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Influence of film thickness on the performance of solid lubricants

Sunil Vishwanath Mahale

INFLUENCE OF FILM THICKNESS ON THE PERFORMANCE OF SOLID LUBRICANTS

BY

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A

THESIS

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Abstract

The object of this investigation was to determine the effect of lubricant film thickness on the coefficient of friction and on the wear life of solid lubricants. Bonded molybdenum disulphide and bonded graphite were the lubricants tested. Dow Corning's LFW-1 and Falex lubricant tester were the two machines used for testing these lubricants.

The lubricants were sprayed on the specimen surface. Pretreatment, spraying and curing were done according to standard or manufacturer's recommended procedures. During the entire research, the procedure for experimentation was followed according to ASTM No. 2625 on Falex and CRC recommendations on LFW-1 machine.

It was found that the film thickness is an important factor in deciding performance of a solid lubricant. The coefficient of friction increases with increase in film thickness for thicknesses in the range tested (0.0017 in. to 0.0007 in.). For a maximum wear life, there is a definite optimum film thickness, above and below which the wear life decreases.

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NOMENCLATURE

Symbol	Significance	Units
D.L.	Direct Load	Lb.
N	Normal Reaction	Lb.
f	Frictional Force	Lb.
P,Q	Constraint Force	Lb.
T	Torque	Lb.-In.
μ	Coefficient of Friction	--
W	Normal Load on Specimen	Lb.
E	Young's Modulus of Elasticity	$\frac{\text{Lb.}}{\text{In.}^2}$
L	Width of Cylinder	In.
D	Outer Diameter of Ring	In.

I. INTRODUCTION

Between one third and one half of all energy produced in the world is presently lost in friction; however, we can expect this figure to shrink in the face of continued research on solid and other lubricants. Even a minor decrease in friction between moving parts can on a world wide basis, bring about a dramatic reduction in total energy lost due to friction. Since the mid 1930's a strong trend has developed toward using higher and higher temperatures in moving parts. The space program has moved the operation of many systems to the void of outer space. One result of these trends has been the development and use of solid lubricants to attain the necessary lubrication of parts at extreme temperatures and vacuum.

A solid lubricant can generally be defined as a material that provides lubrication between two moving surfaces under dry conditions. W. E. Campbell[1] gives a physical explanation of the mechanism of solid friction. He states that when two surfaces are brought in contact under moderate pressure, the real area of contact is only a small fraction of the total interface area. Pressure welding of two surfaces takes place because of high pressure on the small area of contact and, when relative motion takes place, the friction force is predominently that necessary to shear these welded joints. This friction force is tremendously reduced by using a solid lubricant between the sliding surfaces. Alfred[2] has given a scientific explanation for the performance of bonded solids as lubricating materials. Many inorganic solids, such as graphite, are slippery when finely divided. In powdered form, these materials can fill in pores and valleys of mating surfaces providing on-the-spot lubrication for the peaks and ridges that tend to weld together at high loads and low velocities.

To date, solid lubricants are considered for use only where conventional liquid lubricants can not be used, since liquid lubricants normally result in a lower coefficient of friction. In the immediate future, the solid lubricants will not displace the more conventional lubricants but, rather, they will open up new frontiers to operating machinery. The proper application of solid lubricants will permit the successful operation of machinery under conditions of very high temperature, very high vacuum, nuclear radiation, extreme loads and chemically reactive environments.

Alfred[2] discusses the importance of the layered lattice structure of solid lubricants and he emphasizes that it is the major key to the behavior of this type of inorganic solid lubricants. Molecular bonds within each layer are strong, whereas layer to layer bonds are weak. Thus surface asperities have difficulty in penetrating layers, but slide easily over one another because layer-to-layer bonds of the lubricant are readily sheared. In fact, shearing takes place when loads are high, exactly reverse of hydrodynamic lubrication. This unusual characteristic makes solid films well suited to extreme pressure lubrication. Molybdenum disulphide is a layered lattice material. In this compound, the sulphur layers slide easily on one another. But within the individual layers cohesive forces between Mo & S atoms are very strong and hence resist penetration by surface asperities. MoS_2 adheres well to metal surface because the exposed sulphur atoms have strong affinity for metals. Service life of bonded lubricants depends greatly on the bond established between binder and base metal. Three precoating factors improve this bond; cleaning, mechanical roughening and chemical conversion of surface. In addition to providing micro-roughness to the mechanically roughened surface, chemical treatment gives a cushioning layer that further impedes clean metal 'welding' as surface asperities interfere.

M. E. Campbell(3) defines bonded solid lubricants as a series of lubricating materials consisting of friction and wear reducing solids or pigments, such as MoS_2 or graphite, attached to the bearing surface with a adhesive material. The adhesive material may be organic in nature (phenolic, epoxy, polymide resins) or inorganic (sodium silicate, aluminum phosphate, fused ceramic).

NASA Technology Publication Division(4) has published information about different types of bonded lubricants. The organically bonded materials can be used successfully in high load, low speed sliding applications and for moderate speed at loads in antifriction bearing application. Their useful temperature is somewhat limited from approximately $-100^{\circ}F$ to $+400^{\circ}F$. At higher temperature the lubricating solids tend to oxidize and the resin binder rapidly degrades by heat causing film failure from lack of film adhesion. For improved temperature performance, high temperature polymides are used as the binder solution. The solid lubricants with these binders are very heat stable and exhibit useful life at temperatures to $1000^{\circ}F$ in air and to above $1300^{\circ}F$ in an inert atmosphere.

M. E. Campbell(3) also states that inorganically bonded lubricants can be divided into two subgroups, ceramic and nonceramically bonded. Phosphates and silicates are used as binding material for non-ceramic solid lubricants. Principal advantages of these types of lubricants are; (1) they are not subjected to outgassing in a vacuum of 10^{-9} torr and (2) they are compatible with oxygen. Therefore, they are very useful in the space program. However, they do not afford corrosion protection and the film softens when subjected to water or moisture for extended periods of time.

The object of this research was to determine the effect of film thickness on wear life and on the coefficient of friction for solid film lubricants. An attempt is also made to find the relation between the rate of wear and the film thickness.

II. REVIEW OF LITERATURE

There are many references on various aspects of solid lubrication. The majority of these are listed in references (5) and (6). Only those specifically related to this work are discussed here.

Johnson and Sliney[7] present the results of an experimental study on the effect of film thickness on coefficient of friction and wear rate. Lead oxide was used as the solid lubricant and testing was done on a rider-disc type of tester. Tests were conducted at $1200^{\circ}F$ with a load of 1 Kg. on Ni-Cr-Fe alloy rider at a disc speed of 435 fpm. To achieve good repeatability, the disc was ground flat and parallel to within 0.0005 of an inch and then coated accurately with lead oxide after necessary pretreatments. Figure 1 and 2 illustrate the effect of coating thickness on the coefficient of friction and on rider wear. From the graphs it appears that the wear rate continues to increase with film thickness, without limit. The coefficient of friction increases linearly with film thickness.

Alfred[2] states that line contact and area contact methods closely simulate conditions in actual mechanisms. The results of his tests on electrophoratically coated, inorganically bonded MoS_2 lubricant are shown in Fig. 3. Friction coefficient decreases as normal load increases.

