NASA TECHNICAL NOTE

NASA TN D-5420

c.1

NASA TN D-5420

LUBRICANTS FOR INERTED LUBRICATION SYSTEMS IN ENGINES FOR ADVANCED AIRCRAFT

by William R. Loomis, Dennis P. Townsend, and Robert L. Johnson

Lewis Research Center
Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • SEPTEMBER 1969

LUBRICANTS FOR INERTED LUBRICATION SYSTEMS IN ENGINES FOR ADVANCED AIRCRAFT

By William R. Loomis, Dennis P. Townsend, and Robert L. Johnson

Lewis Research Center
Cleveland, Ohio

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

For sale by the Clearinghouse for Federal Scientific and Technical Information
Springfield, Virginia 22151 – CFSTI price $3.00

ABSTRACT

The feasibility of using presently available liquid lubricants in advanced high-speed aircraft engines was studied with both nitrogen inerted recirculating and once-through mist lubrication systems. Three fluids (a mixed ester, a synthetic paraffinic hydrocarbon, and a perfluorinated polymeric fluid) performed satisfactorily with the recirculating system in 3- to 10-hour runs in a full-scale simulated aircraft bearing and seal assembly at outer race bearing temperatures to 700° F (644 K) and with a bulk fluid temperature of 500° F (533 K). The principal problem was excessive leakage of the oil-side bellows face seal. Only one lubricant (synthetic paraffinic hydrocarbon) performed satisfactorily with the once-through mist system at a bearing temperature of 600° F (589 K).

LUBRICANTS FOR INERTED LUBRICATION SYSTEMS IN ENGINES FOR ADVANCED AIRCRAFT

by William R. Loomis, Dennis P. Townsend, and Robert L. Johnson

Lewis Research Center

SUMMARY

The feasibility of using presently available liquid lubricants in advanced high-speed aircraft engines was studied with a nitrogen inerted recirculating lubrication system. A simulated turbine engine sump was used to identify system problems. It incorporated a 4.92-inch (125-mm) ball bearing and 6.33-inch- (161-mm-) diameter face contact seals. These advanced state-of-the-art components were operated at 14 000 rpm with a 3280-pound (14 590-N) bearing thrust load, which resulted in a DN value (bore diameter of bearing in mm times speed in rpm) of 1.75×10^6 and a maximum Hertz surface stress of 197 000 psi (136 000 N/cm^2). In 3-hour screening tests, in which five degassed lubricant candidates were used at a bulk fluid temperature of 500° F (533 K), a dibasic acid ester (MIL-L-7808E) did not provide adequate bearing lubrication at a bearing outer race temperature of 600° F (589 K). However, a mixed ester, a synthetic paraffinic hydrocarbon, and a perfluorinated polymeric fluid performed satisfactorily in short duration runs of 3 to 10 hours at bearing temperatures to 700° F (644 K).

The inerted recirculating lubrication system operated satisfactorily with an oxygen volume concentration of less than 0.5 percent in the nitrogen cover gas. However, seal leakage was so high (>5 standard ft^3/min or 8.5 standard m^3/hr) in a majority of the tests that a flight system of these components would not be practical. The principal problem area was excessive leakage through the bellows-type face contact seal separating the nitrogen gas and oil. Analysis showed that the seal problem was related to thermal deformation.

An inerted oil-mist once-through lubrication system was tested but with less promising results than the recirculating system. Only the synthetic paraffinic hydrocarbon of the five lubricants evaluated performed satisfactorily at a bearing outer race temperature of 600° F (589 K). This system appears limited by its inability to maintain stable bearing temperatures and requires further development.

INTRODUCTION

Demands for higher bulk temperatures of aircraft turbine engine lubricating systems are primarily a result of increases in flight speeds and turbine inlet temperatures (refs. 1 and 2). At aircraft speeds above Mach 2.2, as shown in figure 1, ram air can no longer be used to cool the oil. It is believed that the oil estimated values shown in figure 1 were based on extrapolations of then known measured values. The problem of keeping bulk oil temperatures within limits is then directly related to the amount of cooling that is available from the fuel.

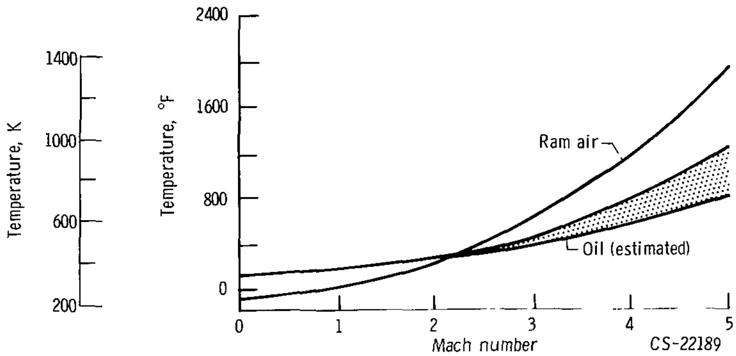


Figure 1. - Effect of Mach number on operating temperatures (from ref. 1).

A serious temperature limitation is placed on the types of lubricants and lubricant systems that can be considered. The ester-base synthetic lubricants of conventional lubricating systems have little margin for potential use at higher temperatures (ref. 2). Thus, there is not only an immediate need for improved lubricants and lubricating systems in uprated engines but advanced engines for high-speed aircraft pose a much greater lubrication system problem because bulk temperatures are expected to be in the 450° to 500° F (505 to 533 K) range.

The BACKGROUND section of this report gives an account of some of the past work in the area of high-temperature lubrication. Particular emphasis is given to the consideration of using an inerted lubrication system. This system might permit the use of presently available lubricants that would be limited by oxygen degradation at the expected higher temperatures in advanced engines. The discussion includes some results from previous lubrication studies in reduced oxygen atmospheres in (1) friction and wear, (2) gear load capacity, and (3) ball bearing tests. The effect of entrained gases in the lubricant is also discussed.

The research reported herein was undertaken to study the feasibility of using the best available high-temperature lubricants in inerted lubrication systems that simulate conditions expected in advanced high-speed aircraft gas-turbine engines. The objectives were as follows:

(1) Design and construct two complete facilities for testing a full-scale gas-turbine mainshaft-bearing-seal compartment

In one facility, an inert recirculating lubrication system is used and in the other, an inert oil-mist once-through system. The test elements are state-of-the-art 4.92-inch (125-mm) ball bearings and 6.33-inch- (161-mm-) diameter face contact seals.

(2) Perform 3-hour screening tests with each of five lubricant candidates in both lubrication test facilities at bearing outer race temperatures in the range of 600° to 800° F (589 to 700 K) at a shaft speed of 14 000 rpm

(3) Perform extended testing of up to 50 hours duration with two selected lubricants in both systems at the maximum operable temperature as determined in the screening tests

Part of the studies reported herein were made by SKF Industries, Inc. under NASA Contract NAS3-6267 (refs. 3 to 6).

