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REPRINT

SOLID LUBRICANT COATINGS

by
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SOLID LUBRICANT COATING STUDIES

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ABSTRACT

Quantitative data on some of the variables affecting the performance of dry-film lubricants are presented. The tests, made on a Falex Lubricant tester, show that optimum performance can be obtained only if the variables affecting their use are known and taken into account.

SOLID LUBRICANT COATING STUDIES

INTRODUCTION

Properly used, solid lubricants are superior to conventional lubricants at high loads and have, in addition, potential value at higher temperatures. Our interest in solid film lubricants was further stimulated by the fact that some of our components must be capable of performing after five years of static storage without relubrication. Solid lubricant coatings have been of considerable value in this area. But optimum performance can be obtained from these relatively new lubricants only if the variables affecting their performance are known and taken into account. Unfortunately, the literature on dry-film lubricants contains almost no quantitative data on these variables. Such data as are available apply to loads of 80,000 psi or above, whereas most Sandia applications are within the 25,000- to 30,000-psi range. This report describes the results obtained on a series of tests conducted in this range. Although still far from complete, the study has shown conclusively that it is possible to control the variables and to secure reproducible quantitative data.

All tests were run with one proprietary lubricant consisting of a mixture of molybdenum disulfide and graphite bonded with a thermosetting resin at a sliding velocity of about 19 ft/min. Tests were made on a Failex lubricant tester, chosen because it makes it possible to vary the load and to measure torque and wear without interrupting the test. Some data were obtained on the effects of each of the following variables: (1) metal pretreatment, (2) hardness, (3) break in or burnishing, (4) surface finish, (5) coating one or both bearing materials, (6) load, (7) humidity, (8) use with nylon bearings, (9) retouching, and (10) use with oil and grease.

THEORY AND BACKGROUND

The excellent lubricating qualities of graphite have long been known; interest in molybdenum disulfide, MoS_2 , as a lubricant is more recent. Both of these materials have a laminar structure. Each lamina of MoS_2 is a sandwich made up of layers of sulphur atoms separated by a layer of molybdenum atoms. The forces holding the atoms in each lamina together are stronger than those attracting the sulphur atoms in one lamina to those in another.¹ This structure is believed to be responsible in part for their lubricating qualities.

Solid lubricants are dispersions of these laminar solids or lubricating pigments in a thermosetting resin. The lubricant is normally sprayed on the bearing surface and then cured at a temperature suited to the resin. The wear and friction characteristics of the resulting dry film vary with the resin used and the molybdenum disulfide-graphite ratio.

From the number of theories in existence, each of which has staunch supporters, it is apparent that the mechanics of friction and wear are complex and little understood. Basically, however, a lubricant has two functions: to reduce or eliminate galling and seizing and thereby increase wear life; and to lower the frictional force between sliding surfaces. In addition, a lubricant may in some applications act as a coolant and also carry debris. Lubrication is usually accomplished by placing a film, either fluid, semifluid or solid, between the two moving surfaces. This study is limited to the solid films.

¹Feng, I. Ming, "Lubricating Properties of Molybdenum Disulfide", Lubrication Engineering, 8 No. 6, p 285 (1952)

A theory which explains many of the phenomena of solid frictions is given by Bowden and Tabor.² An approximation of this theory assumes that when two metals come in contact under load, they weld at a certain number of high points and the friction between them is the resistance to shearing of these welds.

Figure 1 illustrates schematically, the application of this theory to solid lubrication. It shows that the frictional force in a two-material bearing system is dependent on the shear strength of the softer material. In such a system, the frictional force is equal to the shear strength of the softer material times the area of contact: $F = AS$. If hard metals are used as the bearing materials, the area of contact (A) is small but the shear strength (S) is high and, as a result, the frictional force (F) is high. If, however, two hard metals are used but are separated by a thin film of low-shear material, a soft metal or solid film lubricant, both S and A are small and F is correspondingly reduced. A discussion of the application of this theory to solid lubricants is given by Campbell.³

Fig. 1 -- The principle of dry-film lubrication²

²Bowden & Tabor, "The Friction and Lubrication of Solids, Oxford Press, p 112, (1950).
³W. E. Campbell, "Solid Lubricants", Lubrication Engineering, p 9, 195, Aug. (1953).