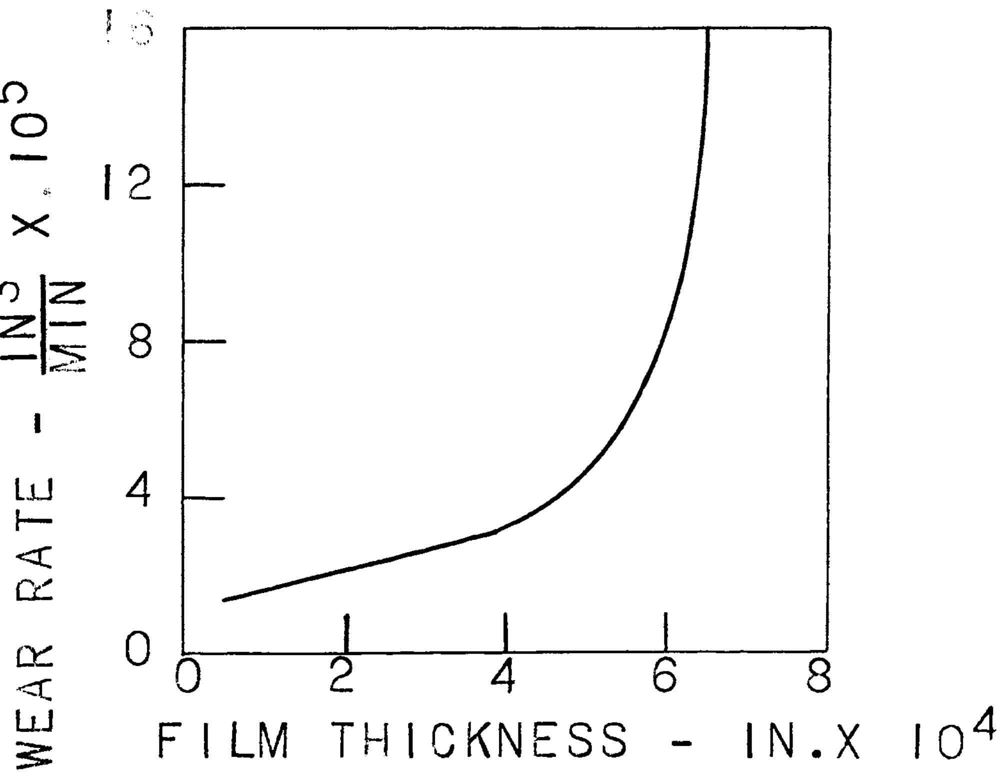

Fig. 1 Film Thickness Versus Wear Rate
for Bonded PbO Lubricant

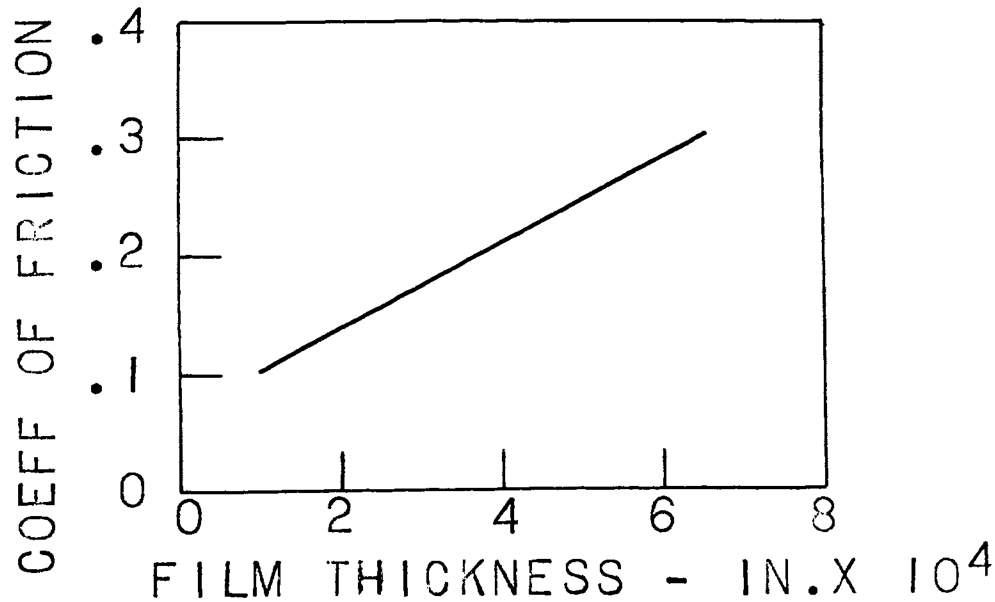

Fig. 2 Film Thickness Versus Coefficient
of Friction for Bonded PbO Lubricant

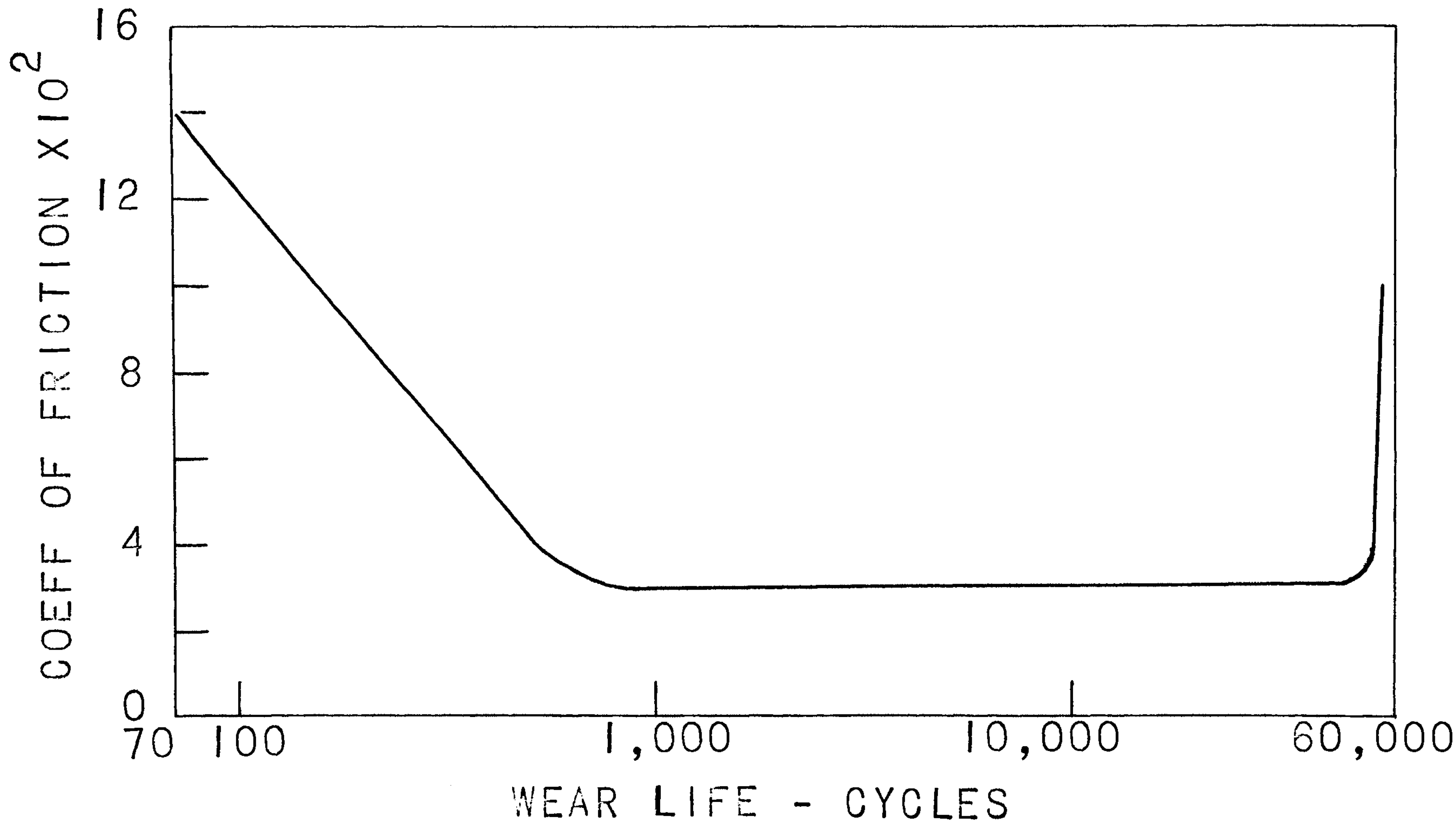

Fig. 3 Wear Life Versus Coefficient of Friction for Inorganically Bonded Lubricant

The flat portion of the curve gives the minimum coefficient of friction for the coating while the sudden upturn of the curve gives wear life.

III. DESCRIPTION OF EXPERIMENTAL EQUIPMENT

A. Dow Corning's LFW-1 Variable Drive Machine

The LFW-1 is a versatile and accurate screening device for evaluation of lubricants and bearing materials in unidirectional motion. It has also been found adaptable for testing under low load, area contact, and oscillating conditions. A photograph of the experimental set up on this machine is shown on Fig. 4. A close up photograph of the ring and block in running condition is shown in Fig. 5.

Schematic representation of actual operation is shown in Fig. 6. Figure 7 is a photograph of two sets of specimens, one unused and the other after testing.

A stationary block is pressed against a predetermined load (maximum 630 lbs.) against a rotating disc. Average Hertz pressure in the line contact area between rectangular specimen and the rotating ring may range up to 11,000 psi. The resulting friction is indicated throughout the test by a dial indicator. A counter records the number of revolutions of the test specimen. A control pointer on the friction indicator can be set for any preselected value of friction and the machine will automatically shut off upon reaching it.

The test shaft of the machine is supported by two roller bearings and the mandrel end of the shaft protrudes through the front panel of the machine where the specimens

Fig. 4 Experimental Set-Up on LFW-1 Machine

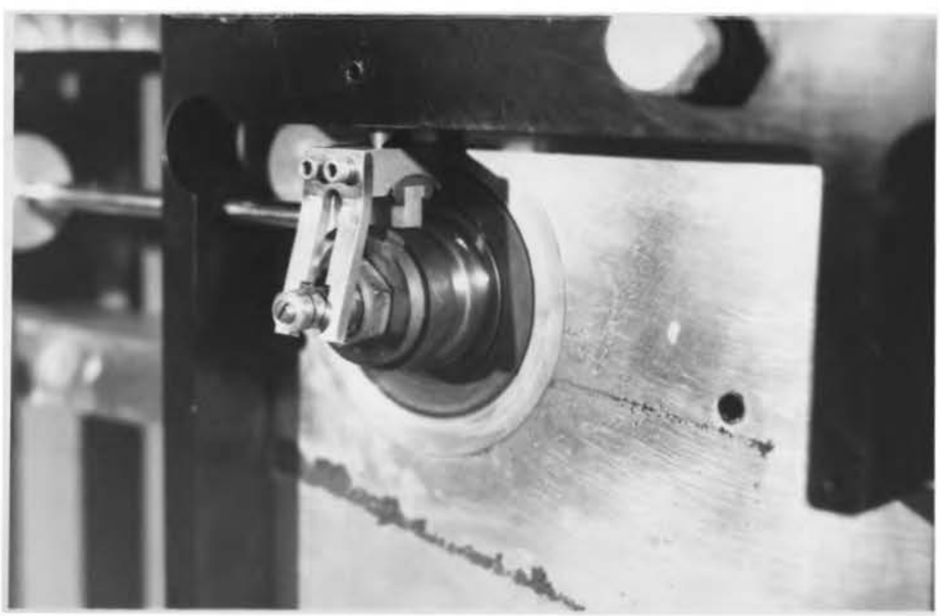

Fig. 5 Close-Up View of Ring and Block Assembly on LFW-1 Machine

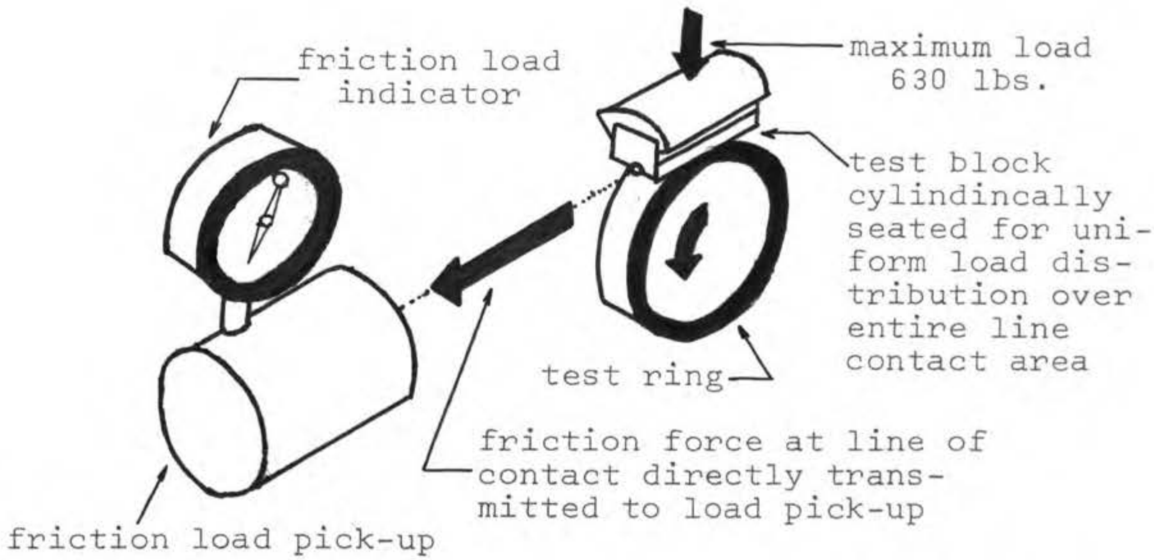

Fig. 6 Schematic Drawing Showing Operational Principle on LFW-1 Machine

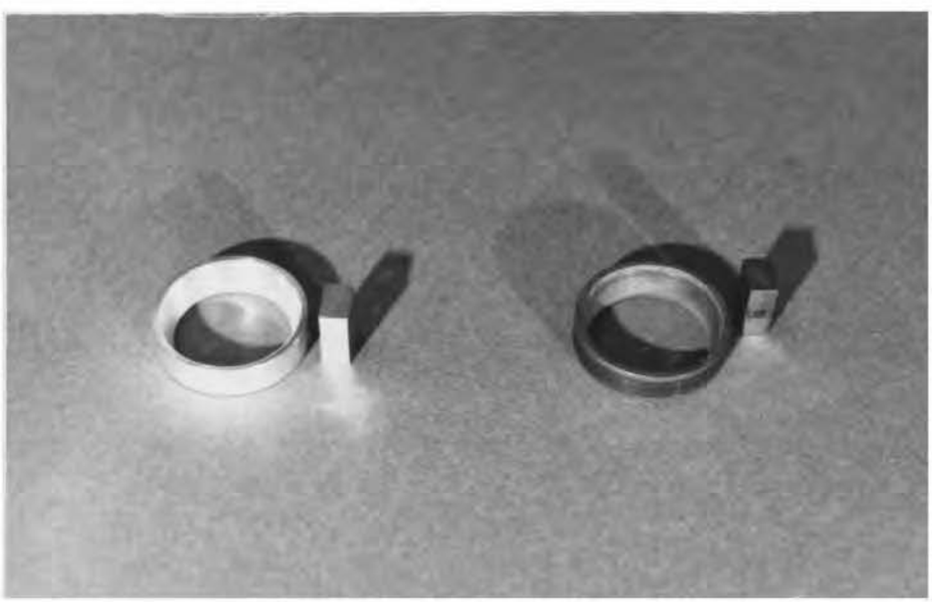

Fig. 7 Photograph of Ring and Block

are mounted. The test block, which is held stationary against the revolving ring, is restrained from horizontal movement by a unique type of holder. The design of this block holder allows the test block to align itself automatically in a manner prescribed by ASTM specifications for compression loaded specimens. This maintains uniform loading throughout the area of contact between the specimens regardless of the force existing between them.

The normal force between the test specimens is produced by hanging dead weights on the lower end of a compound lever system which is designed in such a way as to allow the full value of the friction force to be transmitted to the frictional load pick up device. The compound lever produces leverage of 1:30, due to which a vertical load of 1 lb. produces a normal load of 30 lbs. on the specimens. The friction force indicator reads directly in pounds and is fitted with an infinitely adjustable limit control which allows the operator to preset the value of friction at which the machine will stop. This setting is accomplished without the use of tools by a knurled knob located in the center of the dial face outside of the glass.

A six digit revolution counter is panel mounted in the top of the case. It is driven from the test shaft and is equipped with reset wheel which can be controlled from outside of the case.

Figure 8 is a dimensional drawing of the test ring. The ring is made up of SAE 4620 steel having a Rockwell hardness of HRc 58 to 63. It has a ground face of 0.321 ± 0.0005 in. wide and a diameter of 1.3775 + 0.0001 - 0.0005 in. having eccentricity between the inner and outer surface no greater than 0.0015 in. The surface finish range of the outside diameter surface of the ring is from 5 to 15 micro-in. rms in the direction of motion.

The test block is shown in Fig. 9. It is made up of SAE 01 cold worked 0.250 + 0.0005 - 0.000 in. wide and 0.620 ± 0.0005 in. long having a Rockwell hardness of HRc 30 ± 3. Each block is supplied with the test surface ground to a surface finish of 4 to 8 micro-in. rms, being perfectly square with all outside edges.

