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A rig test to measure friction and wear of heavy duty diesel engine piston rings and cylinder liners using realistic lubricants☆

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Abstract

Laboratory tests to evaluate piston ring and cylinder liner materials for their friction and wear behavior in realistic engine oils are described to support the development of new standard test methods. A ring segment was tested against a flat specimen of gray cast iron typical of cylinder liners. A wide range of lubricants including Jet A aviation fuel, mineral oil, and a new and engine-aged, fully formulated 15W40 heavy duty oil were used to evaluate the sensitivity of the tests to lubricant condition. Test temperatures ranged from 25 to 100 °C. A stepped load procedure was used to evaluate friction behavior using a run-in ring segment. At 100 °C, all lubricants showed boundary lubrication behavior, however, differences among the lubricants could be detected. Wear tests were carried out at 240 N for 6 h at 100 °C with new ring segments. The extent of wear was measured by weight loss, wear volume and wear depth using a geometric model that takes into account compound curvatures before and after testing. Wear volume by weight loss compared well with profilometry. Laboratory test results are compared to engine wear rates.

Keywords: Friction; Wear; Piston ring; Cylinder liner; Diesel engine

1. Introduction

Improved materials or processes for the in-cylinder components of heavy-duty diesel engines have long been sought for better friction and wear performance. The use of laboratory testing to simulate the engine environment has been carried out for many years to save the time, expense and complexity of solely relying on full-scale engine tests during the development process. It is extremely difficult, even with the most well-designed laboratory test, to actually predict performance in the engine since the operating environment is so complex chemically and mechanically [1]. However, a well-designed laboratory test should be able to quickly rank various candidate materials or processes as to their relative performance in an operating engine. Materials which have a long history of use can be used for validation purposes.

In many of the reported laboratory tests measuring piston ring and cylinder liner friction and wear, the lubricant employed is a fully formulated oil in the new condition [2]. Obviously, the oil would not stay in that condition for very long in actual service, since it will degrade with use, and will pick up both external and internal contaminants. It is expected that some of these contaminants and degradation products will have an effect on friction and others on wear and some on both. Despite the fact that used oil represents the typical condition for the ring and liner, little research has been done in this lubricating environment due to the complexity and range of the used oil chemistry. Because of that, there has been a lack of consensus as to the specific or standard composition of a used oil for the purposes of laboratory testing.

The objective of the current project is to define an improved laboratory test to evaluate the friction and wear behavior of ring and liner materials by using more realistic lubricants. The more realistic lubricant can either be oil

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aged in an engine, or synthesized in such a way as to reproduce the behavior of used oil in specific ways. It is a project objective that such a test could become either an ASTM standard or a recommended practice. The purpose of this paper is to summarize the separate test methods for determining friction and wear behavior and demonstrate the use of those methods with typical diesel engine materials in a wide variety of lubricants to demonstrate test sensitivity.

2. Experimental methods

Because of the different measurement requirements for the determination of friction and wear characteristics, separate procedures are required for each. First is a list of materials and procedures common to both tests, followed by the steps unique to the friction and wear test procedures.

2.1. Materials

The piston ring segments used in this study were from an actual heavy-duty top ring (Cat no. 177-7496 for a nominal 13.5 cm bore) sectioned into 25 mm pieces. This particular ring has a plasma-sprayed chromium coating 200 μm thick on a ductile iron substrate. The microstructure of the ring coating is shown in Fig. 1. The ring segment is inserted in a holder, which consists of a portion of the ring groove of the matching piston. That assembly, in turn, is attached to the reciprocating arm of the tester. A flat has been machined in the crown of the holder to allow the ring segment to stand proud by about 2.5 mm. To avoid issues of alignment in matching curvatures of the ring and liner, a flat gray cast iron specimen is used instead. The graphite morphology for the cast iron is shown in Fig. 2. The liner samples were produced in accordance with the ASTM D 1384 glassware corrosion test to determine the corrosivity of engine coolants for typical heavy-duty engine materials. The specimens, originally 25.4×50.8 mm, were cut in half, leaving two 25.4×25.4 mm specimens. The surface was finished with 600 grit SiC paper and rinsed in acetone. This surface finish does not reproduce that of a new plateau-honed cylinder liner wall, but rather the worn area near top ring reversal. This area experiences higher ring and liner loading due to both compression and the combustion event and is under boundary lubrication.

