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Test Method to Evaluate Cylinder Liner-Piston Ring Coatings for Advanced Heat Engines

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TEST METHOD TO EVALUATE CYLINDER LINER-PISTON RING COATINGS FOR ADVANCED HEAT ENGINES

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SUMMARY

A tribological test method has been developed to assess candidate high temperature piston ring and cylinder liner coatings for advanced heavy duty diesel engines. A reciprocating test rig is employed to reproduce the boundary lubricated conditions existing at the piston ring-cylinder liner interface during top ring reversal (TRR). The test specimens are fabricated from commercially available piston rings and cylinder liners to facilitate direct comparisons with current diesel engines. A data acquisition system is used to collect coefficient of friction values. Coating performance is based upon calculated volumetric wear factors of the ring and liner as well as the friction coefficient.

Baseline tests were conducted with chrome coated piston rings sliding on pearlitic gray cast iron cylinder liners while in the presence of a fully formulated engine oil. Friction measurements exhibited excellent repeatability throughout the entire test with average values ranging from 0.079 to 0.082. The mean of the ring and liner wear factors was 7.28×10^{-10} and 2.62×10^{-8} $mm^3/N{\cdot}m$, respectfully and are in good agreement with values reported in the literature. These results indicate that the test method is capable of reproducing TRR conditions at the ring-liner interface of a diesel engine.

1.0 INTRODUCTION

Research on advanced heat engine concepts, such as the low heat rejection engine, has shown the potential for increased thermal efficiency, reduced emissions, lighter weight, simpler design, and longer life compared to current diesel engine designs (ref. 1). These improvements are achieved through the reduction or removal of the cooling system, insulation of the combustion chamber components with ceramics, and converting increased exhaust energy to shaft output power with a recovery turbine.

A major obstacle in the development of a functional advanced heat engine is overcoming the tribological problems resulting from high temperatures at the piston ring-cylinder liner interface, especially at top ring reversal (TRR). TRR is the most critical part of the engine cycle because speeds are low and pressures and temperatures are high, resulting in boundary lubricated conditions. As a consequence, the majority of the wear experienced by the ring and liner occurs at this location (ref. 2).

To avoid scuffing of the ring and liner, typical TRR temperatures in conventional diesel engines are kept near 200 °C by circulating coolant through the cylinder head (ref. 3). However, studies on advanced diesel engine designs have predicted that retaining the heat inside the combustion chamber will result in TRR temperatures in excess of 300 °C with some estimates reaching up to 650 °C (refs. 4 and 5). Such high temperatures impede the use of conventional liquid lubricants and also severely affect the durability of chrome rings and cast iron liners. Tests conducted at only 260 °C have shown that the rings and liners experience wear at least 100 times greater than at conventional TRR temperatures (ref. 6). Both advanced liquid lubricants and improved cylinder liner/piston ring materials that can survive at these extreme conditions are needed. However, evaluating new materials and liquid lubricants using engine tests is costly and offers little control over the test conditions. For these reasons it is important that an accurate bench test simulation be developed in order to identify good coating combinations before effort is expended on engine tests. This bench test approach is not new and a number of rig designs are described in the literature (refs. 6 to 9).

This paper presents a tribological test method designed to simulate TRR conditions to aid in evaluating candidate lubricants and piston ring/cylinder liner materials for advanced diesel engines. The selected test method uses a commercially available pin-on-plate reciprocating wear test rig with specially modified specimens machined from conventional top compression rings and cylinder liners. Loads, speeds, and temperatures are selected to simulate approximate engine wear conditions present at the ring-liner interface at TRR. It is intended that this test setup and procedures be used as a screening tool to select material combinations for engine tests.

To validate the test method, repeated baseline tests were run with conventional chrome coated rings and pearlitic gray cast iron cylinder liners. Data were evaluated to judge the usefulness of the test and to confirm the results with engine experience.

Test Apparatus

Test rig.—A commercially available pin-on-plate reciprocating wear rig, shown in figure 1, is used to conduct the tests. Capabilities of the rig include speeds up to 50 Hz, maximum loading of 250 N, and an adjustable drive mechanism to vary the stroke length from 1 to 15 mm. A piezo-electric transducer attached to the nonmoving support plate measures the frictional force. The rig is also equipped with a contact resistance circuit to study the formation of lubricant films between the specimens. For tests at elevated temperatures, resistive coils embedded in the heater block can heat the specimens via conduction up to 600 °C.

