship:

$$H_S = \frac{Z}{J}\left[\frac{3\mu\, Q\, a\, \mathcal{E}}{8\pi}\right]\omega_s \times 3600 \qquad [1\text{-}29]$$

ω_s, the spinning speed at the non-controlling raceway (inner race) is defined previously in this Appendix.

The total heat generation H_T is defined as the sum

APPENDIX I (Cont.)

of equations [1-24], [1-28], and [1-29] and is given as follows:

$$H_T = H_\ell + H_V + H_S \quad \text{in Btu/hr} \qquad [1\text{-}30]$$

The foregoing analysis, together with standard Hertz stress calculations, has been programmed on a high-speed digital computer which was used to obtain the results given in Enclosure 7.

APPENDIX II

Properties of candidate lubricants (excluding proprietary specifications).

1. Formulations based on refined mineral oils or hydrocarbons:

   a. Esso FN-3157 Oil (MLO 7277 Base)
      Base Stock - 5.3% iso Paraffins, 23.8% one ring Naphthenes
      Antioxidants (in wt.%) - Phenyl Alpha Naphthylamine 2.0
      E. P. Additives (in wt. %) - Tricresylphosphate 1.0
      Foam Inhibition (ppm) - 10
      Kinematic Viscosity

°F	cs
0	10,000
30	1,500
100	79.2
210	8.4

VI (ASTM D567)	74
ASTM Slope	0.759
Pour pt., °F	-30
Flash pt., °F	445
Fire pt., °F	495
Neutrality No.	0.0

Density

°F	
0	0.908
100	0.873
200	0.838
300	0.802
400	0.768
500	0.732

Coefficient of Expansion

°F	10^{-4} cc/cc°F
0	4.0
100	4.2
200	4.4
300	4.5
400	4.9

APPENDIX (Cont.)

Thermal Conductivity

°F	Btu/Hr Ft. °F
0	0.0775
100	0.0732
200	0.0729
300	0.0705
400	0.0682
500	0.0659

Specific Heat

°F	Btu/Lb. °F
0	0.424
100	0.471
200	0.519
300	0.566
400	0.615
500	0.660

b. Kendall Resin 0838 (modified)

Gravity, API	25.7
Flash, COC °F	615
Fire, °F	710
Viscosity SUS @ 100°F	9,305
Viscosity SUS @ 210°F	410
Pour Point °F	+15
Viscosity Index	107.8
Ramsbottom Carbon Residue	1.12%
Aromatic rings per molecule	0.9
Total rings per molecule	5.25

APPENDIX (Cont.)

c. Socony Mobil XRM 112

I. Physical Properties

API Gravity, 60°F	5.9
Density, 68°F	1.0265
Pour Point, °F	10
Flash Point, COC. °F	415
Fire Point, COC	460
Kinematic Viscosity	
cs at 100°F	380.1
cs at 210°F	8.05
Refractive Index, n_d^{20}	1.58
Boiling Range, °F	-
Color	Light Yellow

II. Performance Data

Beta Irradiation Stability

Irradiation in an electron beam at 210°F while stirring under helium showed XRM 112 was quite stable to a dose of $1x10^9$ rads. The viscosity at 100°F showed a 42% increase, and at 210°F it increased 19%. There was very little gas formation during irradiation and no change in flash point after irradiation.

Thermal Stability

After 90 minutes at 700°F under nitrogen, there was no significant evaporation loss or viscosity change.

Lubricating Properties

Under severe wear test conditions in a Modified 4-Ball Machine, Friction and Wear Test, with 60 kilogram predent and 40 kilogram test load and oil temperature of 200°F, XRM 112 gave a scar diameter increase due to wear of 0.49 mm and coefficient of friction 0.141.

Toxicology

Toxicological properties of XRM 112 have not been investigated but it is believed to be about the same as XRM 100. Care should be exercised in its handling and use until more is known of its biological effects.

APPENDIX (Cont.)

d. Kendall Bright Stock 0846

Gravity, °API	27-27.6
Flash, COC °F	550 max.
Fire, °F	625 min.
Viscosity SUS @ 210°F	137-143
Pour Point, °F	+ 15 max.
Viscosity Index	100.0
Conradson Carbon Residue	1.0% Max.
Neutrality No.	Tan-C-0.05 max.
Color, NPA	6 max.

e. Socony Mobil XRM 109F

(These are the same for XRM - 177F)

Kinematic Viscosity	
Cs @ 400°F	4.92
Cs @ 210°F	31.95
Cs @ 100°F	314.1
Cs @ -20°F	96,099
Cs @ -40°F	99,000
Cs @ -65°F	-
Pour Pt., °F	-40
Flash Pt., °F	530
Fire Pt., °F	580
SIT, °F	775
Spontaneous ignition temp.	
Volatility	
6½ hours @ 400°F	-
6½ hours @ 500°F	15

APPENDIX (Cont.)

2. Polyphenyl Ethers

a. Monsanto OS-124

(These are the same for Skylube 600 & MCS 365)

Kinematic viscosity, cs	
100°F	365
210°F	13.1
400°F	2.1
500°F	1.2
700°F	0.65
Viscosity index (ASTM D-567)	-80
Pour Point, °F	+40
Flash Point, °F	550
Fire Point, °F	660
Neutralization No.	0.1
Viscosity at elevated pressure, cs (100°F, gas free)	
Atmospheric	369
200 psig	393
500 psig	434
1000 psig	511
1000 psig (gas saturated)	422
Density, g/cc	
100°F	1.187
300°F	1.100
600°F	0.971
Thermal conductivity, BTU/hr-ft °F	
100°F	0.0768
300°F	0.0733
500°F	0.0695
Specific heat, BTU/lb °F	
100°F	0.368
300°F	0.432
700°F	0.560
Vapor Pressure, mm Hg	
500°F	0.7
700°F	26
800°F	103
Evaporation rate at 400°F, % loss (MIL-L-7808 test method)	0.6

APPENDIX (Cont.)

c. Monsanto OS-138 (MLO-60-231)

Viscosity (cs)

100°F	1831
210°F	24.7
400°F	2.8
700°F	.82

Evaporation (22 hrs)

400°F	0.3% weight loss
500°F	0.2% weight loss
550°F	13.7% weight loss
600°F	34.8% weight loss

Shell 4-Ball Wear Test

400°F	600 rpm 1 hr.	Wear Scar (mm)
	30 Kg	0.973 mm
	50 Kg	1.158 mm

d. Monsanto MCS 293

Viscosity, Density and Compressibility

Temperature °F	°C	Viscosity (cs)	Density (g/cc)	Bulk Modulus* (psi)	(kg/cm2)
0	-18	13,040	-	500,000	35,180
77	25	-	1.195	410,000	28,820
100	38	25.2	1.184	340,000	23,900
210	100	4.13	-	285,000	20,050
300	150	2.0	1.101	230,000	16,200
500	260	0.81	1.017	190,000	13,400
700	370	0.48	0.926		

*Secant, Isothermal 0-7500 psi (0-527 kg/cm^2) range.

Pour Point -20°F(-25°C)

APPENDIX (Cont.)

Vapor Pressure

140 mm Hg at 700°F(371°C)
760 mm Hg at 795°F(424°C)

Flammability Properties

Flash Point	445°F(230°C)
Fire Point	540°F(285°C)
Autogenous Ignition Temperature	940°F(500°C)

Thermal Properties

Temperature		Specific Heat	Thermal Conductivity	
			BTU	K x 10^{-5}
°F	°C	(BTU/lb. °F)	°F ft/hr	Cal/cm °C sec
0	-18	0.312	0.0677	-
100	38	0.347	0.0714	29.7
200	93	0.383	0.0720	29.6
300	150	0.418	0.0690	28.5
400	200	0.452	0.0650	27.0
500	260	0.488	0.0585	24.2

Evaporation Loss (ASTM D-972)

500°F(260°C) at 760 mm	50.7% average
500°F(260°C) at 140 mm	61.0% average

Bearing Rig Test (CRC Procedure - 100 hour test)

	Type 2-1/2	Type 3
Bearing Temperature	550°F(290°C)	600°F(316°C)
Bulk Oil Temperature	500°F(260°C)	550°F(290°C)
Oil Inlet Temperature	450°F(230°C)	500°F(260°C)

APPENDIX (Cont.)

After 100 hours		
Over-all Demerit Rating	63.2	66.1
Change in 100°F(37.8°C) Viscosity	54%	104%
Increase in Acid Number	nil	nil
Consumption Rate	40.8 cc/hr	103 cc/hr
Sludge Formation	2.26 g	8.39 g

Oxidation-Corrosion Test

(Modified FS 791, Method 5308.2; 48 hours at temperature, 5 L/hr air flow)

Temperature	500°F(260°C)	600°F(315°C)
Weight Change of Metals (mg/cm^2)		
magnesium	+0.03	+0.48
aluminum	0	-0.22
titanium	-0.05	+0.16
iron	+0.04	+0.23
copper	-3.97	-8.37
silver	-0.71	-1.39
Change in Viscosity		
at 100°F(37.8°C)	+6.5%	+42.3%
at 210°F(99.0°C)	+4.4%	+20.8%
Neutralization Number		
Initial	0.01	0.01
Final	0.10	0.15

Toxicity

Preliminary screening indicates that this fluid is practically non-toxic when administered orally to rats, percutaneously to rabbits, or by inhalation by rats of vaports produced when the fluid is pyrolyzed on metal surfaces heated to 400°F(260°C) and 700°F(370°C).

APPENDIX (Cont.)

Pour Point, °F (Min.)	-90
Spontaneous Ignition Temperature, °F (Min.)	800
Fire point, F	510
Acidity, mgKOH/g	0.03
Color, NPA	3.5
Emulsion at 130 F, ASTM D1404-56T	Complete Separation
Thermal stability after 48 hours at 600 F in stainless steel laboratory test bomb	
Deposits	None
Lacquer	None
Carbonization	None
Color	Darkening, transparent
Specific gravity at 60/60 F	0.966
Viscosity index, ASTM D567-53	+144
Viscosity (after 72 hours at -65 F), cs	19,000 (no change)

EVAPORATION

Evaporat on loss (during 6-1/2 hours at 400 F), per cent by weight, max 6

b. Heyden Newport Pentalube TP 653A

Spec. Gravity 25/25°C	.9670
Refractive Index 25°C	1.4475
Pour Point °F	-85
Flash Point °F	445
Fire Point °F	483
Spontaneous Ign. Temp. °F	750

Viscosity cs.

400°F	1.00
210°F	3.45
100°F	14.60
-40°F	2,250
-65°F	14,500

APPENDIX (Cont.)

e. Monsanto MCS 353

Kinematic Viscosity, cs

100°F	56-61
210°F	5.9-6.2
400°F	1.3-1.35
600°F	0.62-0.64

Density @ 78°F - 1.198

Pour Pt, +10°F

Solution Pt. +75°F

Boiling Pt. +950°F

Heat capacity and thermal conductivity properties are similar to Monsanto MCS 293

3. Fluorocarbons

a. DuPont PR 143

	PR-143		
	Lot 1	Lot 2	Lot 3
Pour Point, °F	-25	-30	-15
Viscosity, cs. @ 0°F	41,970	47,200	55,400
100°F	285.8	308.7	335.0
210°F	25.7	27.1	29.0
400°F	3.68	3.86	4.11
Viscosity Index	115	115	116
ASTM Slope	0.609	0.602	0.596
Density, g./cc. @ 75°F	1.9051	1.9058	1.9067
210°F	1.7771	1.7793	1.7804

APPENDIX (Cont.)

Evaporation Loss, 6 1/2 hours, 400°F (%)	14.7

Corrosion and Oxidation Test*

MIL-I-9236A, 48 hours @ 500°F	5.1
Evaporation Loss (%)	46.5
Vis. Increase 100°F (%)	22.4
Metals Attack (mg/sq.cm.)	
Titanium	+0.025
Magnesium	Disintegrated
Aluminum	-0.015
Copper	-0.21
Silver	+0.01
Steel	-5.41

*Stabilized with 2% PANA

c. Socony Mobil RM-139A

Flash Pt., °F	485
Fire Pt., °F	560
Autogenous Ignition Temp., °F	790
Pour Pt.	-80
Neutralization No., Mg KOH/g. sample	0.10
Kinematic Viscosity @ -40°F	12,393
@ 100°F	28.40
@ 210°F	5.19
@ 400°F	1.31
ASTM Foam Test, Sequence I	Nil
II	Nil
III	Nil
Evaporation, % 6 1/2 Hours @ 400"F	2.57
@ 450°F	10.01
Panel Coking Test MG, 8 Hours @ 700°F (continuous splash)	49
S.O.D. Lead Corrosion, Mg/in^2	-0.6

Oxidation-Corrosion, 48 Hours @ 450°F

KV_{100}, % change	+80

APPENDIX (Cont.)

Neutralization No., Final	4.0
Sludge, Visual	Nil

Catalyst Metal Weight Change, Mg/cm^2

Ag	-0.03
Al	0.00
Cu	-0.11
Fe	+0.02
Mg	-0.01
Ti	+0.02

High Temperature Bearing Test in Air

Bulk oil - 433°F Oil in - 402°F Bearing - 502°F

Hours	KV_{100}	MN	Hours	KV_{100}	MN
0	28.11	0.16	60	36.22	1.5
10	31.06	0.42	70	37.35	1.6
20	31.98	0.55	80	38.71	-
30	33.18	-	90	40.10	-
40	34.13	-	100	41.37	2.7
50	35.30	0.98			

Oil Consumption, cc/Hour 29

Deposit Demerit Ratings	
Mount	91.5
Boaring	39.0
Overall	22.2

d. Esso Turbo Oil 35

Viscosity at 210°F, cs	7.8
Viscosity at 100°F, cs	37.4
Viscosity at--40°F, cs	12,000
Viscosity at--65°F, cs	-
Viscosity stability, cs at--65°F,%	-
Viscosity index	155
Specific gravity	0.968

APPENDIX (Cont.)

Flash point, °F	480
Pour point, °F	-65
Swell Rubber H, 1 week at 158°F,%	-
Ryder gear test, lb/in	3,100
2 tests, relative rating,%	-
Total acid number	0.1
Oxidation & corr. stability, 72 hr at 347°F	
Weight change, mg/sq cm	
steel, silver, aluminum, magnesium	Complies
copper	Complies
Total acid number increase	0.6
Viscosity change, cs at 100°F,%	+9
Evaporation, 6 1/2 hr at 400°F,%	8
Foaming	Complies
Coking, mg	30
Silver corrosion, 50 hr at 450°F, loss, mg/sq in	0.3
Copper corrosion, 50 hr at 450°F, loss, mg/sq in	0.4
Humidity cabinet life, hr	-
Precipitation number	Zero
Lead corrosion, 1 hr at 325°F, loss, mg/sq in	0.00

APPENDIX (Cont.)

SINCLAIR TURBO-S OIL 1048 IMPROVED

Laboratory Tests	Allison EMS-35H Specification	Typical Tests
Gravity, °API	Determine	19.5
Flash Point, °F	425 Min.	455
KV @ -40°F., cs.	13,000 Max.	9088
KV @ 100°F., cs.	Determine	39.75
KV @ 210°F., cs.	7.5 Min.	8.35
Pour Point, °F.	-60 Max.	Below -75
Acid Number	Determine	0.15
Appearance	Clear	Clear
Low Temperature Stability (-40°F., 72 hrs.)	No Separation, Gelling, etc.	Pass
Corrosion-Oxidation Stability Test (347°F., 72 hrs.)		
Metal Wt. Change, mg./cm.^{2}		
Copper	± 0.4	-0.01
Silver	± 0.2	0.00
Steel	± 0.2	0.00
Aluminum Alloy	± 0.2	0.00
Magnesium Alloy	± 0.2	0.00
Used Oil Tests		
Viscosity @ 100°F., % Change	-5 to +12	7.3
Acid Number Increase	1.5 Max.	1.0
Panel Coking (600°F., 8 hrs.), mg. gain	75 Max.	5.4
Foam Test: Sequence I	100-0-5'	0-0-0
Sequence II	25-0-3'	0-0-0
Sequence III	100-0-5'	0-0-0
Ryder Gear Test, Lbs./in. of Tooth Face Width	3000 Min.	3800
S.O.D. Lead Corrosion (325°F.,1hr.),mg./in.^{2}	-6.0 Max.	-0.62
Evaporation, % Loss (400°F., 6.5 Hrs.)	Determine	8.3

APPENDIX III

ANALYSIS OF ENDURANCE TEST RESULTS

Rules for Estimating Life Parameters for Bearings Run Under Extreme Lubrication Conditions

For the purpose of preparing the maximum likelihood life analysis of Phase II endurance test bearings, as described on the subsequent pages of this Appendix, the endurance results are processed according to the following rules:

1. A bearing is considered failed if any of its elements (except, of course, the cage) suffer fatigue flaking.

2. Both bearings in a test assembly is started on test at the same time. In the event both bearings fail, one is considered a failed bearing and the other, an induced failure, and thus is treated as a suspension.

3. Should only one of two bearings fail in a test, the unfailed bearing is considered a suspended test. It will not be run further.

4. If only the cage of a bearing fails (e.g. excessive bore wear, cracked cage pockets), the cage is replaced and the test continued.

5. If cage failure and bearing fatigue failure exist in the same bearing so that it can be assumed that the cage failure has contributed to the premature failure of another element, then the fatigue failure will be considered an induced failure and treated as a suspended test.

6. Surface distress will not be considered a failure if the bearing can be run further. If, however, surface damage is so severe that the bearing cannot be run further, this mode of failure will be treated the same as a fatigue failure.

7. The existence of surface distress along with a fatigue failure in a bearing will be noted but not considered a reason for designating the failure as an induced one. Such failures will be considered legitimate in calculating life because surface distress is a consequence of the test conditions and may be unavoidable.

APPENDIX (Cont.)

It is recognized that the exclusion rules 2, 4 and 5 tend to bias resultant life estimates towards longer life. Still, it is felt that these rules must be used in order that operational malfunctioning of the test be kept from derating the inherent endurance capabilities of the bearing-lubricant combination. In order to evaluate the overall bearing-lubricant reliability under the current state of operational development, however, a second life estimate is conducted for each test group including all failures which otherwise are considered as suspended tests under exclusion Rules 4 and 5. (Failures of the companion bearing in tests where both bearings fail are still treated as suspended tests, according to Rule 2, even when considering overall bearing-lubricant reliability, since companion bearing failures can be so easily induced by initial failure in a bearing pair under high-speed high-temperature operating conditions.)

APPENDIX (Cont.)