BACKGROUND

General Problem Area

In finding solutions to this general problem of high-temperature lubrication, some attention has been given to unconventional systems, such as powder lubrication (ref. 7), dry films (ref. 8), throwaway schemes (ref. 9), and inerted lubrication systems.

When planning the use of liquid lubricants at high temperatures, the manner in which the lubricant is apt to degrade should be considered. In a conventional open system, lubricant breakdown would probably occur by fluid oxidation. The temperature limit for long-term use of the diester oil formulations most commonly used in present jet engines is about 350° F (450 K). In the absence of oxidation, however, a typical diester formulation has thermal stability to about 575° F (575 K) (from isoteniscope data). These data would suggest that, if oxygen can be kept from the lubrication system of an engine, the lubricant may have useful stability at temperatures over 200° F (111 K) higher than if the system contained substantial amounts of oxygen.

If a lubrication system were essentially free of oxygen, however, a new set of problems might be anticipated. Oxygen in the lubrication system reacts with metals present as well as with the lubricant. The metal-oxygen reaction is very necessary for lubricated parts since the metallic oxides protect the surfaces from seizure during momentary failure of the liquid film. Previous friction and wear studies (ref. 10) conducted in air and vacuum with various alloy combinations showed that the amount of oxygen in the vacuum environment needed to reform the oxide films as they wear away is extremely small, less than 1 ppm by volume. Figure 2 is a plot (from ref. 10) of the coefficient of friction for 52100 bearing steel sliding against itself at various ambient pressures. This figure illustrates the point that, with limited oxygen available, as in an inerted system, more effective lubrication could result than normally encountered in an air atmosphere. However, at the lower pressures ($<10^{-6}$ mm Hg or 1.33×10^{-8} N/cm^2) where oxygen availability was appreciably reduced to less than 1 ppm, the wear surfaces showed evidence of considerable metal transfer. In a closed inert-gas-blanketed lubrication system, it would be expected that a range of oxygen concentrations from 0.01 to 0.50 percent by volume (100 to 5000 ppm) might effectively reduce oil oxidation and yet allow for good lubrication.

Conflicting results on the effect of system operation in an inert atmosphere have been shown by several gear load capacity studies. In one study (ref. 11), this effect was found to vary with the lubricant used. Two mineral oils had greater load capacities in nitrogen environment as compared with operation in air, whereas two MIL-L-7808 (diester based) and two MIL-L-9236 (polyester based) type lubricants had nearly equal gear load capacities in nitrogen and air environments. In another gear study (ref. 12), a significant reduction in gear load carrying capacity for all lubricants tested (mineral oil, polyalkylene glycol, methyl chlorophenyl silicone, 5P4E polyphenyl ether, and MIL-

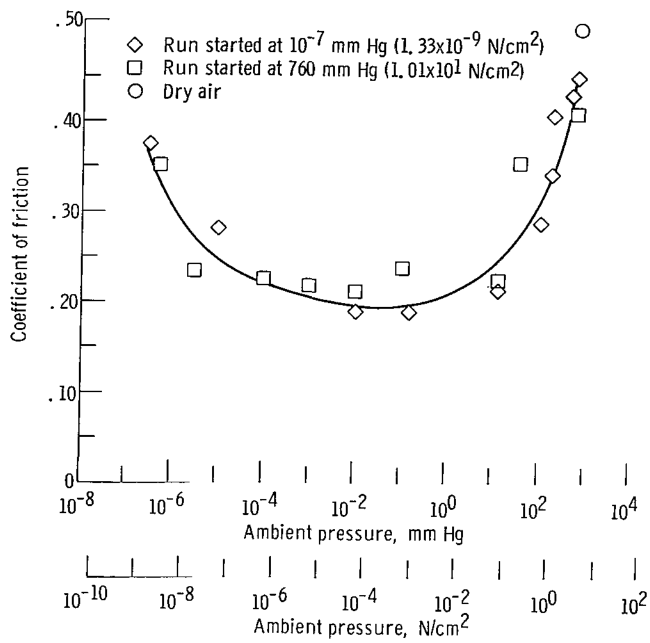


Figure 2. - Coefficient of friction for 52100 bearing steel sliding on 52100 bearing steel at various ambient pressures (from ref. 10). Sliding velocity, 390 feet per minute (2.0 m/sec); load, 1000 grams; temperature, 75° F (297 K).

L-23699 mixed ester) occurred when the air environment was replaced with nitrogen. The nitrogen-blanketed lubricants (ref. 12) showed no increase in gear load capacity at the higher temperature levels; in an air environment, however, an increase in load capacity at higher temperatures was attributed to the formation of lubricant decomposition (oxidation) products on the surface. These decomposition products act as a lubricant.

In ball bearing studies (ref. 13), an improvement in bearing performance occurred when the air environment was replaced with nitrogen for two lubricants (a silicone-diester blend and a MIL-0-6081-A mineral oil). Other studies have been reported in the literature (see ref. 14) on boundary lubrication using inert and/or controlled atmosphere. These studies show considerable disagreement on the effect of oxygen content on friction and wear.

Thus, the literature contains conflicting results on the effect of nitrogen inerting on lubrication capacity. These conflicts indicate a need for additional inert system studies by investigating candidate lubricants under simulated engine operating conditions.

Effect of Entrained Gases in Lubricants

Entrained gases, particularly oxygen, in the lubricant can play a significant role in boundary lubrication, especially at high temperatures. At high surface temperatures, the chemical reaction of oxygen and water in the oil with the metal surfaces can have an effect on lubrication. Degassing of the lubricant in operation could result from low ambient pressures (high altitudes) and high temperature and thereby significantly reduce its ability to lubricate.

The influence of entrained gases on boundary lubrication was determined in a study made previously (ref. 15) using four lubricants in friction and wear experiments. The environment was nitrogen gas at atmospheric pressure. Friction and wear data were obtained on a 5P4E polyphenyl ether, a chlorinated methylphenyl silicone, a paraffinic resin, and a polypropylene at ambient temperatures of 75^{o}, 500^{o}, and 1000^{o} F (297, 533, and 811 K) with each oil in either the as-received (containing normal entrained gases) or degassed conditions. Table I shows the amount of dissolved oxygen and nitrogen in the as-received oils as determined by a gas-chromotography technique described in reference 16. Degassing of three of the oils (silicone, resin, and polypropylene) had no appreciable effect on the friction and wear with René 41 sliding on René 41 (ref. 15). However, the polyphenyl ether appeared to be adversely affected by degassing at all three temperature levels, as shown by the results presented in figure 3. In addition, the friction force trace obtained with degassed polyphenyl ether was markedly "rougher" as compared with the nondegassed fluid (see fig. 4). This "rough" friction trace was probably the result of numerous metal-to-metal contacts occurring through the lubricating boundary film.