The Falex lubricant tester used in these tests consists essentially of a pin rotating between two Vee blocks at a sliding velocity of nineteen feet/min. The loading mechanism on the Falex tester is free to swing about its axis. Torque is measured in inch pounds by a siphon-operated gauge, which resists the motion of the loading mechanism. The most important variable on the block is that of the "V" block angle. This was carefully controlled in all tests.

The diameter of the pins and the angle and surface finishes of the Vee blocks were checked by our Standards Laboratory and routinely checked by our inspection organization. All test specimens were degreased, even before sandblasting. When specimens were sandblasted, they were later flushed with a stream of trichlorethylene before vapor degreasing. All pretreatments were given in conformance with the manufacturers' recommendations. No specimens were touched by hand after sandblasting; rubber gloves and support wires were used during the processing of the specimens.

All specimens were rinsed in thinner prior to mounting in a rotating chuck to be sprayed. Coating thickness was measured with a Pratt Whitney Super Micrometer; the diameter was measured at at least ten points before spraying and after curing of the solid lubricant. The general requirement for coating thickness was 0.3 mil $\pm$ 0.1 mil. However, specimens were evaluated on the basis of wear in addition to coating thickness. When a new batch of specimens was received, their wear life was checked against previous batches under known conditions. If, on the basis of life, the new specimens varied from the old, they were rejected. The rotating specimen was the only coated bearing unless stated otherwise. The test specimens were burnished at a 100 pound jaw load for one minute. Except in those cases where load was a variable, tests were performed at a jaw load of 750 pounds.

Wear life is reported in thousands of revolutions (time in minutes Xrpm of tester) and in time (minutes) to failure. Failure is defined as the point at which the friction doubles and remains at that point or higher for one minute, unless galling occurs earlier.

Except in those instances where wear life was extremely short, reproducibility was of the order of $\pm$ 15%. On tests where wear life exceeded 200 minutes, reproducibility was of the order of $\pm$ 5 - 10%.

In those instances where coefficient of friction as a function of hardness, surface finish or similar variables is presented, the data for short time periods is subject to considerable error, but we feel the indicated trend is significant.

SECTION I

METAL PRETREATMENT

It is advisable to pretreat bearing surfaces before applying solid lubricant. The purpose of the pretreatment is to increase surface area and hence improve adhesion of the solid lubricant to the treated part. In some instances, the pretreatment also offers some protection against corrosion (Section XI).

Proprietary pretreatments for each type of metal are available in large numbers. Since the wear life of the lubricant is to a large degree determined by the pretreatment chosen, quantitative data on these materials are very important. This study includes only a fraction of the available pretreatments, but at least one of each type was studied.

The data and observations on pretreating various metals and alloys are presented under separate headings. In these studies, the alloy being investigated was always used as the rotating pin. The "V" bearings were made of an 1100-type steel with an RMS finish of 5 and a Brinell hardness of 192. Recommendations are tabulated at the end of this section.

Aluminum and Aluminum Alloys

Of the two pretreats tested with aluminum, anodizing, an electrolytically applied oxide, and Alodine 1200, a proprietary conversion film coating of mixed fluorides and phosphates; the latter proved to be far superior. Comparative wear data, Table I, shows that Alodine 1200 increases wear life over anodize by a factor of about 40. These data are in general agreement with the findings of other investigators who report improvements ranging from a factor of 20 to 50.

TABLE I

Pretreatments on Aluminum

750-Pound Load

Rotating specimen aluminum	Wear life thousand revolutions	Coefficient of friction μ
Pretreat Sandblast Degrease Anodize Dry Film	~1.5	0.042
Sandblast Degrease Alodine 1200 Dry Film	67	0.032

Anodizing may be satisfactory for many light load applications. However, if the load exceeds about 1000 psi and a long life is desired, it would be necessary to run development tests to determine whether life requirements can be met.