B. Falex Lubricant Tester

A photograph of the experimental set-up is shown in Fig. 10. Exploded views of the V-blocks and the journal are shown in Fig. 11. The bearing blocks can be inserted in the short lever arms, or jaws, of the load applying mechanism which fits loosely over bifurcated ends of long lever arms. The ratchet wheel is turned by hand until the loading mechanism takes hold, this is indicated by a registration of applied load on the gauge. Additional load can be applied either by turning up the ratchet wheel by hand, or by engaging the load applying arm with the ratchet wheel. The eccentric motion of load applying arm increases the application

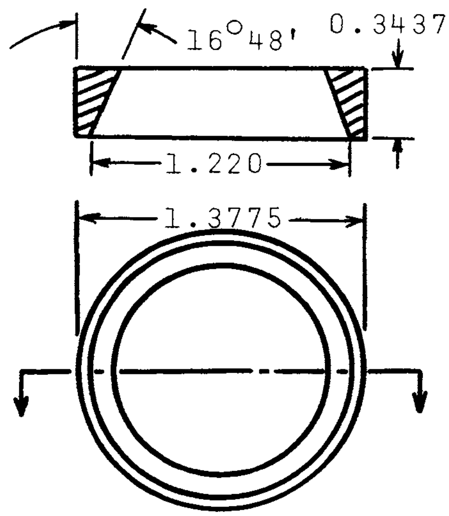

Fig. 8 Dimensional Drawing of Test Ring

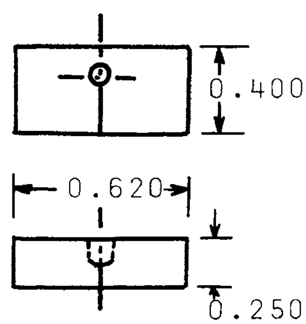

Fig. 9 Dimensional Drawing of Test Block

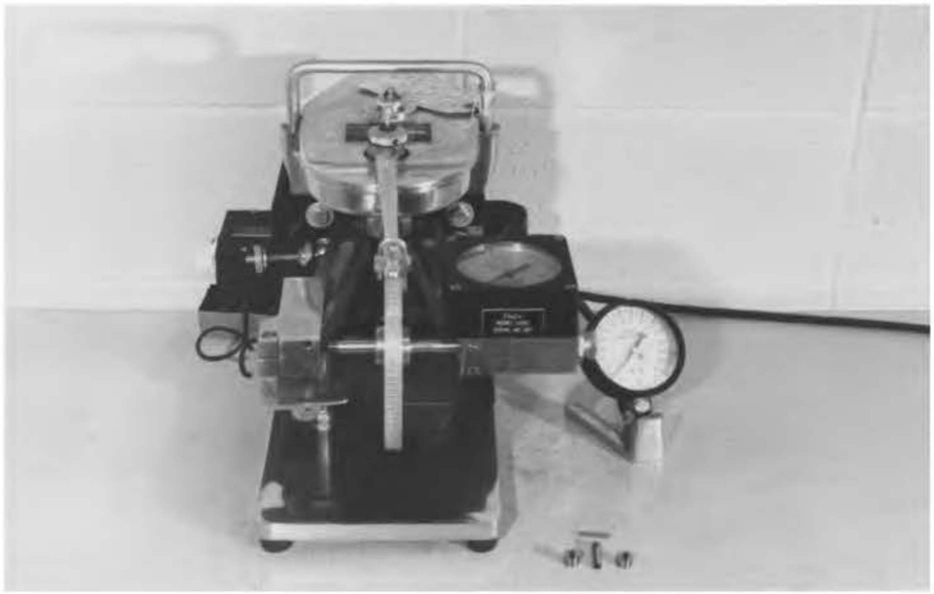

Fig. 10 Experimental Set-Up on Falex Lubricant Tester

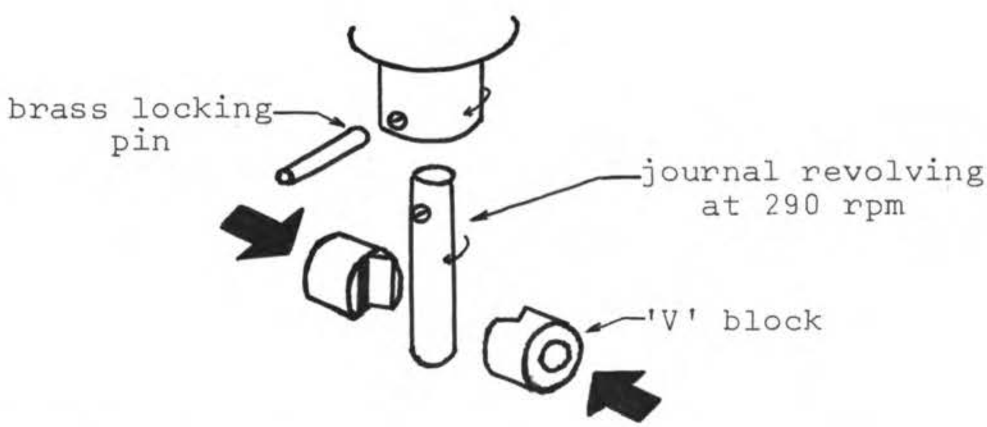

Fig. 11 Exploded View of 'V' Blocks and Journal

of load, one tooth at time. The action is similar to the action of a mechanically operated pair of pliers. The entire mechanism is free to swing about its axis, this tendency to turn being resisted by the syphon operated gauge which registers torque. Instantaneous wear of specimens can be measured from the ratchet wheel. The entire ratchet wheel is divided into 200 divisions, and every 18 teeth on ratchet equals 0.001 in. of wear. Free body diagrams of block and journal are shown in Fig. 12.

The standard test journal for the Falex is 1/4 in. outside diameter by 1-1/4 in. long. It is made up of AISI 3135 steel having a Rockwell Hardness B of 87 to 90 on a ground flat surface with a surface finish of 6 to 10 micro-in. rms. The assembly of journal and blocks is shown in Fig. 12a and free body diagram of journal is shown in Fig. 12b.

The V-blocks are made from AISI C-1137 steel having a Rockwell Hardness C of 20 to 24 with a surface finish of 3 to 6 micro-in. rms. Free body diagram of a block is shown in Fig. 12c.

The locking pin is 1/2 H brass, confirming to ASTM specification B-16 for Free Cutting Brass Rod, Bar, and Shapes for Use in Screw Machines.

Figure 13 is a photograph of two sets of specimens one unused and other after testing.

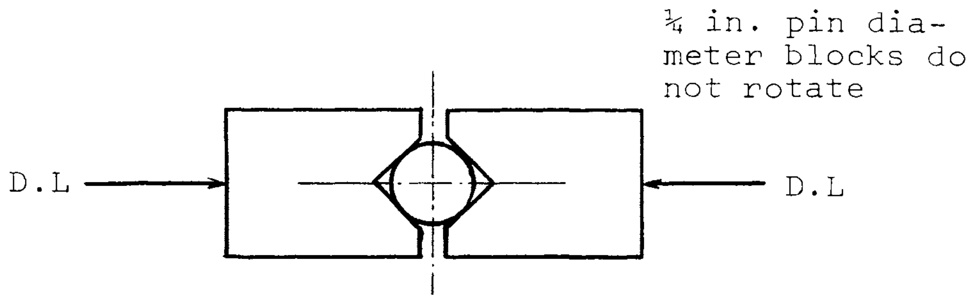

Fig. 12a Assembly of Test Blocks and Journal

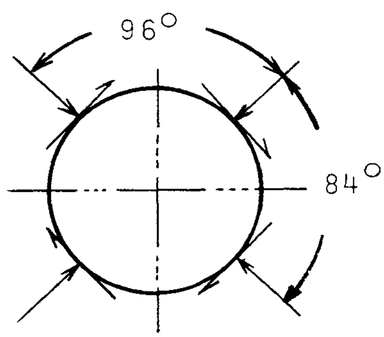

Fig. 12b Free Body Diagram of Journal

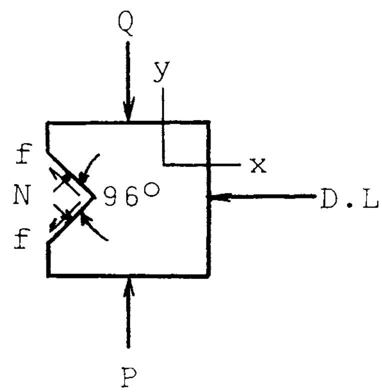

Fig. 12c Free Body Diagram of Test Block

From Fig. b - $\Sigma M_o = 0 = T - 4\,f(\frac{1}{8}) \quad \therefore \; f = 2T$

From Fig. c - $\Sigma F_x = 0 = D.L - 2N\cos 42^\circ \quad \therefore \; N = \frac{D.L}{2\cos 42^\circ}$

$$\mu = \frac{f}{N} = \frac{2T(2\cos 42^\circ)}{D.L}$$

$$= 2.9724\,\frac{T}{D.L}$$

Fig. 12 Coefficient of Friction Calculations on Falex Lubricant Tester

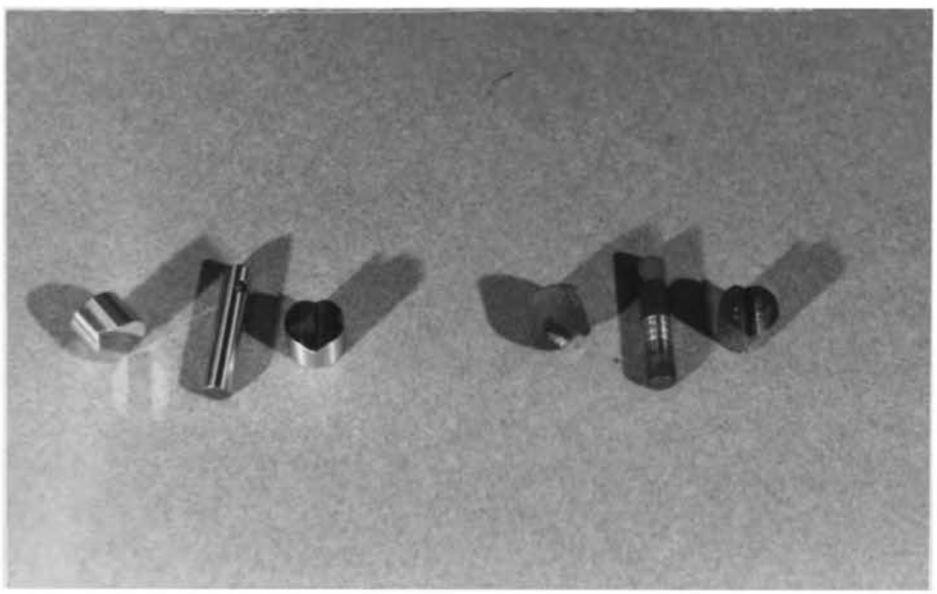

Fig. 13 Photograph of Journal and Blocks

C. Magne Gauge

For measuring lubricant film thickness a Magne gauge was used. A photographic view of this instrument is shown in Fig. 14. Accuracy of this instrument is 0.0001 in. for nonmagnetic film thickness on mild steel base. Measurements are not appreciably affected by the thickness of the base metal if it is over 0.01 in. thick.

The operating principal is that the attractive force between the magnet and specimen is inversely proportional to the film thickness. The force of attraction is balanced by uncoiling a coiled spring. The degree of rotation required is marked on a dial. Knowing the dial reading, film thickness can be determined from calibration curves supplied with the gauge.

