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EVALUATION OF SOLID FILM LUBRICATED AND
UNLUBRICATED GEARS IN DRY HELIUM ATMOSPHERE

(Final Report)

by
D. J. Fessett

30 April 1961

Subcontract 5918-RD-4 Under
Contract AT (10-1)-925

to
KAISER ENGINEERS
Kaiser Building
Oakland, California

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FOREWORD and NOTICES

This report and program was initiated by the Atomic Energy Commission under Contract AT (10-1)-925 to Kaiser Engineers and subcontracted to Western Gear Corporation under Subcontract 5918-RD-4.

Mr. D. J. Fessett, Research Engineer, provided technical supervision and Mr. R. H. Johann, Research Technician, performed the tests. Program supervision was by Mr. M. L. Headman, Manager of Research.

Western Gear Corporation wishes to express their appreciation to all dry film lubricant companies who provided the lubricants for this program.

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ABSTRACT

Evaluation of dry films for application on spur gears for servicing components of a gas cooled nuclear reactor was performed. Commercial dry film lubricants were screened in wear tests to obtain the optimum lubricant which was in turn used in gear tests. A Mo S₂, graphite and sodium silicate binder lubricant (Everlube 811 or equivalent) applied to M 50 tool steel gears resulted in the optimum combination based on stress-cycle tests.

Tests were performed on unlubricated worm gears at simulated loads, speeds and environment of the charging machine for the above reactor to determine wear rates and efficiencies.

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EVALUATION OF SOLID FILM LUBRICATED AND
UNLUBRICATED GEARS IN DRY HELIUM ATMOSPHERE

I. INTRODUCTION:

Efficient operation of gears in a gas cooled reactor requires a lubricant that will not contaminate the helium cooling gas. A program was initiated to determine the performance characteristics of solid film lubricants applied to spur gears under simulated ambient conditions. Design data presented was derived from results of gear and wear specimens tested in helium at 510°F. Wear tests were made on a modified MacMillan race-cube type tester in order to obtain the optimum lubricant. Results of gear tests on the optimum lubricant are defined in terms of cycles to failure based on gear loads, speeds and materials.

Worm gear tests simulating the load, speed and environment of the servicing components of the reactor were performed. In these tests no lubricants were used due to the system requirements.

II. TESTING:

A. TEST EQUIPMENT:

Tests were conducted on modified Mac Millan wear tester, a high temperature spur gear tester and a worm gear tester. All the testers were equipped with heaters and gas facilities for simulating the environment of the servicing machines of the gas cooled nuclear reactor. Description of the individual testers are in the following sub paragraphs.

1. Wear Tester

A modified MacMillan type tester, Figure 1, was used in screening the solid film lubricants at 510°F in a helium environment. The tester is capable of normal loads to 500 lbs and temperatures to 1000°F. Friction forces were measured by strain gages attached to a sensing beam which restrained tangential movement of the test cube. Introduction of helium under pressure in a housing encompassing the test specimens provided the cover gas. Specimen temperature was sensed with a chromel alumel thermocouple in contact with the specimen while the environmental temperature was controlled with a thermocouple adjacent to the contact area of the specimens. Speed was fixed at 850 rpm resulting in a sliding velocity of 306 ft/min. Provisions were made for a warning horn to operate when the friction coefficient exceeded 0.2. Friction force, environment and specimen temperature were simultaneously recorded on strip chart recorders.

The rotating specimen for the wear tester was a standard test cup of 52100 steel manufactured by Timken and processed by a lubricant manufacturer on the outside diameter with their commercial lubricant. A non-lubricated nitralloy cube hardened to 72 Rc acted as the stationary specimen or shoe.

2. Spur Gear Tester

Spur gear tests were conducted on Western Gear's High Temperature Tester, Figure 2. The tester is essentially a closed loop drive having the torque locked into the gears through a differential gear train. Weights applied to the housing of the differential gear train result in an amplified load on the test gears. Speed regulation was through a Reeves variable speed drive coupled to a 10 hp motor. An electronic counter capable of ± 1 rpm accuracy was used in monitoring the speed.