TABLE I. - DISSOLVED OXYGEN AND NITROGEN IN AS-RECEIVED OILS AS DETERMINED BY GAS CHROMOTOGRAPHY (REF. 15)

Lubricating fluid	Gas volume in oil at standard temperature and pressure	Total oxygen and nitrogen in oil	Oxygen in oil	Nitrogen in oil	Ratio of nitrogen to oxygen
	cm^3/liter (or 10^{-3} cm^3/cm^3)	mg/liter (or 10^{-3} mg/cm^3)	mg/liter (or 10^{-3} mg/cm^3)	mg/liter (or 10^{-3} mg/cm^3)	
Polyphenyl ether (five-ring isomeric)	45.9	47.25	15.95	31.30	1.96
Paraffinic resin	42.6	43.80	16.88	26.92	1.59
Chlorinated methylphenyl silicone	137.7	141.85	42.24	99.61	2.36
Polypropylene	88.3	90.80	19.80	71.00	3.59

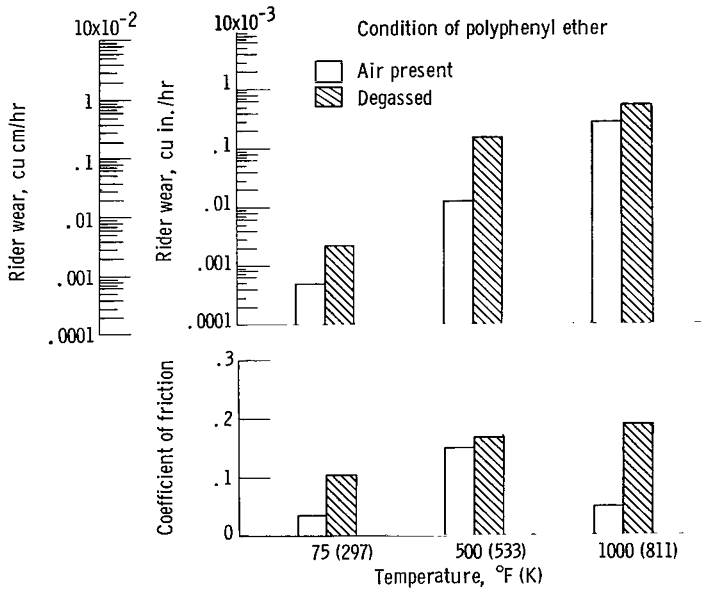
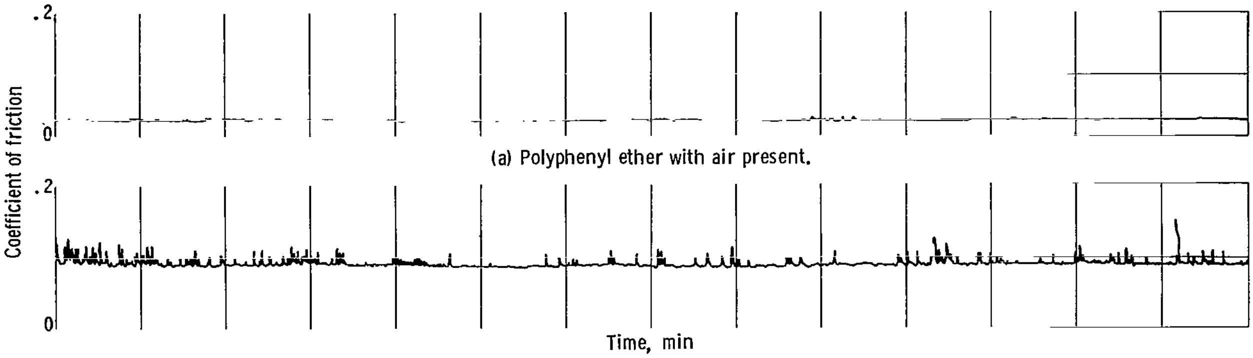


Figure 3. - Coefficient of friction and rider wear for René 41 sliding on René 41 at various ambient temperatures (from ref. 15). Lubricant, polyphenyl ether; flow rate, 0.55 cubic centimeter per minute; atmosphere, nitrogen; sliding velocity, 4600 feet per minute (23.4 m/sec); load, 1000 grams; run duration, 1 hour.



(a) Polyphenyl ether with air present.

(b) Degassed polyphenyl ether.

Figure 4. - Coefficient of friction for René 41 sliding on René 41 (from ref. 15). Lubricant, polyphenyl ether; flow rate, 0.55 cubic centimeter per minute; atmosphere, nitrogen; sliding velocity, 4600 feet per minute (23.4 m/sec); load, 1000 grams; ambient temperature, 75° F (297 K).

A study at Pennsylvania State University (ref. 14) indicated that wear is a function of the amount of oxygen dissolved in the lubricant. In a series of experiments with a modified four-ball tester designed for use at high temperatures with controlled atmospheres, minimum wear values are obtained with oxygen concentrations far below those obtained with air as the atmosphere for a superrefined mineral oil. The amount of oxygen required for minimum wear with this type of fluid increases with increasing load. Also, for lubricants with oxygen-containing molecules, such as the diester fluid, the rate of chemical wear involved in boundary lubrication is not as greatly affected by dissolved oxygen.

Inerted Lubrication System Study

To gain acceptance of a concept such as inerting, it is necessary to demonstrate its feasibility in full-scale hardware. The problem of sealing an inert gas and a lubricant in an engine system and keeping air out is difficult in full-scale hardware. Also, good sealing is critical to realizing an economical system because, if leakage rates are high, the necessary inventory of inerting gas is a weight penalty for the aircraft.

A study was made under NASA contract NAS3-6267 (refs. 3 to 6) to determine the feasibility of using five different lubricants in an inerted recirculating lubrication system at a bulk oil temperature of 500° F (533 K) with bearing outer race temperatures in the range of 600° to 800° F (589 to 700 K). A simulated engine bearing sump apparatus was designed and used in this investigation. The apparatus incorporated 6.33-inch-(161-mm-) diameter face contact seals and a 4.92-inch (125-mm) ball bearing for operation at 14 000 rpm with a 3280-pound (14 590-N) bearing thrust load. The initial bearings and seals were advanced state-of-the-art components selected from different engine development programs. Neither the bearings nor the seals were designed to operate under temperature conditions as severe as those imposed by this program. Only minor changes to those components (e.g., bearing cage clearance) were made for the exploratory studies.