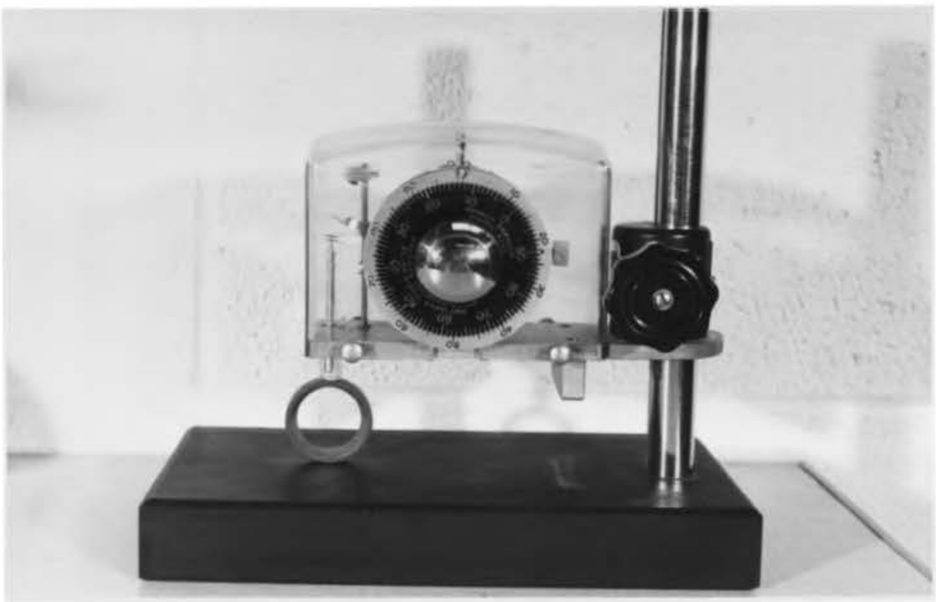

Fig. 14 Measurement of Film Thickness on Magne Gauge

IV. EXPERIMENTAL METHOD

A. Preparation of Specimen and Application of Lubricant

1. Degreasing

Test specimens were degreased in trichloroethylene bath. Then they were suspended in the vapor of boiling benzene for 30 seconds and immersed in boiling methyl ethyl ketone for 30 seconds. After air drying, specimens were again degreased in a trichlorethylene bath and finally wiped with lint free cloth.

2. Phosphating

After decreasing, the specimens were treated with Parco Lubrite #5 solution. The process of treatment consist of immersing the properly degreased work in the Parco Lubrite solution at 180-185°F for approximately 10 minutes to form a uniform protective coating. It is very important to maintain the temperature in the range of 180-185°F. Boiling the solution will cause excessive use of Accelerator, and not keeping it up to proper temperature will result in a nonuniform coating.

Following the Parco Lubrite treatment, the work is immersed in water for 30 to 60 seconds. A cold rinse is satisfactory, however, if the work is to be dried immediately, a hot rinse is preferable to facilitate drying. It is desirable to rinse the work as soon as possible after processing, as the Parco Lubrite solution tends to set up on hot metal making it difficult to rinse. The specimens were oven dried immediately after water rinse. The temperature of the oven was maintained below 225°F. The thickness of the phosphate coat was measured with the Magne gauge.

3. Application of Coating and Measurement of Thickness

The lubricants, namely Electrofilm Lubri Bond 'A' (molybdenum disulphide base) and LaFrance Franlube (graphite base) available in aerosol cans, were sprayed on the surface of the specimens. To achieve uniform coating, the specimens were rotated at approximately 360 rpm. The film was air cured for 8 hours and the specimens were kept in a dissicator for the next 16 hours to eliminate detrimental effects of humidity on the wear life.

The total thickness of phosphate and lubricant coat on the surface of specimen was measured at several points. An average of all these readings was considered as the final thickness.

After complete treatment and measurement of coating thickness, the ring and block of LFW-1 machine were weighed to the nearest 0.1 mg. on an analytical balance. Weighing of the specimens was done again after evaluation on the testing machine.

B. Test Procedure

1. LFW-1 Machine

Before each test, the specimen holder, the tapered section, threaded section, lock nut and washer of the machine were cleaned in benzene and rinsed in methyl ethyl ketone. The threaded portion and brass bushing was lubricated with molykote lubricant. The test ring was mounted on the shaft. The test ring on the shaft was locked with 250 in. lb. of wrench torque. After inserting the specimen block in the quarter segment, the holder was inserted in the seat without marring the surface of the ring. The clearance between lower lever and lower safety bolt was kept between 1/16 to 1/8 of an inch which insures that the lower lever will be horizontal when the bale rod is fully loaded.

The fixed reference of the lever system was aligned before loading the specimen. When the markers were aligned, the friction pin was backed off to reset the dial gauge to zero. This insured that there was no preload on the friction measuring system.

The specimens were initially loaded with 60 lbs. The speed was gradually increased to 72 rpm. The load was then increased so that full load of 630 lbs was applied at the end of 10 minutes. The weights were put on gradually to prevent shock loading. After every load change, the index and fixed reference was readjusted. The speed was rechecked after every load change and adjusted to 72 rpm. The number of revolutions were recorded from the counter. Frictional force was also recorded after each interval of 50 cycles. The test was allowed to continue until the coefficient of friction reached 0.1. The tested specimens were weighed to the nearest 0.1 mg.

The loading conditions were changed while testing graphite lubricant, as the quality was not good compared to Electrofilm lubricant. Load was 30 lbs. initially, and it was increased to 240 lbs. in 8 minutes. Automatic shut off was adjusted for a coefficient of friction equal to 0.2.

2. Falex Lubricant Tester

Before each test, the journal holder and the recess in the loading device were cleaned with benzene and rinsed in methyl ethyl ketone. Solid film coated 'V' blocks were inserted in the recess of the loading device. The solid film coated test pin was inserted in the mandrel and was held in position by a brass shear pin.

After positioning the loading device, the ratchet wheel was turned by hand until the loading mechanism engages (indicated on load gauge). The load applying arm was positioned and drive motor was energied until gauge load of 250 lbs. was reached. The test was continued at the constant load of 250 lbs. The number of teeth required to maintain this load was recorded as instantaneous wear. Average wear rate was determined by dividing the total teeth wear by wear life. The wear life or failure was indicated by a torque rise of 10 in. lb. above steady state value.

V. RESULTS

A. Electrofilm Lubricant - LFW-1 Machine

Condensed data of all tests is tabulated in Table I. The maximum Hertz pressure developed between block and ring (line contact) was 10,900 psi. The minimum coefficient of friction is the lowest value obtained during the entire wear life of operation and it is horizontal part of wear life versus coefficient curve. Figure 15 shows the effect of an increase in load on the frictional force. It can be seen that for increasing thickness, the effect is negligible at lower loads, but becomes considerable at higher loads. Effect of film thickness on wear life can be studied from Fig. 16. The range of coefficient of friction after 10 minutes was 0.0295 to 0.175, but at optimum condition it was 0.03 to 0.07. The drop in coefficient of friction for thicker coatings is considerable. A plot of minimum coefficient of friction versus film thickness is shown in Fig. 17, from which it can be seen that coefficient of friction increased linearly with increase in film thickness. A curve of film thickness versus wear life is shown in Fig. 18. The wear life was maximum at a film thickness of 0.00028 in. At greater thicknesses the wear life dropped drastically. Figure 19, graphically shows the relation between film thickness and wear rate. The wear rate increases linearly at lower thicknesses but the relation becomes exponential at higher thickness. Initial wear in thicker coatings is

Lubricant:	Electrofilm Lubri Bond 'A'
Machine:	LFW-1
Speed:	72 RPM
Max. Load:	630 Lbs. (After 10 Minutes)
Specimen:	Ring and Block
Room Temp.:	80 ± 5 $^{\circ}$F

Thickness in.	Wear Life Cycles	Min. Coeff. of Friction	Wear Rate mg/minute
0.00017	19,360	0.0413	0.108
0.000245	23,729	0.0333	0.114
0.000265	24,105	0.0485	0.119
0.00028	24,204	0.031	0.122
0.000305	23,912	0.0532	0.129
0.00039	21,603	0.0619	0.129
0.000475	16,332	0.0682	0.155
0.00049	14,877	0.0715	0.166
0.0007	2,153	0.0906	0.280