FIGURE 1 - Modified MacMillan Type Wear and Friction Tester

FIGURE 2 - Western Gear's High Temperature Gear Tester

A sealed insulated chamber, which was provided with a tubular heating element capable of 1500°F, housed the test spur gears. Visual examination of the test gears without exposing them to the air was made through a pyrex window on top of the chamber. The control temperature was sensed with a thermocouple located approximately one-half inch from the gear mesh. An indicator attached to the load arm provided an unique means for measuring the change of backlash of the gear system, which in turn was calibrated for the change in backlash or wear of the test gears. Electrical current required to operate the tester was measured with an ammeter, however this value was disregarded since no apparent change in current could be observed during a test cycle.

Test gear geometry of the spur gears was:

- 2 in. pitch diameter
- 12 and 16 diametral pitch
- 3/8 in. face width
- 20° full depth involute tooth form

Materials:

- Nitralloy, 12 D. P.
- M-50 Tool Steel, 12 D. P.
- Halcomb 218 Tool Steel, 16 D. P.

3. Worm Gear Tester

Worm gear tests were performed on a commercial unit modified to accommodate heaters, sealed bearings and a viewing port, Figure 3. Loading was with a prony brake attached to the output shaft. The tester was mounted in a cradled fixture that allowed the input torque and, consequently, the efficiency of the unit to be measured. Input speed was fixed at 225 rpm.

Worm gear geometry was:

- 4 in. pitch diameter
- 10 diametral pitch
- 3/4 in. face width
- 14-1/2° pressure angle

FIGURE 3 - Worm Gear Tester

Material:

440C Stainless Steel, Rc 57

Worm geometry was:

1.25 in. pitch diameter

10 diametral pitch

single thread

0.3142 lead

4°34' lead angle

Material:

Nitralloy 135, Rc 70

B. TEST PROCEDURE:

1. Wear Test

Prior to the friction and wear test the coated test cup was placed in a vacuum chamber at 1 mm Hg for a time period of 5 minutes to simulate the purging environment of the gas cooled reactor servicing machines. After the specimens were installed in the wear tester, a flow of 4 liters/min. of helium was used to purge the test chamber and provide the cover gas. Testing began when the ambient temperature stabilized at $510^{\circ}\text{F} \pm 10^{\circ}$. A normal load of 98.7 lbs (55,000 psi initial contact stress) was applied to the cube until either 20.2×10^4 cycles had elapsed or failure occurred. Failure was arbitrarily chosen as the time when a coefficient of friction of 0.2 was exceeded. If failure was not eminent after 20.2×10^4 cycles, the normal load was increased to 137.4 lbs (65,000 psi contact stress). Frictional force and cube temperature were recorded on strip chart recorders.

2. Spur Gear Test

Spur gear tests were at a constant speed and load. Helium was introduced into the test chamber at a rate of 5 liters/min. and the chamber temperature was maintained at $510^{\circ}\text{F} \pm 10^{\circ}$. Wear readings were

recorded at 15 minute intervals while the gears were rotating. Visual examination of the gears was made every 30 minutes. The test was terminated when a total test gear backlash of 0.002 was exceeded. Various speed-load type tests were run to establish a stress-cycle curve.

Gas chromatographic analyses were made on the gas environment in the spur gear chamber during testing. Certified results made on a Beckman GS-2 Chromatograph indicated there was no more than 0.12% oxygen present in the chamber. Less oxygen was present during the gear tests since any possible leakage during the analysis would increase the measured oxygen content.