A wide range of lubricants was employed for the purpose of determining the sensitivity of the test methods to lubricant condition. They were (a) Jet A aviation fuel, (b) reagent grade mineral oil, (c) new fully formulated heavy-duty SAE 15W40 oil (Cummins Premium Blue API CG-4), and (d) the same oil as (c), aged in an engine with a composition typical of what would be seen after a routine oil drain. Jet A represents a very poor lubricant, which would be expected to produce high friction and wear. Mineral oil is a more highly refined version of a typical base stock without the additive package. The SAE 15W40 oil is typical of commercial heavy-duty oil in the new and engine-aged condition. However, for the final standard, it is anticipated that an engine-aged oil would be used and that the aging process be done in a standard engine test such as ASTM D 6838-02, which is used for American Petroleum Institute (API) certification. These test conditions are closely controlled and relatively reproducible in terms of oil aging.

2.2. Test rig and common running conditions

A Cameron Plint High Frequency Test Rig was used for all testing. The ring segment was mounted in holder as described above and the cast iron flat was mounted in shallow tray containing the lubricant and a thermocouple for temperature control. The experimental arrangement is pictured in Fig. 3. All tests for both friction and wear measurements were conducted with 10 mm stroke at 10 Hz. Samples were mounted to ensure proper alignment by checking perpendicularity of load and ensuring a level reciprocating arm. Shims were employed as necessary to

(a) (b)

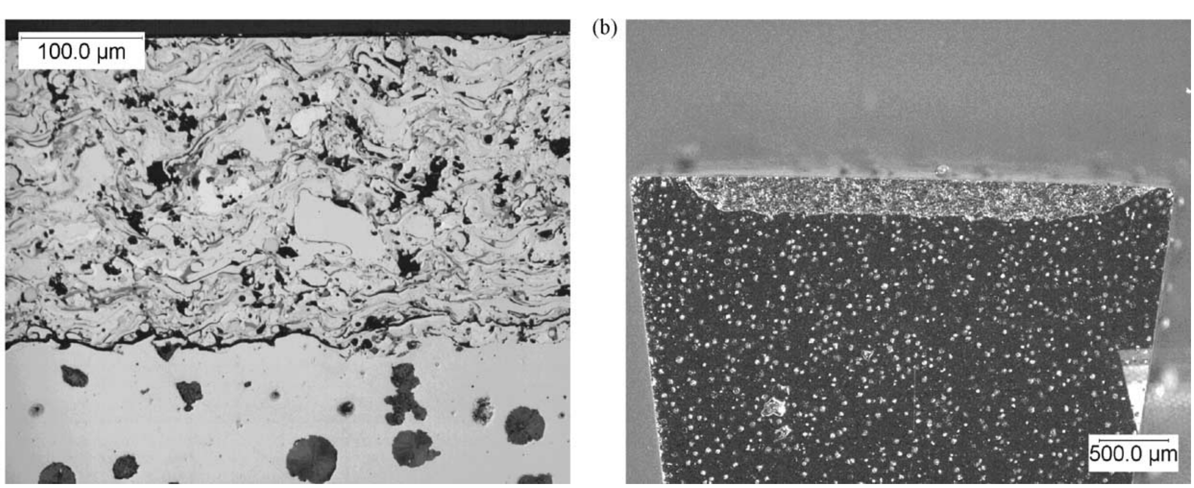


Fig. 1. Cr plasma sprayed ring face. The general microstructure of the coating is shown in (a), while the thickness of the coating is shown in (b).

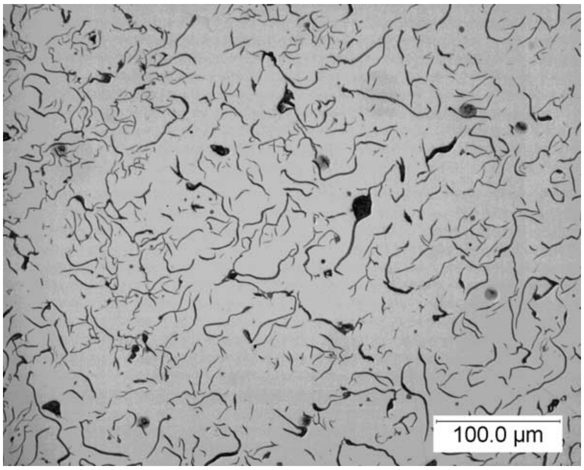


Fig. 2. The graphite morphology of the gray cast iron used for the cast iron flats.

achieve proper alignment as determined by a bubble level. Approximately 10 ml of oil was used in the tray, immersing the cast iron flat and the oil temperatures were controlled to ±1 °C.