Maximum Hertz Pressure: $0.465\sqrt{\frac{WE}{LD}}$

$$= 0.465\sqrt{\frac{630 \times 30\times10^{6}}{0.25 \times 1.3775}}$$

$$= 10{,}900 \text{ psi}$$

Table I. Data for Electrofilm Lubricant on LFW-1 Machine

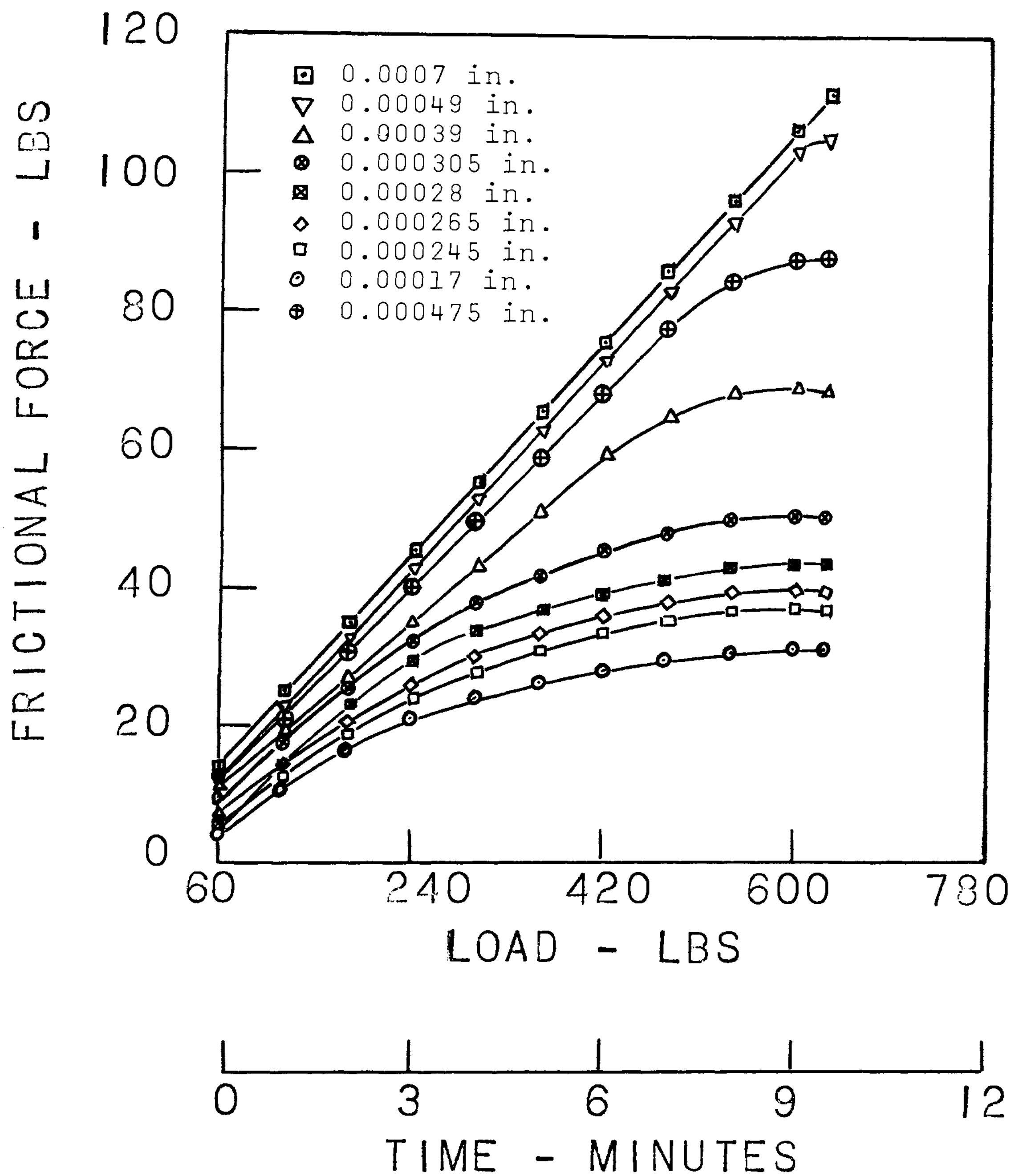

Fig. 15 Effect of Increase in Load on Frictional Force for Electrofilm Lubricant

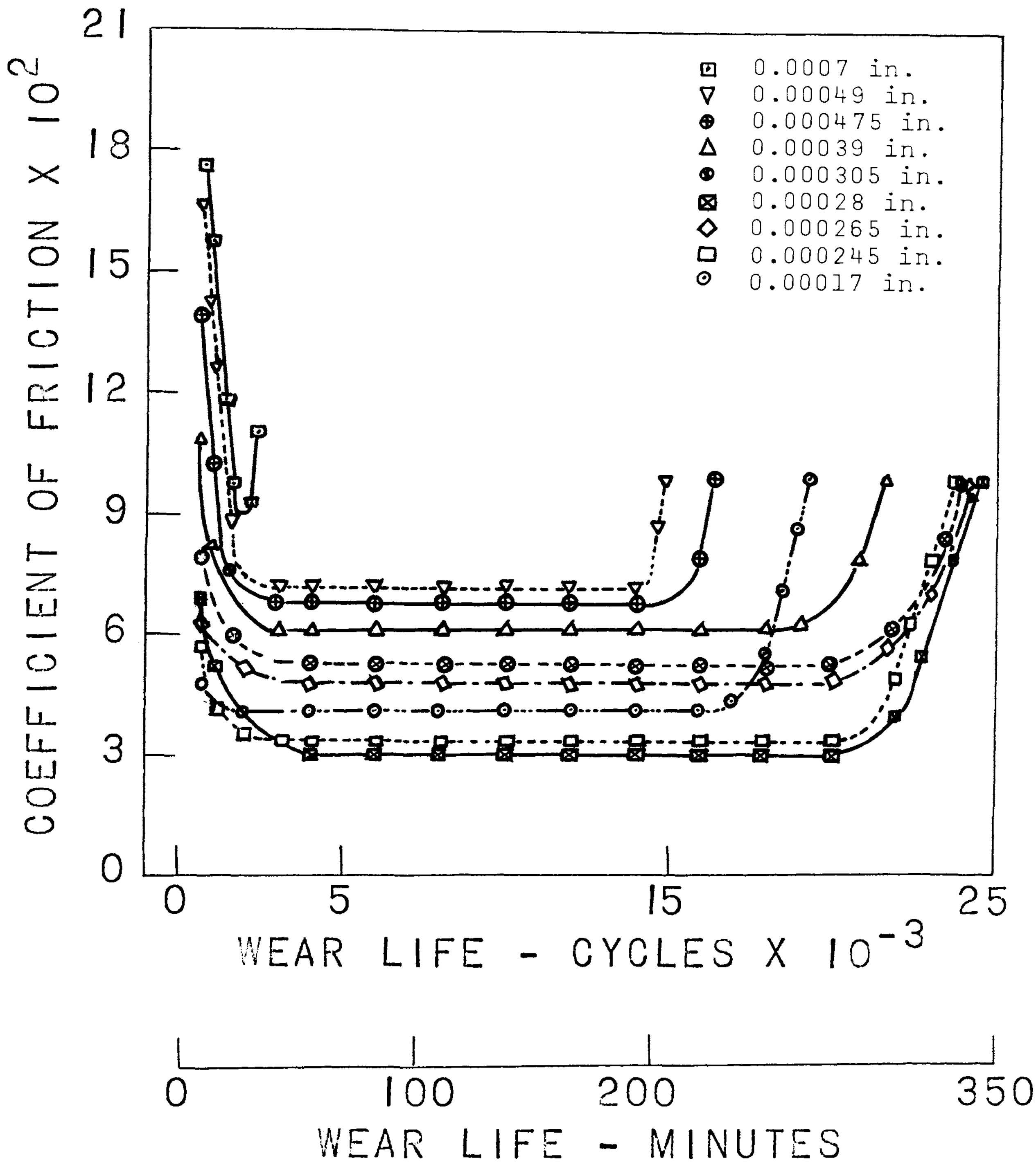

Fig. 16 Wear Life Versus Coefficient of Friction for Electrofilm Lubricant

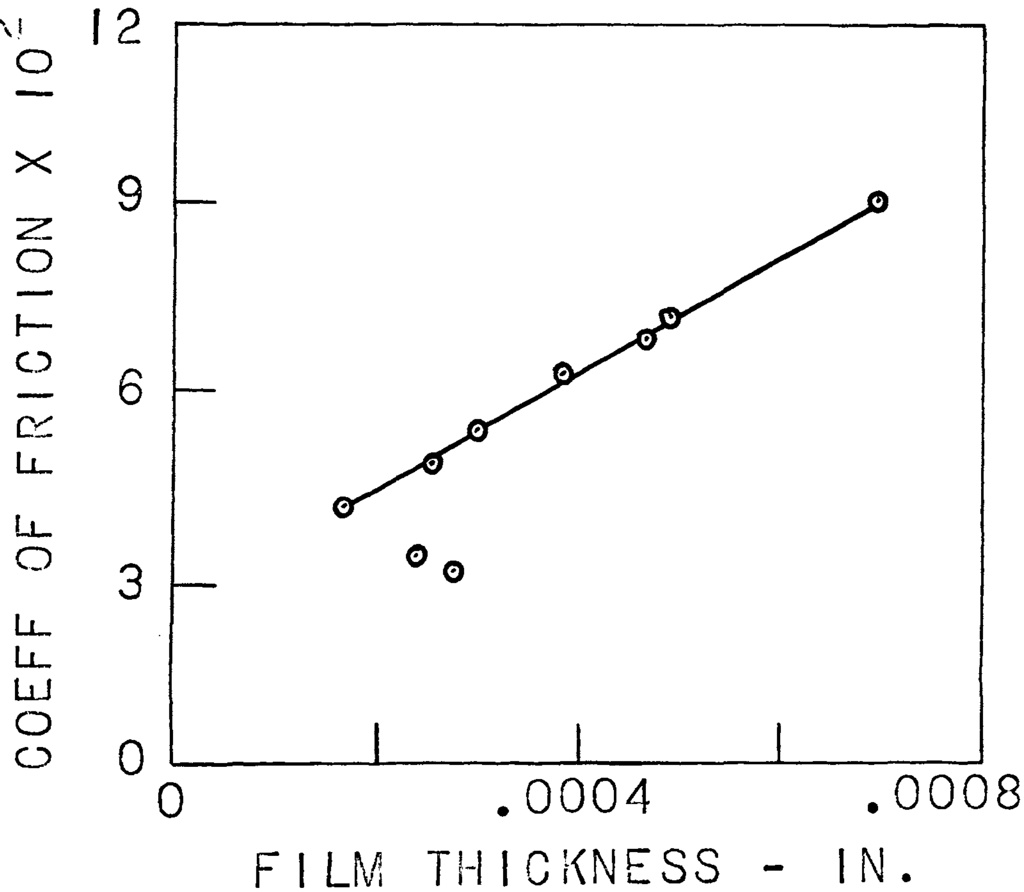

Fig. 17 Plot of Minimum Coefficient of Friction Versus Film Thickness for Electrofilm Lubricant

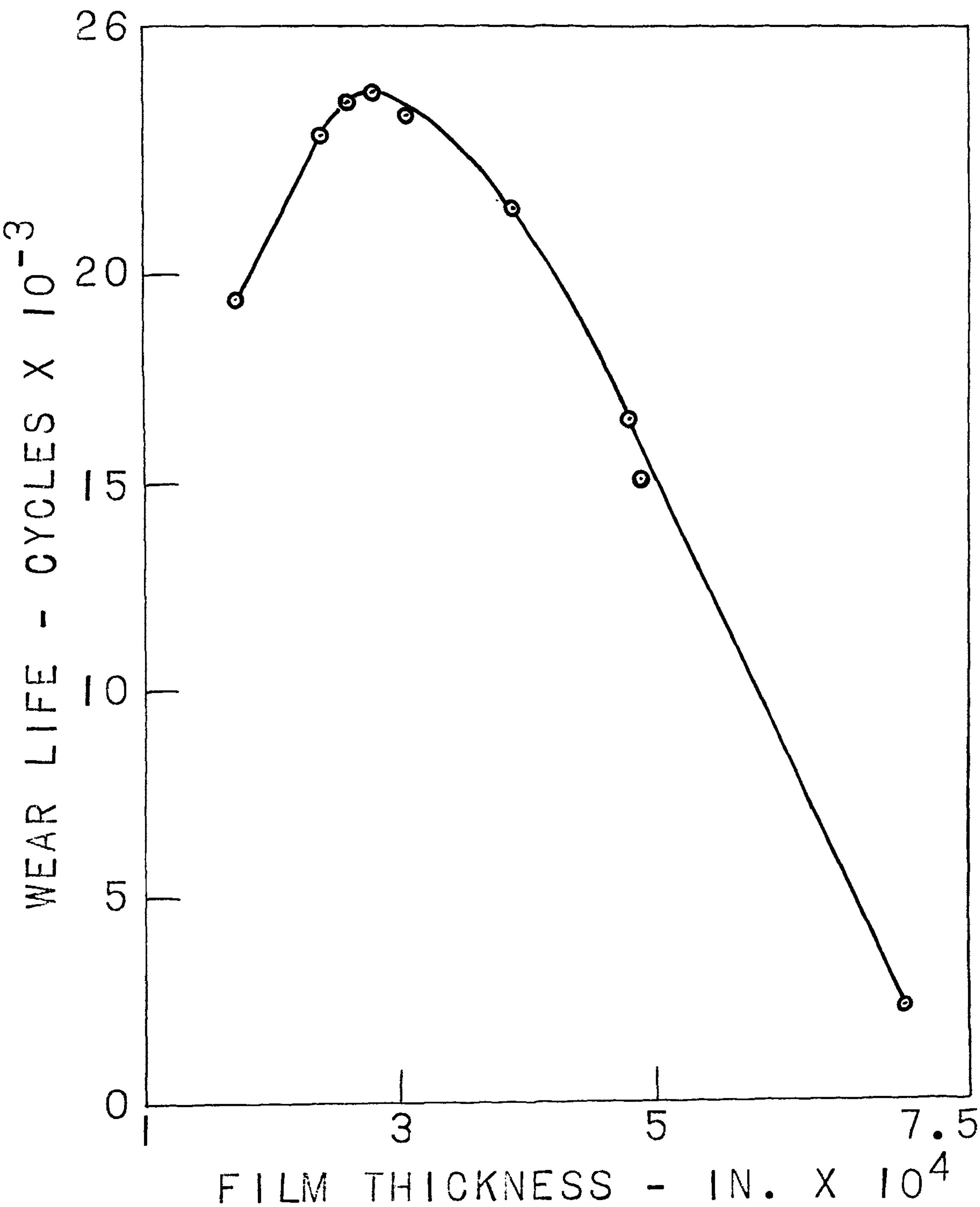

Fig. 18 Effect of Film Thickness on Wear Life for Electrofilm Lubricant

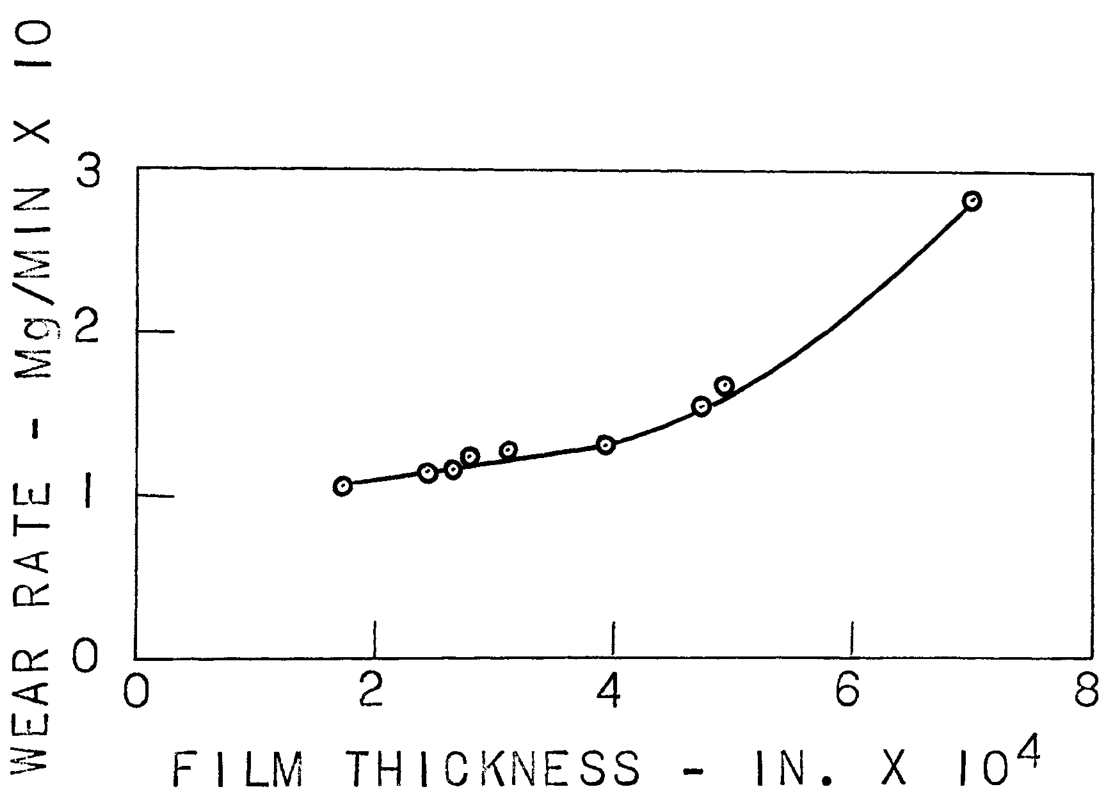

Fig. 19 Effect of Film Thickness on Wear Rate for Electrofilm Lubricant

observed to be high, which in turn effects the load carrying capacity. For continuous operation the film further deteriorates with increase in time. Higher wear rate is the main reason for lower wear life in thicker coatings.