3. Worm Gear Test

Worm gear tests were performed on a Nitralloy 135 worm hardened to Rc 70 and a 440C Stainless Steel (Rc 57) worm gear in an unlubricated condition. Helium was passed over the gears at a flow rate of 5 liters/min. Temperature was maintained at 150°F. Output torque was at 44 in-lbs and the output speed at 5.625 rpm. At approximately eight hour intervals, the gears were exposed to oxygen and backlash readings were taken to determine the wear rate. Total time per test was 150 hours.

III. TEST RESULTS AND DISCUSSION:

A. FRICITION OR WEAR TESTS:

Results of wear tests are presented in graphic form in Figure 4. Cycles to failure were represented by a sharp increase in the coefficient of friction to 0.2. Typical curves showing the coefficient of friction and temperature of the cube vs. cycles of operation in shown in Figure 5. It will be noted the frictional force increased rapidly at the point of pending failure while the temperature rise of the cube lags slightly behind. Also included in Figure 4 are the average coefficients of friction for the various lubricants. There does not appear to be any relation between the average coefficient of friction and the number of cycles to failure.

Molykote Thermo M exhibited the longest life of the lubricants tested, however, due to its organic properties, it was not considered for the gear program. It is generally thought that lubricants without an organic binder would be more stable in a radiation environment. Refer to Table I for the available compositions of the commercial lubricants.

Everlube 811, which is a molydisulfide-graphite composition having an inorganic sodium silicate binder, exhibited the second longest life. Lubricants tested with a phosphorous pre-coat exhibited an increase in life of approximately 6,000 cycles over the nonphosphorous treated specimens. However, the manufacturer of the phosphorous coating did not recommend this product at 500°F for prelong usage. In the wear and friction tests, the duration of time at temperature was very short as compared to the time the lubricant will be subjected to in actual service. To preclude the possibility of detrimental time-temperature effects on the phosphorous pre-coat, it was decided to abandon this pretreatment.

After evaluating the performance of the lubricants against all known operating requirements, Everlube 811 was chosen as the optimum lubricant for further evaluation on gears.

B. SPUR GEAR TEST RESULTS:

Gear test results on nitrided and M 50 gears are shown in Figure 6 and Table II. Failure of the lubricant was arbitrarily selected as the number of cycles when an increase of 0.001 in. wear per tooth occurred (0.002 backlash). This wear value was chosen for two reasons: (1) In general, the solid film lubricant thickness is from .0004 to .0006, therefore, to completely evaluate the lubricant, the thickness and substrate must be completely removed. (2) In the visual examination of the gears having .001 wear the majority of the teeth have started to score and only a few remaining teeth have traces of lubricant on them.

LUBRICANT VS. CYCLES TO FAILURE*

Load: 98.7 lbs from 0 to 20.2×10^4 cycles
 137.4 lbs from 20.2×10^4 to failure

Environment: Helium - Temperature: 510°F

Speed: 306 ft/min., 850 rpm

Specimens: 52100 Timken race (lubricated &
 subjected to vacuum of 1 mm Hg)

135 M Nitralloy cube (non-lubricated)

*Failure: Coefficient of friction exceeds .2

FIGURE 4

FIGURE 5

COEFFICIENT OF FRICTION & TEMPERATURE VS. CYCLES

TABLE I.
DRY FILM COMPOSITIONS

LUBRICANT	COMPOSITION
Everlube 620	Molybdenum Disulfide Powder in a Resin Binder.
Everlube 811	Mo S ₂ , Graphite and Sodium Silicate Binder.
NAMC-23A	Mo S ₂ , Graphite and Sodium Silicate Binder.
Molykote X15	Mo S ₂ , Graphite and Sodium Silicate Binder.
Molykote Thermo M	Mo S ₂ , Graphite and Silver in Epoxy Resin.
Molykote X-106	Mo S ₂ in Epoxy Resin.
Dry Lube	Mo S ₂ in Various Resin.
Poxy Lube #75	Mo S ₂ in Epoxy Resin.
Micro Seal	Synthetic Graphite in Ceramic Base Binder.
Electrofilm 4253-W 2	Graphite, Silver, Indium in Resin Binder.
Electrofilm 2006	Graphite, Mo S ₂ in Silicon Binder.
Electrofilm 77-S	Mo S ₂ in Resin Binder.
Electrofilm 2008	Electrofilm 4253-W 2 and Electrofilm 2006.