Data Acquisition System.—To facilitate long term tests, a computer data acquisition system collects and stores frictional force data once every minute for the entire duration of the test. A displacement sensor, directed towards the motor drive shaft, is used to trigger the acquisition system to take a quick burst of 10 data points over a 200 μs interval at the midpoint of the stroke. The system software then calculates the friction coefficient by dividing the average of the 10 friction force values by the applied normal load.

2.0 Preparation of Baseline Test Specimens

Shown in figure 2 is an example of the ring and liner specimens used in the baseline tests. The liner specimen was cut from a commercially available honed gray cast iron cylinder liner with a bore of 130 mm and a wall thickness was 8.6 mm. The ring specimen came from a SAE 9254 SS tapered face keystone piston ring that had its outer diameter surface covered with a 250 μm thick electroplated chrome coating. The ring's crown radius was measured on a few randomly chosen specimens and found to be between 15 to 19 mm.

Liner Specimen.—The liner specimens were prepared by first cutting the liner radially into 60 mm wide sections. Next, the sections were cut at 22.5° intervals resulting in sixteen arc segments. To achieve rigid support and perpendicular contact between the ring and liner, a smooth flat surface was prepared on the back of the specimen by removing 3.9 mm of the outer surface with a band saw measured at the centerline and then grinding to the final thickness of 4.5 mm. One hole at each end was then drilled to accommodate mounting screws as shown in figure 2. During machining, care was taken not to scratch or damage the honed surface.

Ring Specimen.—Fabrication of the ring specimen began by cutting a piston ring into eleven 35 mm long pieces measured along the inner radius. Engines that use the rings and liners from which the test samples were made typically produce maximum pressures of 13.4 to 15.1 MPa at TRR. To simulate this pressure loading on the ring, combined with the ring's expanding force, a contact length of 6 mm and a normal load of 192 N were selected. This contact length, as shown in figure 2, was produced by using Electrical Discharge Machining (EDM) to remove all but a 6 mm section of the ring's outer diameter.

To secure the ring specimen properly to the carrier head, a special fixture was designed and fabricated. This fixture was very similar to the one described in reference 5.

Wear Test Setup and Procedure: Prior to testing, both specimens were ultrasonically cleaned first in heptane and then in ethyl alcohol; each cleaning lasting for 10 min.

The test parameters, listed in table I, were chosen to simulate the severe temperatures and pressures present at TRR during engine operation under steady-state full load conditions.

Setup for the baseline tests was initiated by securing the liner specimen in the steel boat with mounting screws as shown in figure 1. Next, the ring specimen was placed in its adapter, lowered until it made contact with the liner specimen, and then the steel boat was adjusted until the midpoint of the ring coincided with the centerline of the liner. The stirrup was then positioned on the carrier head and the temperature controller activated. The temperature of the specimens was allowed to stabilize at 200 °C before positioning the lubricant drip feed system. When lubricant reached the test area, the rig was started and the load applied until reaching 192 N. Each test was run for 24 hr.

Wear Analysis.—The wear on the ring and liner was quantified by determining the wear factor for both specimens. The wear factor is defined as:

$$K = \frac{V}{N \cdot x} \tag{1}$$

where

V volume of removed material, mm^3

N normal load at the sliding contact, N

x sliding distance, m

The wear produced on the ring was in the form of a rectangular scar that extended for the entire contact length. To properly calculate the wear factor, the volume of material removed during a test has to be determined. This was accomplished by first using an optical microscope to measure the width of the wear scar at six equal intervals along the length of the scar. This width data, along with the ring's crown radius value, was sufficient information to calculate the cross-sectional area of the ring portion that was removed. The volume of removed material is then just the average area multiplied by the 6 mm contact face.

For the liner, the volume was determined by taking surface profiles along the length of the wear scar with a stylus profilometer at four locations, equally spaced apart, along the scar width. A typical surface profile, shown in figure 3, indicates that the depth of the wear scar is on the same order of magnitude as the liner's original honed surface finish. As can be seen, the wear scar is located between points 2.2 and 17.9 mm of the profile length resulting in a scar of a little under 16 mm. The cross-sectional area of the profile was calculated with the help of a graphics program that was specifically written to analyze output files from the profilometer. The average of the four areas was then multiplied by the wear scar width to obtain the volumetric wear.

RESULTS AND DISCUSSION

Friction.—Coefficient of friction vs. time plots for the five tests, given in figure 4, illustrates the repeatability of the test method. The average friction values for the five experiments are summarized in table II and ranged from 0.077 to 0.082.