Maximum Likelihood Estimation of Fatigue Life Parameters

In evaluating bearing (or bearing element) endurance tests, a problem arises from the fact that failures will occur or the testing of a bearing will be suspended for reasons other than the inherent fatigue failure of the bearing (or element) under test, so that the bearing lives must be estimated in a manner which takes appropriate account of these extraneous failures. In (35), a method originated by Lieblein-Zelen (42) was used in its modified form involving multiple subgroup randomization.* A closer analysis of the Lieblein-Zelen method has recently shown it to be correct and of satisfactory efficiency for uncensored samples (no unfailed test elements) and for samples truncated at a single life value exceeding all failure lives (test group without extraneous failures terminated at a preselected time). The method is not, however, directly applicable to the situation where extraneous failures (of non-test elements) occur at lives not exceeding the longest fatigue life of a test element, and the expedient suggested in (43) to deal with discontinuances is now recognized as questionable.

McCool and Tallian (44) have recently developed a method of maximum likelihood estimation for the parameters of a Weibull distribution which is applicable to any censored sample (involving discontinued tests or extraneous failures) provided only that the times at which censoring occurs are independent of the fatigue life of the failed test elements. As described in (44), this new method consists of the following principal steps:

(a) The likelihood function pertaining to the lives of a censored sample of endurance tested bearings or elements, of which m_1 have failed, is written as

$$\log \mathcal{L} = \sum_{k=1}^{m} \log f_i(L_k) + \sum_{j=m_1+1}^{m} \log[1 - F_i(L_j)] + R \qquad [1]$$

---

* The Lieblein-Zelen method as applied to bearing endurance life evaluation is discussed in more detail by Tallian (43).

APPENDIX (Cont.)

where:

$f_1(L)$ is the probability density function of the life distribution of the element,

L_k is the life at failure of the k-th failed element (taken in random order),

$F_1(L)$ is the cumulative distribution function of the life distribution of element,

L_j is the life at which testing of the j-th unfailed element was discontinued, and

R is a term containing only quantities independent of the element life distribution.

It is assumed that the test element has a Weibull life distribution (with unknown scale parameter L_{10} and a dispersion parameter e, but with zero minimum life).

Then, from the definition of a Weibull distribution (43)

$$f_1(L) = \frac{Ke}{L_{10}^{e}} L^{e-1} \exp\left\{-K\left(\frac{L}{L_{10}}\right)^{e}\right\} \qquad [2]$$

$$F_1(L) = 1 - \exp\left\{-K\left(\frac{L}{L_{10}}\right)^{e}\right\} \qquad [3]$$

where

$$K = \log \frac{1}{0.90} = 0.105361$$

APPENDIX (Cont.)

Substitution of Eqs. [2] and [3] into Eq. [1] yields the explicit formula for the likelihood function.

It is shown in (44) that an estimator with many desirable properties, the so-called maximum likelihood estimator can be obtained for the parameters L_{10} and e by finding values of these parameters which maximize the likelihood $\mathcal{L}$ defined by Eq. [1]. This "maximum likelihood" method of estimation is applicable to the situation where competing failure mechanisms operate (i.e., where elements other than the failed elements are also subject to failure) provided only that the lives of elements other than the failed elements are statistically independent of the failed element lives. In case a test is arbitrarily terminated, it is necessary to require that the termination rule be independent of the lives of the test elements, i.e., that it be established without benefit of information regarding the outcome of the test itself. These requirements are usually fulfilled in bearing fatigue tests.

As shown in (44), the determination of the L_{10} and e values maximizing [1] is best accomplished by computing the partial derivatives $\partial \log \mathcal{L} / \partial L_{10}$ and $\partial \log \mathcal{L} / \partial e$ and setting

$$\left.\begin{aligned} \frac{\partial \log \mathcal{L}}{\partial L_{10}} &= 0 \\ \frac{\partial \log \mathcal{L}}{\partial e} &= 0 \end{aligned}\right\} \qquad [4]$$

with suitable verification that the L_{10} and e values satisfying Eq. [4] give indeed a maximum of $\mathcal{L}$.

Eq. [4] is a system of two simultaneous equations in L_{10} and e. It cannot be solved analytically. A computer program was, therefore, written which supplies a solution by the iterative Newton-Raphson method. Using an IBM 1620 computer, a solution for a group of lives can be obtained within a few minutes. Given L_{10} and e, it is then possible to compute L_{50} from the formula

$$L_{50} = \left(\frac{\log 2}{k}\right)^{1/e} L_{10} = (6.57881)^{1/e} L_{10} \qquad [5]$$

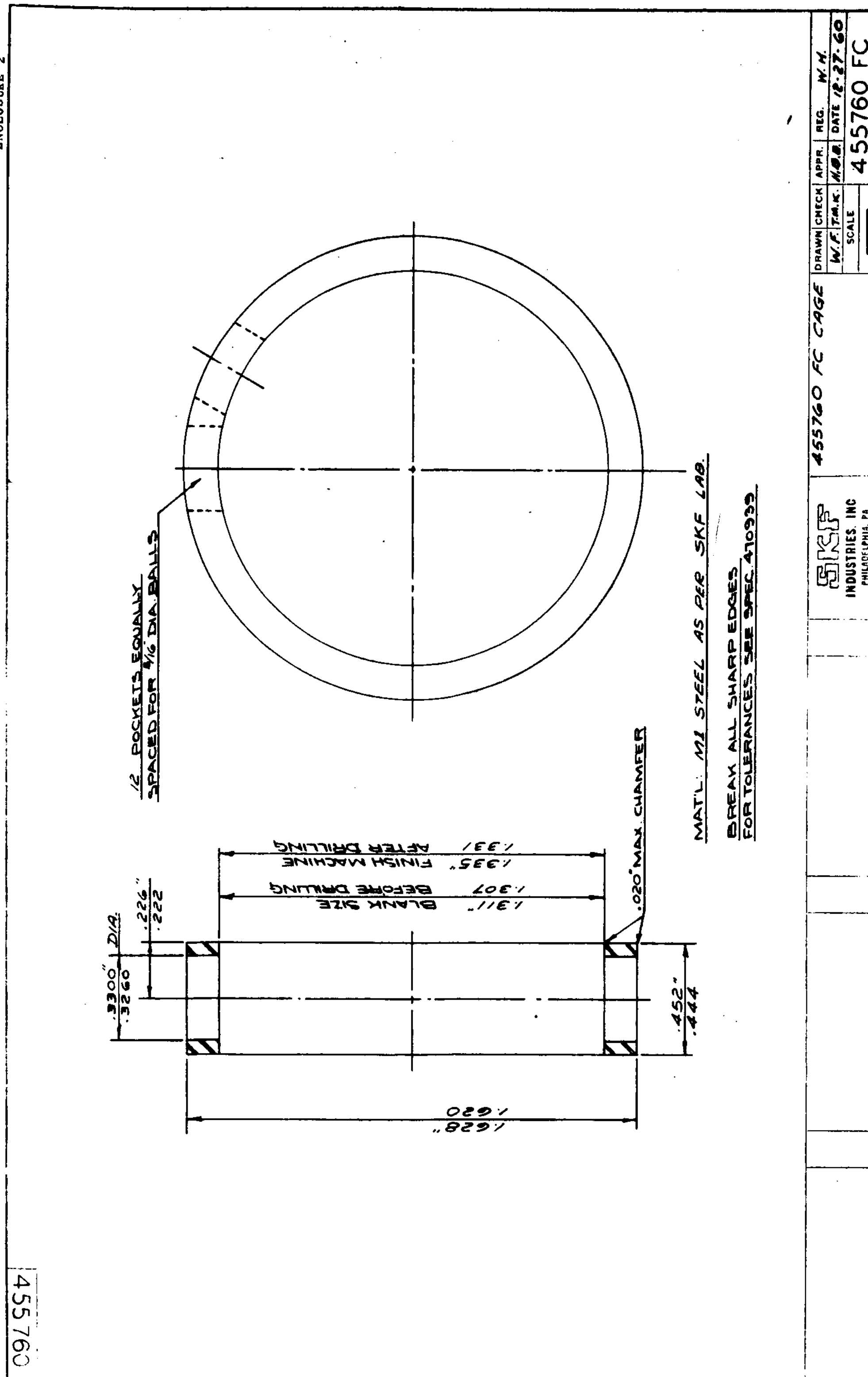
APPENDIX (Cont.)

The maximum likelihood method of estimation does not directly yield confidence limits. The one-sigma confidence limits of L_{10} for a test group of 30 bearings were obtained by a Monte Carlo method described in (44) which operates essentially as follows: Sets of random numbers having a Weibull distribution with zero minimum life, a known value of L_{10}, and several known values of e are numerically generated. These samples of Weibull distributed numbers are subjected to censoring by a method simulating the censoring occurring in the life tests considered. Each censored sample of numbers is then subjected to the maximum likelihood estimation method described above, giving numerous estimates of the parameter L_{10}.

The standard deviation σ_{L10} of this group of L_{10} estimates is computed and expressed as a fraction σ_{L10}/L_{10} of the population value of L_{10} and as a function of the population Weibull slope e. Given the maximum likelihood estimates for L_{10} and Weibull slope e for a bearing life test group, it is now possible to read from the Weibull graph the value of σ_{L10}/L_{10} applicable to the estimated value of e. For any test group containing 30 bearings, σ_{L10}/L_{10} is read multiplied by the L_{10} estimate for that group, and added and subtracted from L_{10} to form upper and lower one-sigma confidence limits. For groups of size other than 30, a correction proportional to the square root of the group size is applied to σ_{L10}. This method of obtaining confidence limits is admittedly approximate since the censoring conditions used in the Monte Carlo study are not exactly the same as those encountered in the life tests. It has been shown, however, (44) that the effect of minor differences in truncation on the confidence band for L_{10} is small as long as no discontinuance occurs prior to L_{10}.