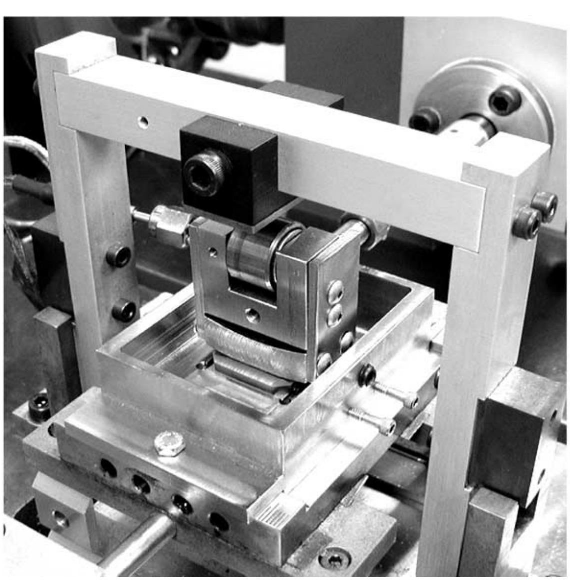
2.3. Friction test procedure

Friction tests were conducted with a run-in ring segment and a freshly prepared cast iron flat. Approximately 1 h at 240 N was required to condition the ring segment. An alternate method for run-in involved stepping the load from 20 to 240 N and back down for about 1 min at each load until any hysteresis in the friction force disappears. Achieving repeatability of the friction force at each applied load defined run-in for the purpose of this study. The use of a new ring segment provided unstable friction results, as seen in Fig. 4, comparing new and conditioned specimens which were run in the engine-aged oil at room temperature. In the case of the new ring segments, run-in would be occurring to different extents with the progressive loading and this does not represent typical engine conditions. An estimation of Hertzian stresses in the new and run-in condition is summarized in Table 1. For purposes of the calculations, a run-in wear scar diameter of 3 mm was assumed, which was typical after conditioning. The bench test stresses were also compared to estimations of ring and liner stresses in the engine. As can be seen, the use of the flat cast iron counterface, instead of a conformal surface, provides a stress multiplication which ensures boundary lubrication and accelerated wear.

If testing at elevated temperatures, the oil was heated to the desired temperature and allowed to stabilize before starting the test. During heating, the oil was stirred with the ring and ring holder assembly, unloaded, while oscillating at approximately 1 Hz to ensure temperature uniformity. The load was applied, starting at 20 N, in 20 N increments to a maximum of 240 N and the average friction force was measured for each load. The time at load depended on obtaining a stable level of friction force. Occasionally, intermediate loads were rechecked for repeatability. Once running-in was accomplished, repeatability was very good.

The data was plotted in a Stribeck curve format to determine the lubricating regime for each lubricant at various temperatures. The traditional Stribeck curve plots friction coefficient against a reduced parameter (viscosity × speed/pressure). A more complete description of the Stribeck curve and its ability to distinguish the various lubrication regimes can be found in [3]. For these tests, the average reciprocating speed was constant and the viscosity effects were not considered, so the Stribeck plots presented

(a)



(b)

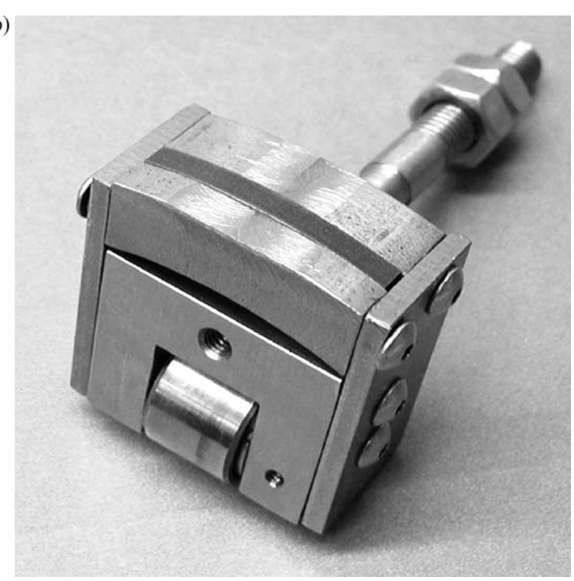


Fig. 3. Overall view of the fixture holding the ring and cast iron flat segments is shown in (a), while (b) shows a closer view of the ring segment holder.

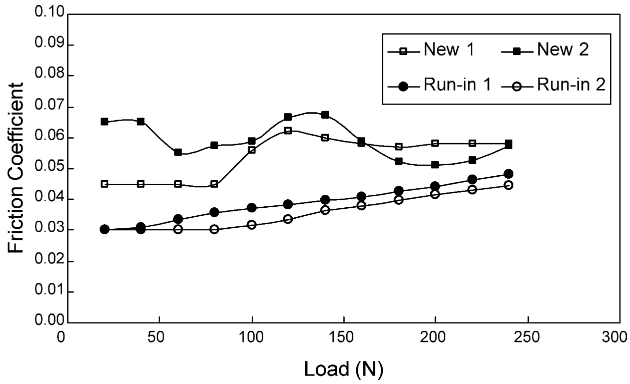


Fig. 4. A comparison of the friction coefficient variation with load for a new and used ring segment tested in engine-aged 15W40 oil at room temperature. Each curve represents one set of measurements at different loads.

here only involved the effect of load. If Hertzian contact is considered, the pressure is proportional to the load to the one-third power [4].