APPARATUS AND PROCEDURE

A schematic diagram of the simulated turbine engine sump employing an inerted lubricating system is shown in figure 5. Heaters on the bearing and housing outside

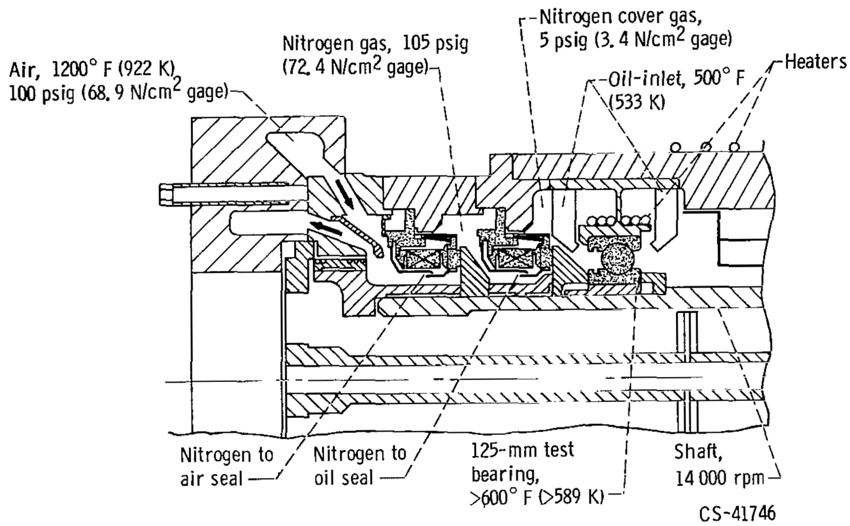


Figure 5. - Bearing and seal assembly in simulated engine sump of inerted lubrication system.

diameter were intended to permit operation to a bearing outer race temperature of 800° F (700 K) and were used to control test cavity temperature. Hot air in the hollow main shaft was used to ensure the heat flow path from the inner to outer test bearing races by maintaining an inner race bearing temperature at least 10° F (5.6 K) higher than that of the outer race. Two face contact seals (bellows secondary) formed the sealing system. Nitrogen (oxygen content less than 10 ppm) was introduced between the two seals at 105 psig (72.4 N/cm^2 gage). The seal between the nitrogen gas and the bearing sump was subjected to a pressure differential of 100 psi (68.9 N/cm^2) with the nominal pressure of the bearing compartment at 5 psig (3.4 N/cm^2 gage). The seal between the hot air (at 1200° F or 922 K and 100 psig or 68.9 N/cm^2 gage) and nitrogen was subjected to a pressure differential of 5 psi (3.4 N/cm^2).

Total seal gas (nitrogen) leakage in excess of 5 standard cubic feet per minute (8.5 standard m^3/hr) across both seals was arbitrarily considered unsatisfactory seal operation. Leakage from the interseal cavity to the bearing test chamber was monitored by injecting a known small quantity of helium into the nitrogen supplied to this cavity. Monitoring was carried out by a mass spectrometer unit drawing samples from both the interseal and bearing cavities. Total oxygen volume concentration of the leakage gases was not to exceed 0.5 percent.

The test bearing was a split-inner-ring angular-contact ball bearing, which is a type widely used in aircraft turbine engines. This design permits a maximum ball complement (because of separable inner-ring halves) and supports thrust load in either direction. The separable ring also permits the use of a precision-machined one-piece cage that is required for high-speed, high-temperature operation. The test bearing has a bore diameter of 4.92 inches (125 mm) and a nominal mounted operating contact angle of 26°. This bearing runs at the test speed of 14 000 rpm (DN = 1.75×10^{6}) and a thrust load of 3280 pounds (14 590 N) (maximum Hertz surface stress of 197 000 psi or 136 000 N/cm^2). These conditions are typical of engine practice. For operating temperatures up to 600° F (589 K), consumable-electrode vacuum-melted (CVM) M-50 tool-steel rings and balls were used. At higher temperatures, CVM WB49 tool steel was used for the bearing rings and CVM M-1 tool steel for the balls. The cages are of an outer-ring piloted design and were constructed of silver-plated M-1 tool steel. The bearings had nominal 51.6-percent inner ring and 52.1-percent outer ring conformities, 4-microinch (0.10-μm) root-mean-square maximum cross-groove roughness, twenty-one 13/16-inch- (20.6-mm-) diameter balls, and 0.0068- to 0.0080-inch (0.173- to 0.203-mm) unmounted internal radial looseness.

The selection of test lubricants was made on the basis of published performance, property data, and expected performance under the experimental operating conditions. Temperature-viscosity data are given in figure 6. The following lubricants were used:

(1) Dibasic acid ester formulated with proprietary additives

This is a reference fluid that meets MIL-L-7808E specification. The viscosity variation with temperature (curve 1 of fig. 6) shows an extrapolated value of 0.64 centistoke (0.64×10^{-6} m^2/sec) at 600° F (589 K).

(2) Mixed ester-based formulated fluid with improved thermal stability over MIL-L-7808E-type fluid

It has an estimated viscosity of 1.17 centistokes (1.17×10^{-6} m^2/sec) at 600° F (589 K) (curve 2 of fig. 6).

(3) Synthetic paraffinic hydrocarbon containing an antiwear additive and having an estimated viscosity of 2.4 centistokes (2.4×10^{-6} m^2/sec) at 600° F (589 K) (curve 3 of fig. 6)

(4) Unformulated modified polyphenyl ether (C-ether) having an estimated viscosity of 0.6 centistoke (0.6×10^{-6} m^2/sec) at 600° F (589 K) (curve 4 of fig. 6)

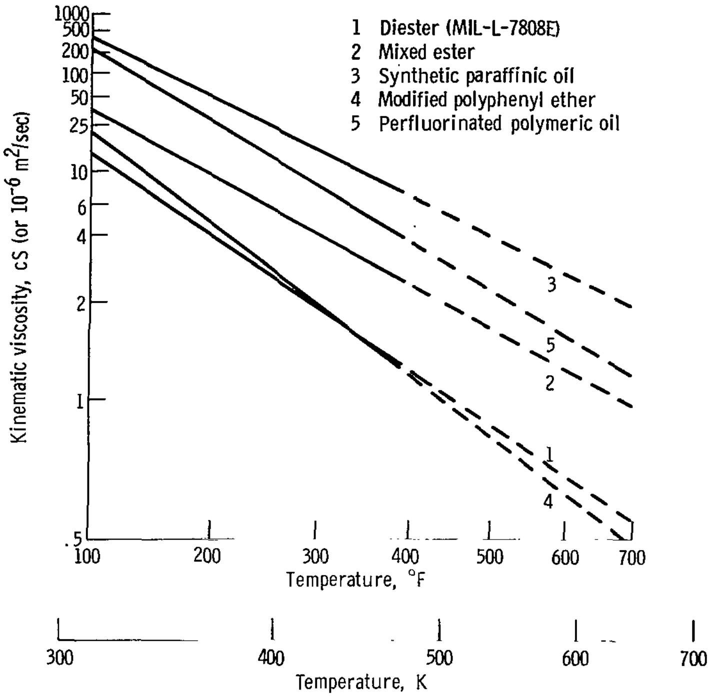


Figure 6. - Viscosity-temperature relation for recirculating oils. (Extrapolated values beyond 400° F (478 K) shown as dashed lines.)