ENCLOSURE 2

STANDARD DESIGN M-1 STEEL CAGE

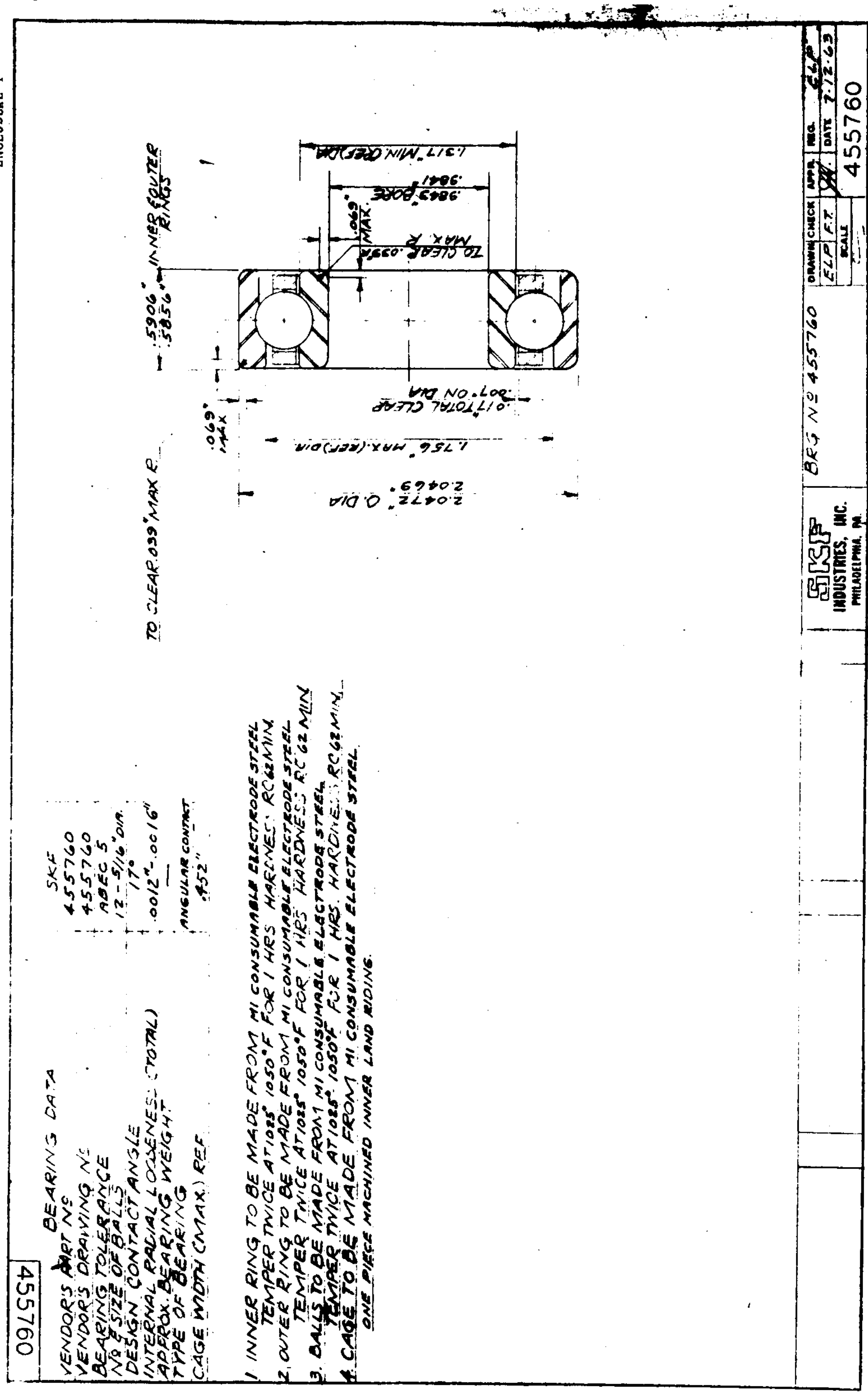


AL65T068

ENCLOSURE 1

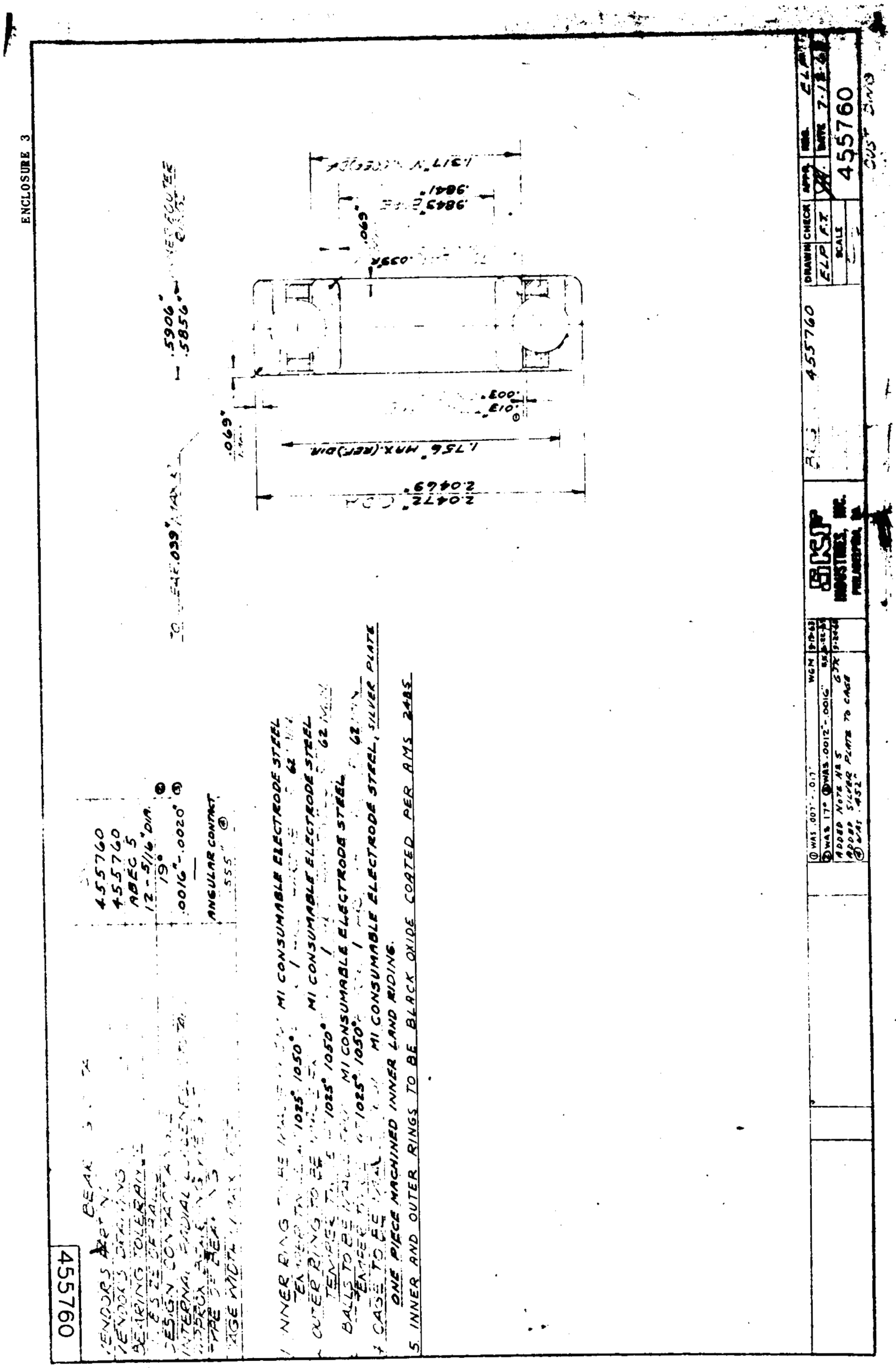
STANDARD DESIGN 17° CONTACT ANGLE 7205
TEST BEARING MADE OF M-1 STEEL

ENCLOSURE 1



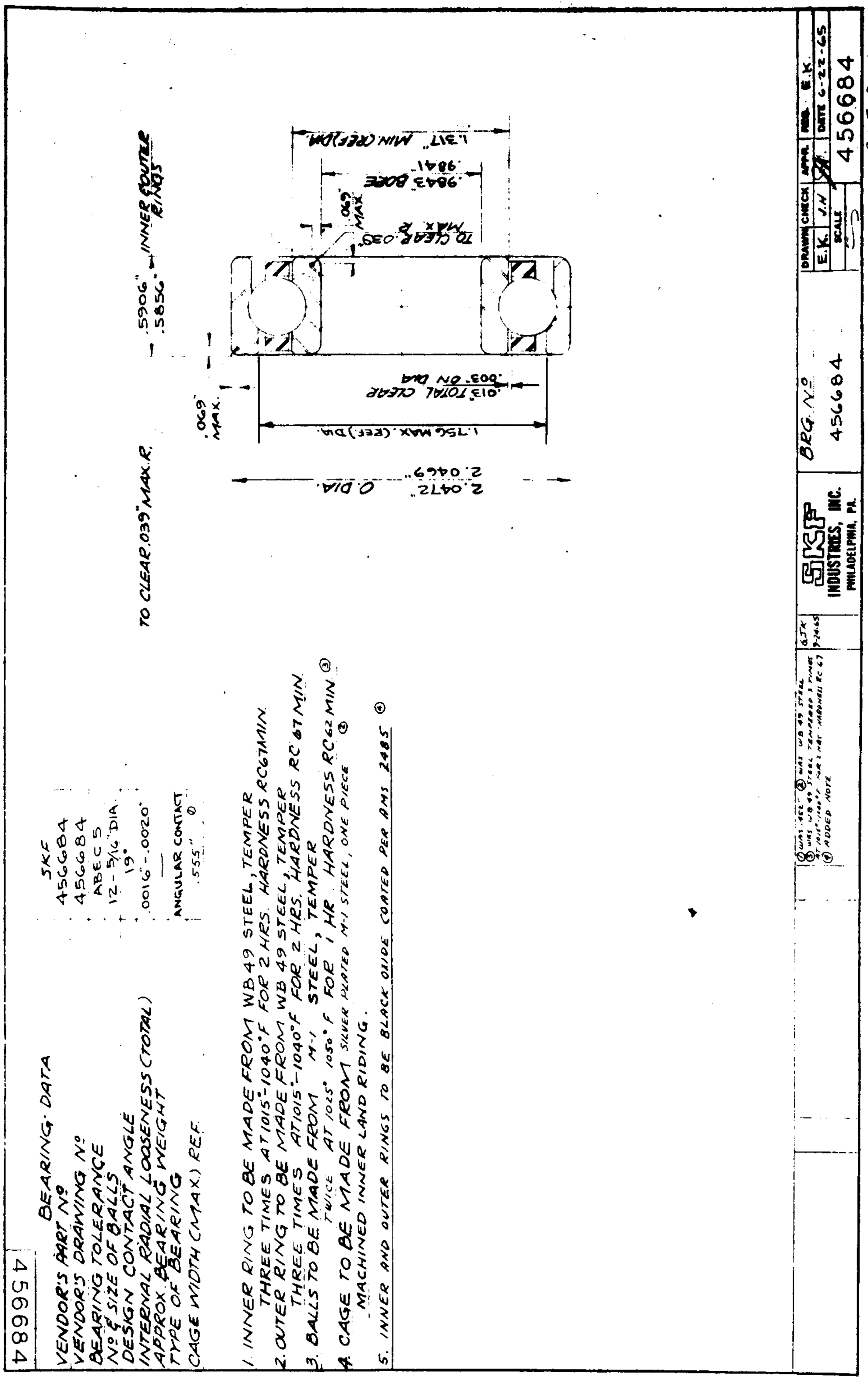
ENCLOSURE 3

SPECIAL 19° CONTACT ANGLE 7205 TEST
BEARING MADE OF M-1 STEEL



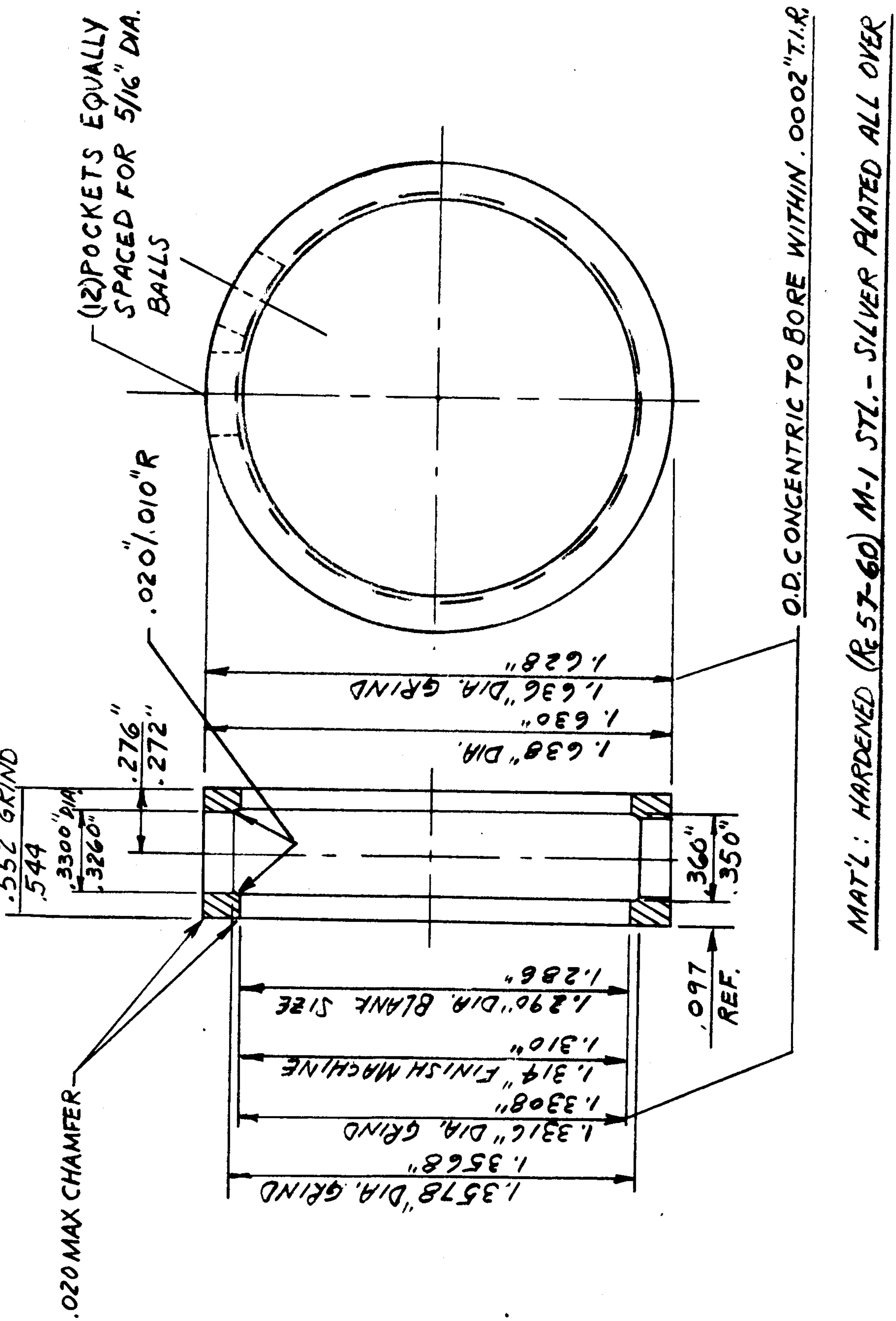
ENCLOSURE 4

SPECIAL 19° CONTACT ANGLE 7205 TEST
BEARING MADE OF WB49 STEEL



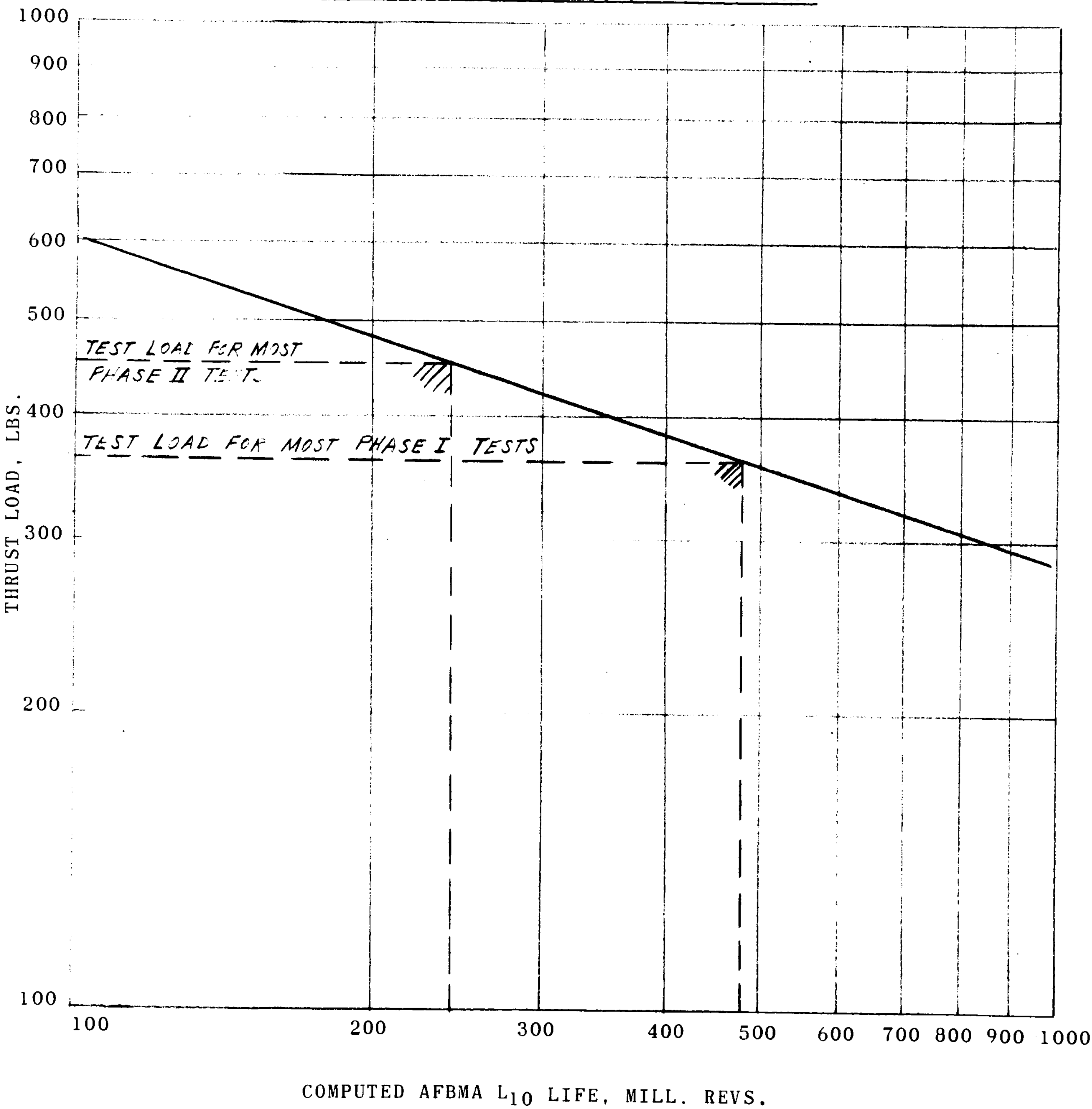
ENCLOSURE 5

SPECIAL SILVER PLATED M-1 STEEL CAGE



ENCLOSURE 6

LOAD-LIFE RELATIONSHIP FOR M-1 (#455760)
AND WB49 (#456684) TOOL STEEL BEARINGS



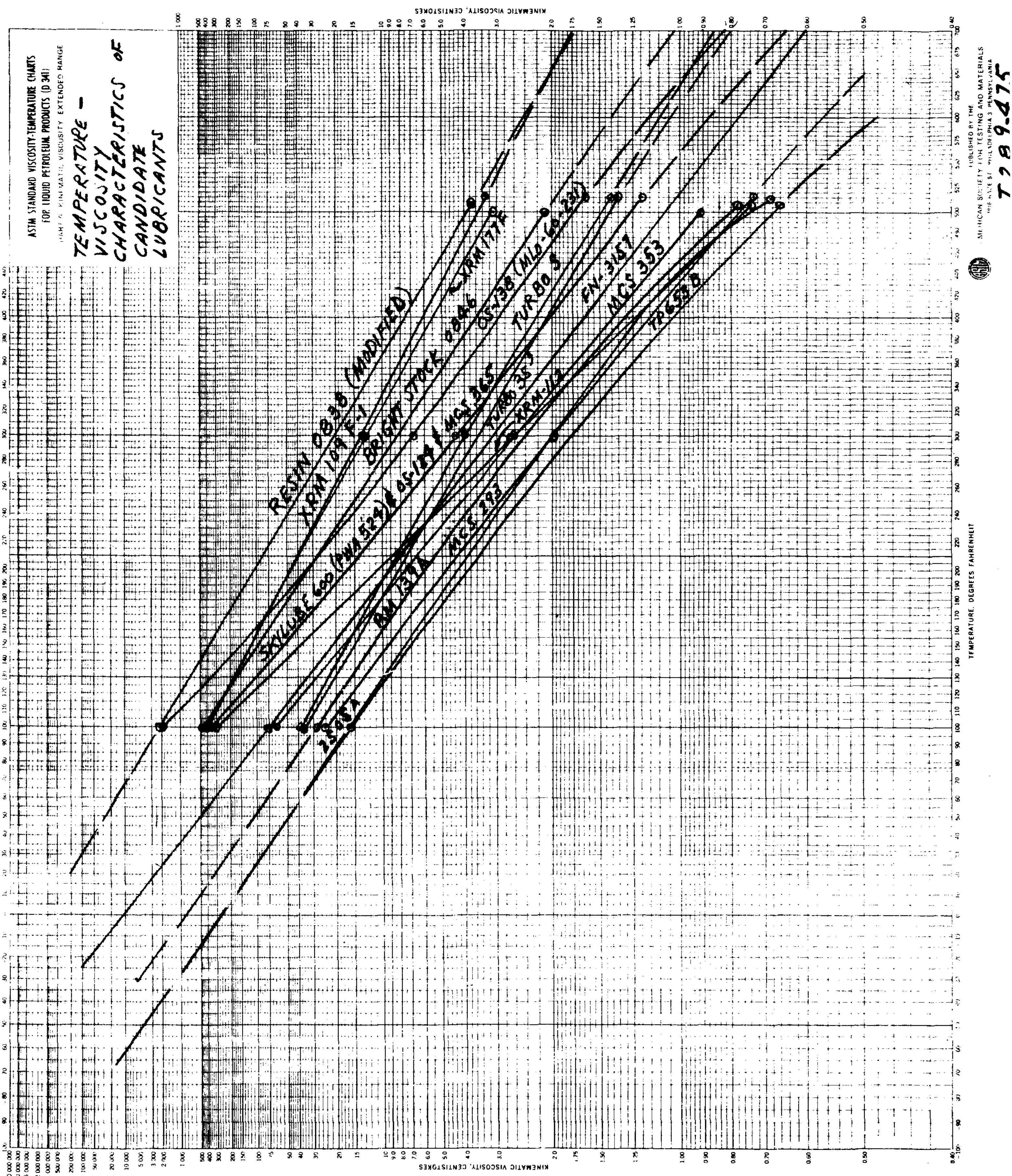
ENCLOSURE 7

COMPUTED DYNAMIC CHARACTERISITCS OF TEST BEARINGS

Test Conditions:	17° Contact Angle							19° Contact Angle					Modified Design
Shaft Speed, RPM	20,000	20,000	20,000	20,000	20,000	20,000	20,000	20,000	20,000	20,000	42,800	42,800	42,800
Thrust Load, LBS.	150	295	365	450	580	725	918	365	725	918	365	459	459
Computer Operating Parameters:													
Inner ring contact angle, degrees	17.9	19.1	19.1	20.1	20.9	21.7	22.5	21.0	25.8	26.4	24.0	24.0	22.9
Outer ring contact angle, degrees	15.4	17.6	17.9	19.0	20.0	20.9	21.9	19.5	24.6	25.4	16.9	18.0	17.3
Axial bearing deflection, in.	.0012	.0017	.0020	.0021	.0024	.0026	.0029	.0017	.0018	.0021	.0017	.0018	.0018
Ball centrifugal force, LBS.	6.3	6.4	6.4	6.4	6.4	6.4	6.5	6.4	6.5	6.5	30.0	29.9	31.0
Inner semi-major contact axis, in.	.0302	.0367	.0360	.0413	.0443	.0472	.0503	.0369	.0433	.0464	.0369	.0394	.0349
Outer semi-major contact axis, in.	.0259	.0309	.0322	.0346	.0370	.0393	.0418	.0314	.0365	.0390	.0336	.0354	.0406
Normal inner ring ball load, LBS.	38.8	73.4	91.2	107.1	133.6	161.8	197.8	83.2	137.4	170.6	70.9	89.9	93.8
Normal outer ring ball load, LBS.	44.9	79.4	97.2	113.1	139.6	167.8	203.8	89.2	143.3	176.5	98.8	117.5	122.8
Maximum inner contact stress, KPSI	189.4	242.8	268.1	278.9	301.9	323.0	346.5	258.8	309.2	333.7	219.9	243.7	268.9
Maximum outer contact stress, KPSI	183.2	228.7	249.1	260.5	281.1	300.2	321.5	241.5	287.5	309.7	232.6	249.7	238.7
Spin-to-roll ratio on inner	0	0	0	0	0	0	0	0	0	0	0.22	0.21	0.20
Cage speed, RPM	8035	8060	8064	8077	8089	8100	8113	8082	8151	8163	17490	17473	17371
Ball rolling speed, RPM	39518	39642	39661	39727	39783	39839	39902	39749	40088	40150	86028	85940	84091
Ball axis orientation angle, degrees	22.4	23.8	23.8	25.1	26.1	27.0	28.1	26.1	31.9	32.7	14.1	15.1	14.4
Spinning heat generated, BTU/HR.	102.6	210.0	258.0	338.0	451.0	591.0	787.0	260.5	563.5	745.0	562.0	714.0	625.0
Total heat generated BTU/HR.	578.4	1046.0	1332.5	1728.5	2163.7	2809.2	3729.9	1152.2	1998.8	2619.4	2773.0	3424.3	3446.0
Minimum friction coefficient required to prevent gyro slip	.014	.008	0.007	.006	.005	.004	.004	.008	.006	.005	.021	.018	.017
Bearing life (Lundberg-Palmgren), mill revs.	3350	519.0	250.5	170.8	88.9	50.3	27.8	332.0	77.0	40.6	328.5	171.9	124.8
Life of inner ring contact, mill revs	4235	691.0	301.0	205.0	106.3	60.0	33.1	401.5	92.9	48.6	431.0	212.4	138.5
Life of outer ring contact, mill revs	13130	2343.0	1188.0	806.0	426.0	244.0	135.5	1523.0	356.0	189.5	1134.0	670.0	920.0
EHD oil film thickness on inner for 2.5 cs viscosity lubricant 10^{-6} in.	4.5	4.4	4.4	4.3	4.3	4.2	4.2	4.4	4.3	4.3	7.7	7.6	7.6
Ball gyroscopic moment, in-LBS.	0.18	0.19	0.19	0.20	0.21	0.22	0.23	0.21	0.26	0.26	0.55	0.58	0.59

*19° contact angle (nominal unmounted), but with the ball-race conformities changed from 52.2% to 53% on the inner ring and from 53.2% to 52.3% on the outer ring, all other design parameters identical to those in Enclosures 3 and 4.