Morphology.—Upon completion of each test, both the ring and liner exhibited worn areas that had a smooth, glossy finish which was a good indication that a fine polishing wear mode was present. To further investigate this assumption, the midpoint of the liner and ring wear areas were analyzed with a scanning electron microscope (SEM). The photomicrographs taken by the SEM can be seen in figure 5. Both specimen wear areas demonstrated faint wear lines in the direction of sliding which was consistent with a mild abrasive wear mechanism. Also included in the liner image were honing marks and probable surface fatigue cracks.

To better evaluate the test method's effectiveness at simulating the wear that occurs at TRR, a top compression ring and a cylinder liner were obtained from a real engine that was undergoing a major overhaul. The ring and liner operated in the engine for almost 400 000 miles and were identical to the hardware that was used to make the test specimens. SEM photographs of the ring and the liner at TRR are shown in figure 6.

If ring specimens in figures 5 and 6 are compared, they both exhibit surface characteristics consistent with a polishing wear mechanism. However, the engine ring also contains cracks and micropitting on the chrome coating's surface. The cracks most likely were caused by exposure to the repetitive thermal shock loads produced by combustion. Pinpointing the mechanism responsible for the micropits is a little more difficult but may be a result of spalling due to thermal cycle fatigue and/or microwelding. The absence of thermal cracks and micropitting on the baseline ring can probably be attributed to a number of factors, including the short duration of the test, constant temperature being used, and the drip feed system keeping the test area well lubricated. Aside from the micropits and cracks, the ring surface exhibits a smooth surface that is consistent with mild abrasive wear.

As shown in figures 5 and 6, both the laboratory and engine liners contain surface fatigue cracks and signs of light abrasive wear in the direction of ring motion indicting that both experienced similar wear mechanisms.

Ring and Liner Wear

The wear factor results for the rings and liners are given in table II and also shown graphically in figure 7. The ring wear factors were repeatable and ranged from 5.23×10^{-10} to 1.23×10^{-9} $mm^3/N \cdot m$. The liner wear factors were also repeatable and are on the order of 10^{-8} for each of the five tests.

For comparison purposes the wear factor was also calculated for the engine ring and liner. See the Appendix for a detailed description on how these wear factors were calculated. The bar chart indicates that there is excellent corroboration between the engine ring wear factor and the baseline ring wear factors. The baseline wear factors for the liner specimens, however, are about an order of magnitude greater than the wear factor for the engine liner. The discrepancy between wear factors may be explained by realizing that, under boundary lubrication, a new liner experiences an initial high rate of wear known as run-in followed by a lower steady-state wear rate (ref. 10). Rings, on the other hand, start out with a highly polished surface finish and are less susceptible to break-in effects. Because of the short duration of the tests (24 hr), the liner specimens were still undergoing run-in and thus the reason for the high wear factor values. Alternately, after 400 000 miles, the engine liner was operating in the steady-state wear rate period resulting in a lower wear factor.

The bar chart also illustrates the tendency for the liner wear factor to be greater than the wear factor for the ring. This is not an unexpected result. The purpose of a piston ring is to provide blow-by and oil control. If the surface profile of the ring were to wear excessively, the ring would lose its ability to perform its functions effectively. Therefore a chrome coating, which is harder than cast iron, is placed on the ring to insure that most of the wear occurs mainly on the liner. This systems approach is corroborated by the wear factor results for the engine ring and liner.

CONCLUDING REMARKS

The wear test and procedures presented here demonstrate excellent wear data repeatability. It is shown that the ring and liner specimen wear surface morphologies are quite similar to real engine experience near TRR. Furthermore, the wear factors for the laboratory and engine rings and liners were similar. Wear results for the baseline liner specimens are an order of magnitude greater than the wear on the engine liner. However, since the ring wear is of greater concern, the corroboration exhibited between the laboratory and engine rings suggests that the bench tester and established procedures can be conveniently used to screen materials and lubricant candidates for heavy duty applications. Finally, for lubricated tests of advanced materials, wear factors, friction, and surface morphology play an important role in determining optimum combinations for enhanced engine performance and life.

APPENDIX

In order to corroborate the test wear factor results with conventional diesel engines seen on the road today, a ring and liner were obtained from an engine that operated for approximately 400 000 miles. Wear measurements were made on the ring and liner, as discussed below, and the values used to calculate wear factors.