Tests were conducted at room temperature (about 20 °C), 40 and 100 °C. The latter two temperatures also correspond to viscosity index measurements and 100 °C is close to the average engine operating temperature.

The friction force for each lubricant was also measured as a function of temperature during cool down from 100 °C at low (20 N), medium (100 N) and high (200 N) loads.

2.4. Wear test procedure

A new ring segment and freshly abraded cast iron flat were used for each wear test. Each sample was ultrasonically cleaned and weighed to an accuracy of 0.1 mg prior to testing. The ring curvature and transverse profile were measured using a profilometer to establish the initial geometry. The samples were installed in the test rig, the alignment was checked as described above and the oil was added. The oil was heated to 100 °C before the application of the load. A load of 240 N was applied in a slow continuous manner (about 1 min ramp up) and the average friction force was monitored during the course of the test. The test duration was typically set for 6 h, although longer duration tests may be needed to produce measurable wear if using high wear resistance materials. After testing, the ring segment and cast iron flat were removed, ultrasonically cleaned, and reweighed. The ring segment wear scar diameter was measured optically at 100× and the profile remeasured, both transverse and parallel to the reciprocating direction using the profilometer. The cast iron flat wear scar was also measured optically in at least three locations to determine the average width. Profile traces transverse to the reciprocating direction were taken in at least three locations to determine the average scar depth and shape. Profile traces in the direction of motion were taken to determine the curvature of the scar ends.

A geometric model was developed to calculate the wear volume of both the ring segment and cast iron flat that takes into account the compound curvature of the ring segment and the curvature of both wear scars. Details of this model are described in Appendix A. This model is a refinement of more commonly used models, which assumes the wear scars to be flat. The post-test profile measurements serve as input to the model to calculate the wear volume. Therefore, the wear of each sample can be expressed either as (a) a weight loss (if measurable), (b) a wear coefficient, or (c) the depth of wear. The last measure is important since it can be used to compare rig test results with engine test data. The individual component wear and the total system wear are reported for each test.

Table 1
Estimation of the Hertzian stresses in the bench test compared to the engine

	Unworn	Worn (3 mm scar width)
Bench test (240 N load)	354 MPa	174 MPa
Engine (4000 N load)	51 MPa	42 MPa
Multiplication factor	7	4

3. Results and discussion

3.1. Friction testing results

Figs. 5 and 6 summarize the effect of load, temperature, and lubricant on the friction behavior of the conditioned, run-in ring and new cast iron flat. The data is presented in a Stribeck type format to show the effect of load to establish lubrication regimes. Fig. 5, for a new fully formulated heavy-duty lubricant, shows that, even at room temperature, either mixed or boundary lubrication conditions exist. This is likely due to the relatively small contact area leading to large stresses, even at lighter loads. At high load, there is no significant increase in friction coefficient from 25 to 100 °C, possibly due to the friction modifiers added to a multi-viscosity oil. The reproducibility of the friction coefficient

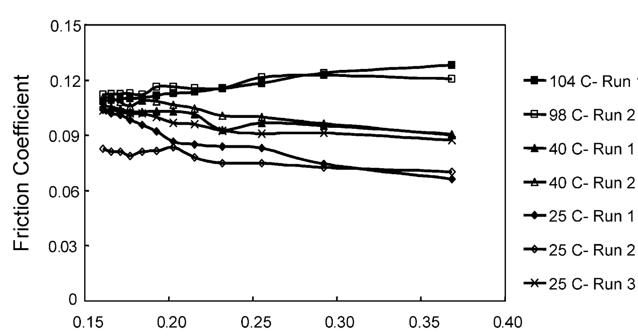


Fig. 5. The effect of temperature and load on the friction coefficient for new 15W40 oil.

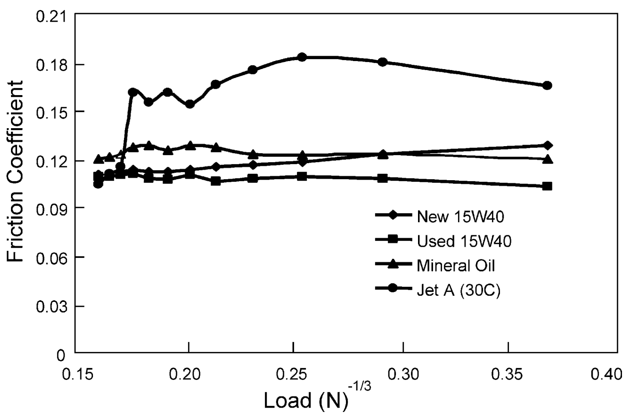


Fig. 6. A comparison of the friction coefficient variation with load for a variety of lubricants at 100 °C (Jet A was tested at 30 °C).