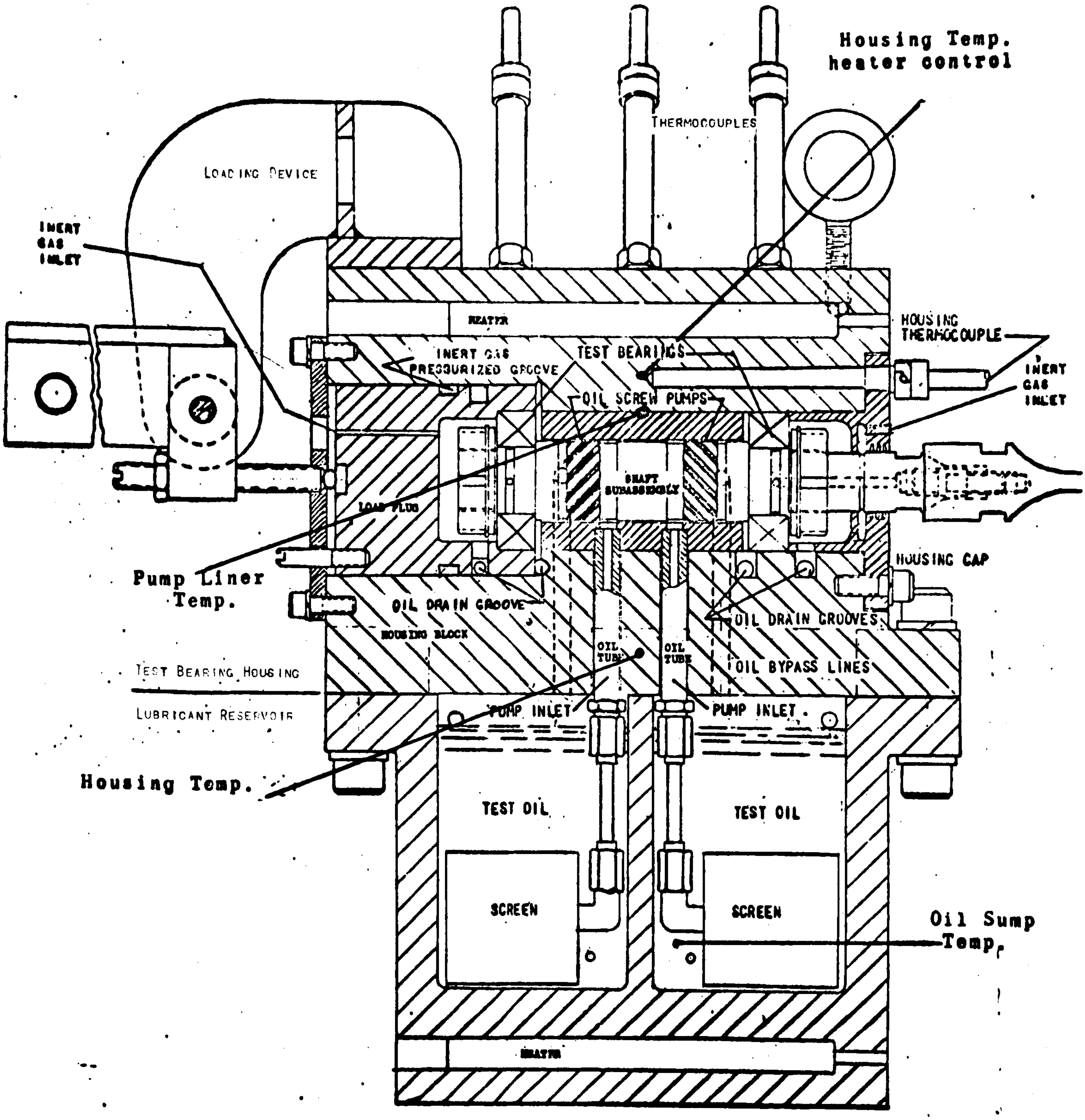
ENCLOSURE 8



RESEARCH LABORATORY SKF INDUSTRIES, INC.

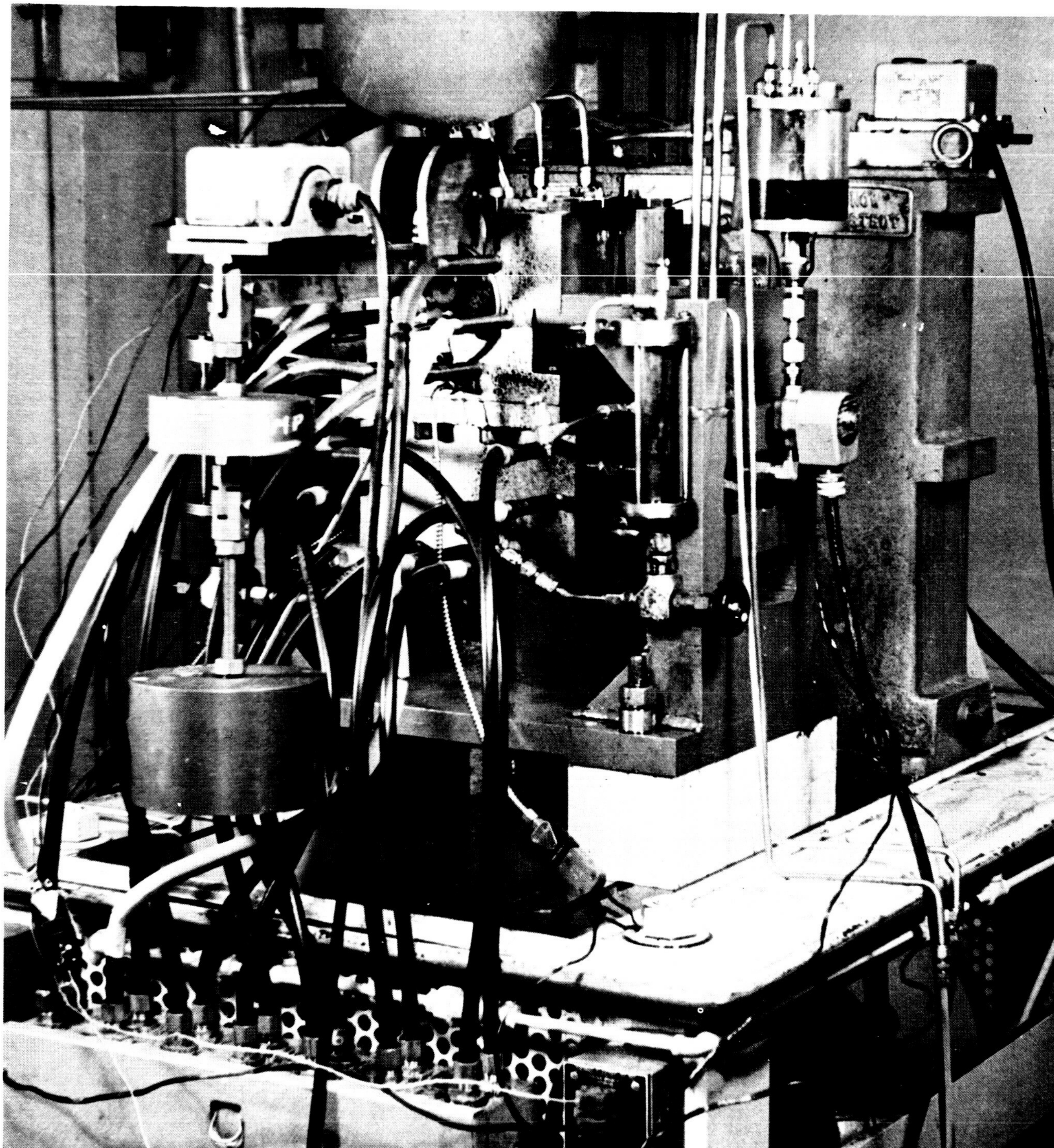
ENCLOSURE 9

LAYOUT SKETCH OF HIGH-SPEED HIGH-TEMPERATURE TESTER



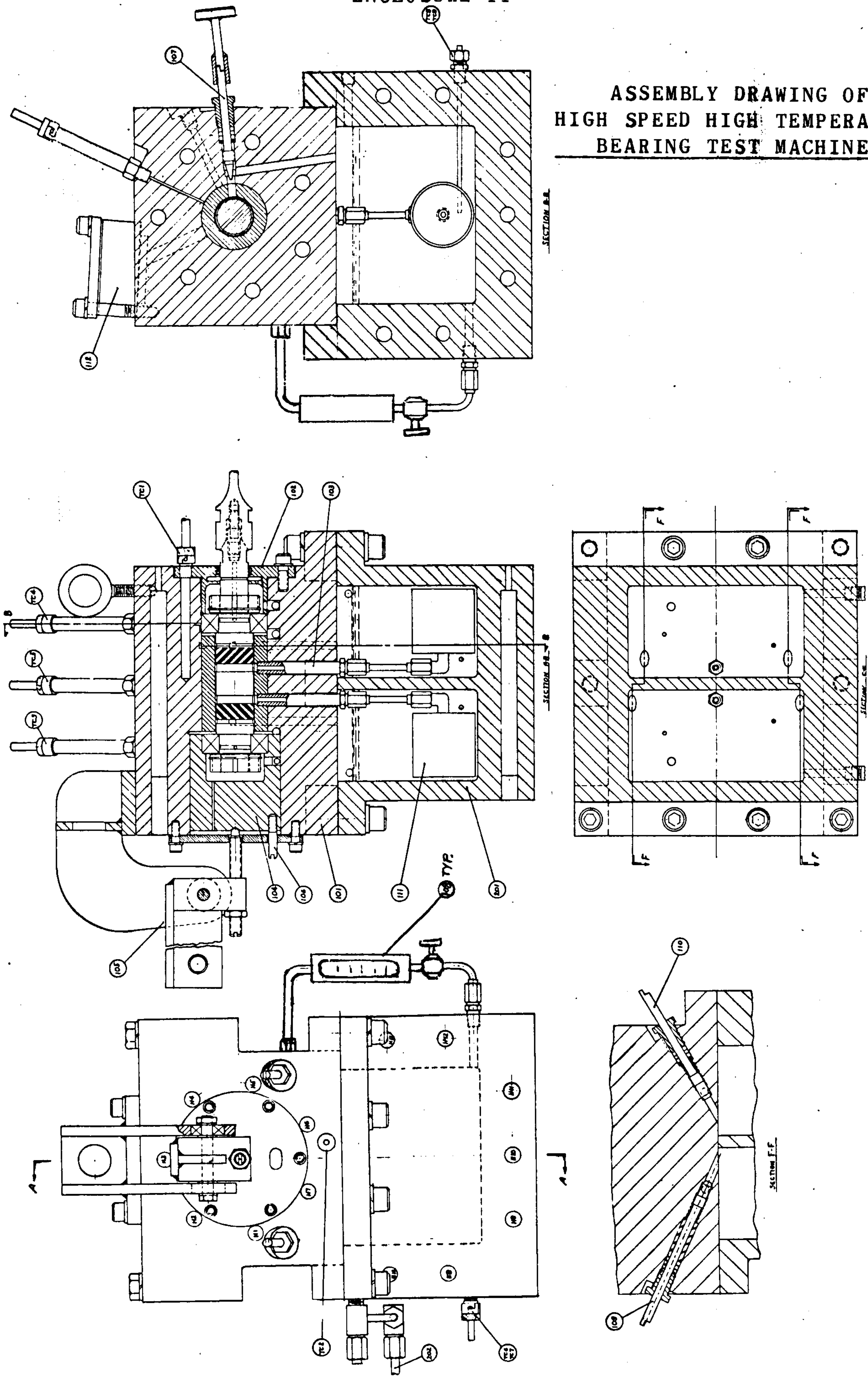
ENCLOSURE 10

HIGH SPEED HIGH TEMPERATURE BEARING TEST MACHINE



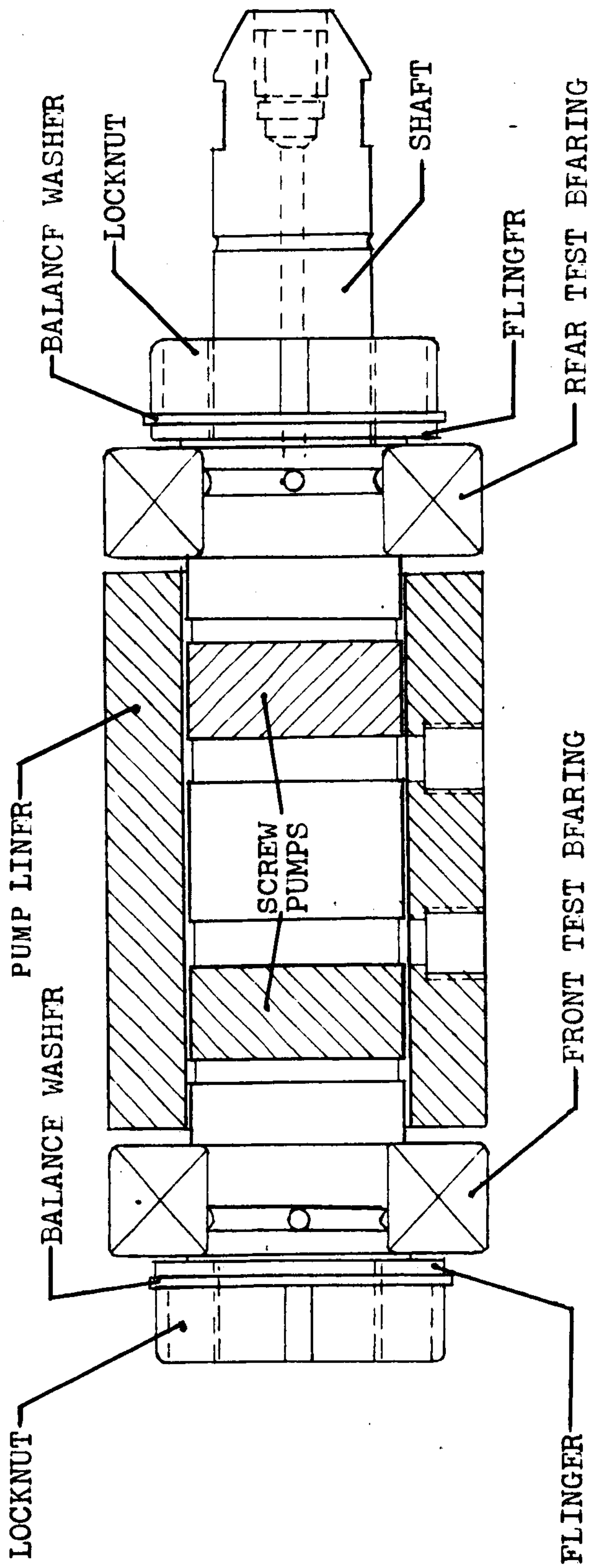
ENCLOSURE 11

ASSEMBLY DRAWING OF HIGH SPEED HIGH TEMPERATURE BEARING TEST MACHINE



ENCLOSURE 12

SHAFT SUB-ASSEMBLY



ENCLOSURE 13

CONTROL PANEL FOR
HIGH-SPEED HIGH-TEMPERATURE TESTER

HEATERS — DC MOTOR — MG
ON
OFF
SPEED CONTROL
LOW, HIGH PRESS.
INDICATORS
RESET

DC MOTOR
SPEED—AMPS— VOLTS—HOUR METER

PRESSURE
GAGES

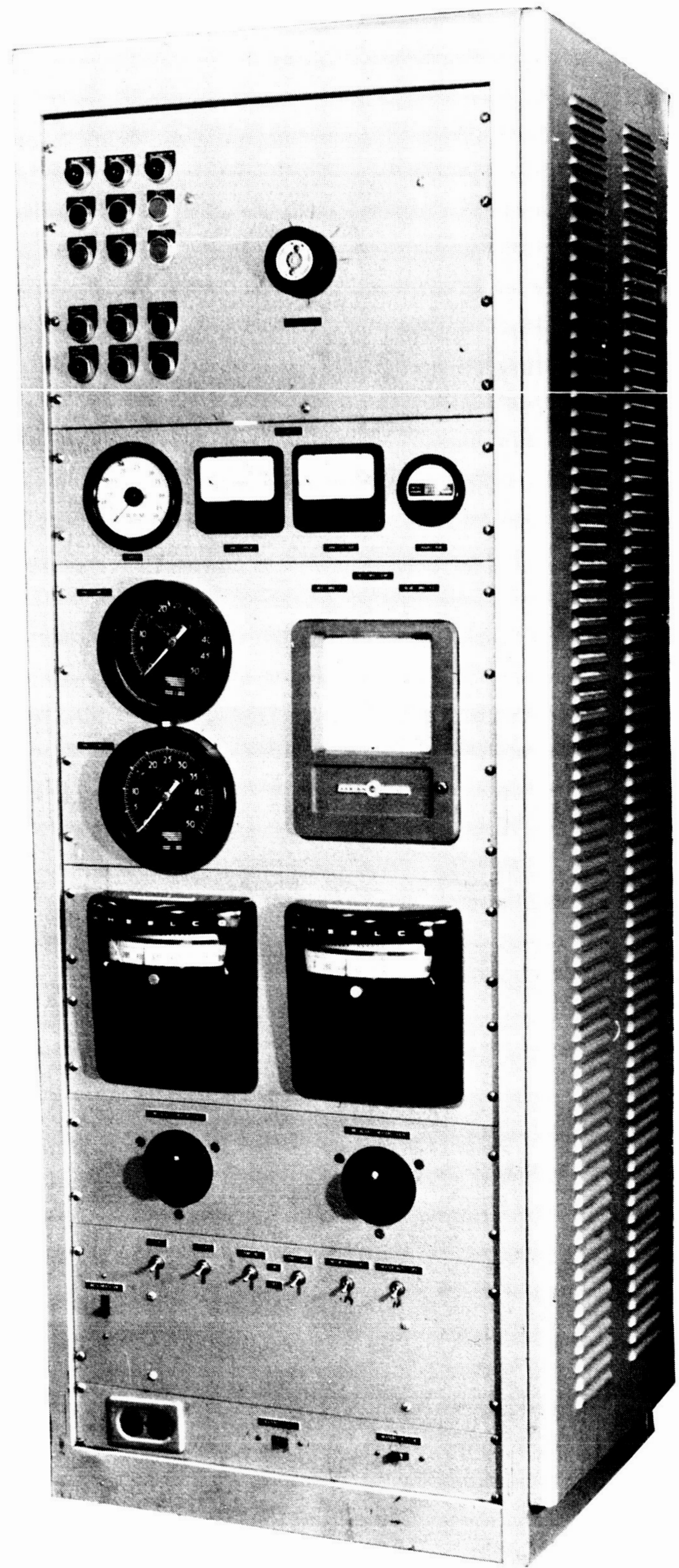
BEARING
TEMPS.

