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A Review of Sub-Scale Test Methods to Evaluate the Friction and Wear of Ring and Liner Materials for Spark- and Compression Ignition Engines

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Metals and Ceramics Division

A Review of Sub-Scale Test Methods to Evaluate the Friction and Wear of Ring and Liner Materials for Spark- and Compression Ignition Engines

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Preface

Energy resources are of strategic interest to the United States, and the transportation sector is a major consumer of energy. In 1998 the transportation sector accounted for about 27.7 % of the total U.S. energy consumption.* The United States Department of Energy, Office of Energy Efficiency and Renewable Energy, Office of Transportation Technologies, is sponsoring programs to develop new scientific and technological strategies that will improve the efficiency of vehicles and therefore reduce the consumption of vital U.S. energy resources.

Friction, lubrication, and wear (tribology) issues impact the energy efficiency of vehicles in many ways, and therefore it is important to develop new design concepts, lubrication strategies, and tribomaterials (materials whose functions involve friction or wear) for engine and drive-train components. The current work supports that aim.

Testing is a significant component of materials development. Full-scale vehicle tests or instrumented engine test cell programs can be very expensive, and industry seeks to reduce the cost of obtaining engineering design and selection data for materials, lubricants, and coatings. Any useful subscale test (simulative test) must rank materials in the same order of merit as they would behave in the end-use applications, which in the present case are piston rings and cylinder bores. This study was conducted to review past laboratory-scale test methods and to assess their validity for ranking materials and lubricants for use in engines. It concludes with a summary and recommendations for future research.

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* *Transportation Energy Data Book*, Edition 19, Oak Ridge National Laboratory report 6958 (1999), p. 2-4.

Executive Summary

A review was conducted of past laboratory-scale test methods and to assess their validity for ranking materials and lubricants for use as piston and liner materials in compression-ignition (CI) and spark-ignition (SI) engines. Most of the previous work was aimed at simulating SI engine environments. This report begins with a discussion of the numerous factors that can affect the validity of an approach to simulating engine conditions in a laboratory. These include not only mechanical, chemical and thermal factors, but also human factors as regards how the vehicle is operated and maintained. The next section provides an annotated review of open literature publications that address the issues of laboratory simulation of engine components. A comparison of these studies indicates a lack of sufficient standardization in procedures to enable a systematic comparison of one publication to another. There were just a few studies that compared several laboratory test methods to engine test results, and these indicated that some test methods correlate, at least qualitatively, better than others. The last section provides a series of recommendations for improving the accuracy and validity of laboratory-scale simulations of engine behavior. It became clear that much of the engine wear damage occurs during start-up when the engine is cold, and this calls into the question the usefulness of test methods that attempt to simulate steady-state running conditions. It is recommended that a new standard test method, perhaps developed with the help of the ASTM wear and erosion committee, be developed. It would use cold start-up conditions in the presence of degraded oil, or simulated degraded oil.

1.0 Introduction

The performance of mechanical components is limited by the capabilities of the materials from which they are made. Machine designers must make compromises in performance for a number of reasons. Sometimes, engineering materials of low enough cost are not available to meet the demanding requirements of new equipment designs and less than optimal materials must be used. Sometimes using a premium grade material or surface treatment is the only way to meet the design requirements. The latter tends to be the case for high-performance military applications or high-end automobiles and trucks. Compromises between performance and cost are particularly common in the highly-competitive ground transportation industry.

Diesel engines, which by most measures are more fuel-efficient than spark ignition engines, comprise the major propulsion systems for on-highway heavy vehicles, like Class 8 trucks. Diesel engine technology has been driven in recent years by increasing pressure from two sources: (1) the desire to improve fuel economy, and (2) the need to meet increasingly stringent emissions requirements. Designers have used a broad spectrum of strategies to improve performance and lower emissions. Widening the choice of structural materials offers designers more opportunity to optimize engines and meet these two important goals. Some of the needs for new materials involves the use of *tribomaterials* – i.e., materials whose functions in some way involve friction, lubrication, and surface damage resistance. Both the materials used for friction- or wear-critical parts and liquid lubricants are broadly considered to be tribomaterials.