To study the effect of wear on the solid film and on the surface of the ring, microscopic photographs of the ring surface were taken at different stages. Figures 20 to 23 are the photographs at 100 magnification taken by an ordinary microscope. Photographs from an electron microscope at 1000 magnification are exhibited in Fig. 24. The original surface of the ring after perfect degreasing at 100X and 1000X can be seen in Fig. 20 and 24a respectively. The white lines in black background are due to machining roughness. For improved wear life, the surface should be perfectly smooth, which can be achieved by sand blasting of the specimen. Figure 21 and 24b are the photographs taken just after coating the surface with solid lubricant. In these figures can be seen the various size particles of the components contained in the lubricant. Fig. 22 and 24c respectively are the photographs at 100X and 1000X of the ring surface at optimum performance, as defined as the condition of minimum coefficient of friction. The lubricant has been compacted and spread by the rubbing action of two mating surfaces. Photographs at 100X and 1000X of the surface at failure of the solid lubricant film are shown in Fig. 23

Fig. 20 Bare Ring Surface after Degreasing at 100X

Fig. 21 Ring Surface after Coated with Solid Film Lubricant at 100X

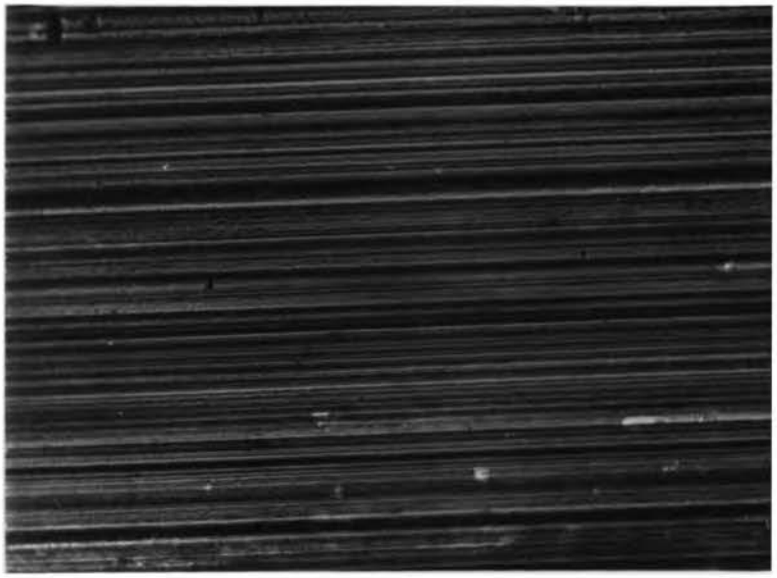

Fig. 22 Ring Surface at Optimum Performance
Taken at 100 Magnification

Fig. 23 Ring Surface at the End of Wear Test
(100X)

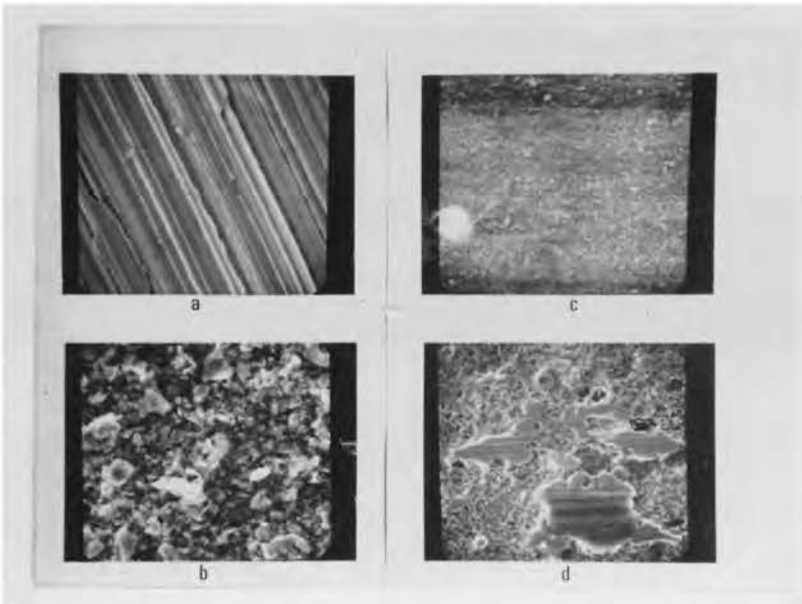

Fig. 24 Microscopic Photographs of the Ring Surface from Electron Microscope (1000X)

a. Bare Ring Surface after Degreasing

b. Ring Surface after Coated with Solid Film Lubricant

c. Ring Surface at Optimum Performance

d. Photograph of the Surface at the End of Wear Test and 24d respectively. For these tests lubricant failure was defined as the point at which the coefficient of friction was equal to 0.1. Definite portions of the ring surface from which the lubricant has been completely removed can be seen in these photographs. The corresponding metal-to-metal contact resulted in the higher coefficient of friction.

B. Graphite Lubricant - LFW-1 Machine

Table II lists the important data obtained during these tests. The maximum Hertz pressure developed was 6750 psi. at the line of contact. Effect of increase of load for a constant speed of 72 rpm is shown in Fig. 25. For greater thicknesses the curve tends to straight line with increasing slope. A higher coefficient of friction at same load and thickness was obtained in comparison with Electrofilm lubricant. Effect of film thickness on wear life can be studied from Fig. 26. Even at low Hertz pressure, low wear life indicates the importance of quality of bonding material. After eight minutes, the range of coefficient of friction was from 0.13 to 0.26 but at optimum condition it dropped down to 0.075 to 0.17. Relation between film thickness and wear life is shown in Fig. 27. Maximum wear life of 7952 cycles was obtained for a film thickness of 0.00031 in. The minimum coefficient of friction for this thickness is 0.1125 in. For practically the same film thickness (0.00028 in.) of Electrofilm lubricant, the maximum wear life of 24,204 cycles obtained at minimum coefficient of friction of 0.048.

Lubricant: LaFrance Franlube Graphite

Machine: LFW-1

Speed: 72 RPM

Max. Load: 240 Lbs. (After 8 Minutes)

Specimen: Ring and Block

Room Temp.: 80 ± 5°F

Thickness in.	Wear Life Cycles	Min. Coeff. of Friction	Wear Rate mg/minute
0.00013	3621	0.073	0.115
0.00022	7259	0.0938	0.144
0.00031	7952	0.1125	0.158
0.000435	5665	0.14	0.216
0.00059	2749	0.171	0.332