HEATER CONTROLS
1-7 8-14

FINE ADJUSTMENT
HEATERS
6-7 13-14

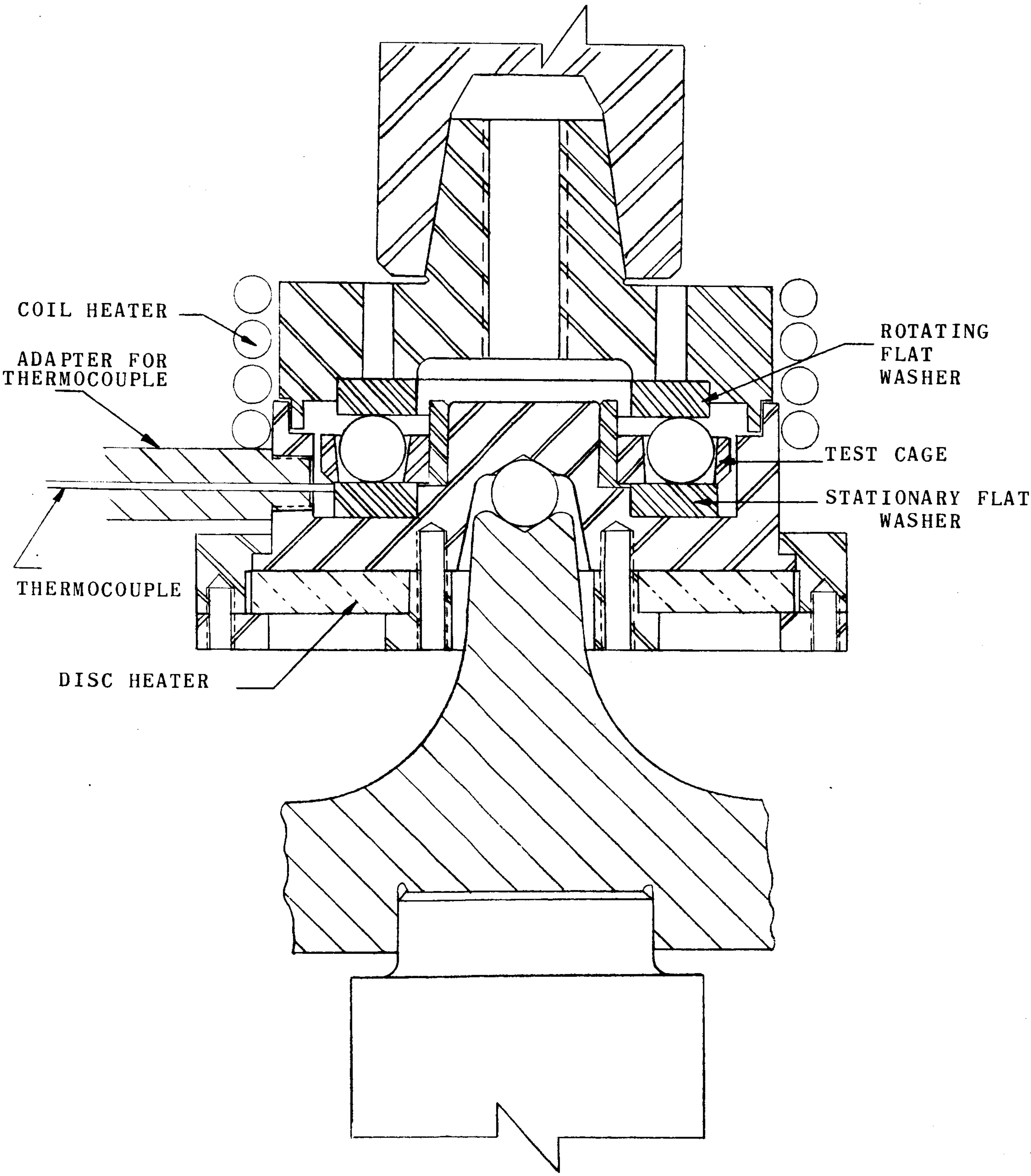
INSTRUMENT SWITCHES

HEATER SWITCHES



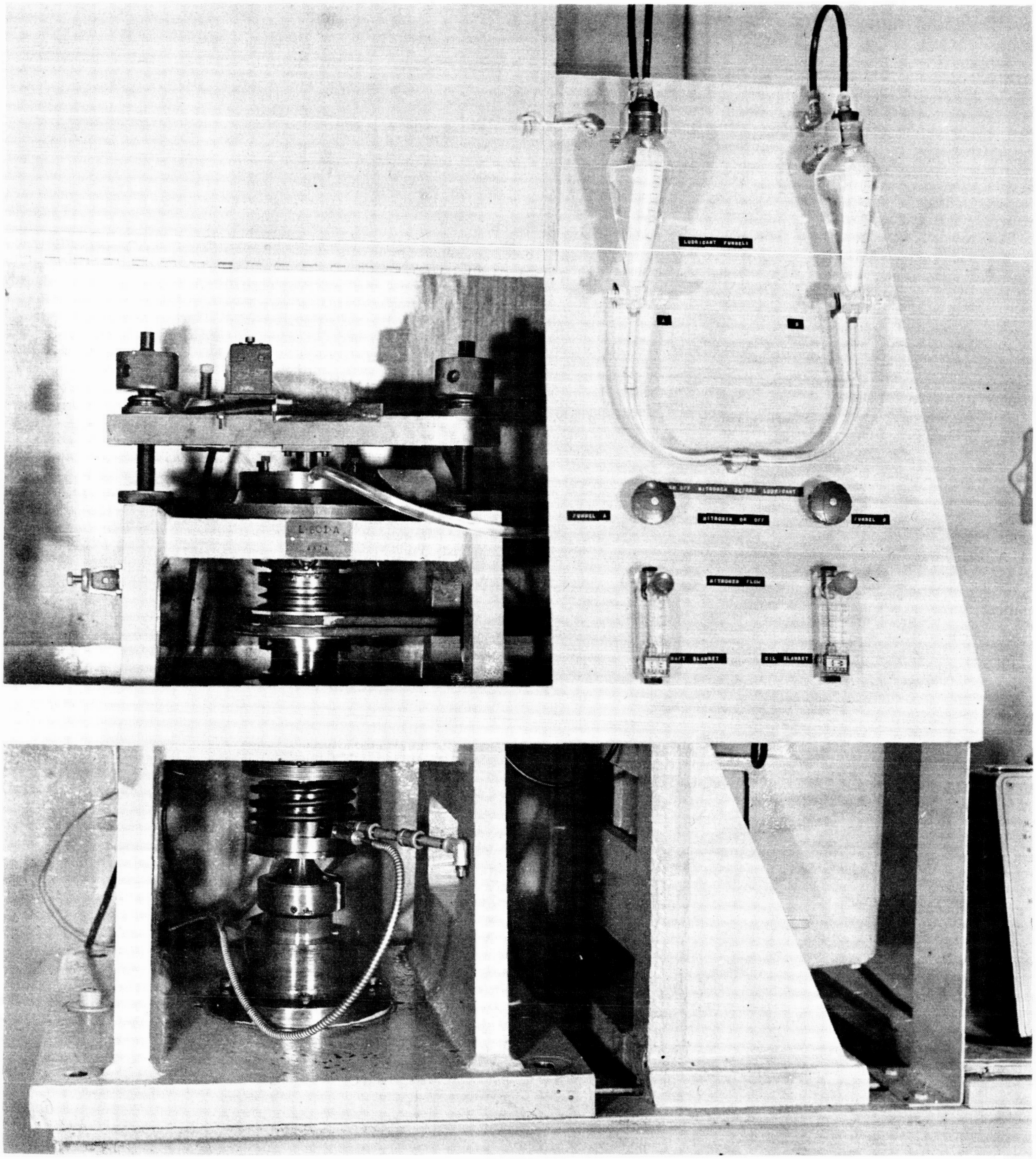
ENCLOSURE 14

CAGE WEAR STUDY TEST HEAD

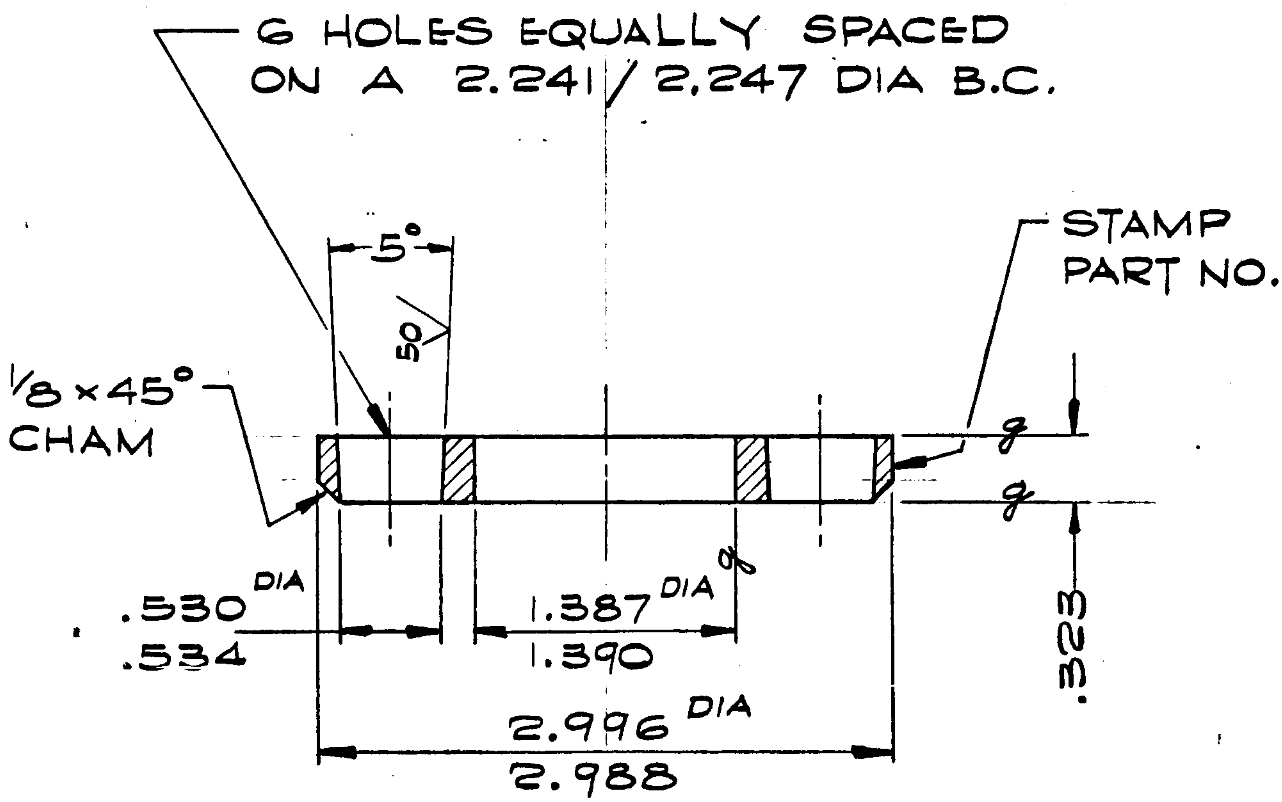


ENCLOSURE 15

CAGE COMPATIBILITY TESTER



ENCLOSURE 16



REMOVE ALL SHARP EDGES

ENCLOSURE 17

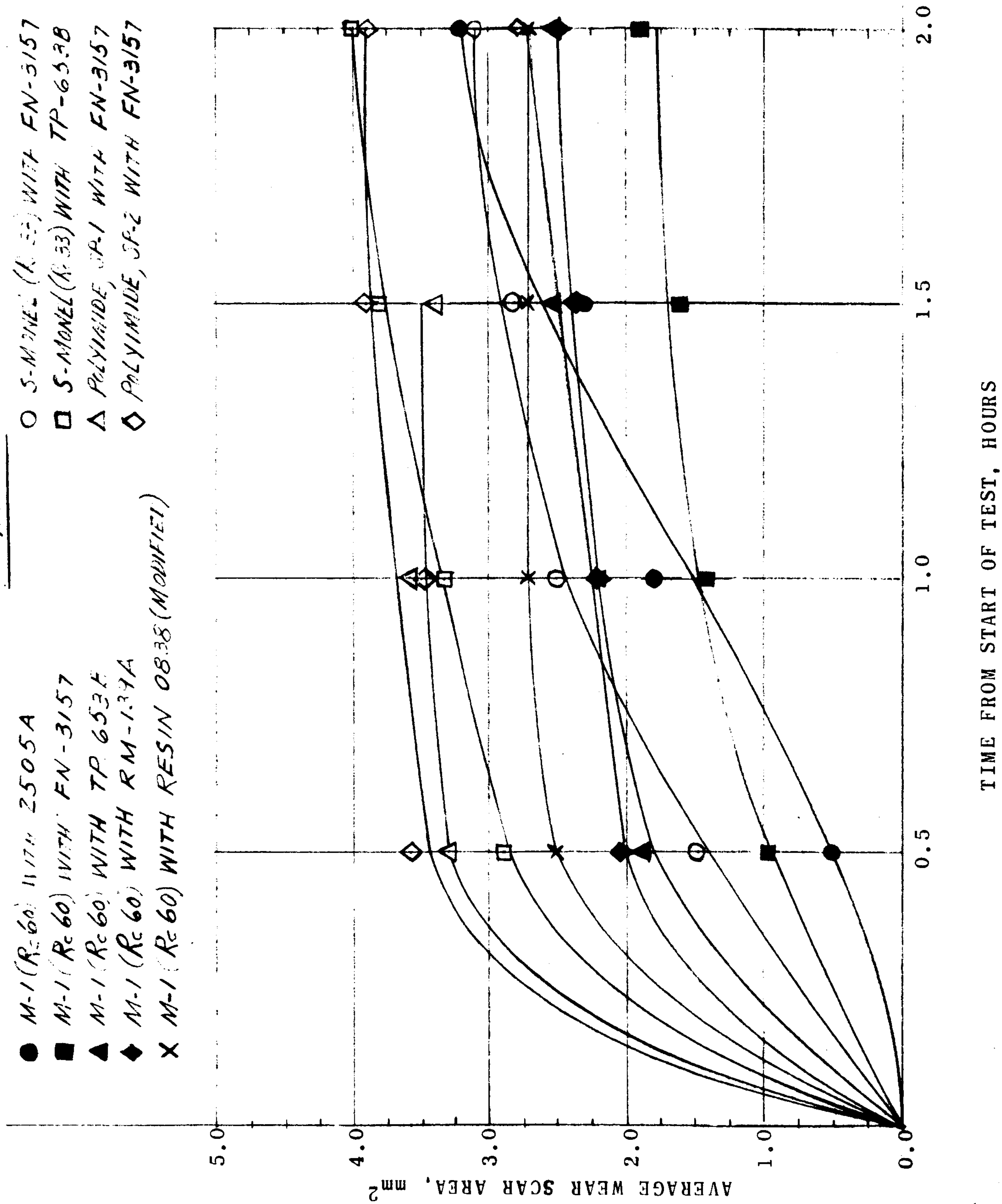
CAGE COMPATIBILITY TEST RESULTS FOR ALL LUBRICANT-CAGE MATERIAL COMBINATIONS TESTED

AVERAGE WEAR SCAR AREA (MM^2) AFTER ONE-HALF HOUR RUNNING TIME AT INDICATED TEMPERATURE (°F)

	HYDROCARBONS						ESTERS			POLYPHENYL ETHER	FLUOROCARBON
CAGE MATERIALS	FN-3157	FN-3157 MoS_2	XRM112	0838 RESIN	0846 BRIGHT STOCK	0846 B.S. & MoS_2	2505A	TP6538	RM139A	OS-124	PR-143
M-1 (RB 96)	8.1 @ 500	-	-	-	-	-	13.3 @ 500	5.6 @ 500		4.4 @ 700	-
M-1 (Rc 40)	4.4 @ 500 11.4 @ 700	-	7.5 @ 700	4.5 @ 500 2.2 @ 700	-	-	1.31 @ 500	2.9 @ 500	3.4 @ 500	6.4 @ 700	-
M-1 (Rc 60)	1.0 @ 500 2.6 @ 700	7.3 @ 700	6.9 @ 700	2.5 @ 500 1.0 @ 700	-	-	0.1 @ 500 4.4 @ 700	1.9 @ 500	2.1 @ 500	1.6 @ 700	1.4 @ 700
S-MONEL (RB 85)	5.2 @ 500	-	11.2 @ 700	2.5 @ 500 1.5 @ 700	-	-	5.5 @ 500	-	3.4 @ 500	3.1 @ 700	-
S-MONEL (Rc 33)	1.5 @ 500 7.2 @ 700	-	6.2 @ 700	-	0.9 @ 700	1.2 @ 700	5.1 @ 500 3.1 @ 700	2.9 @ 500	3.8 @ 500	0.62 @ 700	6.8 @ 700
HAYNES 25 (Rc 50)	9.2 @ 500	-	-	-	-	-	17.3 @ 500	7.4 @ 500	-	7.8 @ 700	-
HIDUREL (RB 80)	10.5 @ 500	-	-	-	-	-	6.5 @ 500	11.7 @ 500	-	5.8 @ 700	-
POLYIMIDE, SP-1	3.3 @ 500	-	-	-	-	-	-	-	-	12.2 @ 700	-
POLYIMIDE, SP-2	3.6 @ 500	-	-	-	-	-	-	-	-	-	-
440 CM (Rc 52)	1.8 @ 500	-	-	-	-	-	-	-	-	4.4 @ 700	-
440 CM (Rc 57)	3.4 @ 500	-	-	-	-	-	-	-	-	3.5 @ 700	-