When new materials, lubricants, or surface treatments are developed, it is important to be able to evaluate their potential to improve engine performance in a fast, efficient, and cost-effective way. That requirement poses a significant challenge for the test engineer because small changes in the way materials are exposed to the mechanical, chemical, and thermal aspects of their surroundings can affect their friction and wear behavior. Therefore, the ultimate challenge becomes one of identifying and controlling the key factors needed to enable materials, lubricants, and coatings to be tested in the laboratory in such a way that directly correlates with their performance in the end application. The degree of confidence by engineering decision-makers in laboratory test data must be based on laboratory-field correlations.

1.1 *Factors Involved in Simulations*

In the present context, the term *simulation* refers to a physical test that attempts to duplicate the key factors in an engine, not a mathematical simulation that generates results based on a set of starting assumptions and boundary conditions. Figure 1 schematically depicts six main categories of effects that must be addressed in developing useful laboratory-scale tests for candidate engine materials and lubricants. Many of the factors listed in each box are interrelated. For example, the sliding speed and contact pressure can combine to produce frictional heating. Frictional heating, in turn, can alter the tribochemistry of the exposed surfaces. Thus, a change in one factor can affect several other factors. This degree of complexity makes it difficult or impossible to conduct friction or wear experiments in which only one parameter is treated as the independent variable and everything else is ‘held constant.’

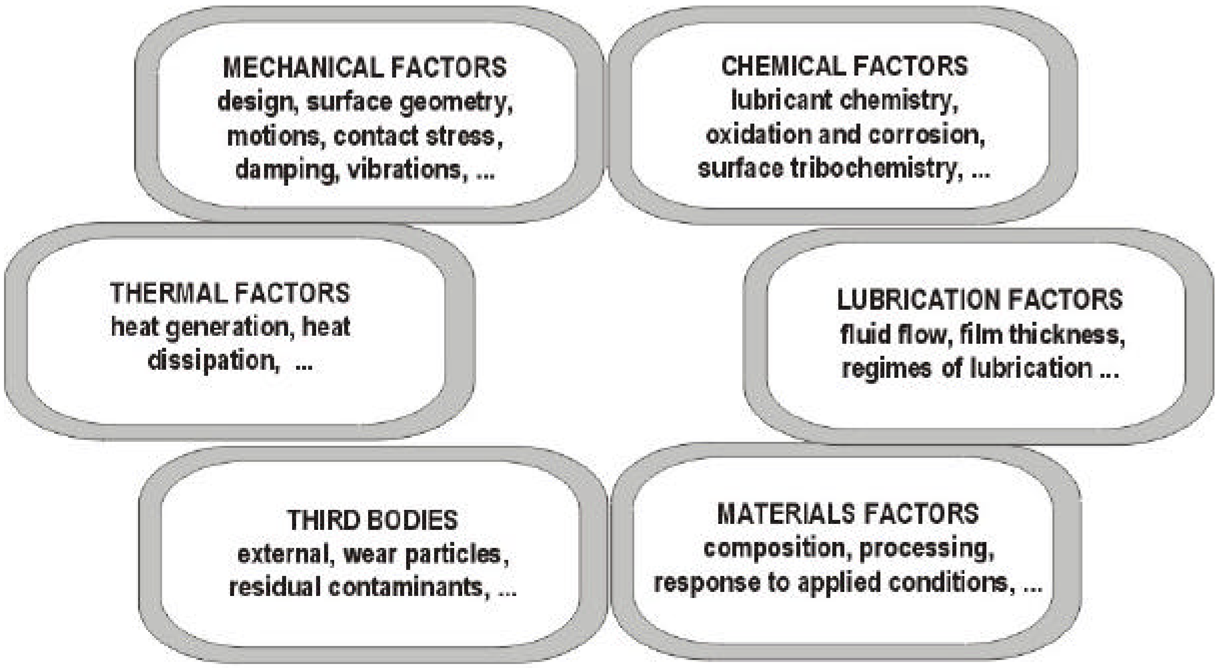

Figure 1. General categories of factors that must be considered in the course of developing accurate simulations of friction and wear-critical parts in diesel engines.

1.1.1 *Mechanical factors* include much more than just the stresses applied to surfaces and the speed of relative motion. They include the periodicity or intermittency of relative motion, the stiffness and damping capacity of the system, and the manner in which surfaces meet at macro and microscales. Despite heroic attempts to control the quality of the engines we build, we still live in an imperfect world. Machines are designed and built by imperfect people from imperfect materials in imperfect factories, and the result of this is phenomena like residual assembly stresses, surface finish defects, contamination by hard particles, and alignment errors in the assembled components. Some of these effects are subtle and difficult to detect during manufacturing, but they can nevertheless have an effect on the life and performance of an engine. Simulating such subtle effects is not easy and can widen the gap between laboratory testing and field response.