(5) Unformulated perfluorinated polymeric fluid with an estimated viscosity of 1.6 centistokes (1.6×10^{-6} m^2/sec) at 600° F (589 K) (curve 5 of fig. 6)

RESULTS AND DISCUSSION

Inerted Recirculating System Studies

A series of 3-hour lubricant screening tests was attempted in the inerted recirculating lubrication system. The tests were made to explore the feasibility of using the test fluids in such a system. The running conditions were at expected environmental conditions of a bearing sump in an advanced engine. The runs were also for the purpose of identifying problems. These runs were not of sufficient duration to document fully the potential problems that might accrue from extended operation with partial lubricant films. Such partial or discontinuous films might be expected at the high surface temperatures for the bearings and oil seals of these runs.

A compilation of the screening test results is given in table II, and measurements of viscosity and neutralization number for each test lubricant before and after each

TABLE II. - SUMMARY OF TEST RESULTS IN RECIRCULATING OIL APPARATUS

[Nitrogen blanketed except where noted.]

Fluid	Run time, hr	Nominal test bearing temperature		Nominal oil flow		Seal leakage		Oxygen in test pump, percent	Salient conditions of test components	Results and conclusions
		^{o}F	K	gal/min	m^3/hr	standard ft^3/min	standard m^3/hr			
Diester	1.8	600	589	4 to 5	0.9 to 1.1	1.5 to 10.0	2.5 to 17.0	Not measured	Oil degraded; bearing glazed and heavy cage pocket wear	Bearing failed; oil not suitable for 600^{o} F (589 K)
Mixed ester	3.4	750	672	1.0	0.23	1.0 to 3.0	1.7 to 5.1	0.04	Some oil degradation; slight bearing cage wiping and pocket wear	Borderline passing at 750^{o} F (672 K); satisfactory for 700^{o} F (644 K) 3-hr runs[a]
	0.2	<780	<689	1.0	0.23	6.0 to 10.0	10.2 to 17.0	0.04	Oil degradation starting at 750^{o} F (672 K); bearing surface distress and heavy cage wear	Bearing failed because of oil degradation
Synthetic paraffinic hydrocarbon	5.4	700	644	1.0 to 1.5	0.23 to 0.34	2.0 to 3.0	3.4 to 5.1	0.06	All components satisfactory	Satisfactory for 700^{o} F (644 K) 3-hr runs[a]
	1.8	750	672	1.0	0.23	2.0 (except at end of test)	3.4	0.09	Oil viscosity decreased	Oil seal liftoff caused excess leakage and apparatus fire
Modified polyphenyl ether	3.0	600	589	1.2 to 2.0	0.27 to 0.45	7.0 to 11.0	11.9 to 18.7	0.12	Some oil degradation; scoring of seal carbons caused excess leakage	Required higher oil flow to stabilize bearing temperature[a]
Modified polyphenyl ether (open atmoshere)	3.2	600	589	1.0	0.23	2.0 to 12.0	3.4 to 20.4	20.0	Oil seals somewhat scored; dark particles in oil but no coking	Higher temperature testing not recommended because of oil oxidation[a]
Perfluorinated polymer	3.7	700	644	2.0	0.45	15.0 to 23.0	25.5 to 39.1	0.05	Bearings discolored but otherwise unchanged; seals good but leakage high	Oil slightly corrosive; required higher oil flow to stabilize bearing temperature[a]
	0.7	740 (800 attempted)	667 (700 attempted)	0.5 to 2.0	0.11 to 0.45	5.0 to 10.0	8.5 to 17.0	0.05	Bearing surface smeared; oil viscosity increased and oil seal worn	Bearing failed; higher oil flow and seal leakage than desired

[a]Run complted in desired 3-hr minimum period without component failure.

TABLE III. - LUBRICANT VISCOSITY BEFORE AND AFTER

SCREENING RUNS AND ACID NUMBER CHANGE

[Nitrogen blanketed except where noted.]

Fluid	Bearing test temperature		Operating time, hr	Viscosity at 100° F (311 K), cS (or 10^{-6} m^2/sec)		Acid number change during run
	°F	K		Before[a]	After	
Diester	600	589	1.8	12.05	12.3 to 12.8	100 percent increase
Mixed ester	600	589	3.3	38.9	37.4	No change
	700	644	3.3	38.9	46.3	Slight decrease
	750	672	3.4	38.9	38.3	Slight decrease
	800	700	0.2	38.9	45.0	Slight increase
Synthetic paraffinic hydrocarbon	600	589	3.0	443.3	471.8	No change
	700 to 750	644 to 672	7.2	443.3	360.4	No change
Modified poly-phenyl ether	600	589	3.0	24.8	24.9	No change
Modified poly-phenyl ether (open atmosphere)	600	589	3.2	24.8	26.5	No change
Prefluorinated polymer	600	589	3.0	267.9	347.4	No change
	700	644	3.7	267.9	392.3	No change

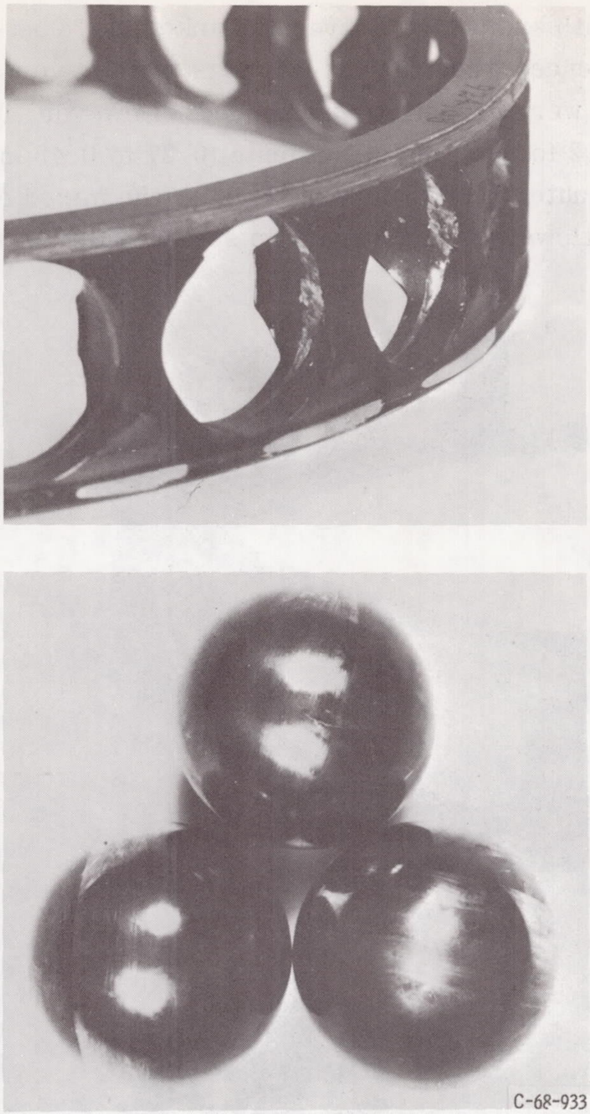
[a]Values on new degassed fluids.

screening run are shown in table III.