Steel (Other than Stainless)

Ebonol S, a proprietary conversion coating of oxide, and phosphate conversion coatings were tested on steel. The phosphate conversion gave far better results than did the Ebonol S, Table II.

Zinc Plate (Over Steel Pins)

Results of the tests indicate that zinc plate should not be used for loads above a few hundred psi. Cadmium plating will be checked later, but no significant improvement over zinc is expected. If zinc is to be used, see Table VII for the proper pretreatments.

TABLE II

Pretreatments on Mild Steel

Rotating Specimen 3135 Steel Brinell 137	Wear life thousand revolutions	Coefficient of friction μ
Sandblast Degrease Ebonol S Dry Film	11.9	0.037
Sandblast Degrease Phosphate Dry Film	69.6	0.027
Degrease Phosphate Dry Film	56.6 Max.	

Nickel Plate (on Steel Pins)

The data obtained on nickel plate indicate that at least 0.5-mil thickness should be used. Wear and friction data on 0.2- and 0.5-mil Ni-Plate are reported in Table III.

TABLE III

Pretreatments on Nickel Plate

Rotating Specimen Steel Nickel Plated	Wear life thousand revolutions	Coefficient of friction μ
0.2-Mil Ni. Light Sandblast Degrease Dry Film	5.8	0.043
0.5-Mil Ni. Light Sandblast Degrease Dry Film	8.4	0.043

A comparison of wear data for nickel plate with those for aluminum and steel, Tables I and II, indicates that a better treatment for nickel plate is needed. However, wear life is greatly increased if both bearings are coated with solid lubricant. This technique will be discussed in Section IV.

Copper and Copper Alloys

The pretreatments studied on copper and copper alloys were Ebonol C, a commercially available oxide coating, and indium flash. On brass pins, indium flash proved to be inferior to Ebonol C. The data presented in Table IV show that Ebonol C is so superior to indium flash as to make the latter hardly worth consideration for the loads used.

TABLE IV
Pretreatments for Copper

Rotating Specimen	Wear life thousand revolutions	Coefficient of friction μ
Brass Indium Flash		
Indium-Dry Film	Immediate Failure	
Light Sandblast Indium Flash Dry Film	7.3	0.053
Light Sandblast Degrease Ebonol C Dry Film	31.9	0.039

Stainless and Corrosion-Resistant Steel

In tests of stainless steel, Types 302 and 402, the best results were obtained with Type 402 (see Table V). Performance was considerably improved when both bearings were coated with solid lubricant.

TABLE V
Pretreatments for Stainless Steels

Rotating Specimen (as described)	Wear life thousand revolutions	Coefficient of friction μ
Stainless Steel 302 Sandblast Degrease Dry Film	3.8	0.048
Stainless Steel 402 Sandblast Degrease Dry Film	7.5	0.043
Stainless Steel 402 Against Coated "V" Bearing	43	0.043 Lowest Value

Titanium and Titanium Alloys

The only pretreatment used for titanium alloys was a light sandblast and degrease. Further investigations will be conducted on pretreatments for titanium and its alloys. However, the only safe generalization is that if one of the bearing materials is titanium, both bearings should be coated with solid lubricant. Titanium versus titanium performed very poorly, even with both bearings solid coated (see Table VI).

Magnesium and Magnesium Alloys

No data were obtained on magnesium or its alloys. The recommended pretreatment in Table VII was taken from a supplier's bulletin. Investigation of pretreatments for magnesium is scheduled for a later data.