1.1.2 *Chemical factors* include the manner in which chemical reactions play a part in friction and wear behavior. Under rubbing contact, chemical reactivity and kinetics are known to change. The formation of acids in lubricants can accelerate wear. Reactivity varies strongly with temperature. Engine materials can dissolve in fuels and lubricants, changing their chemistry over time. Deposits, like carbon, build up on contact surfaces.

1.1.3 *Lubrication factors* include identifying the appropriate regime(s) of lubrication experienced by the components of interest and the degree to which the laboratory test must replicate that/those regime(s). Lubricants also serve to remove heat and harmful particles. A simulation should replicate the regime of lubrication and account for the changes in viscosity of the lubricant during the operating cycle.

1.1.4 *Materials factors* involve the composition, processing, and microstructures of the materials that must be lubricated or protected from wear. Surface treatments and coatings are used to protect materials if full separation by the lubricant cannot be ensured.

1.1.5 *Third bodies* are particles that affect wear or lubrication. Three-body wear is, in fact, a form of abrasive wear in which loose particles in the interface between bodies result in the progressive loose of surface material. The sources of third bodies include dust from the environment, wear particles, sand left over from engine block casting operations, and soot particle agglomerates that happen to get entrained in the lubricant. Simulating third body effects is challenging because their effects have to be balanced with the other influences affecting the dominant wear mode.

1.1.6 *Thermal factors* include the generation and dissipation of heat in the engine components. Among other things, temperature affects the tribochemistry of the lubricant and its ability to support the contact pressure (viscosity). Much of the wear of engines is generated when the engine is cold, at start-up, so wear should not be assumed to be a linear function of operating time. Rather the instantaneous wear rate is expected to change significantly over the course of an engine's useful lifetime.

1.1.7 *The Human Factor*. Arguably the most important, yet uncontrolled, variable to be addressed in seeking a correlation between laboratory results and field performance is the human factor. Designers have a difficult time accounting for the whims and eccentricities of the driver and the use to which a vehicle is put once it leaves the dealer's control. The author of a popular paperback book puts it this way:

> "Anyone can drive a car for 200,000 miles if the engine and transmission are rebuilt every 75,000 miles. There's no trick to that. But there is a trick to driving 200,000 miles without a major repair."
>
> Bob Sikorsky (1997)
> Drive It Forever, ATG Media.

The influence of human factors is manifested in three ways:

1. The driving profile and operating environment for the vehicle. This includes things like the average length of each trip, the number of trips per day, and the part of the country that the vehicle operates in (ambient temperature, weather conditions, and topography).

2. Operator technique. This includes aspects like the aggressiveness of the driver, the driver's tendency for rapid accelerations and decelerations, and whether the driver warms up or idles the vehicle before starting to drive.

3. Care and maintenance practices. This includes oil change intervals, expertise of the person conducting engine repairs, engine adjustments, and related factors. If a vehicle is driven for long distances (compared with frequent short trips), when wear rate is least severe, the oil change interval can be extended.

Appendix A, compiled from two recent reviews, lists the scientific and engineering factors involved in simulations in greater detail.

1.2 *Levels of Testing Scale*

The German standard DIN 50 322 lists six levels of tribotesting that range from field trials (Level I) to simple specimen coupon tests (Level VI). This system can be further simplified into five levels as follows:

Level A. **Vehicle on- and/or off-road tests**
Level B. **Full-size dynamometer test stand** (entire vehicle)
Level C. **Full-scale engine tests** (engine test cells)
Level D. **Sub-assembly tests** (full-scale mating parts)
Level E. **Coupon tests** (sub-scale tests, part sections or simple coupons)

The cost per test tends to increase from Level E to A, but the degree to which the test variables can be controlled tends to increase in the opposite direction.

The primary concern in developing and using Level D and E tests to select materials and lubricants is to establish a direct engineering linkage to Level A performance. It is possible that an intermediate level of correlation will be necessary. For example, it may be possible to correlate Level E to Level C and Level C to Level A. Correlation issues comprise the focus of this review.