$$\text{Maximum Hertz Pressure} = 0.465\sqrt{\frac{WE}{LD}}$$

$$= 0.465\sqrt{\frac{240 \times 30\times10^{6}}{0.25 \times 1.377}}$$

$$= 6750 \text{ psi}$$

Table II. Data for Graphite Lubricant on LFW-1 Machine

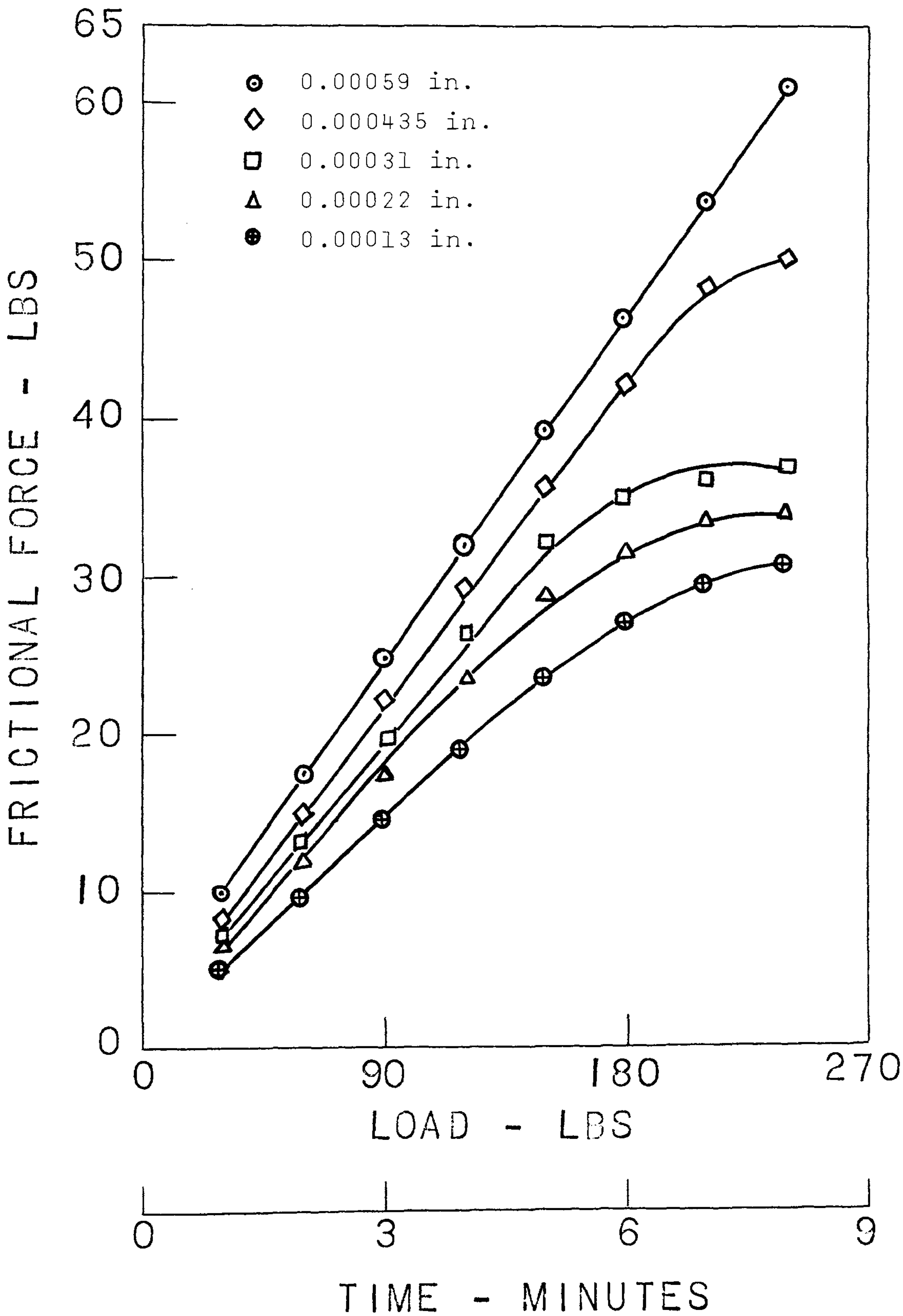

Fig. 25 Effect of Increase in Load on Frictional Force for Graphite Lubricant

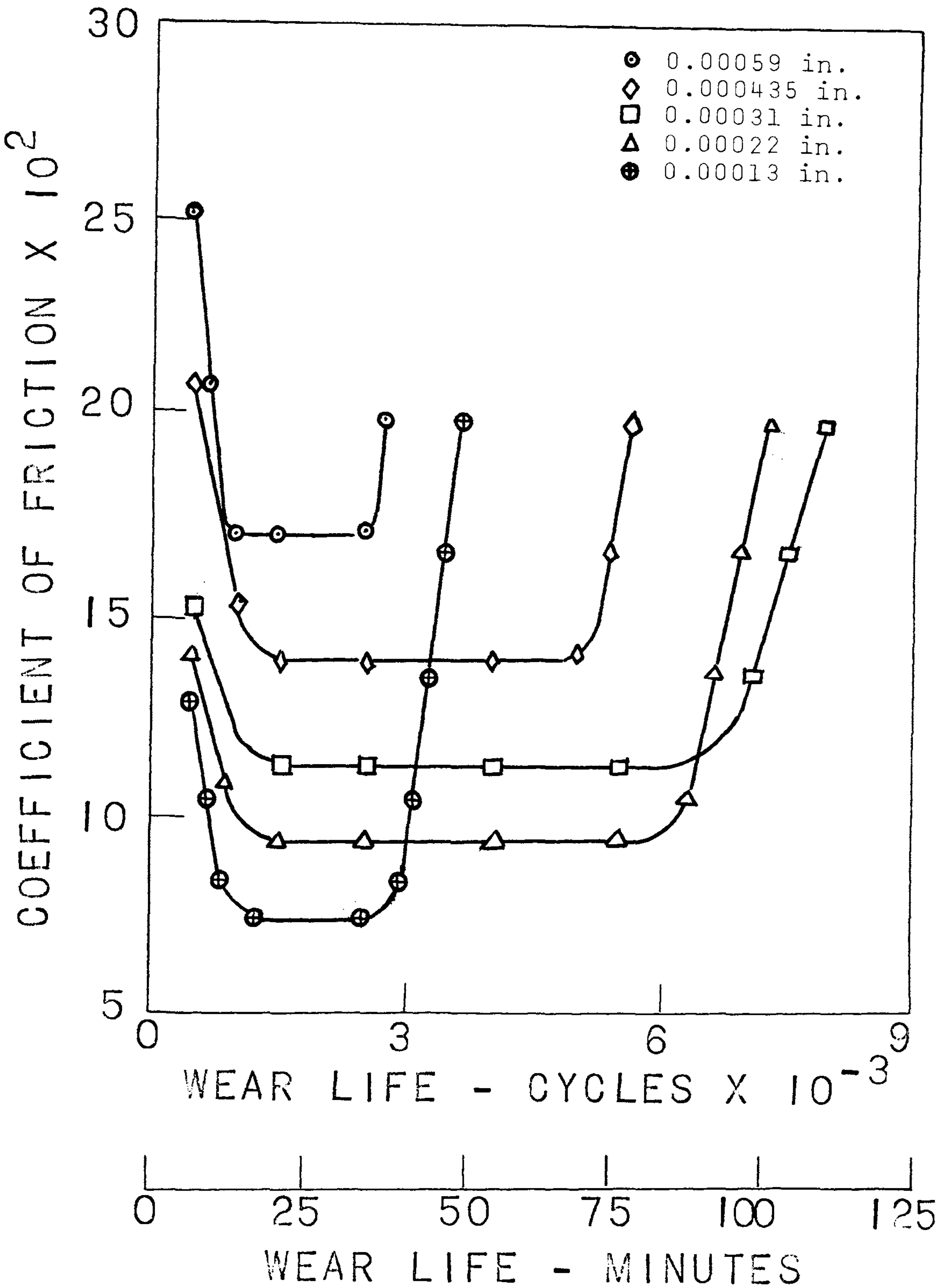

Fig. 26 Wear Life Versus Coefficient of Friction for Graphite Lubricant

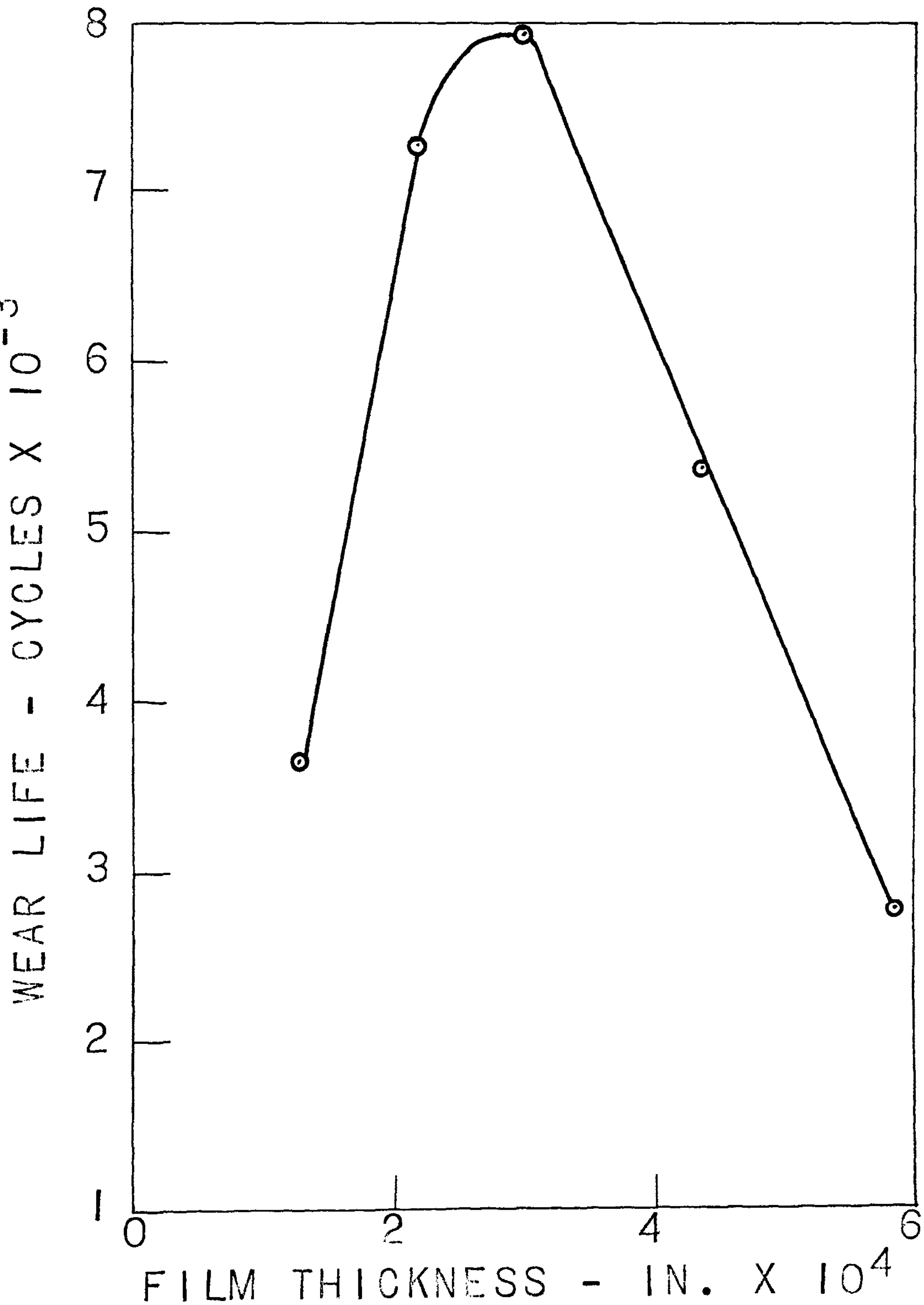

Fig. 27 Effect of Film Thickness on Wear Life for Graphite Lubricant

It demonstrates that the high coefficient of friction for graphite has affected the wear life even at low contact pressure. The effect of film thickness on minimum coefficient of friction is shown in Fig. 28. The friction coefficient increases linearly with increase in film thickness. The graph of wear rate versus film thickness is shown in Fig. 29. The minimum rate of wear is found to be 0.115 mg/ minute at a film thickness of 0.00013 and the maximum rate was 0.171 mg/minute at a thickness of 0.00059 in.

C. Electrofilm Lubricant - Falex Tester

The results of the tests for electrofilm on Falex tester are shown in Table III. Influence of film thickness on wear life is shown in Fig. 30. Maximum wear life of 179 minutes was obtained at film thickness of 0.00036 in. Effect of film thickness on wear rate is shown in Fig. 31. Wear rate was measured as teeth per minute and it is expressed in terms of in. per minute, where 1 tooth per minute is equivalent to 5.56×10^{-5} in. per minute. Wear rate at 0.00098 in. thickness was 5.17×10^{-5} in. per minute, and at 0.00014 in. it was 1.53×10^{-5} in. per minute. These figures show that the wear rate is considerably higher at greater thicknesses.

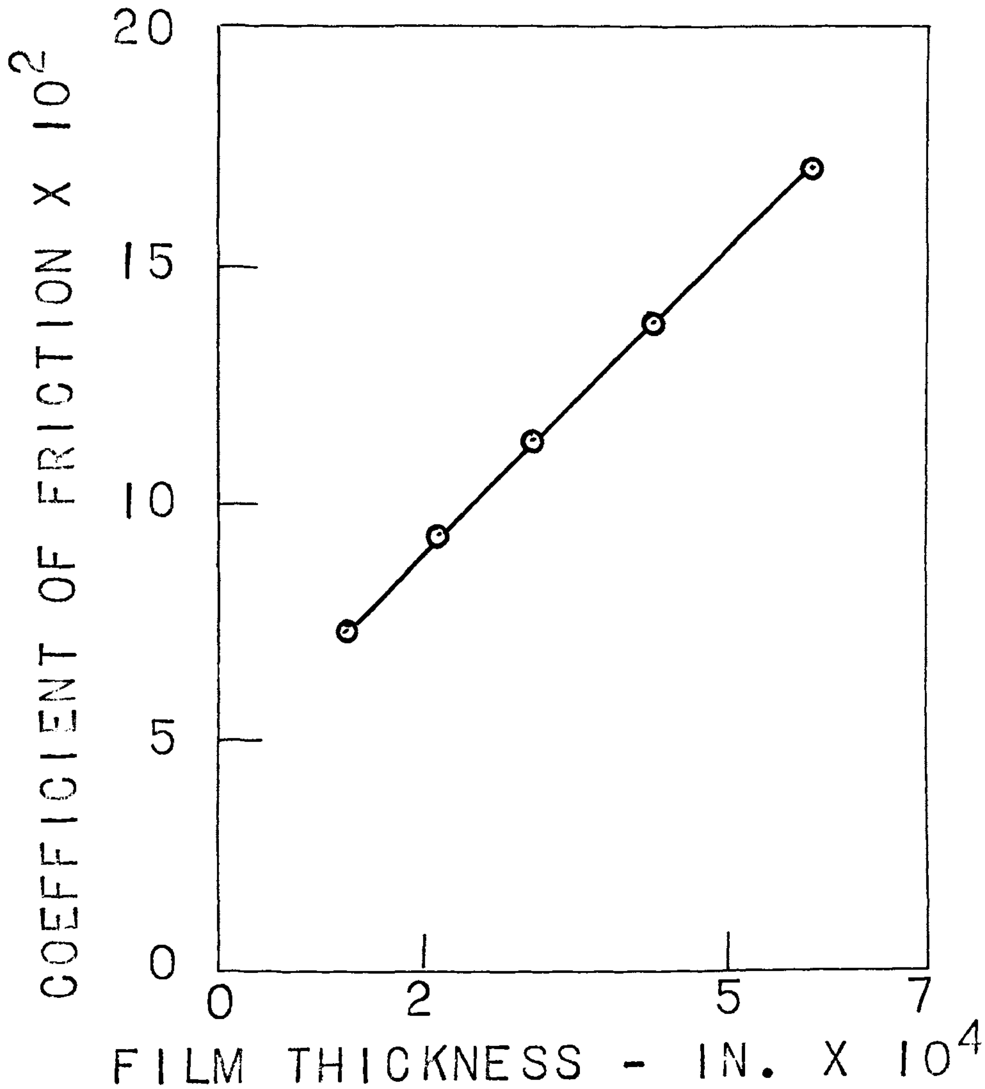

Fig. 28 Plot of Minimum Coefficient of Friction Versus Film Thickness for Graphite Lubricant