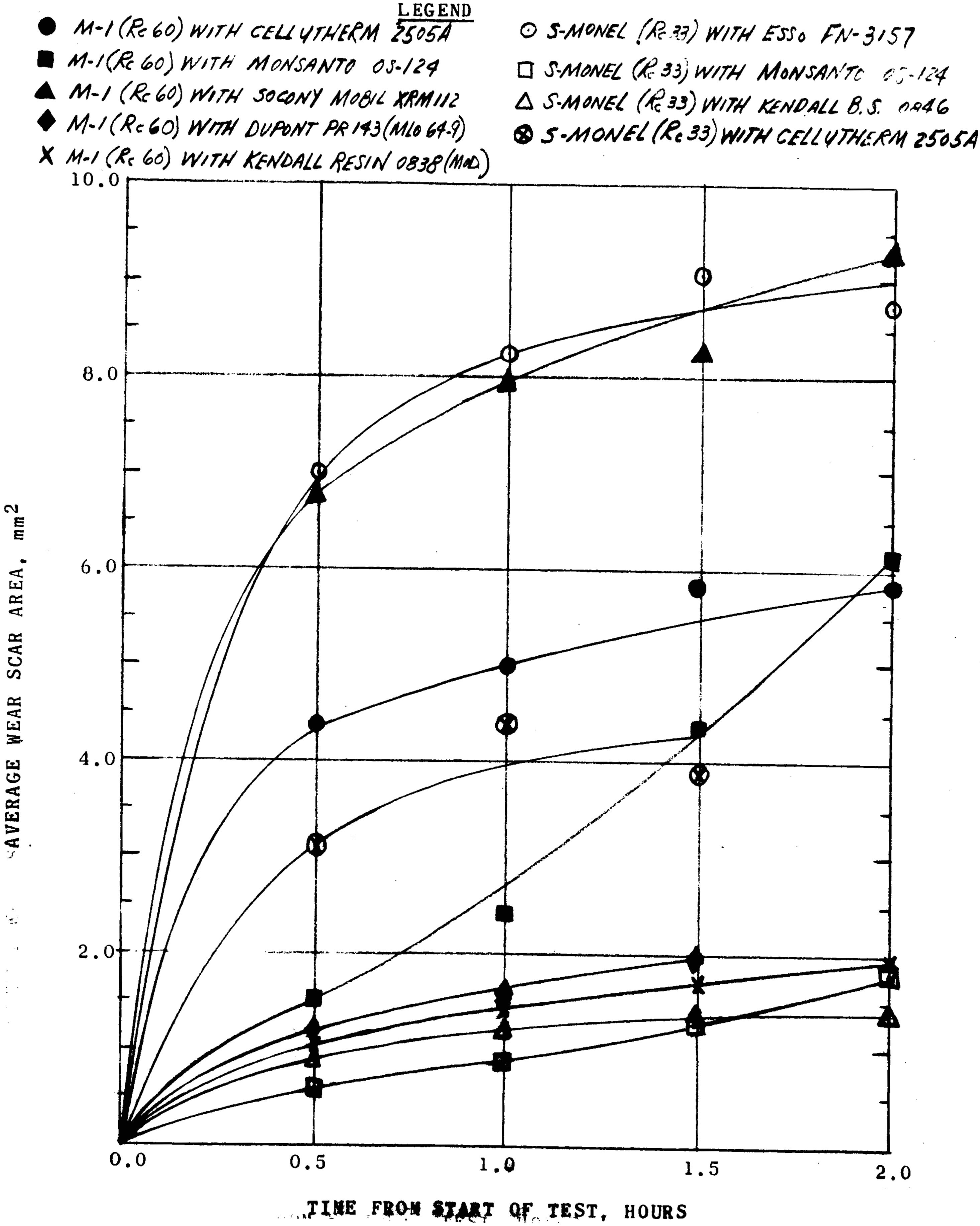
ENCLOSURE 18

CAGE COMPATIBILITY TEST RESULTS OF THE MOST WEAR RESISTANT MATERIAL COMBINATIONS AT 500°F, 1000 LBS. LOAD AND 1200 RPM



ENCLOSURE 19

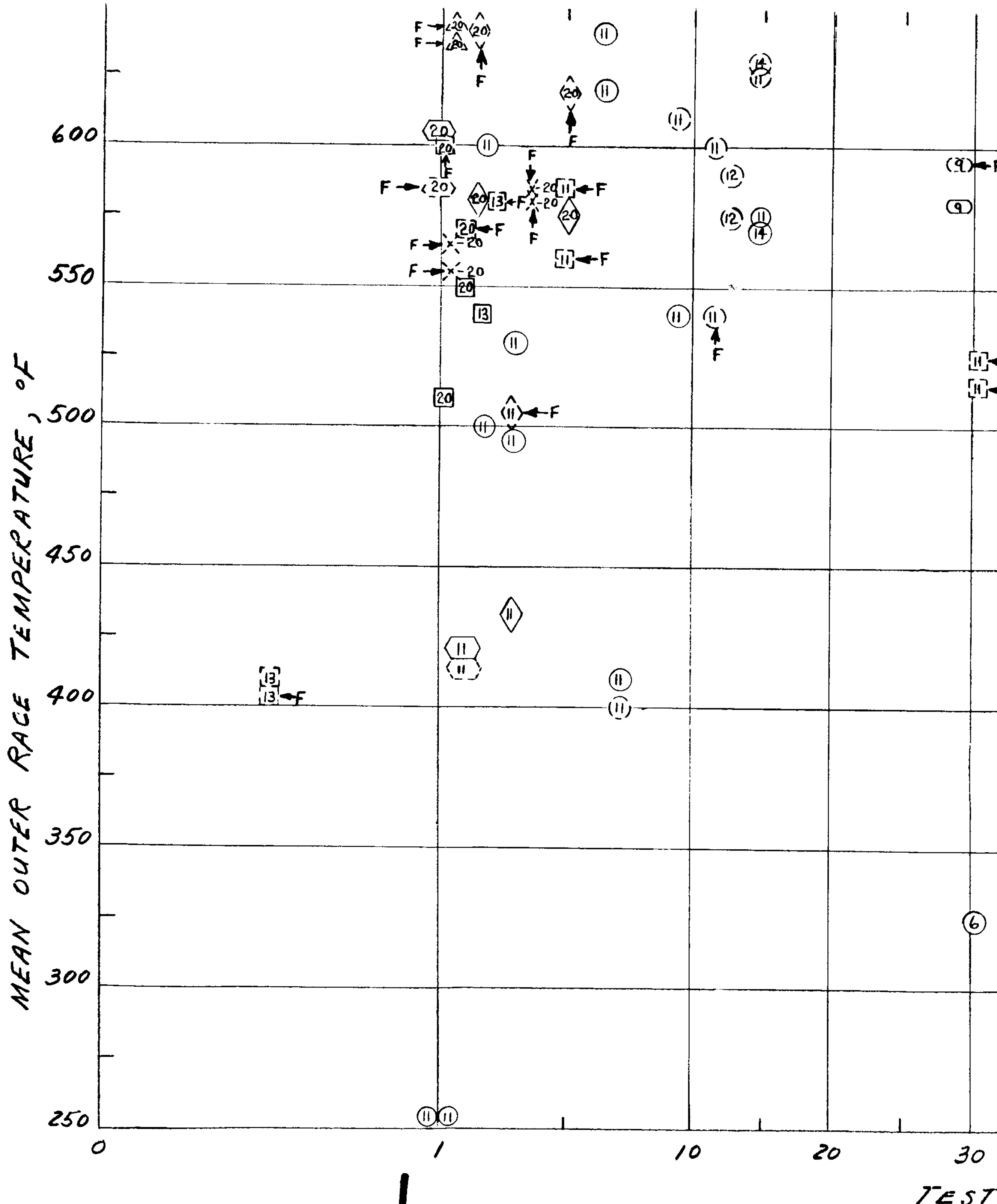
CAGE COMPATIBILITY TEST RESULTS OF THE MOST WEAR RESISTANT MATERIAL COMBINATIONS AT 700°F, 1000 LBS. LOAD AND 1200 RPM



RESEARCH LABORATORY SKF INDUSTRIES, INC.

ENCLOSURE

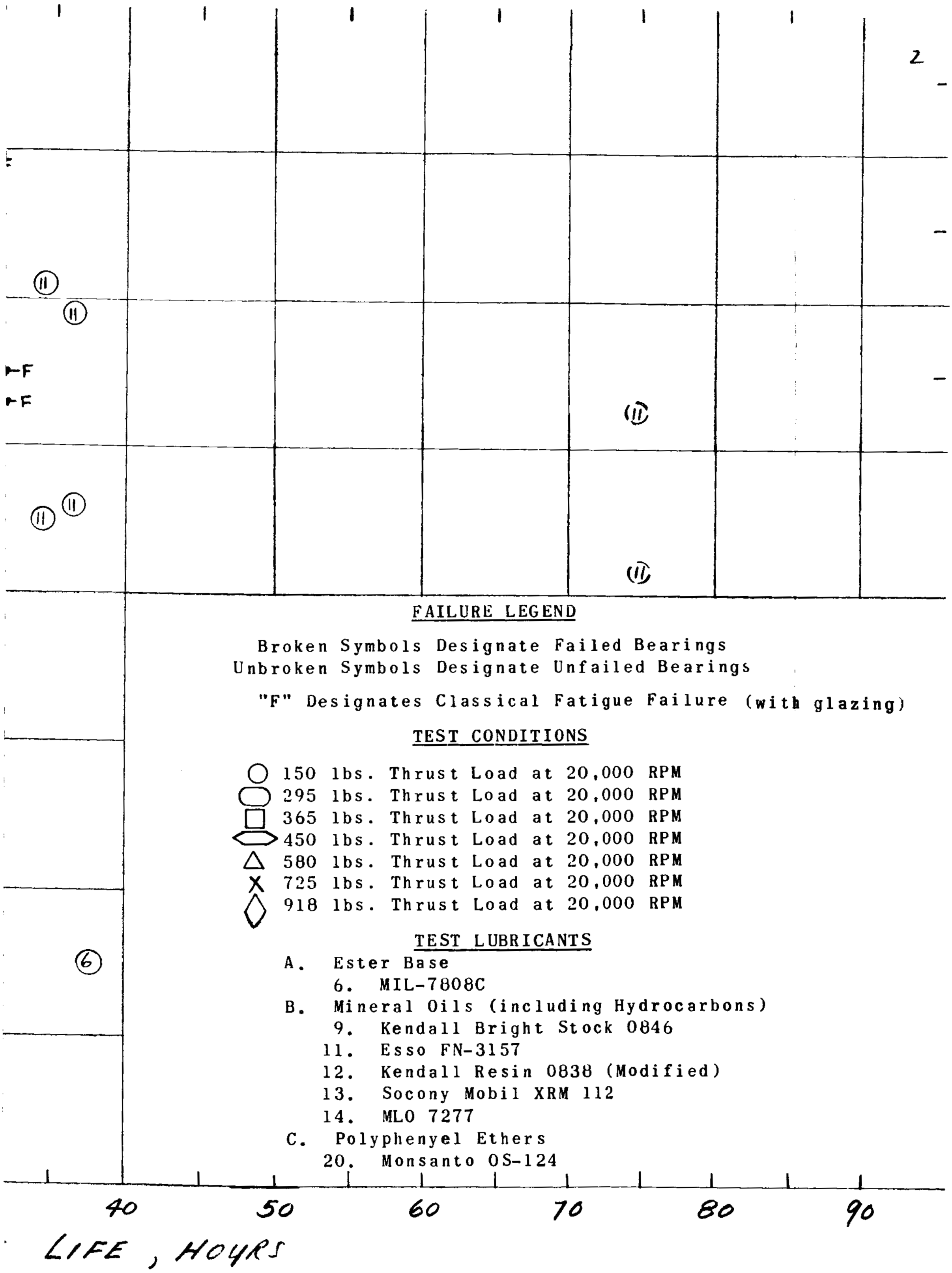
SUMMARIZED TEST RESULTS OF CV
STEEL BEARINGS AT SPEEDS UP TO 20,
680°F AND THRUST LOADS



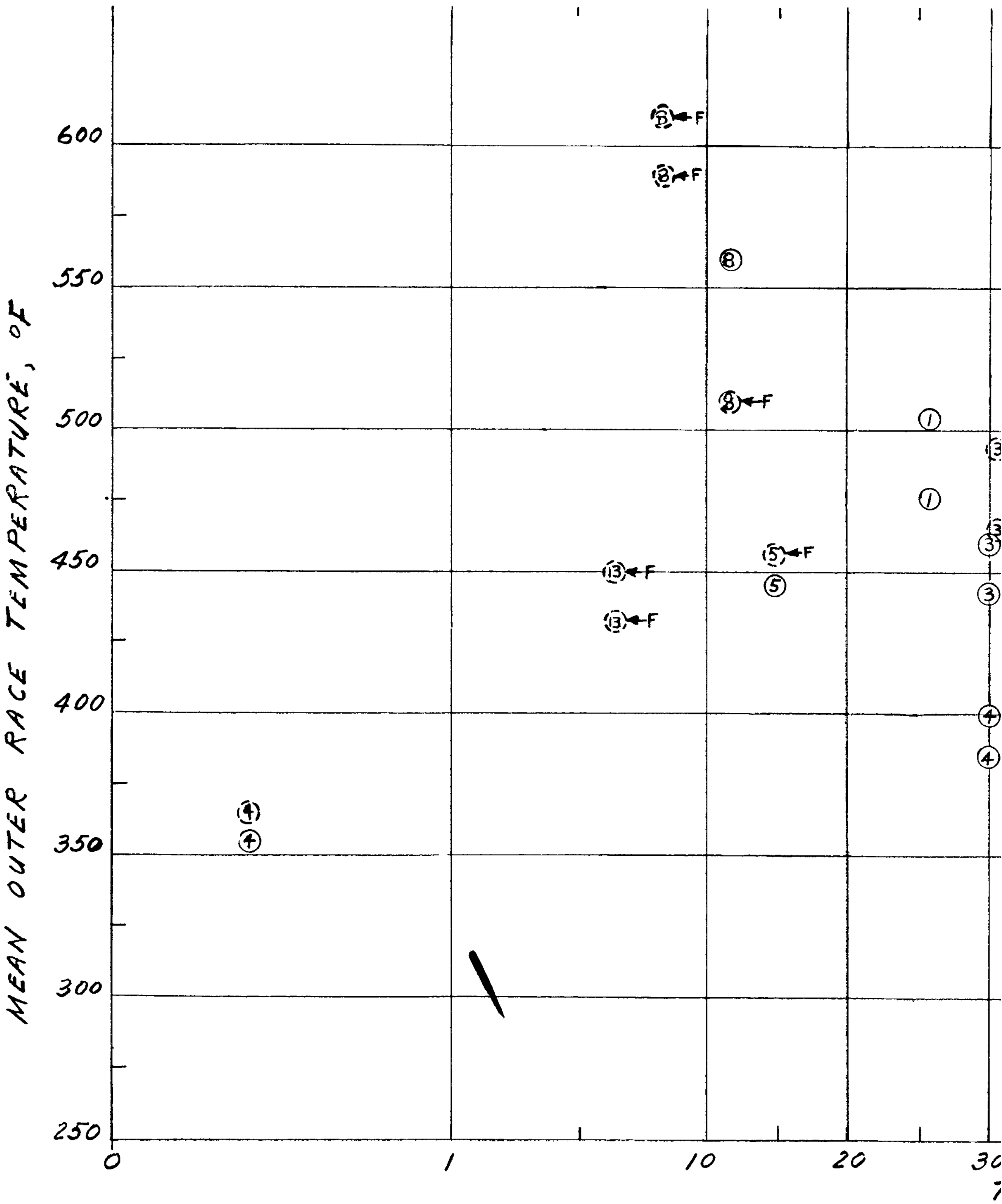
RESEARCH LABORATORY SK

20

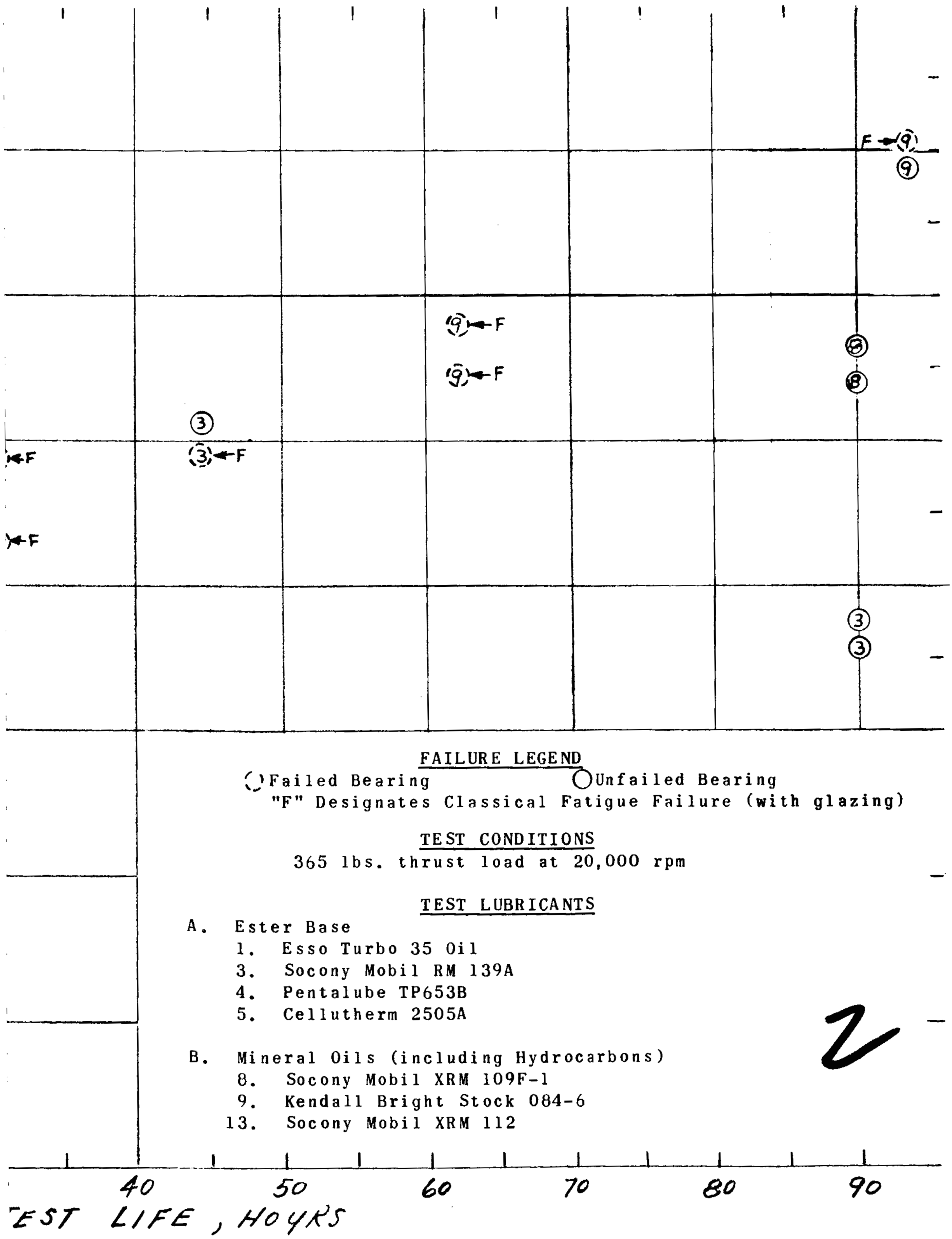
'M M-1 (#455 760) TOOL
000 RPM. TEMPERATURES UP TO
; UP TO 918 LBS.