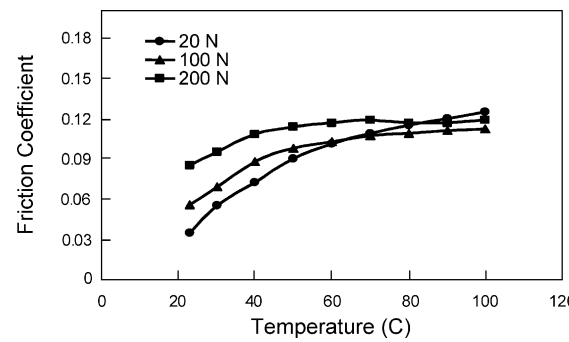


Fig. 7. Relationship between friction coefficient and temperature for new 15W40 oil.

in this rig test is very good except for the results at 25 °C. As a result of this reproducibility at higher temperatures, subtle differences in the performance of different lubricants can be detected.

Fig. 6 is a summary of the 100 °C friction data for the four different lubricants tested in this study with the exception of Jet A, which was tested at 30 °C. As would be expected, Jet A aviation fuel with the lowest viscosity and no additives, produced the highest friction coefficients for all but the highest loads, but the increase was not substantially higher. The reason for the relatively small difference is unclear, nor is the reason for the reduction of friction coefficient at higher loads. Mineral oil was comparable to the new fully formulated oil except at higher loads, where it had a more difficult time producing an effective boundary film. The used fully formulated oil had the lowest friction coefficient of the group. This oil, although typical of the aging produced in a routine service interval, was by no means degraded to the point where it was losing functionality.

Lower friction coefficients can be achieved by shearing the long chain polymer viscosity improvers. This shearing occurs very rapidly in service because of running through gear pumps, small oiling orifices, and sliding surfaces, and is, therefore, typical of a lubricating oil in actual use. Another possibility for reduced viscosity in actual service can be due to fuel leaking into the oil from bad fuel injectors. In the current study, the precise cause for the reduction of the friction coefficient could not be determined directly.

The effect of temperature on new fully formulated oil is shown in Fig. 7. The friction coefficient changes very little with either load or temperature over the range of approximately 60–100 °C. From a testing point of view, this implies that strict temperature control may not necessary when testing lubricants in this condition. The engine-aged fully formulated oil showed a slightly stronger temperature dependence, although there is little difference between 90 and 100 °C. The temperature dependence of mineral oil looked very similar to the used oil albeit with a slightly higher friction coefficient along the whole range of temperatures, most likely due to the absence of additives which would produce a more effective boundary film.

In summary, for the friction behavior, the stepped load approach is effective in identifying the sensitivity of the friction coefficient to load and temperature and allows for a determination of lubrication mode. Small but definite differences among a wide range of lubricants can be reliably measured, which will facilitate the evaluation of a variety of used lubricants for the selection of a standardized composition.

3.2. Wear testing results

Wear tests were conducted with the same lubricants as in the friction testing using the procedure described in Section 2. New ring and cast iron flat segments were used for each test in order to measure wear volumes and depths. It is important that wear is described as a system characteristic; that is, considering both the wear of the individual components and that of the total system. If this is not done, a new ring face material may show lower wear than current conventional materials but cause enhanced wear in the counterface, making it unsuitable for use.

Fig. 8 is a summary of the wear as measured by weight loss of the ring and cast iron flat in the different

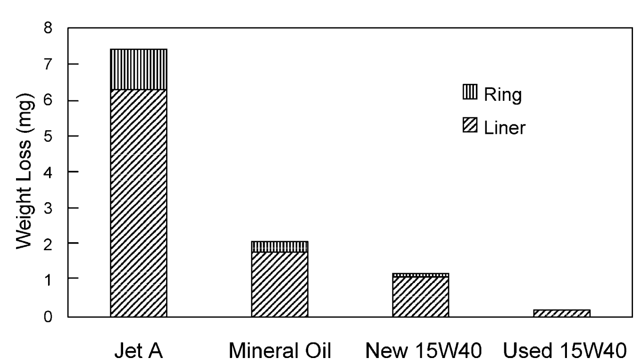


Fig. 8. Ring and cast iron flat weight loss after wear testing in various lubricants.

lubricants. These results have not yet been repeated, so it is not known what the variability may be. The relative amounts of wear correlate with the effectiveness of the lubricants, with Jet A showing 3–4 times the wear of the mineral oils. Used 15W40 oil showed the least wear, as it had been conditioned by prior use, lowering the friction coefficient as described above. The cast iron flat showed much more weight loss than the ring because a larger wear scar is produced compared to the smaller contact area of the ring face. Also, the cast iron flat is softer than the plasma sprayed Cr plating, although quantifying these hardness differences is difficult due to the inhomogeneity and lack of ductility of both materials.