Typical tooth wear vs. cycles on spur gears at various loads operating at 300 rpm are shown in Figure 7. It will be noted the wear rates are initially greater due to lapping of the high spots on the tooth surface, however, the wear decreases to a normal rate corresponding to normal wear of the lubricant. Upon removal of the lubricant film, which results in metal to metal contact, the wear rate increases.

Tooth wear is illustrated in Figure 8 showing the comparison of a gear prior to failure and one beyond the point of failure. Scoring did not occur until the entire working area of a gear tooth had its lubricant worn off. The effect of base material on the performance of the lubricant is apparent in Figure 6. Everlube 811 applied to M 50 tool steel gears shows an appreciable advantage over the same lubricant applied to nitrided steel gears. Surface hardness of the nitrided steel was Rc 70 while hardness of the M 50 steel was Rc 57. In an extended test period beyond failure, pitting was observed in the pitch line region of the M 50 gear and severe wear occurred on the addendum of the driving gear. Equal gear test results were obtained from two other lubricants having the same composition as Everlube 811, namely; Molykote X15 and NAMC-23A.

An increase in speed from 300 to 1000 rpm did not appreciably effect the cycles to failure as shown in Figure 6. Also, the appearance of the lubricant on the working surface of the teeth did not show any dependence on the various sliding velocities that occur on the gear tooth surface. In general, the lubricant was removed in radial strips extending from the root to the top land of the tooth.

Inaccuracies in the mounting of the gears will cause end loading on the gear teeth, which is readily apparent when the lubricant begins to wear. Premature failure usually occurs when the gear teeth have an uneven load on them causing concentrated surface stresses.

Tests were conducted to determine the effect of coating one surface as compared to coating both mating surfaces. Nitrided gears (12 D. P.) were used in this investigation. Tests were also conducted on a gear having a mirror like finish vs. a coated gear. Results indicated that gears having lubricant on both surfaces exhibited a longer life. Surface finish of the uncoated gears did not effect the lubricant life as shown in Figure 9.

Results of tests conducted on 16 pitch Halcomb 218 gears were inconclusive because the change in backlash, which was used as the criteria in evaluating the lubricants, was not indicative of the severe wear apparent on the dedendum and addendum of the gear teeth. The severe wear was due to the relatively low hardness of 52 Rc on the gear teeth.

FIGURE 6

TABLE II.
SPUR GEAR TESTS

LUBRICANT	Gear Material	Load (lbs/ in of face)	Speed rpm	Cycles to Lube Failure
Everlube 811 vs Everlube 811	Nitralloy	200	300	525, 000
Everlube 811 vs Everlube 811	Nitralloy	300	300	220, 000
Everlube 811 vs Everlube 811	Nitralloy	400	300	180, 000
Everlube 811 vs None	Nitralloy	200	300	160, 000
Everlube 811 vs None	Nitralloy	400	300	75, 000
Everlube 811 vs Polished	Nitralloy	400	300	83, 000
Microseal vs Microseal	Nitralloy	300	300	75, 000
Microseal vs Microseal	Nitralloy	400	300	50, 000
Everlube 811 vs Everlube 811	M-50	200	300	940, 000
Everlube 811 vs Everlube 811	M-50	300	300	600, 000
Everlube 811 vs Everlube 811	M-50	400	300	350, 000
Everlube 811 vs Everlube 811	M-50	500	300	210, 000
Everlube 811 vs Everlube 811	M-50	300	1000	490, 000
Everlube 811 vs Everlube 811	M-50	500	1000	250, 000
Everlube 811 vs Everlube 811	M-50	500	1000	270, 000
NAMC-23A vs NAMC-23A	M-50	300	1000	720, 000
Molykote X15 vs Molykote X15	M-50	300	1000	740, 000