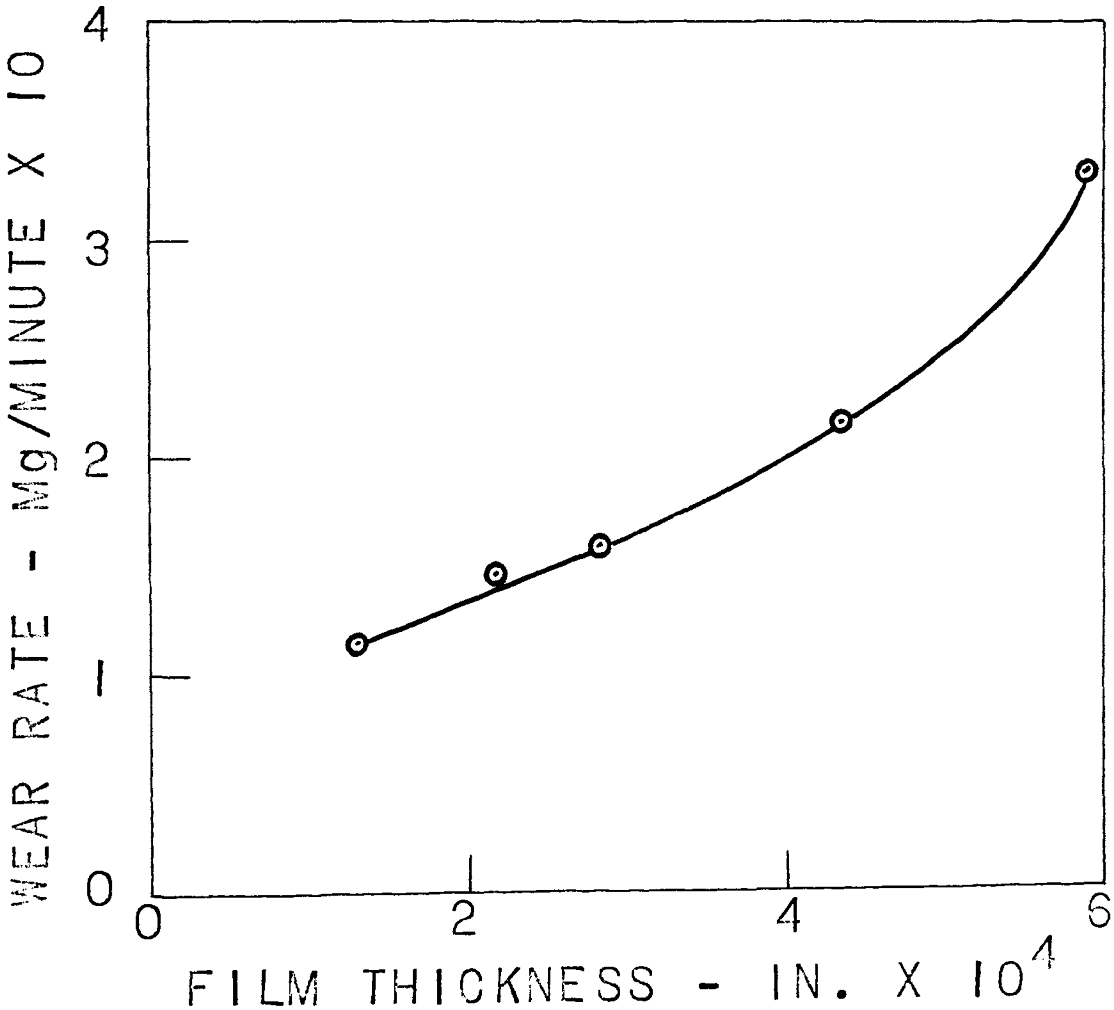

Fig. 29 Effect of Film Thickness on Wear Rate for Graphite Lubricant

Lubricant: Electrofilm Lubri Bond 'A'

Machine: Falex

Speed: 290 RPM

Max. Load: 250 Lbs.

Specimen: 'V' Blocks and Journal

Room Temp.: 80 ± 5°F

Thickness in.	Wear Life Minutes	Wear Rate (in./Min.)x10^{6}
0.00014	103	15.3
0.00021	152	16.7
0.00027	169	17.8
0.00036	179	18.9
0.00051	167	20.6
0.00059	157	23.4
0.00069	137	28.4
0.0008	115	35.0
0.00098	78	51.7

Table III. Data for Electrofilm Lubricant on Falex Machine

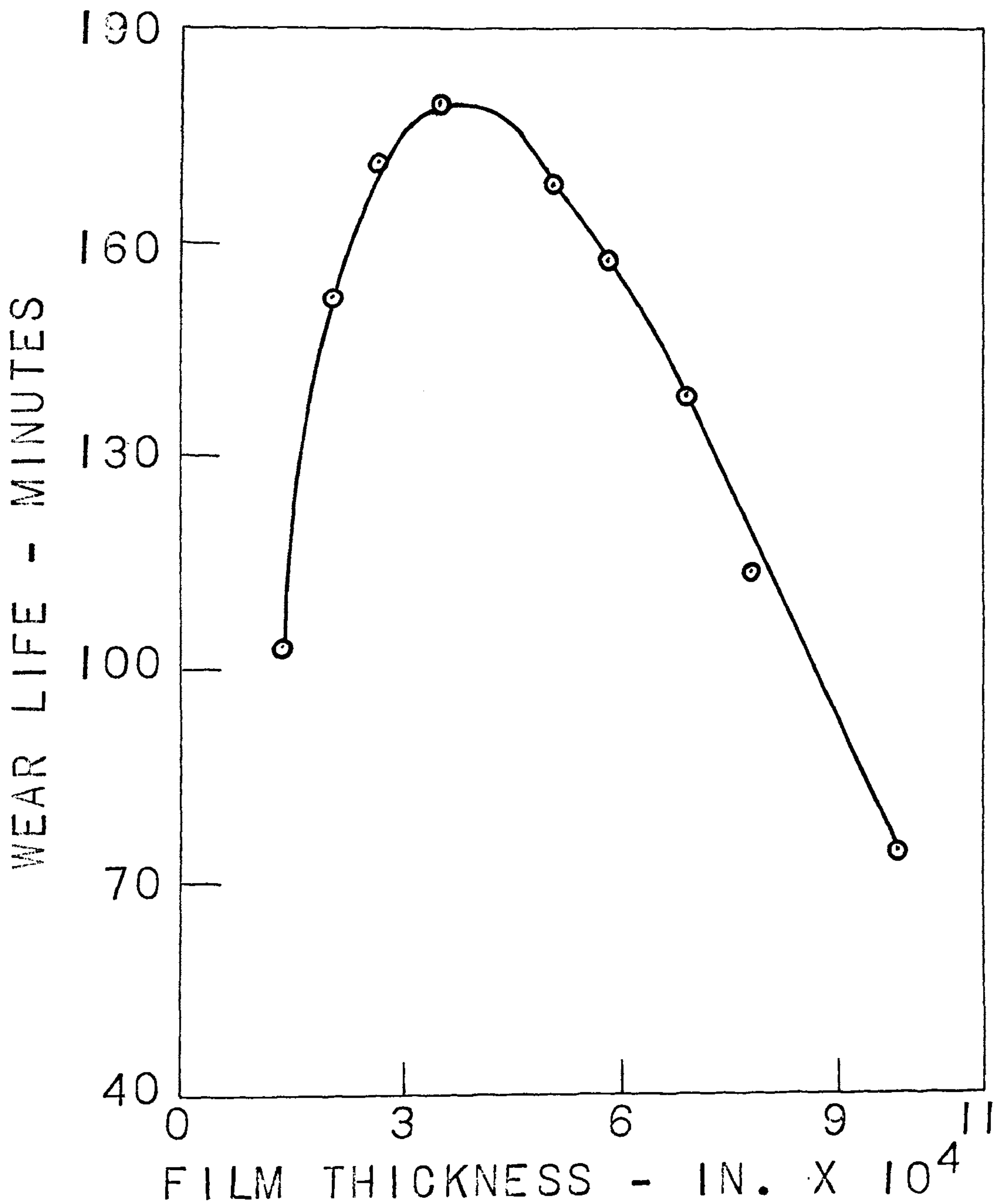

Fig. 30 Effect of Film Thickness on Wear Life of Electrofilm Lubricant

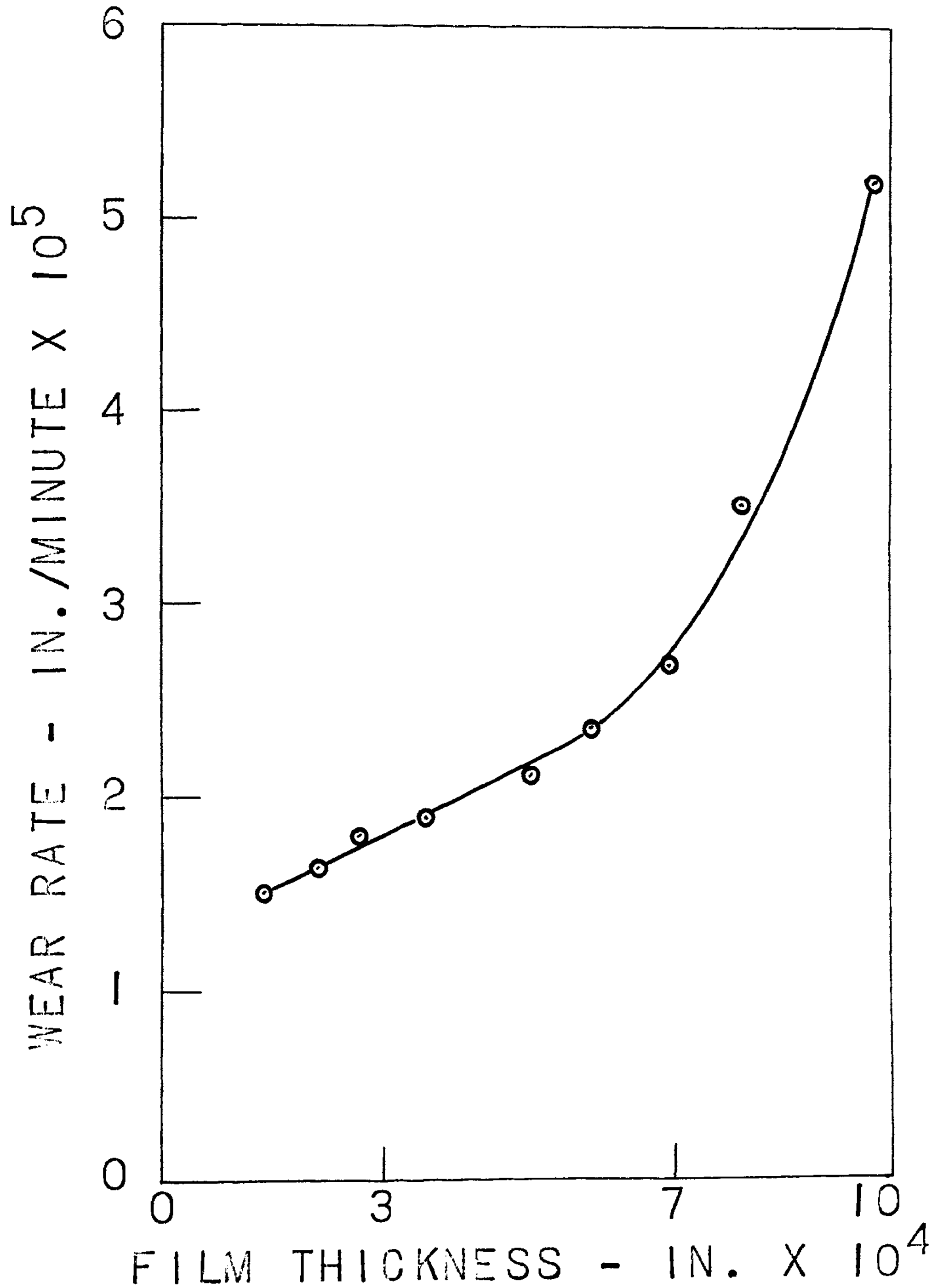

Fig. 31 Effect of Film Thickness on Wear Rate for Electrofilm Lubricant

VI. CONCLUSION

The film thickness is a very influential parameter governing the performance of a solid lubricant. After examining the results it can be concluded that, there exists an optimum film thickness below and above which the wear life decreases. For the molybdenum disulfide lubricant this optimum thickness was found to be 0.00028 in. and for the graphite it was 0.00031 in. It also can be concluded that the minimum coefficient of friction increases with increasing film thickness.

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VIII. VITA

Sunil Vishwanath Mahale was born on July 16, 1945 in Bombay, India. He received his primary and secondary high school education in Bombay. He has received his college education from Parle College and Victoria Jubilee Technical Institute. He received a Bachelor of Engineering degree in Mechanical Engineering with honors in August 1967 from the University of Bombay. After his graduation he worked for one year in Design and Development Department of Crompton Parkinson Ltd. of India.

Since September 1968, he has been working for an M.S. in Mechanical Engineering at the University of Missouri-Rolla.