A surface profile taken of the liner near TRR is shown in figure 8 and clearly demonstrates the amount of wear that resulted from 400 000 miles of service. As a frame of reference, TRR is located at approximately the 23 mm point of the profile length. Notice that it is possible to distinguish the three classic lubrication regimes that the ring experiences as it travels near TRR. Following the trace from left to right, the ring is in hydrodynamic lubrication until reaching a distance of ~18 mm. from TRR whereupon a decrease in the fluid film thickness caused by reduced ring velocity initiates elastohydrodynamic lubrication to become dominant and wear increases. This regime lasts for ~10 mm until the ring is ~8 mm from TRR and boundary lubrication takes over. Not unexpected, most of the wear occurs in this region.

To quantify the amount of wear, four traces were taken of the liner at TRR, each at 90° intervals, and the cross-sectional area measured. The profile in figure 8 is typical of the traces that were generated. The area measurements were used to obtain a wear volume by averaging the four area values together and then multiplying the area by the circumference of the liner bore. The wear volume for the liner was found to be 130.7 mm^3.

The wear volume for the used ring was determined by measuring the thickness of the remaining chrome coating and comparing it to the chrome thickness of a new ring. Using an optical microscope, the chrome thickness for the new ring was found to be 0.22 mm compared to 0.18 mm for the used ring. Therefore, the net coating thickness loss was 0.04 mm. Knowing the ring's crown radius the cross-sectional area of the removed coating was found to be 0.089 mm^2. Finally, the wear volume was obtained by multiplying the area by the ring's circumference, yielding a value of 36.45 mm^3.

The next step was to estimate the sliding distance to be used in calculating the wear factors. The value for the sliding distance was found by first recognizing that, for a 4-stoke engine, combustion occurs during every other cycle. Since temperatures and pressures are at their highest during combustion, a majority of the ring and liner wear occurs at this time. Therefore, the sliding distance should only be associated with this moment in the cycle. From a direct measurement of the liner profile in figure 8, the sliding distance is ~18 mm. Hence, for a truck that averaged 50 mph at a speed of 1200 rpm and traveled for 400 000 miles, the sliding distance is 9.455×10^6 m. The manufacturer has reported that at rated speed and at full load, the cylinder pressure produces a total radial force of 13 000 N. at the ring and liner interface. However, this does not represent typical normal running conditions for the engine since most of the life of the engine is spent operating on highways. For this case a more realistic ring load is closer to 2/3 of full load conditions. Assuming this load remains constant throughout the 18 mm sliding distance, the ring and liner wear factors were calculated with equation (1) to be 4.44×10^{-10} and 1.60×10^{-9} $mm^3/N\cdot m$, respectfully.

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TABLE I.—WEAR TESTING PARAMETERS

Speed	20 Hz
Load	192 N (32 N/mm)
Temperature	200 °C
Time	24 hr
Stroke length	15 mm
Lubrication	15W-40, oil drip 1 drop every 20 sec

TABLE II. —WEAR TEST RESULTS FOR THE CHROME RING AND CAST IRON LINER

Test number	Average ring wear factor, $10^{-10}\ \frac{mm^3}{N \cdot m}$	Average liner wear factor, $10^{-8}\ \frac{mm^3}{N \cdot m}$	Average friction coefficient
Engine	4.44	0.16	---
1	7.08	4.54	0.081
2	12.3	2.39	0.082
3	5.23	2.12	0.082
4	5.24	1.66	0.079
5	6.53	2.39	0.079

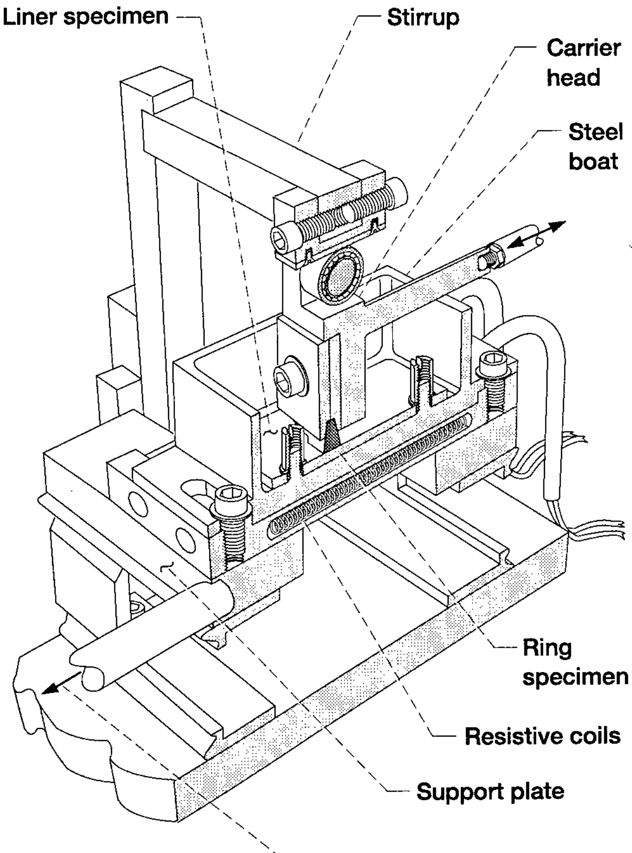

Figure 1.— Schematic of the reciprocating wear rig.