TABLE VI

Pretreatments for Titanium

Rotating Specimen (as described) Solid Filmed	"V Block	Wear life thousand revolutions	Coefficient of friction μ
Titanium	Titanium (bare)	1	0.064
Titanium	Titanium (solid filmed)	2	0.043
Titanium	4140* (rms 25)	2	0.025
Titanium	4140* (solid filmed)	5	0.053
4140* (500 lbs)	Titanium (solid filmed)	255	0.05
4140*	Titanium (bare)	186	--
Aluminum and Soft Steel	Titanium (bare and coated)	25	0.06

A summary of the appropriate pretreatments to be used on various materials prior to the application of solid lubricants is furnished in Table VII.

*Low alloy steel, hardness R_c 45

TABLE VII

Preferred Pretreatment		
Steel (Other than Stainless)	Aluminum and Aluminum Alloys	Copper and Copper Alloys
Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I	Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I	Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I
Degrease MIL-T-7003	Degrease MIL-T-7003	Degrease MIL-T-7003
Phosphate MIL-C-12968, Type A MIL-C-490C, Gr. I Apply solid lubricant	Alodine 1200 MIL-C-5541 Apply solid lubricant	Ebonol C MIL-C-495 Iridite 7 (**) Apply solid lubricant
Cure (*)	Cure (*)	Cure (*)
Cadmium or Zinc Plate (if used)	Stainless Steel	Nickel and Chrome Plate (based on data obtained on nickel)
	Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I	Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I
Degrease MIL-T-7003	Degrease MIL-T-7003	Degrease MIL-T-7003
Phosphate MIL-C-12968, Type B MIL-C-490C, Gr. I Apply solid lubricant	Apply solid lubricant	Apply solid lubricant
Cure (*)	Cure (*)	Cure (*)
Titanium	Magnesium (XX)	
Light Sandblast 120 grit or vapor hone MIL-C-490A, Gr. II, Type I	Degrease and Corrosion Removal per MIL-T-7003	
Degrease MIL-T-7003	Sand or Grit Blast	
	Dichromate per MIL-M-3171, Type 3	
Apply solid lubricant	Apply solid lubricant	

NOTE: If the parts to be sandblasted are extremely oily or greasy, they should be degreased before they are sandblasted.

(*) According to manufacturer's recommendations.

(**) Pretreat for magnesium has not been investigated. These data were obtained from a solid lubricant supplier.

Sandblast

It has been observed that regardless of the metal to be coated with solid lubricant or of the particular pretreatment chosen, a light sandblast (120 grit or finer) of the bearing surface in advance of pretreatment improves wear life considerably. However, it was also established that heavy sandblasting can reduce wear life. One series of samples which was given heavy sandblast had to be discarded because their performance was so poor. With steels, it was found that more reproducible results were obtained when the specimens were sandblasted before being solid coated than when the phosphate was applied without sandblasting.

SECTION II

EFFECT OF HARDNESS OF BEARING MATERIALS

The data presented in Figs. 2 and 3 graphically illustrate two points regarding relative hardness of bearing materials.

1. Until the rotating bearing becomes harder than the uncoated stationary bearing, the coefficient of friction becomes lower as the hardness of the rotating coated bearing is increased. When the rotating coated bearing becomes harder than the uncoated stationary bearing, the coefficient of friction increases slightly.

Fig. 2 -- Coefficient of friction, μ , versus hardness

2. When very hard, uncoated "V" blocks are used, the coefficient of friction drops as the hardness of the rotating coated bearing increases. Note that when both bearing materials are hard, μ is very low.

Fig. 3 -- Coefficient of friction versus hardness

These data agree with results which one might predict from an examination of the theory described in Figure 1. The data in Figure 4 is in agreement with data presented by Ralph Crump at the Army, Navy Industry Lubricants Conference in San Antonio, Texas, November 6, 1956.

The study showed that other things being equal the length of wear life increases directly with the degree of hardness of the bearing material. This is in agreement with data reported by Ralph Crump.

SECTION III

BREAK-IN OR BURNISHING

Parts sprayed with solid film usually have a dull rough appearance before they are used. When subjected to use under heavy loads these parts burnish down to a finish of about 4-8 micro inches.