TOTAL WEAR OF GEAR SET VS. CYCLES

Specimens: 2 inch - 12 pitch - 20° full involute spur gear
Material - M 50 (Rc 57)
Lubricant - Everlube 811

Test Conditions: Helium environment - 510°F
Speed - 300 rpm
Hertz compressive stress - noted

FIGURE 7

After Failure

Before Failure

FIGURE 8 - Typical Surface Appearance of Spur Gears

FIGURE 9

LUBRICATED VS. NON-LUBRICATED GEARS C. WORM GEAR TESTS:

Program requirements dictated that worm gear tests be conducted without any form of lubrication on the worm gear set. This was to establish wear rates and efficiency for reactor service machine gears that would be required to be decontaminated with a strong detergent. Tests were conducted at loads equivalent to a Hertz stress of 11,750 psi and a sliding velocity of 73.6 ft/min. Wear of the test gears measured as the change in backlash is shown in Figure 10.

In the first test the contact area at the start was approximately 50% while in the second test the contact area was over 90%. This would account for the initial rapid wear rate of the first test which decreased to a constant value of .000017 in/hr. In the second test the wear rate was initially low, however, after approximately 70 hours of testing, the wear rate increased to a constant value of .000037 in/hr. It is believed this increased wear rate

FIGURE 10

was due to the decreased efficiency of the bearings thereby increasing the output torque and the load on the gear teeth. An efficiency curve of the entire unit vs test time is shown in Figure 11. It is shown that after 70 hours on the second test the efficiency decreased from approximately 13.3 to 9.5% where it remained for the balance of the test. The bearings were grease - packed, however, the majority of the grease was washed from them in MIL 250 thinner to prevent contamination of the system. At the conclusion of the second test the worm assembly was removed and the friction torque of the failed worm gear bearings measured. The friction torque of the unloaded bearings on the output worm gear shaft was 11 in-lbs at the conclusion of the tests.

Typical tooth surface finish of the worm gear and worm after the 150 hour test is shown in Figure 12 were extreme scoring and galling is evident. Galling is the advanced stage of scoring where metal is instantaneously welded and then ripped from the surface leaving plateaus and valleys. Upon examination of the worm gear there were two distinct areas of wear, one was heavily galled and scored as in Figure 12 and the other heavily scored. Hardness readings were obtained and it was found that the galled area corresponded to a 58 Rc hardness and the scored area corresponded to a 56 Rc hardness. Also, circular tooth thickness change due to wear was approximately .002 on the harder surface and .003 on the softer surface.

Tooth thickness measurements were made on the worm and worm gear and there was found to be approximately 57% wear on the worm gear and 43% on the worm from the first test, and 46% wear on the gear and 54% on the worm from the second test.

It was of interest that a high pitch audible sound was emitted from the loaded worm gear set at the initial stage of the run-in of the gears.

FIGURE 11

Worm Gear

Worm

FIGURE 12 - Surface Appearance After 150 Hour Test - Unlubricated

IV. CONCLUSIONS and RECOMMENDATIONS:

A. DRY FILM LUBRICANT ON SPUR GEARS:

Everlube 811 or an equivalent lubricant applied to M 50 spur gears (Rc 57) exhibited the longest life of any of the material-dry film lubricant combinations tested. The following equation which was drawn to include the shortest life cycle test is recommended for the above lubricant-gear combination.

$$N = 10^{\frac{518000-S}{74000}}$$

Where: N = Number of cycles for removal of lubricant

*S = Hertz compressive stress (psi)

*Pitch line velocity $\leq$ 524 ft/min.

S $\leq$ 140,000 psi

S $\geq$ 50,000 psi

A high degree of accuracy corresponding to a Class III gear should be maintained to insure the longest lubricant life. End loading or misalignment will create highly stressed areas and therefore reduce the lubricant's life.