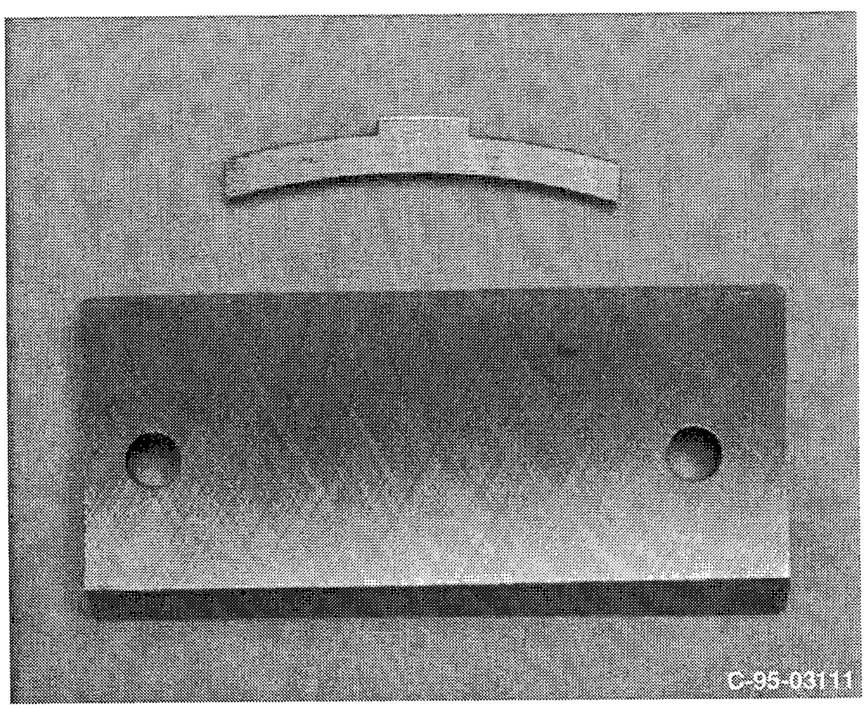

Figure 2.—Chrome coated ring and cast iron liner test specimens.

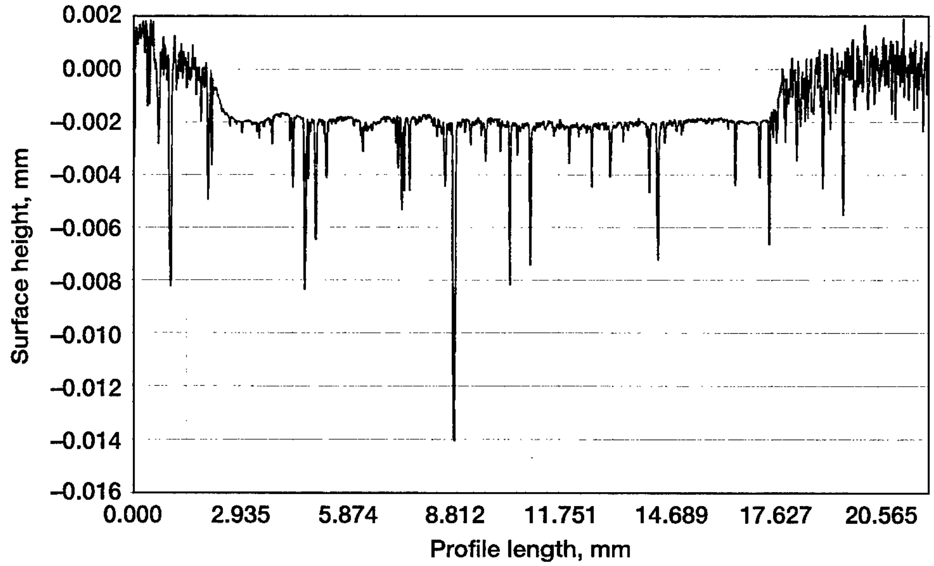

Figure 3.—Worn surface profile of liner specimen.