As the part burnishes, the coefficient of friction decreases slightly. Figure 4 illustrates the friction change with break-in or burnish for the material pair used. The length of time before the friction drop occurs depends upon the load, speed, and bearing combinations used. X-ray diffraction studies indicated a random orientation of MoS_2 before burnishing. However, after the specimens had been burnished, the material had undergone an orientation similar to that reported by I. Ming Feng⁴ and Campbell and Kozak.⁵

Fig 4 -- Wear and coefficient of friction versus time

⁴Feng, I. Ming, "Lubricating Properties of Molybdenum Disulfide", Lubrication Engineering, 8, No. 6, p 285, (1952)
⁵W. E. Campbell and R. Kozak "The Wear of Carbon Brushes in Dry Atmosphere" Trans. ASME, 70, p 491, (1948)

Data also show that if the solid film has been burnished at high loads, friction is slightly reduced when the same lubricant is subsequently used at lower loading (see Fig. 5). Curves A and B resulted from tests made with the same lubricant, bearings, and loadings. After the dry film had been burnished through the loadings depicted by Curve A, the cycle was repeated. The reduced coefficient of friction in the second cycle is shown by Curve B. This reduction is presumably the result of burnish obtained during the first cycle.

Fig. 5 -- Coefficient of function versus load

SECTION IV

EFFECT OF SURFACE FINISH

Interesting data were obtained on the effect of surface finish when only one bearing has been treated with solid film. In Section II it was shown that the softer bearing should be treated with dry film. This, naturally, suggested the investigation of the effect of surface finish. It was found that with a 3135-type steel with a hardness of Rockwell B85 rotating against uncoated 4130 "V" blocks, $R_c 55$, of various finishes, wear life was a log function of surface finish (Fig. 6, Curve A). When both bearing materials are of hard metal, the effect of surface finish on wear life is slightly less important (Fig. 6, Curve B).

Curves A and B in Fig. 7 also show that the coefficient of friction is a function of surface finish. These data suggest that if the hardness of the two bearing materials differs greatly, it may be desirable to coat both bearings as a means of obtaining maximum wear life rather than to choose the more expensive alternative of obtaining a very fine finish. It is less expensive to apply a coat of solid film than to obtain a 10-20 RMS finish (see Section V).

Fig. 6 -- Wear life versus surface finish

SECTION V

EFFECT OF COATING BOTH BEARING MATERIALS

Coating both bearings was found to minimize the importance of surface finish. It has been suggested that a light sandblast be used on the moving bearing; but when both bearings are coated, the stationary bearing should be given the recommended pretreatment without the light sandblast. This has been tested only with 1100-type steel of the hardness in the order of Brinell 192. Probably the effect would be different with extremely hard steels.

Wear life is increased when both bearings are coated. The friction is higher initially, at all loads, but in most cases the friction finally reduces to the approximate value obtained on the same bearing pair with only the moving member coated. This reduction in friction usually occurs at a point when 15 - 40% of the total life has been expended. There is some evidence that the time required for this friction reduction is dependent on the hardness of the bearing materials e.g. hard bearing materials exhibit the reduction in friction earlier (percentage wise) in the life.

SECTION VI

EFFECT OF LOAD

Load Versus Wear Life

This study showed that as the load is increased the wear life is shortened. The general effect is that doubling the load reduces the life 30 to 50%.

Fig. 8 -- Wear life versus load

Load Versus Coefficient of Friction

Most published data seen by the author before commencing this investigation reported coefficient of friction values below 0.1, usually between 0.04-0.02. These are valid values for the materials tested and the loads and speeds encountered. But note that these values were obtained with case-hardened materials, polished mating bearings, loads between 50,000-100,000 psi, and sliding velocities of 20 feet per minute and faster (see Section VIII). An engineer faced with a design problem in which conditions were the same as those given except that the loads were of the order of a few hundred psi could not figure on a basis of a μ of 0.02-0.04.

The data in Fig. 9 were obtained on the Falex tester. Two separate load gauges were used for these tests and the sylphon gauge which indicates torque in inch pounds was carefully calibrated.