The principal results obtained for each of these degassed fluids are as follows:

(1) The MIL-L-7808E type lubricant was not suitable, even with inert blanketing, at an outer race bearing temperature of 600° F (589 K). Inadequate lubrication was considered to have caused the bearing distress that included excessive wear of the cage and balls. Figure 7 shows the appearance of the cage and balls after test; heavy wear was encountered on these components. High oil flow rates of 4 to 5 gallons per minute (0.9 to 1.1 m^3/hr) were required to stabilize bearing temperature. The neutralization number of the lubricant increased significantly during the run, indicating fluid degradation, but there were only minor coke deposits in the test cavity. It should be noted that this oil is limited to a bulk temperature of about 350° F (450 K) when operating in an air environment.

(2) The mixed ester lubricant performed satisfactorily in the short tests at bearing temperatures up to 750° F (672 K). At a bearing temperature of 650° F (616 K), this



C-68-933

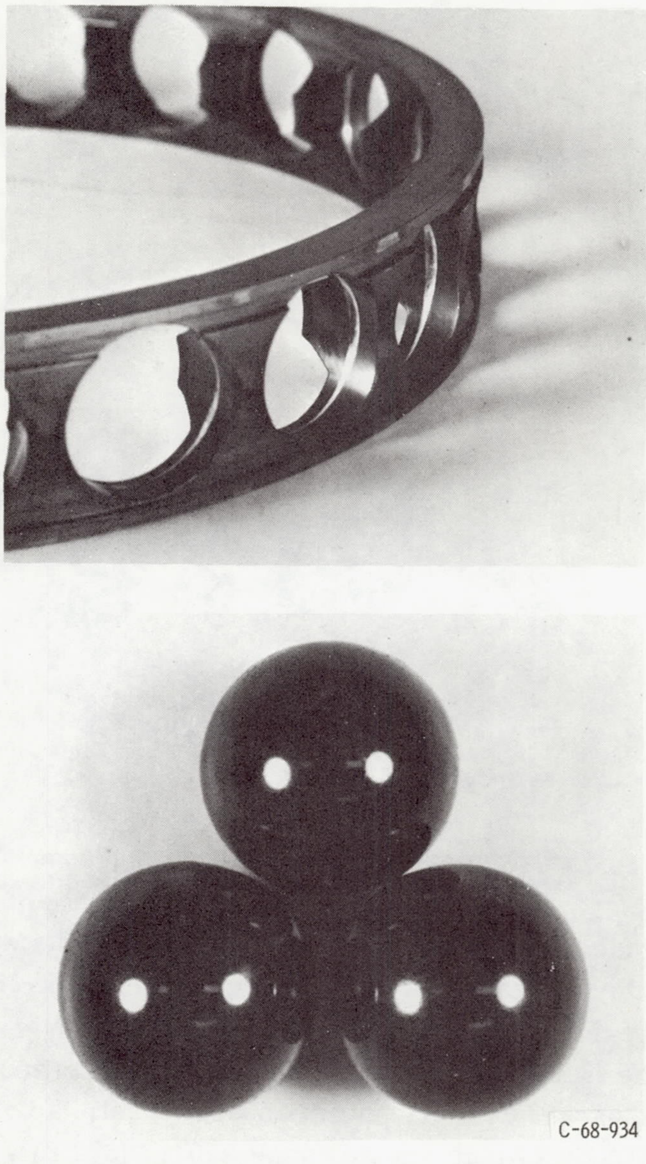
Figure 7. - 4.92-Inch (125-mm) bearing cage and balls after 1.8-hour inerted run at 600° F (589 K) with diester (MIL-L-7808E) lubricant. Bearing experienced heavy cage wear and wear on balls.

fluid performed satisfactorily for about 10 hours before the run was stopped as a result of malfunctioning of the oil-side seal. Mass spectrometer data on the oils revealed that oil degradation at a bearing temperature of 750° F (672 K) would preclude its use for longer periods of time than 3 hours, and its use should be limited to lower temperatures.

(3) The synthetic paraffinic lubricant was operated satisfactorily at bearing temperatures up to 700° F (644 K). An attempted run at 750° F (672 K) was aborted after less than 2 hours because of excessive leakage of the oil-side test seal. Mass spectrometer data on the oil from the 750° F (672 K) run indicated that significant fluid decomposition

had started after 1/2 hour of operation.

(4) The modified polyphenyl ether (C-ether) lubricant performed satisfactorily at 600° F (589 K) both with and without nitrogen blanketing. Post-test inspection after both runs revealed no evidence of coking; the bearings were in very good condition (fig. 8), and the oil properties were essentially unchanged. However, it had been necessary to provide oil flows of 1.2 to 2 gallons per minute (0.27 to 0.45 m^3/hr) as compared with 0.5 to 1 gallon per minute (0.11 to 0.23 m^3/hr) for the mixed ester and paraffinic lubricants. These higher flows were necessary to stabilize the bearing temperature and prevent

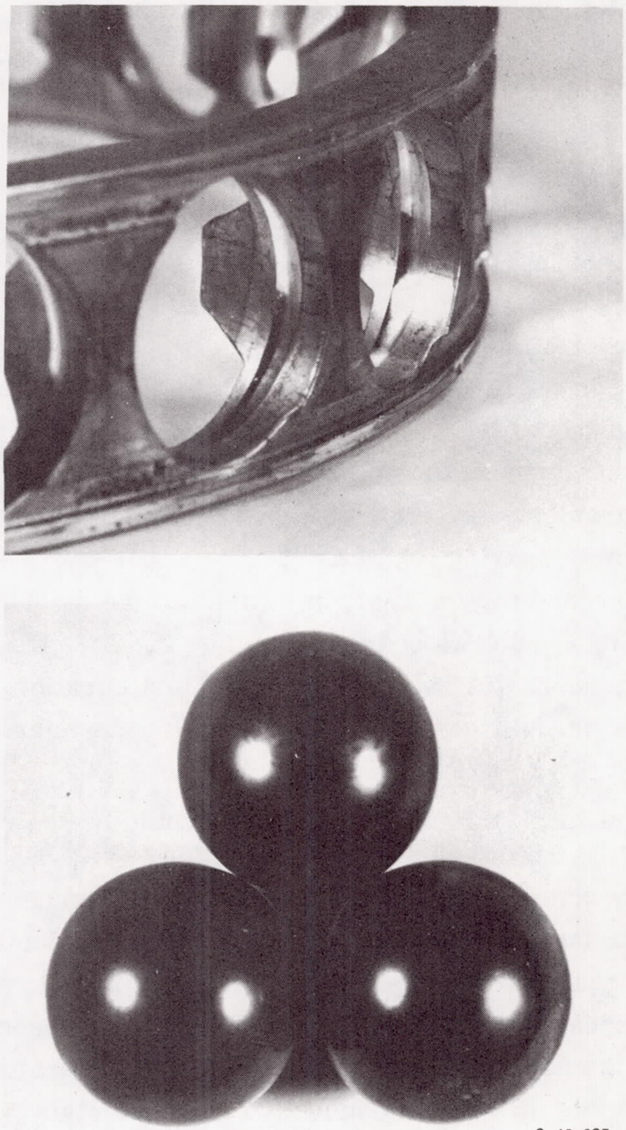


C-68-934

Figure 8. - 4.92-Inch (125-mm) bearing cage and balls after 3.2-hour open atmosphere run at 600° F (589 K) with modified polyphenyl ether lubricant. Bearing experienced moderate cage pocket wear.

excessive temperature excursions and are considered less desirable for practical operations. Higher temperature testing was suspended because of the relatively poor bearing cooling experienced with the lubricant.