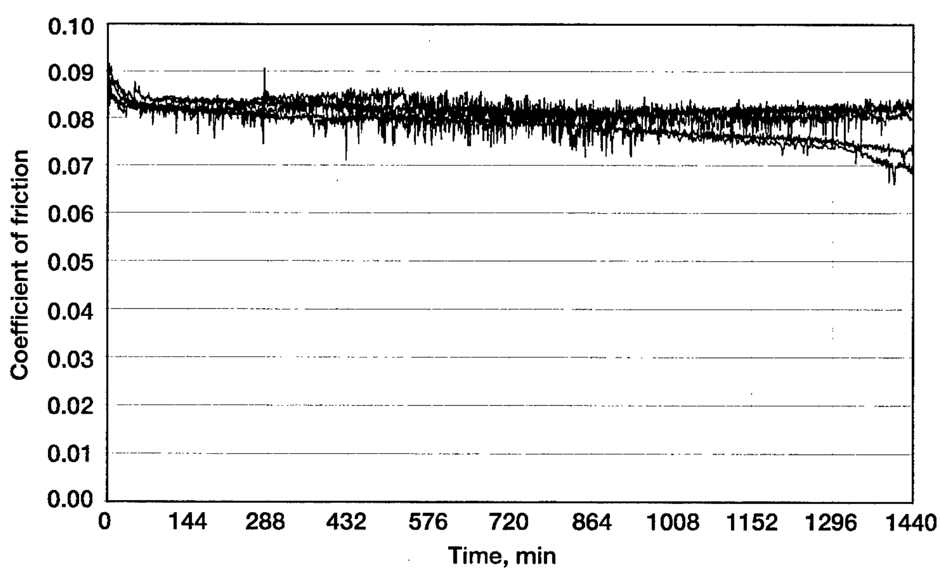

Figure 4.—Baseline friction vs. sliding time traces for chrome coated rings sliding against cast iron liners.