B. NON-LUBRICATED WORM GEARS:

A Hertz stress of 11,750 psi and a sliding velocity of 73.6 ft/min. on a 440C Stainless worm gear (Rc 57) and a Nitralloy 135 worm (Rc 72) will result in a wear rate of approximately .000017 in/hour at 100% contact area. Efficiency of the above worm gear set was approximately 13%. Refer to the Appendix for equations pertaining to Hertz stress, sliding velocity and efficiency of worm gears.

*See Appendix

APPENDIX I.

SPUR GEAR EQUATIONS

From "Practical Gear Design" by D. Dudley

$$S_{\max} = 5715 \sqrt{\frac{W}{d} \left(\frac{mg + 1}{mg} \right)}$$

Where: $S_{\max}$ = Hertz compressive stress for
steel vs steel on 20° pressure
angle spur gear, psi
 W = Tangential load/in. of face, lbs/in
 d = Diameter of gear, in.
 mg = Gear ratio

$$V_p = \frac{d \pi N}{12}$$

Where: V_p = Pitch line velocity, ft/min
 d = Pitch diameter of gear, in.
 N = Revolutions per minute

WORM GEAR EQUATIONS

From "Gears" by H. E. Merritt

$$S_c = \pi F_i / D$$

Where: S_c = Surface Compressive Factor
 F_i = Tangential load/in. of face, lbs.
 D = Pitch Diameter of worm gear, in.

$$S_{max} = 2,250 \sqrt{S_c}$$

Where: S_{max} = Compressive stress for steel vs steel, psi

$$V_s = 0.262 \text{ d.n. sec } \lambda$$

Where: V_s = Sliding velocity, ft/min.
 λ = Lead angle of the worm
 d = Diameter of worm, in.
 n = Revolutions per minute of worm

$$E = \frac{100 (1 - \mu \tan \lambda)}{1 + \mu \cot \lambda}$$

Where: E = Efficiency, %
 μ = Coefficient of friction
 λ = Lead angle of the worm

APPENDIX II

NON-LUBRICATED SPUR GEAR TESTS

Introduction

Tests were required on unlubricated spur gears operating at 16.5 rpm and having a load equivalent to 30,000 psi Hertz applied. Test temperature was 250°F and the test environment was helium.

Test Apparatus

Due to the light loads, a special spur gear tester utilizing a prony brake loading system was used, see Figure I. Speed was fixed at 16.5 rpm. Helium was introduced into the chamber at a rate of 5 cu. ft./hr which is equivalent to a complete change every minute. Temperature was obtained with a strip heater located below the test gears. A controller used in conjunction with a chromel-alumel thermocouple located approximately 1/2 in. from the gear mesh controlled the temperature within $\pm 5^\circ\text{F}$.

Testing

A 12 pitch, 2 in. Nitralloy 135 spur gear was mated with a 12 pitch, 2 in. Stainless Steel 440C gear. The Nitralloy gear had a surface hardness of 70 Rc and the 440C Stainless Steel gear a hardness of 57 Rc. The Hertz load of 30,000 psi was equivalent to a torque of 27.56 in-lbs on the test gears.

Total test time was 150 hours. At approximately 24 hour intervals, the gears were stopped and exposed to air for approximately 1 hour. Visual examination and backlash readings were made periodically and results recorded.

Figure I - Light Load Spur Gear Tester

Test Results

After 150 hours of testing, there was no measurable wear on the gear teeth. The only change was a burnished surface where the teeth made contact as shown in Figure II. Two 150 hour tests were performed with all conditions being the same except the gears were reversed resulting in Nitralloy 135 gear being the driver in the first test and the 440C Stainless gear being the driver in the second. Results of both tests were identical.

Nitralloy 135

440C Stainless

Figure II - Unlubricated Test Gears After Testing