(5) The perfluorinated polymeric fluid was tested successfully to temperatures of 700° F (644 K), but higher oil flow rates were required to achieve bearing temperature stability than with the mixed ester or the synthetic paraffinic lubricants. Although the test bearings were essentially unchanged by the operation (fig. 9), there was some discoloration resulting from corrosivity of the fluid at temperatures above 600° F (589 K).



C-68-935

Figure 9. - 4.92-Inch (125-mm) bearing cage and balls after 3.7-hour inerted run at 700° F (644 K) with perfluorinated polymeric lubricant.

Several attempts at 800° F (700 K) bearing runs were unsuccessful; seal leakage rates were in excess of 5 standard cubic feet per minute (8.5 standard m^3/hr), and a test bearing failed.

Only minor oil coking occurred during most of these runs and was not sufficiently serious to affect bearing or seal performance. Longer term runs of 50 hours with the mixed ester and the synthetic paraffinic lubricants were not completed because of repeated oil-side seal malfunctions, which were also the limiting factors in a majority of the 3-hour screening studies.

Analysis revealed that the seal malfunctions were not related primarily to the inerting gas or the thermal stability of the fluids, but rather to thermal deformations, especially in the oil-side seal, which accounted for about 90 percent of the leakage. A more complete discussion of the shaft seal problem is given in reference 17. A continuing contract effort is intended to provide new and improved mainshaft seals for advanced air-breathing propulsion. This work is under NASA contract NAS3-7609, and the first phase of this study is reported in reference 18.

The bearings performed well beyond their expectation during these short-term runs with the various classes of lubricants. Four general classes of failures were observed:

(1) Excessive wear of balls and cage resulting from inadequate lubrication in the ball path (such as with MIL-L-7808E oil)

(2) Gross smearing in the ball path considered to have resulted from loss of internal bearing clearance

(3) Cage wiping (smearing) on the outer lands

(4) Heat-transfer problems (such as with the C-ether fluid) that resulted in difficulty maintaining stable bearing temperatures

A good general reference for the analysis of the failure mechanisms in highly loaded rolling and sliding contacts, such as characterized by the first three of these listed failures, is given in reference 19. Since the longest test duration was about 10 hours in this study of inerted system lubrication, no conclusions regarding longer term operation can be drawn.

Another cause of smearing and spalling failures could be the result of inadequate elastohydrodynamic (EHD) film thicknesses. Tallian, et al. (ref. 20) reported that asperity contact frequency and duration depend on EHD film thickness and are predictable from the statistics of surface microgeometry. The ratio between film thickness and composite root-mean-square roughness of the rolling surfaces is an important parameter governing wear and lubrication behavior in the EHD regime. Since all the lubricants in this study did not have the same viscosity at the bulk oil temperature of 500° F (533 K) (fig. 6), the failure of some oils and the success of others might be attributable to viscosity as it affects EHD film thickness, particularly. For example, the synthetic paraffinic lubricant gave the most promising results in these studies, and it had the highest viscosity of all the oils studied at 500° F (533 K).

In only one case (MIL-L-7808E fluid) could bearing failure definitely be attributed to lack of lubricating ability of the lubricant. A major difficulty in the other runs was with bearing cooling, which appears to be a result of poor surface wetting at elevated temperatures. Table IV shows the initial decomposition temperature of the lubricants (as measured by the isoteniscope) and the inerted system screening run temperatures attained. The limited data available do not indicate any clear-cut correlation between fluid thermal stability and the ability of the lubricant to operate satisfactorily at higher temperatures in short runs. Further efforts will be required to establish if such a relation does exist.

TABLE IV. - LUBRICANT INITIAL DECOMPOSITION TEMPERATURE

(BY ISOTENISCOPE) AND SCREENING TEST RESULTS

Fluid	Initial decomposition temperature		Screening run temperature			
	^{o}F	K	Highest successful, ^{o}F	Highest successful, K	Failure run, ^{o}F	Failure run, K
Diester (MIL-L-7808E)	575	575	(a)	(a)	600	589
Mixed ester	(b)	---	750	672	750 to 800	672 to 700
Synthetic paraffinic	600	589	700	644	750	672
Modified polyphenyl ether	675	631	600	589	(c)	(c)
Perfluorinated polymeric	780	689	700	644	720 to 780	656 to 689

[a]No successful runs at 600^{o} F (589 K).
[b]Isoteniscope data not available.
[c]No runs attempted above 600^{o} F (589 K).

Inerted Oil-Mist Once-Through System Studies

In addition to recirculating system studies, an effort was made to study the feasiblity of using conventional lubricants in an oil-mist once-through simulated-aircraft inerted lubrication system. Five lubricants were tested in an inerted once-through system in 3-hour screening studies at the same operating conditions as the recirculating experiments. The lubricants were (1) synthetic paraffinic hydrocarbon, (2) diester of MIL-L-7808E type, (3) polyalkylene glycol, (4) polyolefin, and (5) polyester. Two of these oils (the paraffinic hydrocarbon and the diester) were the same oils evaluated in the inerted recirculating system. The paraffinic hydrocarbon ran successfully at 600^{o} F (589 K) outer race bearing temperature, but all other runs failed by loss of internal clearances in the bearings. Excessive lubricant decomposition products were found for all lubricants except the paraffinic hydrocarbon. There was no evidence of lubrication distress in the ball tracks of mist test bearings. These results indicate that substantially higher mist-rig operating temperatures may be possible with the paraffinic hydrocarbon. Such operation would require optimized design (internal clearances and other bearing geometry), more efficient application of lubricant mist to minimize friction heat sources, and careful control of gas-mist flow to achieve stable bearing temperatures. Further developmental studies of this concept are being made to achieve more efficient application of lubricants.