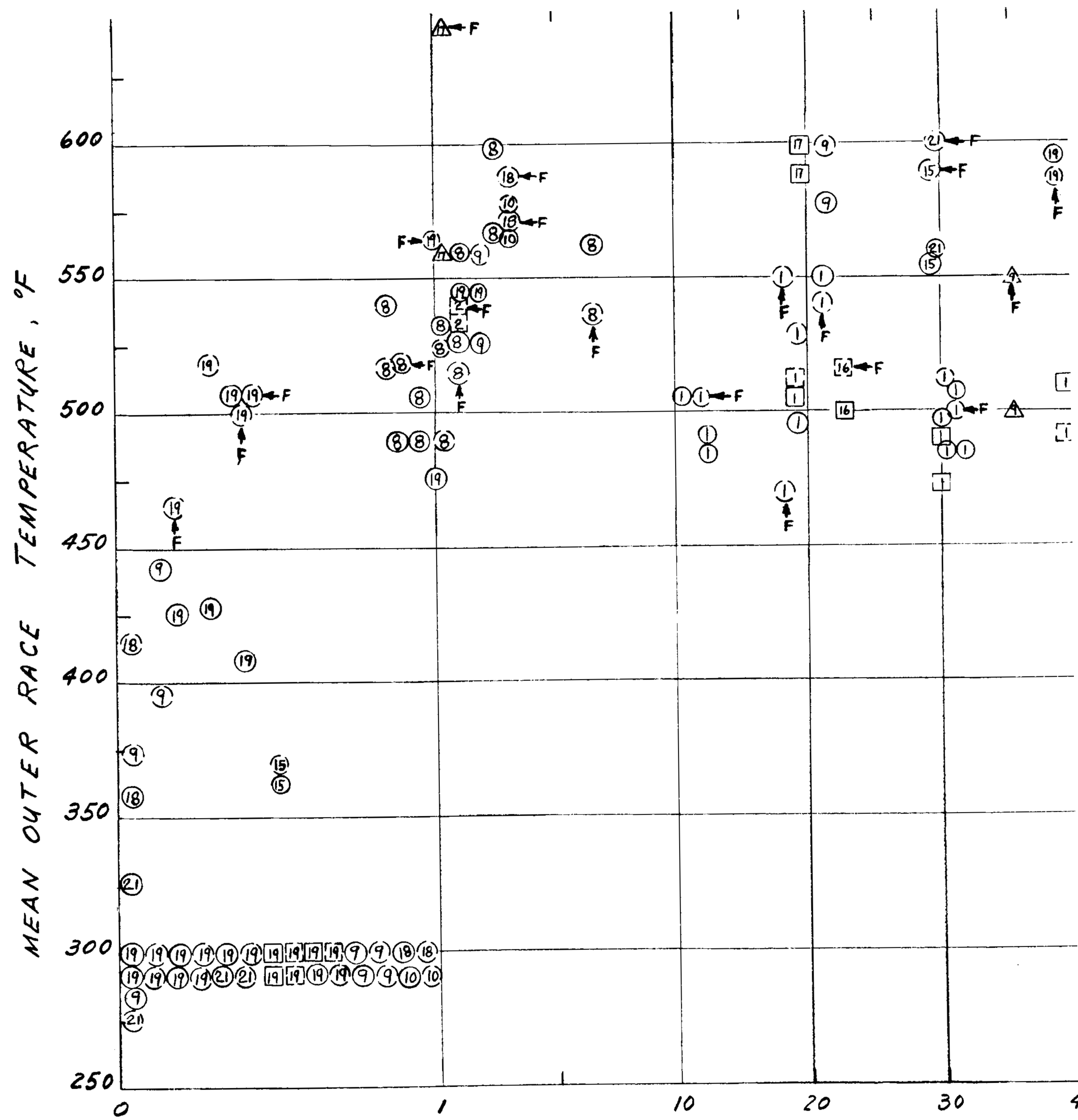


SUMMARIZED TEST RESULTS OF CV
STEEL BEARINGS AT SPEEDS UP TO 20,
610°F and THRUST LOADS



M WB49 (#456 684) TOOL
000 RPM. TEMPERATURES UP TO
OF 365 LBS.

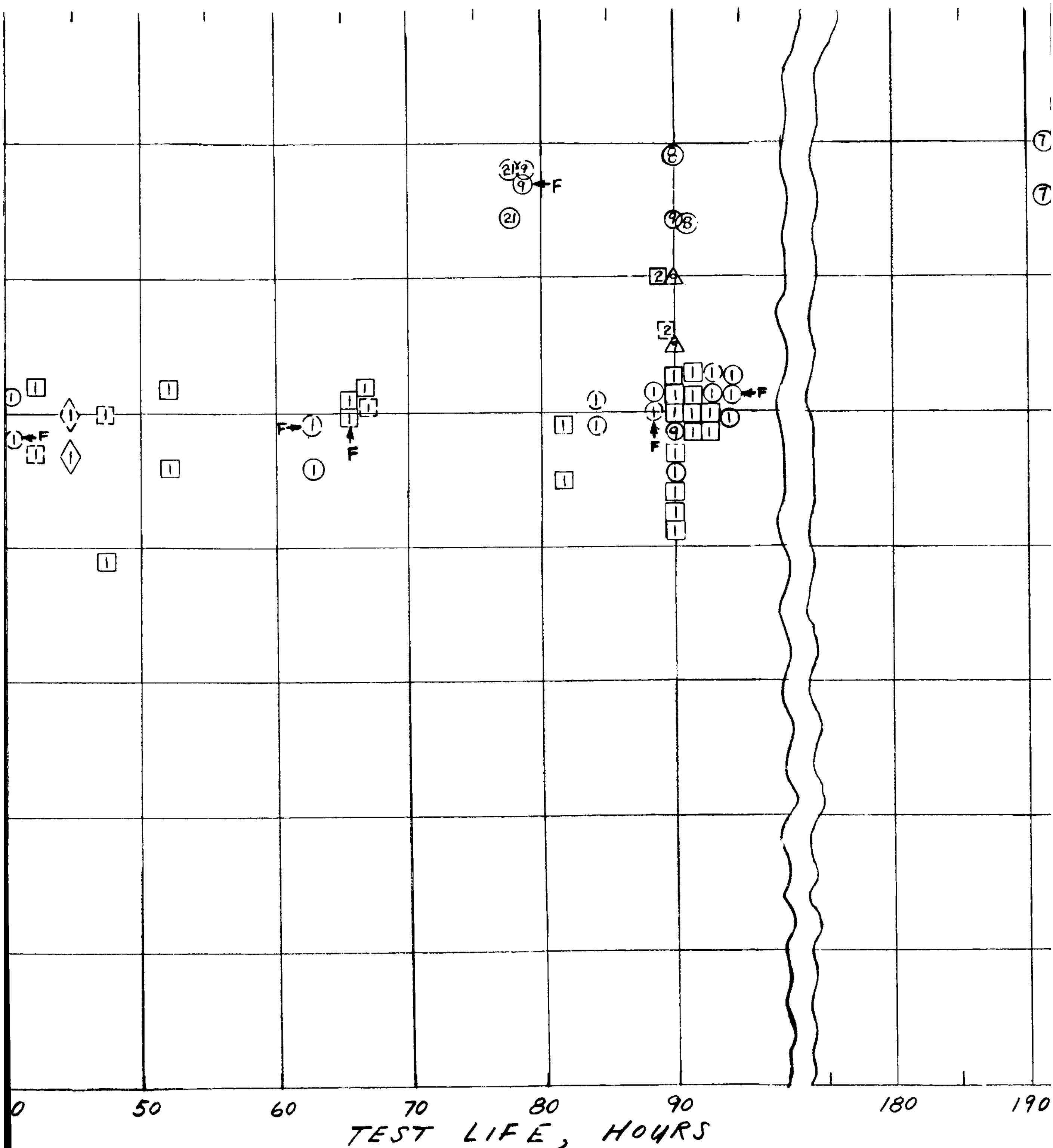


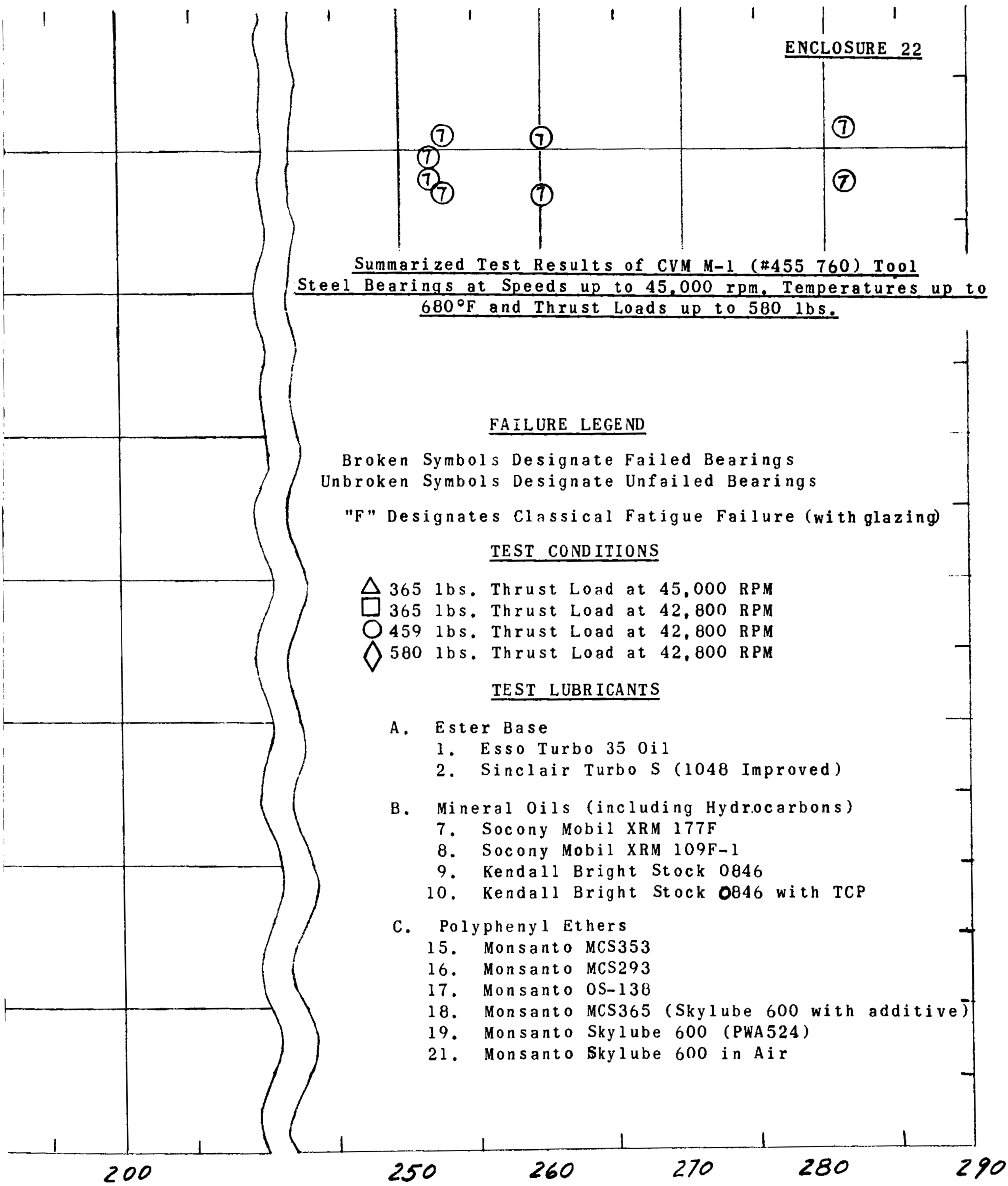


MEAN OUTER RACE TEMPERATURE, °F
600
550
500
450
400
350
300
250
0
1
10
20
30
F

ENCLOSURE 22

SUMMARIZED TEST RESULTS OF CVM M-1 (#455 760) TOOL STEEL BEARINGS AT SPEEDS UP TO 45,000 RPM. TEMPERATURES UP TO 680°F AND THRUST LOADS UP TO 580 LBS.





ENCLOSURE 22

Summarized Test Results of CVM M-1 (#455 760) Tool Steel Bearings at Speeds up to 45,000 rpm, Temperatures up to 680°F and Thrust Loads up to 580 lbs.

FAILURE LEGEND

Broken Symbols Designate Failed Bearings
Unbroken Symbols Designate Unfailed Bearings

"F" Designates Classical Fatigue Failure (with glazing)

TEST CONDITIONS

△ 365 lbs. Thrust Load at 45,000 RPM
□ 365 lbs. Thrust Load at 42,800 RPM
○ 459 lbs. Thrust Load at 42,800 RPM
◇ 580 lbs. Thrust Load at 42,800 RPM

TEST LUBRICANTS

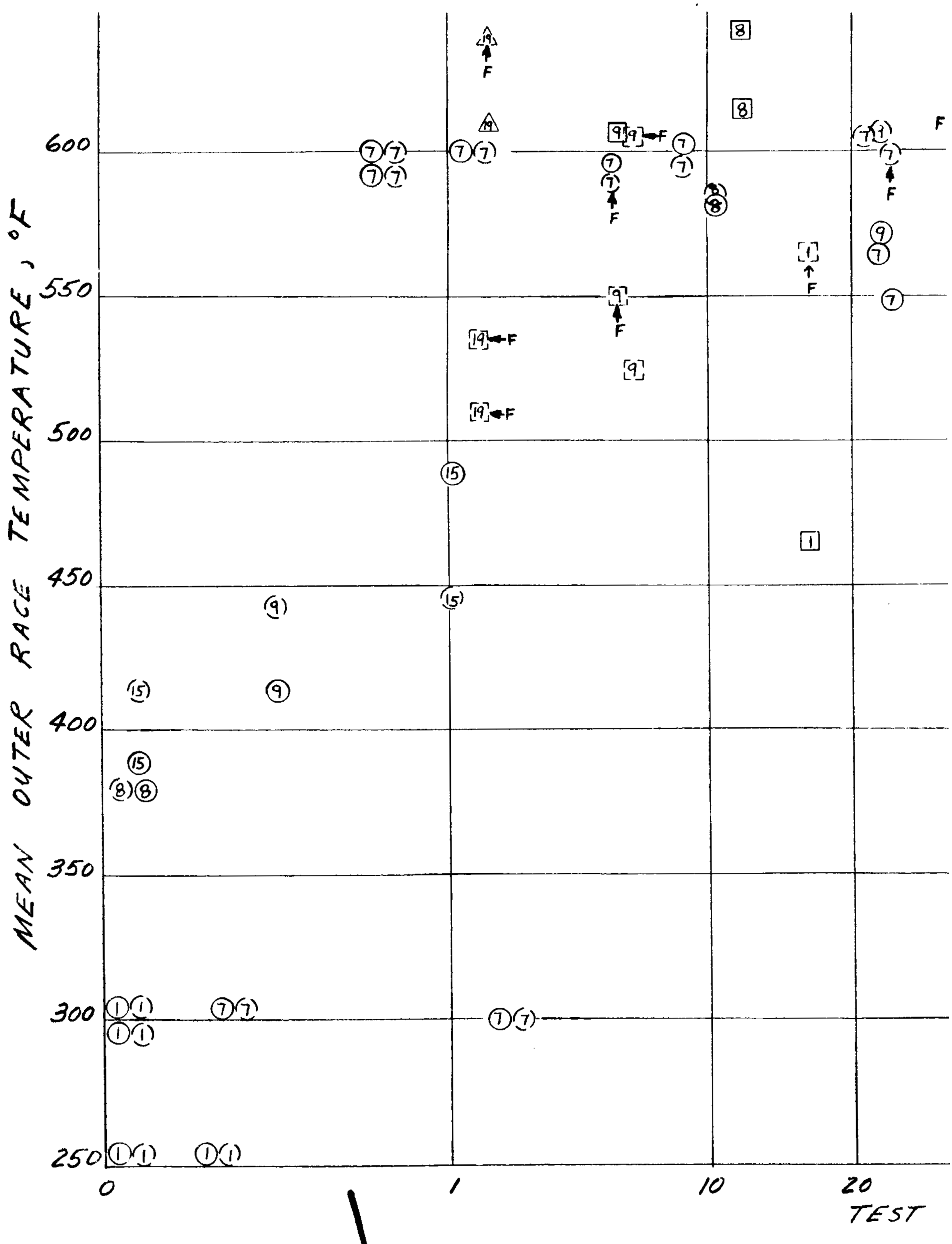
A. Ester Base
   1. Esso Turbo 35 Oil
   2. Sinclair Turbo S (1048 Improved)

B. Mineral Oils (including Hydrocarbons)
   7. Socony Mobil XRM 177F
   8. Socony Mobil XRM 109F-1
   9. Kendall Bright Stock 0846
   10. Kendall Bright Stock 0846 with TCP

C. Polyphenyl Ethers
   15. Monsanto MCS353
   16. Monsanto MCS293
   17. Monsanto OS-138
   18. Monsanto MCS365 (Skylube 600 with additive)
   19. Monsanto Skylube 600 (PWA524)
   21. Monsanto Skylube 600 in Air

2

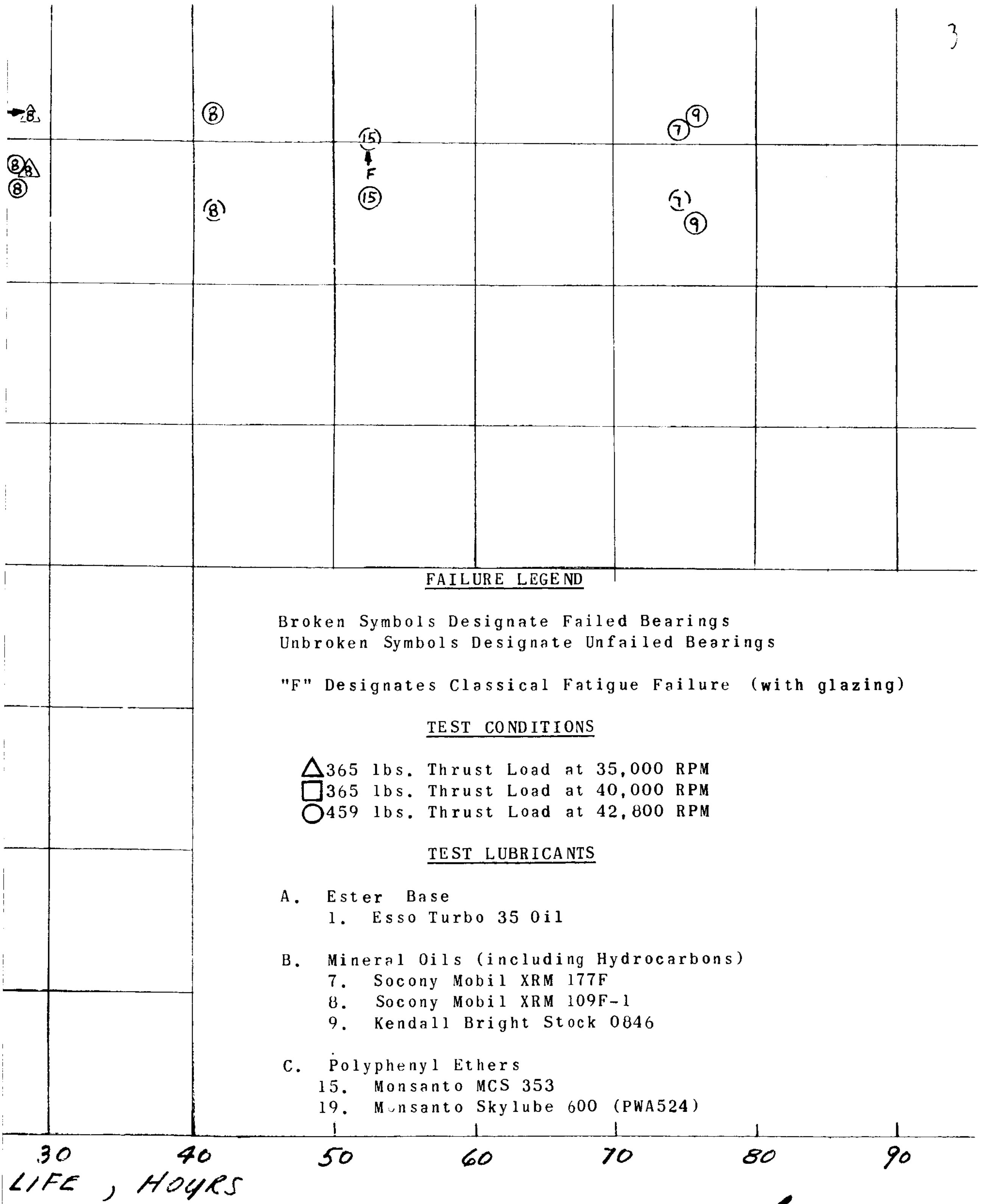
SUMMARIZED TEST RESULTS
STEEL BEARINGS AT SPEEDS UP T
642°F AND THRUST



RESEARCH LABORATORY

SURE 23

OF CVM WB49 (#456 684) TOOL
0 42,800 RPM. TEMPERATURES UP TO
LOADS UP TO 459 LBS.