Profile traces and wear scar width measurements after testing allowed for a wear volume to be calculated using the geometric model described in Appendix A. These wear volumes were then compared to the volume of material lost by weight change through a density calculation. The void density of the plasma sprayed coating was approximately 10% as measured by image analysis and was used to correct the Cr loss measurement. This comparison is presented in Fig. 9. WL refers to the volume as determined by the weight loss and PT refers to the volume calculated using profile traces and the geometric model. As can be seen, there is fairly good agreement between the two methods and this is further illustrated by the correlation plot in Fig. 10. This can be considered as a validation for the geometric model, since the weight loss measurement involves fewer assumptions. However, there are test conditions, which may produce such low wear that a weight loss measurement becomes impractical and profilometry would be more sensitive. This test configuration allows for either type of measurement.

From the point of view of the engine hardware, wear is better expressed in dimensional terms. For example, if the ring face plating thickness is about 200 μm, as is the case here, then the time needed to produce a wear depth 200 μm from the face can be used to define the effective life of the part. Both conventional engine wear testing and radioactive tracer methods, such as surface layer activation, determine wear as a rate of change of a critical dimension. From the profile traces of these tests, a depth of wear has been determined for both parts and a wear rate calculated by dividing the wear depth by the time of test. This data is shown in Fig. 11 and the wear rates determined by this method can then be compared to engine measurements reported in [5,6].

A comparison between rates measured in this bench test and those measured in a running engine using surface layer activation is summarized in Table 2. First, the ring wear rates in the laboratory test are about an order of magnitude higher than in the engine. This is desirable for an accelerated test so that results may be obtained in a reasonable test time. One reason for the accelerated rate in the laboratory test is due to higher and relatively constant contact stresses as compared to the cyclic loading of the ring against the liner during engine operation. Although the ring is in constant contact with the liner in an engine, the load on the ring varies due to the compression, combustion, and expansion of the gases in the combustion chamber [7].

The liner in an engine experiences maximum wear near the top ring reversal position, where the loading is highest for the reasons described above [7]. Furthermore, the speed is the lowest at this region, reducing the oil film thickness. Using surface layer activation with localized irradiation at this position, liner wear rates measured in an operating engine typically are about 25–50% of the ring wear rate [5,6]. The laboratory tests, therefore, substantially overestimated the relative wear rates of the cast iron compared to the ring. All of the reasons for this are still unclear, although there are several possible factors to be considered. First, the engine in the mentioned references made use of an electrodeposited Cr plated ring in contact with a cast iron liner similar in microstructure to the cast iron flat used here.

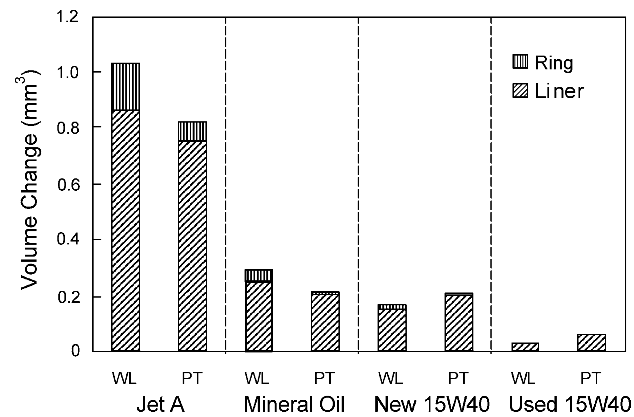


Fig. 9. The wear volumes of the ring and cast iron flat segments calculated using the geometric model.

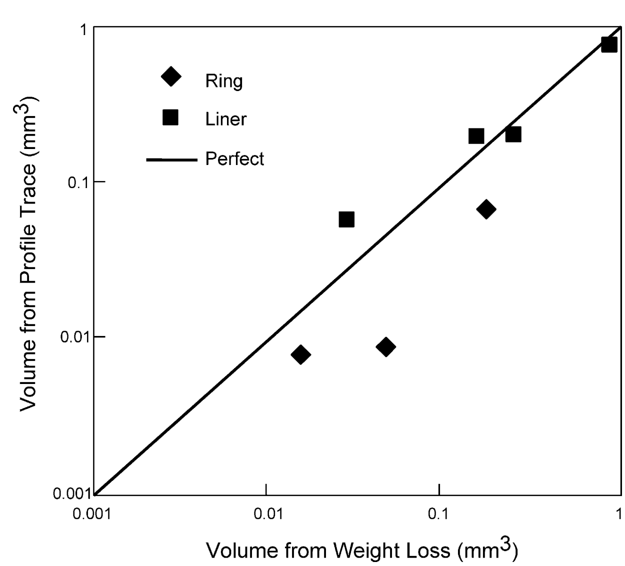


Fig. 10. Correlation between the wear volume calculated from weight loss with that calculated from profile traces.