SUMMARY OF RESULTS

Studies were made of a lubrication system that incorporated a nonoxidizing cover gas (nitrogen). The experimental apparatus was a simulated advanced high-speed aircraft gas-turbine engine sump operating at 14 000 rpm with full-scale components. A 4.92-inch (125-mm) ball bearing (3280-lb or 14 590-N thrust load) and 6.33-inch- (161-mm-) mean diameter face contact seals were used in 3-hour screening studies with five lubricant candidates. The main purposes of these studies were to explore the feasibility of using an inerted system and to identify problems with 500° F (533 K) bulk fluid temperatures and 600° to 800° F (589 to 700 K) bearing temperatures. The experimental data and analysis revealed the following:

1. Dibasic acid ester (MIL-L-7808E) lubricant is not suitable for 600° F (589 K) bearing temperature in inerted operation because of inadequate lubricating ability. However, a mixed ester and a synthetic paraffinic fluid performed satisfactorily at a bearing temperature of 700° F (644 K) in short-duration runs of 3 to 10 hours. Perfluorinated polymeric fluid also performed satisfactorily at a bearing temperature of 700° F (644 K), but higher oil flow was required than with the mixed ester and the paraffinic fluids to maintain a stable bearing temperature. Also, there was evidence of corrosive attack on the bearing at higher temperatures with the fluorocarbon. A modified polyphenyl ether (C-ether) fluid operated satisfactorily at a bearing temperature of 600° F (589 K) but was not run at higher temperatures because of apparent heat-transfer problems that may be related to surface wetting deficiencies. No conclusions regarding longer term operation of inerted system lubrication can be drawn from this study.

2. The inerted recirculating lubrication system operated satisfactorily; low oxygen volume concentartion ($<$0.5 percent) was maintained during operation. Seal leakage was frequently so high, however, that a flight system of these components would be impractical. This leakage was attributed to thermal deformation of the sealing face.

3. Excessive leakage through the oil seal was a limiting factor in a majority of the lubricant studies.

4. An inerted oil-mist once-through lubrication system was tested but with less promising results than with the recirculating system. Only the synthetic paraffinic hydrocarbon of the five lubricants evaluated ran satisfactorily at a bearing outer race temperature of 600° F (589 K). This system concept requires further developmental studies.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 11, 1969
126-15-02-27-22.

REFERENCES

1. Bisson, Edmond E.; and Anderson, William J.: Advanced Bearing Technology. NASA SP 38, 1964, p. 2.
2. Reynolds, H. W., Jr.: Turbine Oil and Test Method Development Needed for Curren and Future Engines. Presented at the USAF-SWRI Turbine Engine Lubrication Conference, Sept. 13-15, 1966.
3. Hingley, C. G.; Southerling, H. E.; and Sibley, L. B.: Supersonic Transport Lubrication System Investigation. Rep. AL65TO38, SKF Industries, Inc. (NASA CR-54311), May 20, 1965.
4. Hingley, C. G.; and Sibley, L. B.: Supersonic Transport Lubrication System Investigation. Rep. AL65TO77, SKF Industries, Inc. (NASA CR-54312), Nov. 20, 1965.
5. Rhoads, W. L.; and Sibley, L. B.: Supersonic Transport Lubrication System Investigation. Rep. AL66TO32, SKF Industries, Inc. (NASA CR-54313), May 20, 1966.
6. Rhoads, W. L.; and Sibley, L. B.: Supersonic Transport Lubrication System Investigation, Phase 1. Rep. AL67TO60, SKF Industries Inc. (NASA CR-54662), Sept. 18, 1967.
7. Macks, E. J.; Nemeth, Z. N.; and Anderson, W. J.: Preliminary Investigation of Molybdenum Disulfide-Air-Mist Lubrication for Roller Bearings Operating to DN Valves of 1×10^6 and Ball Bearings Operating to Temperatures of 1000° F. NACA RM E51G31, 1951.
8. Sliney, H. E.: Bearings Run at 1250° F with Solid Lubricants. Space/Aeronautics, vol. 35, no. 3, Mar. 1961, pp. 91-100.

9. Macks, F.: Lubrication Reference Manual for Missile and Space Vehicle Propulsion at Temperatures Above 700° F. Tribo-Netics Labs. (WADC TR-58-638, Vol. 1, Pt. 1) DDC No. AD-213474), Jan. 1959.

10. Buckley, Donald H.; Swikert, Max; and Johnson, Robert L.: Friction, Wear, and Evaporation Rates of Various Materials in Vacuum to 10^{-7} mm Hg. ASLE Trans., vol. 5, no. 1, Apr. 1962, pp. 8-23.

11. Baber, B. B.; Lawler, G. W.; Smith, H. R.; Beane, G. A.; and Ku, P. M.: Gear Lubrication in Inert Gas Atmospheres. ASLE Trans., vol. 3, no. 1, Apr. 1960, pp. 142-148.

12. Baber, B. B.; Anderson, E. L.; and Ku, P. M.: Effects of Lubricants, Metals, Temperature, and Atmospheric Environments on Gear Load-Carrying Capacity. Paper 67-Lub-27, ASME, Oct. 1967.

13. Nemeth, Z. N.; and Anderson, W. J.: Effect of Air & Nitrogen Atmospheres on the Temperature Limitations of Liquid & Solid Lubricants in Ball Bearings. Lubr. Eng., vol. 11, no. 4, July-Aug. 1955, pp. 267-273.

14. Klaus, E. E.; and Bieber, H. E.: Effect of Some Physical and Chemical Properties of Lubricants on Boundary Lubrication. ASLE Trans., vol. 7, no. 1, Jan. 1964, pp. 1-10.

15. Buckley, Donald H.; Johnson, Robert L.; and Brainard, William A.: Influence of Entrained Gases and a Halogen Additive on Boundary Lubrication of Four Oils at Temperatures to 1000° F. NASA TN D-2686, 1965.

16. Petrocelli, James A.; and Lichtenfels, Dean H.: Determination of Dissolved Gases in Petroleum Fractions by Gas Chromatography. Anal. Chem., vol. 31, no. 12, Dec. 1959, pp. 2017-2019.

17. Johnson, Robert L.; Loomis, William R.; and Ludwig, Lawrence P.: Performance and Analysis of Seals for Inerted Lubrication Systems of Turbine Engines. NASA TN D-4761, 1968.

18. Parks, A. J.; McKibbin, R. H.; and Ng, C. C. W.: Development of Main Shaft Seals for Advanced Air Breathing Propulsion Systems. Rep. PWA-3161, Pratt & Whiteny Aircraft (NASA CR-72338), Aug. 14, 1967.

19. Smalley, A. J.; Murray, S. F.; Christensen, H.; and Cheng, H. S.: Review of Failure Mechanisms in Highly Loaded Rolling and Sliding Contacts. Rep. MT167TR23, Mechanical Technology, Inc., May 1967. (Available from DCC as AD-657337.)

20. Tallian, T. E.; McCool, J. I.; and Sibley, L. B.: Partial Elastohydrodynamic Lubrication in Rolling Contact. Elastohydrodynamic Lubrication. Inst. Mech. Eng., London, 1965, pp. 122-137.

NASA-Langley, 1969 —— 15 E-4380