It will be noted that while the coefficient of friction decreases with increasing load, the wear increases. While no completely satisfactory explanation for this apparent anomaly has come to our attention, it appears possible that an increased load would affect the rate or degree to which individual lamellar particles become or remain oriented with the director of motion. In the limiting case degradation of the film may be the determining factor of wear life.

Fig. 9 -- Coefficient of friction versus load

SECTION VII

HUMIDITY

As far as could be found, no data have been obtained previously on the effect of humidity on solid film lubricants. All data presented here were obtained with the same solid film compound and with the same bearing materials. All variables were controlled.

Another factor considered, but not varied, was the storage period at a given humidity prior to evaluation. In this series of tests a "soak" period of at least eight hours was allowed. Although data taken at higher humidities were somewhat erratic, they show that friction increases with humidity to about 50 per cent and is constant from 50-75 per cent RH. No data were obtained at humidities higher than 75 per cent RH.

Effect of Humidity on Friction

Peterson and Johnson,⁶ have investigated the lubricating properties of MoS_2 at high humidities and in the presence of water. Their data show an increase in friction at humidities about 10 per cent, but they report lower friction values at high humidities (75 per cent) than at humidities of 50-60 per cent.

The friction trend obtained in this study is shown in Fig. 10. Data on solid filmed 3135-type steel versus uncoated 1100-type "V" bearings of established wear life show that if a drop of water is placed on the bearing surface during operation, the friction rises abruptly and then falls to about the same as the original value. However, the wear life was always reduced by a factor of 3-5. Other specimens were placed in water overnight, removed, dried, and tested. Friction and wear data on these specimens were the same as for those that had never been placed in water. These data indicate the possible effect of oxidation on the uncoated bearing.

Fig. 10 -- Coefficient of friction versus relative humidity

⁶Peterson, M. P. and Johnson, R. L., NACA, TN 30-55, (1953)

Humidity on Wear

An appreciable increase in wear life occurs between 10 per cent and 25 per cent RH; however, although load appears to be a significant factor, the wear life dropped to a new low at 40 per cent RH (Fig 11). At 75 per cent RH and 750 pounds normal load, the wear life obtained was approximately the same as at 40 per cent RH.

More detailed information on the effects of humidity on friction and wear is needed. If the solid lubricant or the materials and surface finishes were varied, an entirely different behavior might occur. The part played by oxidation is not well enough understood to preclude this possibility. In addition, if the pigment (dry lubricant) were varied, or if more than one pigment were present and the ratios were varied, or if the resin were changed and the pigments held constant, the effect of humidity on friction and wear could be expected to differ from the data presented here.

Fig. 11 -- Wear life versus relative humidity

Table VIII shows that μ for solid filmed steel versus nylon is 0.46 as compared to 0.14 for steel to nylon lubricated.

TABLE VIII

The Effect of Lubricants on Nylon in Contact with Steel

100-Pound Load	Stationary $\longrightarrow$	or "V" Block
Rotating Specimen	Nylon (dry)	Nylon (oiled)
Steel (dry)	0.27	0.14
Steel (solid filmed)	0.46	

SECTION IX

RETOUCHING SOLID LUBRICANT SURFACES

The effect of proper pretreat was well established on tests conducted with an air-dry, aerosol-type solid lubricant compound. On the basis of 100 per cent for the sandblast plus phosphate pretreat, observed wear data were as shown in Table IX.

TABLE IX

Wear as a Function of Various Retouching Techniques

Pretreat	Comparative Wear Life in Per Cent
Degrease Only	15-25
Sandblast Only	60
Sandblast and Phosphate	100

From the above data it appears that if the dry film is to be retouched because of chipping, the operator should know the importance of thoroughly cleaning the surface before applying the lubricant. It may also be wise to abrade the surface with a very fine emery cloth before cleaning and coating it.

If the surface to be retouched is not subject to heavy wear, no difficulty need be expected. If, however, the area to be retouched must undergo heavy wear, the engineer must make his decision on the basis of the extent of damage and the conditions of load and speed. Although no absolute formula can be given for this, limited retouching would probably be satisfactory in most light load applications.