When reviewing the literature of laboratory simulations of ring and liner performance, seven basic considerations were kept in mind:

Consideration	Level A - Performance	Level E - Performance
Contact stress level	difficult to measure in running engines; often calculated based on geometric and operating assumptions; varies with time and wear-in; may vary from engine to engine and cylinder to cylinder	generally easy to control in laboratory testing systems; conformal contact enables constant stress; counterformal contact generally produces decreasing stress with time due to wear
Heat flow	depends on engine configuration and cooling system; varies during start-up; may vary from cylinder-to-cylinder; affects internal stresses from differential thermal expansion of adjacent parts	tends to reach a steady state condition and is repeatable from test to test
Residual stress level	varies with the assembly procedure and heat treatment of the part; residual stress distributions exist in cast and wrought parts; difficult to measure in full-size parts (neutron-based methods)	difficult to simulate in the laboratory
Part alignment	can have significant effect on wear pattern and friction during start-up and running-in	affects the repeatability of results, affects uniformity of contact stress, important effects in conformal testing
Surface features	critical for identifying the dominant	an important indication of the accuracy of

	form(s) of wear; can be hidden by deposits or lubricant residues	the simulated wear behavior
Lubricant degradation	oxidation and contamination with water and fuel changes behavior; lubricant residence time in hot areas can trigger changes; lubricant properties in the sump are not the same as those in the ring-liner contact zone; oil changes	difficult to simulate correctly in the laboratory; difficult to duplicate the form of soot, fuel residues, wear debris, and water build-up
Constancy of conditions	field use produces a wide range of variable operating conditions; even relatively tight test protocols are difficult to reproduce exactly due to weather, road conditions, traffic, etc.	most often run under constant operating conditions; the development of complex protocols involving variable loads, speeds, temperatures, start/stop, and changing oil conditions is rarely if ever done at this level of testing

2.0 Annotated Literature Review of Ring/Cylinder Bore Studies

Information for this review was gleaned from a variety of sources. Each citation briefly summarizes the approach taken when producing the simulation. Only a small number of these studies compared the results of their work with similar materials or lubricant performance in fired engines. That made it difficult to assess the validity of the simulation. In most cases, only a few of the aspects of the engine environment were matched in the design of sub-scale tests. Such features as piston ring/cylinder alignment during engine assembly, and cylinder-to-cylinder variations in running engines were not generally considered in laboratory simulations.

The following standard format was selected for this review to facilitate the comparison of published studies.

[Ref #] Author(s)	Publication details - date, volume, etc.
Title of Publication	
Components being simulated	Level of Simulation (A, B, C, D or E)*
Applied load or contact pressure	Rationale for selection
Applied motion	Rationale for selection
Applied contact speed	Rationale for selection
Materials/surface treatments	
Lubricant(s)	Test temperature/environment
Correlation of results with actual components (if any)	
Other comments or discussion:	

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* see listing and description in Section 1.2

Summary List of References that are Outlined in the Following Section:

[1] R. J. Sloan, U.S. patent # 5,007,284 (1991)
[2] G. F. Al-Khalidi and T. S. Eyre, *Tribology International* (1987) Vol. **20** (1), 18-24
[3] K. F. Dufrane and W. A. Glaeser Proc. Intl. Conf. on Wear of Matl's (1987), ASME, pp. 285-291.
[4] K. F. Dufrane, W. A. Glaeser, and A. R. Rosenfeld, Report (1988) ORNL/Sub/84-00216/1
[5] S. E. Hartfield-Wunsch, S. C. Tung, and C. J. Rivard, SAE Technical paper (1993) # 932693.