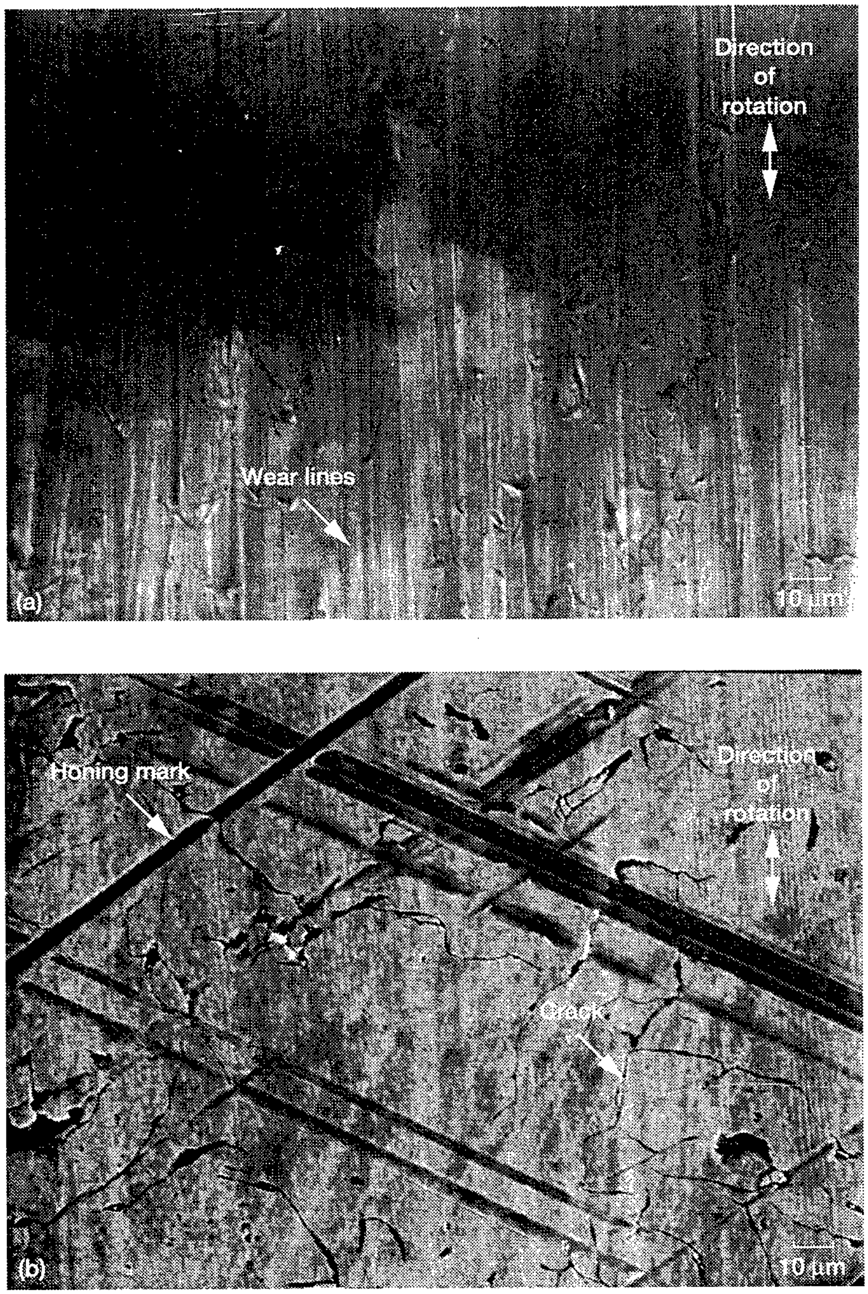

Figure 5.—SEM photos of lab test specimens. (a) Ring. (b) Liner.

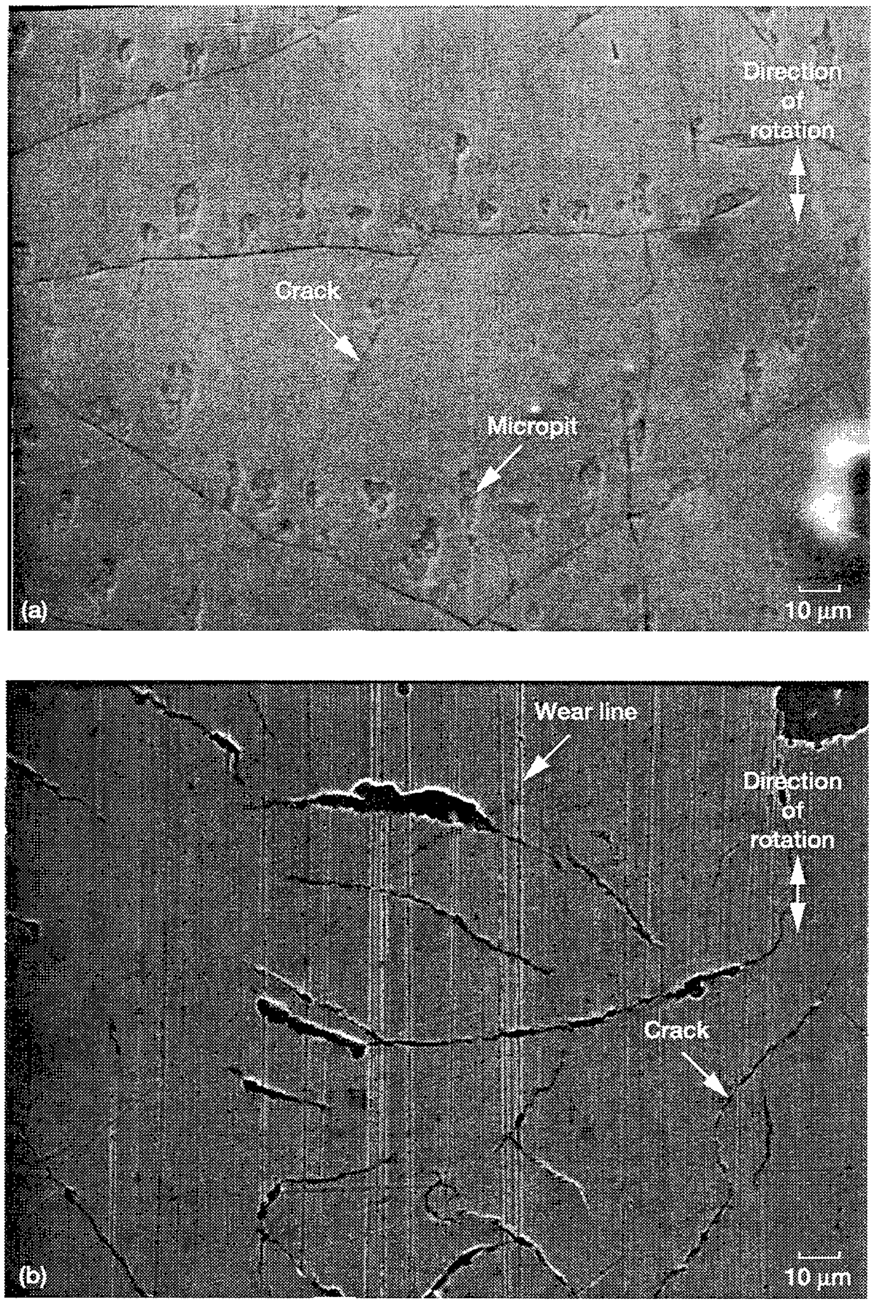

Figure 6.—SEM photos of engine tested hardware. (a) Ring. (b) Liner at TRR.