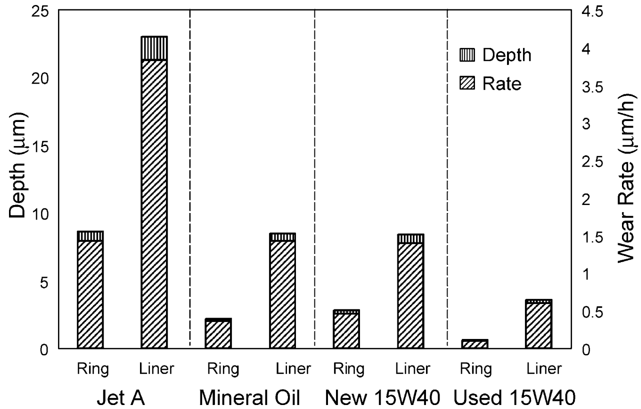


Fig. 11. Wear depth and wear rate of ring and cast iron flats calculated from profile traces.

The laboratory test used a plasma sprayed Cr plated ring, which would ordinarily be used with an induction hardened liner surface. The use of the plasma sprayed ring segment against unhardened cast iron may have contributed to higher iron wear rates. Also, the effect of lubricant chemistry in the top ring reversal area and the stability of a solid film boundary layer produced at higher temperatures may contribute to lower wear in the engine environment. The most difficult conditions to reproduce in a laboratory test are the variations in pressure, temperature and gas composition, which occur in the combustion chamber. In future work, an attempt will be made, through controlling the applied load, to shift the relative wear rates to better reflect engine experience. The friction and wear test methods described here will be applied to oils either engine-aged or artificially aged to standardize the test fluid. The engine-aged oil will come from ASTM standard oil aging tests.

4. Summary and conclusions

This paper describes laboratory test methods to measure the friction and wear characteristics of ring and liner materials in heavy duty engines. A wide variety of lubricants ranging from Jet A aviation fuel to fully formulated heavy duty lubrication oil were used to determine the ability of the test methods to detect variations in lubricant performance. Friction behavior was determined by using stepped loading. This approach allows for the determination of lubrication mode as well as the range of friction coefficients encountered in service. Wear tests at fixed load accelerated the wear rates of the ring and cast iron flat by at least an order of magnitude over that observed in an operating engine; however, the relative liner wear rate compared to the ring was much higher than encountered in the engine.

Table 2
Comparison of bench test ring and liner wear rates to the engine

	Ring wear rate (μm/h)	Liner wear rate (μm/h)
Bench test	0.1–0.5	0.6–1.4
Engine test	0.005–0.05	0.002–0.02
Multiplication factor	10–20	70–300

Acknowledgements

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Appendix A. Wear volume models of the ring and liner

A.1. Wear volume of the ring

A schematic drawing of a hypothetical piston ring with compound curvatures is illustrated in Fig. A1. The original ring and crown radii are denoted by r_o and R_o, respectively. When the ring sliding against a flat specimen with comparable or lower hardness, the worn surface on the ring is not flat but has compound curvatures, with ring radius r_w and crown radius R_w. The top view and side view of the wear scar on the ring are shown in Fig. A2, with scar length l and scar width w. The wear volume V_{ring} can be obtained by

$$V_{ring} = f_r(w,l,r_o,R_o) - f_r(w,l,r_w,R_w) \tag{A1}$$

where f_r is defined below and can be solved by numeric integration

$$f_r(w,l,r,R) = 2\int_0^{l/2}\left[a^2 \arccos\left(\frac{b}{a}\right) - b\sqrt{a^2-b^2}\right]\mathrm{d}y \tag{A2}$$

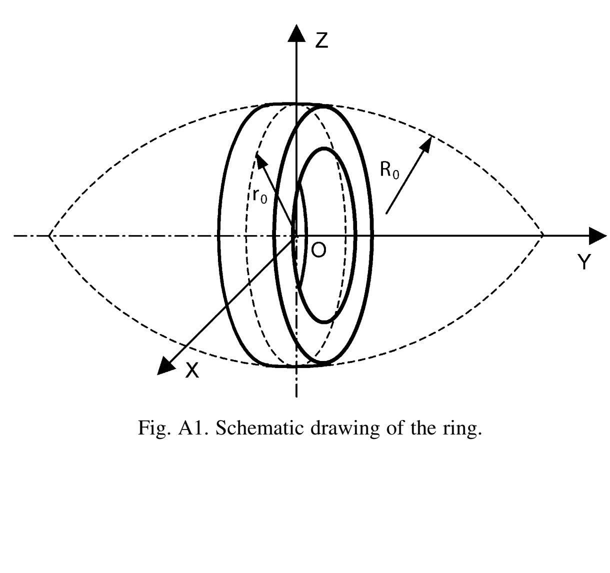

Fig. A1. Schematic drawing of the ring.