FAILURE LEGEND

Broken Symbols Designate Failed Bearings
Unbroken Symbols Designate Unfailed Bearings

"F" Designates Classical Fatigue Failure (with glazing)

TEST CONDITIONS

△365 lbs. Thrust Load at 35,000 RPM
□365 lbs. Thrust Load at 40,000 RPM
○459 lbs. Thrust Load at 42,800 RPM

TEST LUBRICANTS

A. Ester Base
   1. Esso Turbo 35 Oil

B. Mineral Oils (including Hydrocarbons)
   7. Socony Mobil XRM 177F
   8. Socony Mobil XRM 109F-1
   9. Kendall Bright Stock 0846

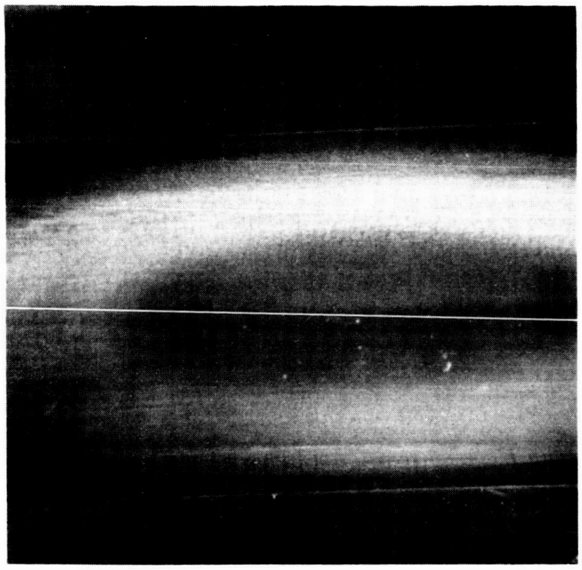
C. Polyphenyl Ethers
   15. Monsanto MCS 353
   19. Monsanto Skylube 600 (PWA524)

V

SKF INDUSTRIES, INC.

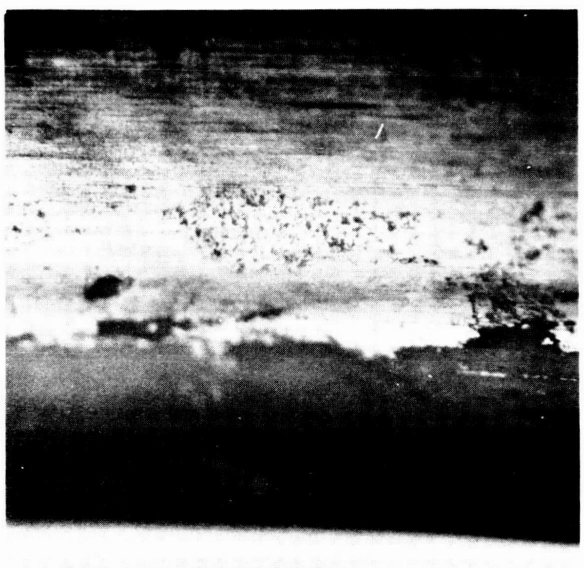
ENCLOSURE 24

TYPICAL APPEARANCE OF TESTED BEARING RACES SHOWING NORMAL, GLAZED, GLAZED AND PITTED, AND GLAZED AND FLAKED CONDITIONS



a. Normal (459#, 42,800 rpm, Bright Stock 0846, 500°F, 231.1 mill. revs.)

b. Glazed (365# load, 20,000 rpm, OS-124, 555°F, 2.2 mill. revs.)

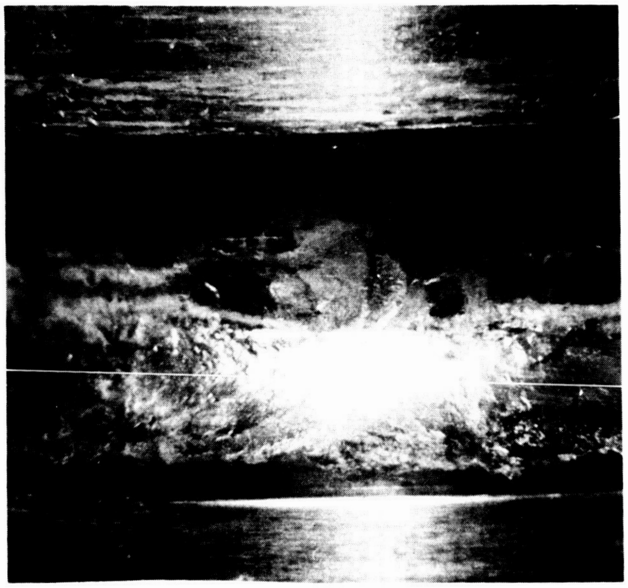


c. Glazed and Pitted (150# load, 20,000 rpm, FN-3157, 570°F, 78.4 mill. revs.)

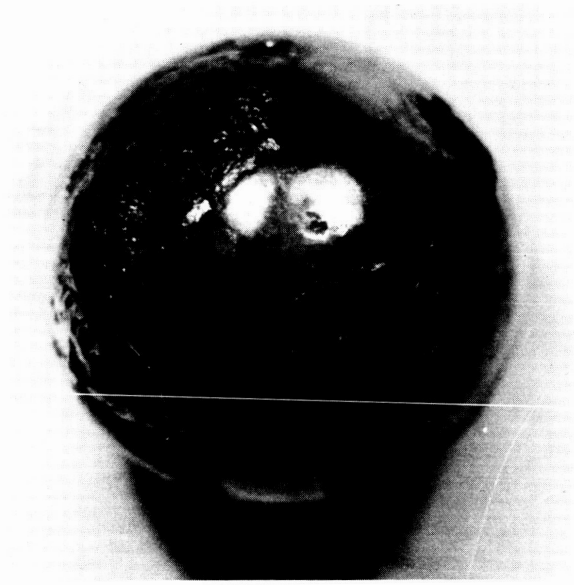
d. Glazed and Flaked (365# load, 20,000 rpm, RM-139A, 495°F, 35.2 mill. revs.)

ENCLOSURE 25

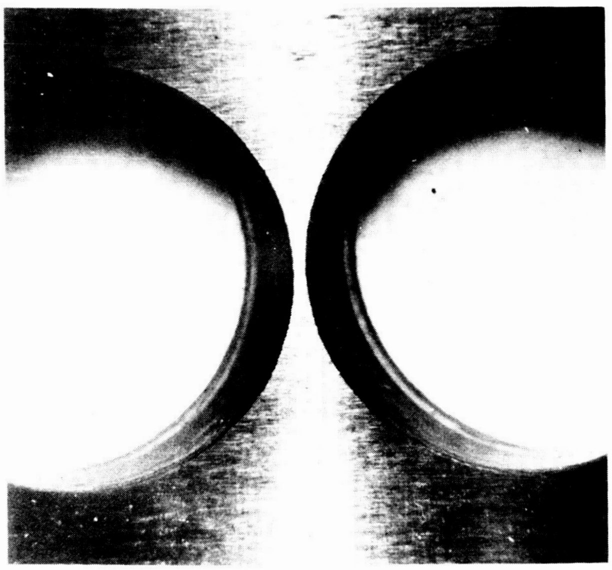
TYPICAL APPEARANCE OF TESTED BEARINGS SHOWING SMEARING, BALL FLAKING, AND CAGE WEAR



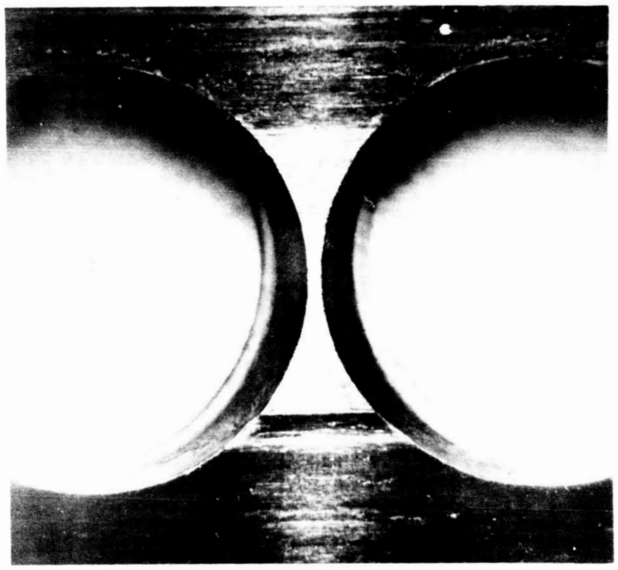
a. Smearing (365# load, 20,000 rpm, XRM 112, 410°F, 0.6 mill. revs.)



b. Flaked Ball (365# load, 20,000 rpm, Bright Stock 0846, 522°F, 74.2 mill. revs.)



c. Cage (No wear) (365# load, 20,000 rpm, RM 139A, 440°F, 36.0 mill. revs.)



d. Cage (Showing wear) (365# load, 20,000 rpm, 2505A, 455°F, 17.9 mill. revs.)

ENCLOSURE 26

LIMITING TEMPERATURE AND LUBRICATING CAPABILITIES OF CANDIDATE LUBRICANTS IN 7205 BEARINGS UNDER 365 LBS. THRUST LOAD

(ALL TESTS CONDUCTED IN N_2 ENVIRONMENT EXCEPT WHERE NOTED)

LUBRICANT	TEST SPEED, 10^3 RPM	MAXIMUM TEMPERATURE TESTED WITH THE LEAST SURFACE DISTRESS, °F	BEARING CONDITION AFTER TEST	ESTIMATED MINIMUM VISCOSITY FOR SATISFACTORY BEARING OPERATION AT TEMP., CS	MINIMUM COMPUTED SAFE RATIO OF EHD FILM THICKNESS TO COMPOSITE SURFACE ROUGHNESS* (h/σ)
HYDROCARBONS:					
ESSO FN-3157	20	500	GLAZED	> 1.07	> 0.54
KENDALL RESIN 0838 (MODIFIED)	20	580	SMEARED & COKED	> 2.60	> 1.09
SOCONY MOBIL XRM 112	20	450	GLAZED & FLAKED	> 0.94	> 0.35 - 0.43
KENDALL BRIGHT STOCK 0846	42.8	580	SERVICEABLE	1.51	1.28 - 2.08
SOCONY MOBIL XRM 109F-1	42.8	590	SLIGHTLY GLAZED	2.45	1.68 - 2.10
SOCONY MOBIL XRM 177F	42.8	610	SERVICEABLE	< 2.10	< 2.01 - 2.94
POLYPHENYL ETHERS:					
MONSANTO OS-124	20	550	GLAZED	> 1.05	> 0.37
MONSANTO OS-124	42.8	600	SLIGHT GLAZING	1.15	0.97
MONSANTO SKYLUBE 600	42.8	540	GLAZED	> 1.08	> 0.69
MONSANTO SKYLUBE 600 (IN AIR)	42.8	570	SLIGHTLY GLAZED	0.94	0.82
MONSANTO MCS 365	42.8	570	GLAZED & FLAKED	> 0.94	> 0.91
MONSANTO MCS 293	42.8	500	SLIGHTLY GLAZED	0.78	0.74
MONSANTO MCS 353	42.8	585	SUPERFICIALLY PITTED	> 0.70	> 0.62 - 0.97
ESTERS:					
CELLUTHERM 2505A	20	445	GLAZED	> 1.0	> 0.45
HEYDEN NEWPORT TP653B	20	400	SLIGHTLY GLAZED	1.03	0.45
SOCONY MOBIL RM 139A	20	440	SERVICEABLE	1.08	0.45
ESSO TURBO OIL 35	42.8	515	SERVICEABLE	1.36	1.08 - 1.77
SINCLAIR TURBO S (TYPE 1048)	42.8	550	SLIGHTLY GLAZED	1.26	1.03

*AT INNER RING BALL CONTACT. RANGE OF h/σ VALUES APPLIES TO TEST BEARINGS HAVING DIFFERENT MEASURED SURFACE ROUGHNESS.

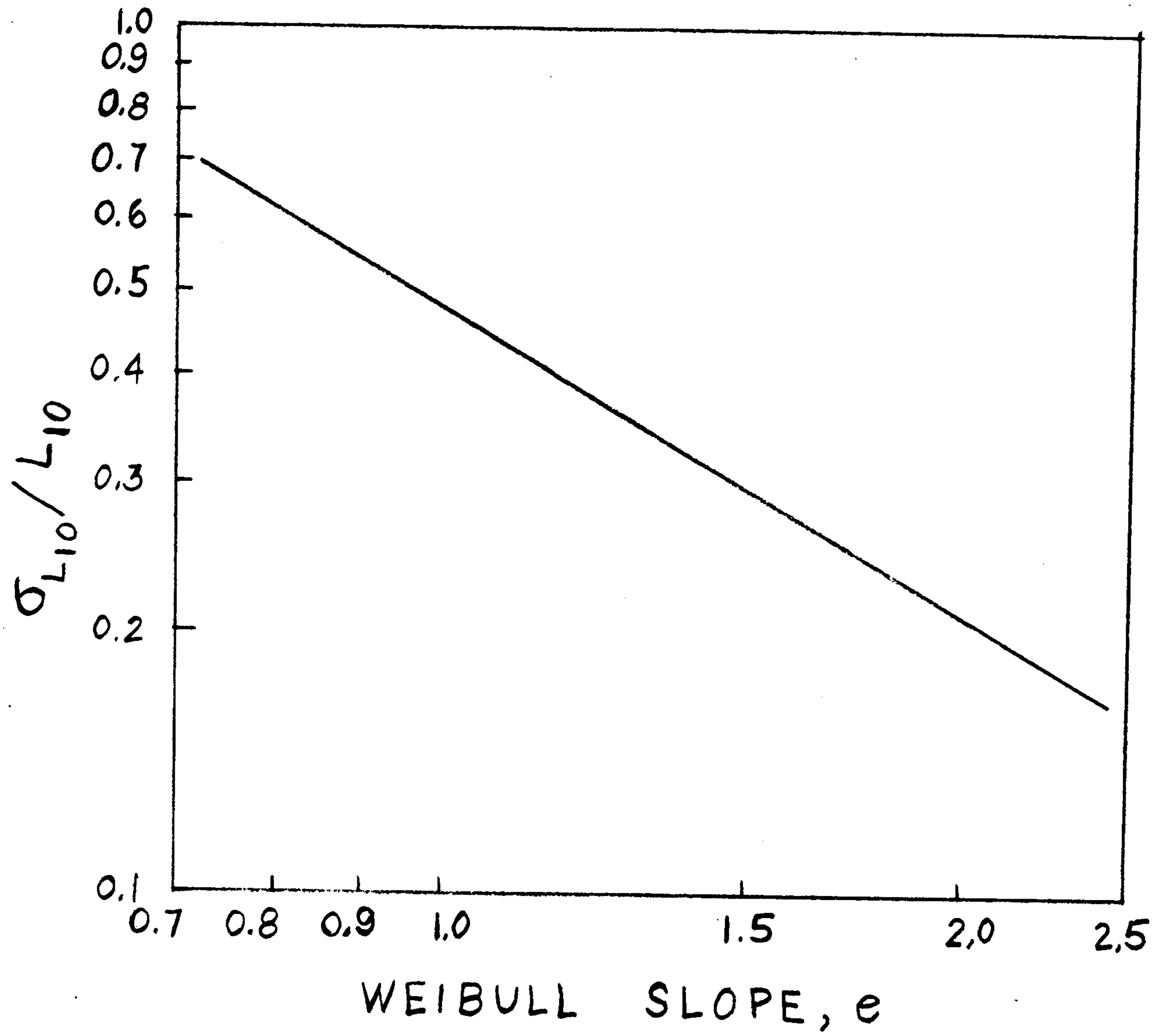
ENCLOSURE 27

UNFAILED CVM M-1 STEEL BEARINGS AFTER RUNNING 726.5×10^6 REVOLUTIONS AT 42,800 RPM, A MEAN TEMPERATURE UP TO 601 °F AND UNDER 459 LBS. THRUST LOAD WITH CIRCULATING SOCONY MOBIL XRM 177F OIL IN AN N_2 BLANKET

BEARING NO. 391 ON LOAD END FROM RUN NO. E82

BEARING NO. 392 ON DRIVE END FROM RUN NO. E82

ENCLOSURE 28



STANDARD ERROR OF MAXIMUM LIKELIHOOD ESTIMATES OF L_{10} FROM A TRUNCATED SAMPLE OF SIZE 30

APPENDIX (Cont.)

Surface Tension, dynes/cm.	17.8	-*	-*
Foaming (ASTM D-892 mod.), ml. foam/min. to collapse @ 75°F	10/0	-*	-*
200°F	0/0	-*	-*
75°F	5/0	-*	-*
Volatility (ASTM D-972 mod.), % @ 500°F in 6-1/2 hours	4.5	2.4	1.4
Volatility, % by Thermogravimetric Analysis @ 464°F	0	0	0
680°F	10	7.8	5.5
739°F	25	18.0	16.2
792°F	50	39.6	34.6
829°F	75	65.3	56.0
889°F	100	99.4	93.2
Linear Decomposition Rate, %/day @ 790°F by Isoteniscope	13.1	12.7	4.9
Ultraviolet Spectrum, 10 mm. cell	Transparent above 2100 Å		
Infrared Spectrum	Typical of a nonfunctional, highly fluorinated compound		
F^-, p.p.m.	<1	<1	<1
Acidity, p.p.m. HF	3	<1	1

*Not expected to differ from PR-143 Lot 1.

4. Ester Base Lubricants

a. Celanese Cellutherm 2505A

Description: Trimethylopropane Esters (Meets MIL-L-9236B)
Viscosity, cs

@ -65°F	19000
@ 100°F	16.0
@ 210°F	3.55
@ 400°F	1.25

Flash Point, °F (Min.) 460