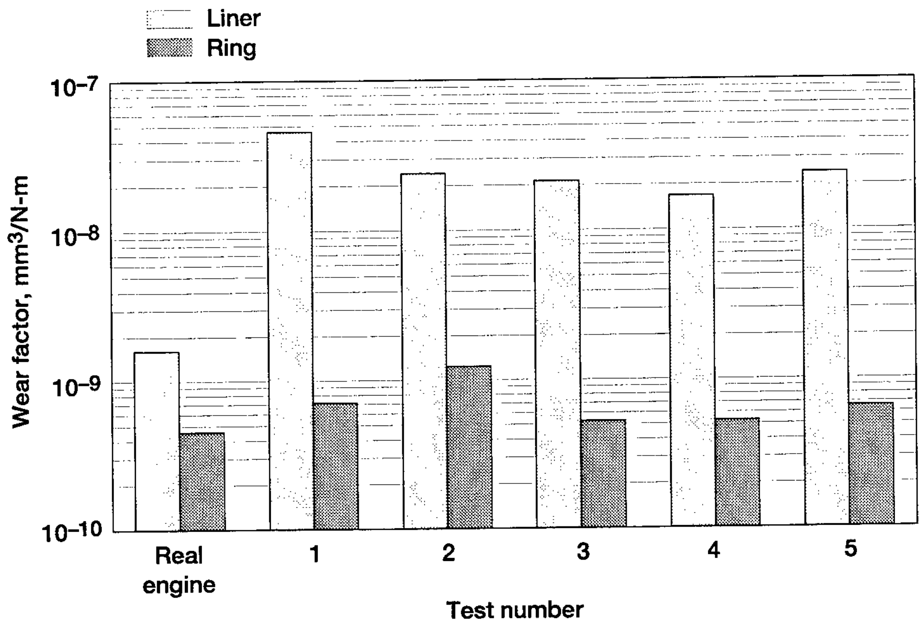

Figure 7.—Baseline wear factors for chrome coated rings sliding against cast iron liners.

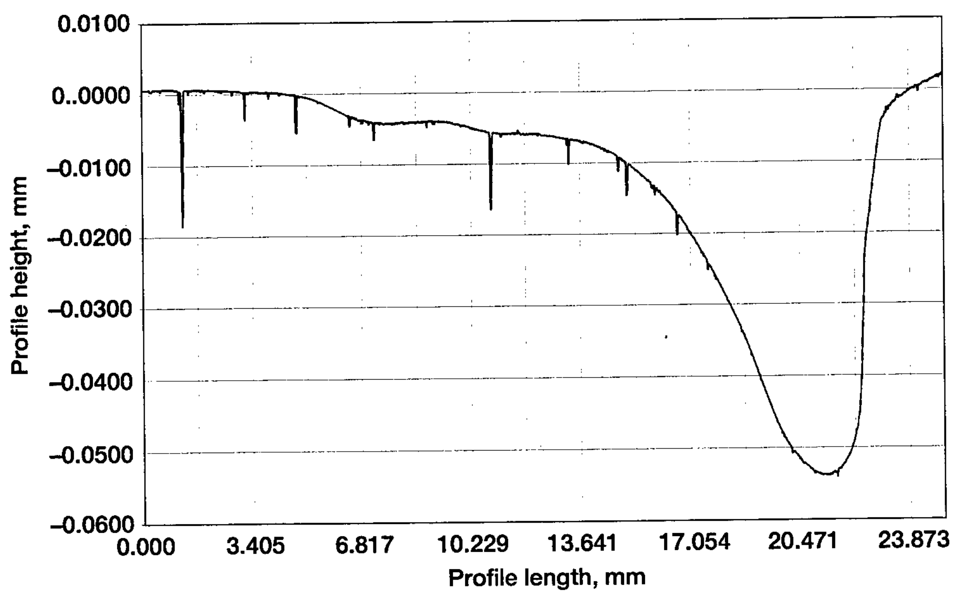

Figure 8.—Surface profile of a cylinder liner near TRR. The liner was obtained from an engine which operated for 400,000 miles.

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13. ABSTRACT *(Maximum 200 words)*

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