[6] G. C. Barber and K. C. Ludema *Wear* (1987) Vol **118**, pp. 57-75
[7] M. G. S. Naylor, Oak Ridge National Laboratory Technical Report (1992) ORNL/Sub/87-SA581/1
[8] M. P. Volarovich, *Wear* (1958/9) Vol. **2**, pp. 203-216.
[9] E. Wacker, Metallurgical Aspects of Wear (1979), pub. by DGM E.V., pp. 247-267
[10] D. J. Patterson, S. H. Hill and S. C. Tung, *Lubrication Engineering* (1993), February, pp. 89-95.
[11] S. Venkatesh, *Wear* (1973), Vol. **25**, pp. 65-71
[12] S. H. Hill, S. E. Hartfield-W∫nsch, and S. C. Tung (1996), *Tribology Trans*., Vol. **39** (4), 929-935.
[13] V. D. N. Rao, D. M. Kabat, D. Yeager, and B. Lizotte, Concepts in New Engine and Component Design, SAE Special Pub. SP-1225 (1997); also SAE Paper 97009.
[14] V. D. N. Rao, D. M. Kabat, R. Rose, D. Yeager, R. Brandt, and D. Y. Leong, SAE Paper (1997) SAE 970008
[15] V. D. N. Rao, D. M. Kabat, H. A. Cikanek, C. A. Fuinari, and G. Wuest, SAE Paper (1997) SAE 970023 (in Applications for Aluminum in Vehicle Design, SAE Spec Pub-1251)
[16] P. I. Lacey and R. T. Stockwell (1999) *Tribology Trans*., Vol. **42** (1), 192-201.

Annotated Reference List:

[1] Ralph J. Sloan U.S. patent # 5,007,284 (1991)
Piston Ring and Liner Wear Simulator and Method of Using Same
piston ring and cylinder liner wear — Level E – piston rings and liner segments
pneumatic pressure – variable* — up to ~ 650 N to sim. turboch. diesel ring loads
Reciprocating motion — same as ring-on-liner motion
Applied contact speed — up to 700 rpm (limited by inertia of apparatus)
Materials/surface treatments — various plated/coated rings and grey cast iron liners
Lubrication: none or engine oil — T up to 550^{o} C (max. for advanced diesels)
Correlation with Cummins V-903 production engine, un-cooled Cummins NTC 250 engine
Other comments: Instrumented with crank angle and friction force measurements. Other data comparisons were with Cameron-Plint TE-77 reciprocating tester to check ability to discriminate between oils. *Typical load: 125 N/cm of ring width.
Sample test results:

Sloan machine data:

Ring Material	Liner Material	Load (N/mm)	Speed, Test time (rpm, h)	Temper-ature (C)	Lubricant, supply rate (ml/h)	Friction Coefficient	Liner Wear Rate (mm^3/N-m)
Cr-plate	grey cast iron	124	266, 2	25	mineral oil, 1	0.05	1.9 x 10^{-6}
Pl. Spray Cr_2O_3	Pl. Spray Cr_2O_3	124	266, 2	427	none	0.55	< 3 x 10^{-7}
TiC/CaF_2 coating	Cr_3C_2 coating	124	266, 2	427	none	0.58	4.5 x 10^{-5}

Cameron Plint machine data:

Ring Material	Liner Material	Load (N/mm)	Speed (rpm)	Temper-ature (C)	Lubricant (in tray)	Friction Coefficient	Liner Wear Rate (mm^3/N-m)
Cr-plate	grey cast iron	8	600	25	mineral oil	0.16	2.3 x 10^{-6}
Cr-plate	grey cast iron	20	1200	25	mineral oil	0.06	1.4 x 10^{-8}
TiC/CaF_2 coating	Cr_3C_2 coating	20	360	360	none	1.45	1.9 x 10^{-5}

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[2] G. F. Al-Khalidi and T. S. Eyre *Tribology International* (1987) Vol. **20** (1), 18-24
Bore-polishing – identification and simulation
cylinder bores of diesel engines — Level E – piston ring segment on liner segement

Load: 300-1500 N in 300 N increments not given
Reciprocating motion piston on liner motion
8.3 cycles/min for up to 20 h not given
Materials/surface treatments
Two oils – one promoting bore polish. 80^o C constant immersion
Correlation with actual components: Observed features similar to those in a 'Tornado' diesel engine.
Other comments: Characteristics of bore polishing features in a Tornado diesel engine: 'light polish' = mirror finish overlaid on the honing pattern (Ra = 0.55 μm), 'medium polish' = mirror finish showing faint original honing pattern (Ra = 0.15 μm), 'heavy polish' = mirror finish with no traces of the original honing pattern (Ra = 0.08 μm)

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[3] K. F. Dufrane and W. A. Glaeser Proc. Intl. Conf. on Wear of Matl's (1987), ASME, pp. 285-291.
Wear of Ceramics In Advanced Heat Engine Applications
Rings and liners Level E – cylinders (in a simulated piston) against opposing flats
Up to 7.7 N/mm Assume 35 N/mm is typical for a diesel engine
Reciprocating cylinders on flats to simulate a crowned ring on a liner
Speeds 500-1000 rpm for 1.5 hr rationale not given
Series of self-mated monolithic ceramics
Dry and with SAE 10W oil 290-540^o C with diesel exhaust blown through
Correlation with actual components: not given, except for reference to 'typical' value of the wear coefficient for a Cr-plated ring on cast iron.