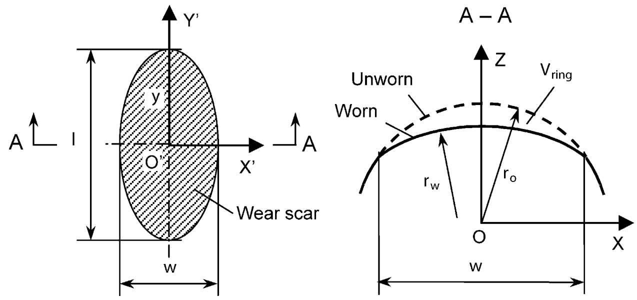

Fig. A2. Schematic drawings of the wear scar on the ring.

$$a = \sqrt{R^2 - y^2} - R + r \tag{A3}$$

$$b = \sqrt{r^2 - \frac{w^2}{4}} \tag{A4}$$

A.2. Wear volume of the cast iron flat specimen

A schematic drawing of the wear scar on the flat cast iron specimen is illustrated in Fig. A3. L_s is the stroke length of the reciprocating test. The full length of the wear scar on the flat is the stroke length (L_s) plus the wear scar length (l) on the ring. The flat cast iron has the same scar width (w) as the ring. The wear scar on the flat is composed of three segments, the cylindrical middle and the compound-curvature ends with radii of R_f and r_f.

The wear volume of the cylindrical scar middle, V_{f2}, can be directly calculated by

$$V_{f2} = L_s\left[r_f^2 \text{ arc sin}\left(\frac{w}{2r_f}\right) - \frac{w}{2}\sqrt{r_f^2 - \frac{w^2}{4}}\right] \tag{A5}$$

The wear volumes of the compound-curvature scar ends, V_{f1} and V_{f3}, may be obtained by

$$V_{f1} = V_{f3} = \frac{1}{2}f_r(w, l, r_f, R_f) \tag{A6}$$

where f_r has been defined by Eq. (A4) in Section A.1.

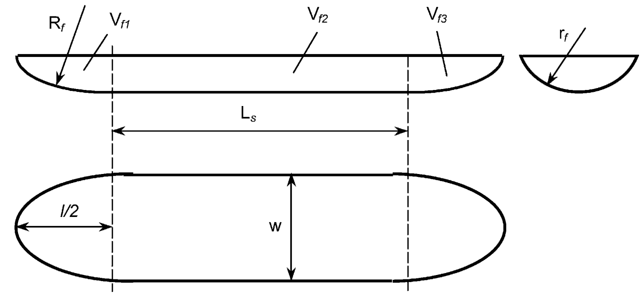

Fig. A3. Schematic drawing of the wear scar of the cast iron flat.

Thus, the whole wear volume of the flat cast iron is

$$V_{\text{flat}} = 2V_{f1} + V_{f2} \tag{A7}$$

References

[1] Truhan JJ, Kodali P. A comparison of rig and engine testing to measure in-cylinder wear in heavy duty diesel engines. Proceedings of the fourth international filtration conference. San Antonio, TX: Southwest Research Institute; 2001 p. 76.

[2] Blau PJ. A review of sub-scale test methods to evaluate the friction and wear of ring and liner materials for spark and compression ignition engines, ORNL/TM-2001/184. Oak Ridge National Laboratory Nov 2001.

[3] Cheng HS. Lubrication regimes. In: ASM handbook, vol. 18. Friction, lubrication and wear technology. Materials Park, OH: ASM International; 1992. p. 89–97.

[4] Young WC. Roark's formulas for stress and strain, 6th ed. New York: McGraw-Hill; 1989 p. 650–1.

[5] Truhan JJ. The classification of lubricating oil contaminants and their effect on wear in diesel engines as measured by surface layer activation. Proceedings of the first international filtration conference. San Antonio, TX: Southwest Research Institute; July 1996 p. 239.

[6] Truhan JJ, Covington CB. The use of direct method surface layer activation to measure wear in diesel engines. Second international congress and exposition for mobilization technology, Paper SAE 93162. Sao Paulo, Brazil: Society of Automotive Engineers; Oct 1993.

[7] Kodali P, Truhan JJ, Richardson D. A study of cylinder liner wear. Proceedings of the third international filtration conference. San Antonio, TX: Southwest Research Institute; Sept 1999 p. 68.