SECTION X

EFFECT OF OIL AND GREASE ON SOLID LUBRICANT

A number of solid lubricated test specimens were stored in MIL-G-3278 grease for 30 days and removed. Half of these specimens were degreased and the other half were wiped so that only a film of grease remained.

The specimens that were not degreased failed immediately at the test loads. Those samples that were degreased showed normal wear life. From this experiment it appears that the deterioration results from the presence of grease at high loads and temperatures rather than from damage to resin during storage.

Degradation of solid lubricant coatings occurred when they were tested while coated with the following fluids and greases:

1. Dimethyl silicones
2. MIL-O-6085A
3. Chlorinated silicone
4. Dimethyl silicone-type grease
5. Petroleum base oil

Present data are inadequate to permit much generalization on the effect of grease on dry films. It is, however, probably safe to state that at high loads grease and solid lubricant should not be used in combination. In those instances where solid coatings seemed to perform better when used with grease, the loads and speeds were low and the friction values high. This suggests if the coefficient of friction is too high with a solid lubricant coating and the coefficient of friction is of paramount importance, it would be wise to investigate the use of oil or grease rather than of a solid lubricant coating plus a liquid lubricant.

SECTION XI

CORROSION

Tests on samples which were exposed to adverse atmospheres indicate that very little resistance to corrosion is conferred by a coating of the solid lubricants. Results were somewhat erratic and quantitative data do not appear to be significant. It is suggested that corrosion protection be otherwise provided for in designs which incorporate a solid coating.

CONCLUSIONS

Solid lubricants have many advantages, especially under conditions of high load and temperature. But the variables affecting their performance are so complex that much remains to be learned about them. The study reported here will be continued as time permits. Use of a different solid compound is expected to result in more change in the data on the effect of humidity than on the other data reported here.

A better pretreatment is presently needed for nickel, chromium plate, stainless and corrosion-resistant steels, and titanium and its alloys. This problem will be studied and data reported as available. Friction and wear data on combinations of materials can be obtained from the laboratory if such factors as load and speed are furnished.

The principal conclusions of this study to date are presented in the following list of statements:

1. Increasing the load decreases the coefficient of friction, but it is relatively constant over a wide load range.
2. Increasing the load decreases the wear life. Doubling the load approximately halves the wear life.
3. Increasing the speed decreases the coefficient of friction, but it is relatively constant over a wide range.
4. Increasing the speed decreases the wear life, an effect which is probably dependent on actual bearing temperatures.
5. Ambient temperature (68-100°F) has little effect on the coefficient of friction.
6. The greater the difference between the hardness of the two materials, the higher the coefficient of friction.
7. The harder the two bearing materials, the lower the coefficient of friction.
8. As burnishing takes place, the coefficient of friction decreases slightly.
9. The coefficient of friction increases with humidity to about 50 per cent RH but remains fairly constant from 50 to 75 per cent RH. Optimum wear life is obtained at 20-30 per cent RH. Water has a deleterious effect on a solid filmed bearing in action, but samples that were immersed in water performed satisfactorily if dried before use.
10. If only one bearing is coated:
 - a. The softer bearing should receive the solid film.
 - b. The moving member should be coated when practical unless it is much the smaller.
 - c. The coefficient of friction increases with the surface roughness of the uncoated material.
 - d. The effect of the surface roughness of the uncoated bearing is minimized by using harder bearing materials or preferably by coating both bearings.
11. Coating both bearing materials:
 - a. Gives a higher coefficient of friction, but
 - b. Yields a longer wear life and, is to be preferred when frictional requirements do not prohibit.
12. Coating both bearings is recommended with soft materials (brass, bronze, etc.) and with aluminum, stainless steel, and titanium.

13. Oil and grease should not be used over solid film at high unit loadings. Where grease and oil improve operation at low loading, dry film was probably a poor choice of lubricant.

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