Selected data follow:

'Ring' Material	'Cylinder' Material	Lubricant	Test Temperature (C)	'Ring' Wear Coefficient, k (dimensionless)
Cr	cast iron	SAE 30	100	5 x 10^{-9} reference*
Cr	cast iron	SAE 10W SF/CC	20	2 x 10^{-6}
SiC	SiC	none	20	fracture – no measurement
SiC	SiC	SAE 10W SF/CC	20	3 x 10^{-6}
Si_3N_4	Si_3N_4	none	20	9 x 10^{-4}
Si_3N_4	Si_3N_4	SAE 10W SF/CC	20	2 x 10^{-6}
YPSZ	YPSZ	none	20	8 x 10^{-4}
YPSZ	YPSZ	SAE 10W SF/CC	20	2 x 10^{-4}

* Authors claim to be "typical diesel truck engine experience"

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[4] K. F. Dufrane, W. A. Glaeser, and A. R. Rosenfeld Report (1988) ORNL/Sub/84-00216/1
Studies of Dynamic Contact of Ceramics and Alloys for Heat Engines
Rings and liners Level E – cylinders (in a simulated piston) against opposing flats
Up to 950 N, up to 50 N/mm Assume 35 N/mm is typical for a diesel engine
Reciprocating cylinders on flats to simulate a crowned ring on a liner
Speeds 500-1000 rpm for 1.5 hr rationale not given
Series of self-mated monolithic ceramics
Dry and with poly-alphaolefin oil most tests at 260^o C with diesel exhaust blown through
Correlation with actual components: not given, except for reference to 'typical' value of the wear coefficient for a Cr-plated ring on cast iron.
Other comments: More detailed version of Ref # Contains additional friction and wear data.

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[5] S. E. Hartfield-Wunsch, S. C. Tung, and C. J. Rivard SAE Technical paper (1993) # 932693.
Development of a Bench Wear Test for the Evaluation of Engine Cylinder Components and the Correlation with Engine Test Results

Ring/cylinder bore	Level E – ring segment and bore segment
80N (7 MPa)	intended to represent typical pressure; scar width x ring width
Reciprocating motion	similar to piston and bore
600 rpm, 10 mm stroke, 5 hrs	turn-around events per unit time similar to engine

Thermally-spray modified aluminum alloy liners, cast iron (baseline), Cr and Mo plated rings.
SAE 5W-30 API SG oil — Temperature 120^o C to minimize oil degradation.
Correlation with actual components: compared results to a 200 hr, 4 cylinder engine dynamometer test.
Other comments: Thermal sprayed coating materials were not well described – only as “A, B, C, and D”, so it was not possible to correlate results with composition.

The following comparative wear test rankings were obtained:

Alloy	Bench Test Average Wear Depth (μm)	Bench Test Wear Ranking (1 = best)	Engine Test Average Wear Depth (μm)	Engine Test Wear Ranking (1 = best)
A	5.2	1	6.0	1
B	24.1	4	15.0	4
C	6.1	2	8.5	3
D	12.1	3	8.0	2

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[6] G. C. Barber and K. C. Ludema *Wear* (1987) Vol **118**, pp. 57-75
The Break-in Stage of Cylinder-Ring Wear: A Correlation between Fired Engines and a Laboratory Simulator

Piston ring/cylinder wall	Level E – ring segment on cylinder segment
Applied load or contact pressure	Rationale for selection
Reciprocating, slider crank	simulates ring and cylinder bore
350 cycles/min x 19 mm stroke	speed within the capabilities of the test machine

Materials/surface treatments: Mo-filled Cadillac top rings on coarse pearlitic cast-Fe.
Lubricant: 0.04 ml drip at 3 min interv. — Room temperature
Correlation with actual components: Examined 18 cylinder liners taken from fired Cadillac and Chevrolet engines. Engine run time from 20 min to 94,000 miles. Claimed damage features very similar in simulated and fired engine.
Other comments: No observable surface damage below contact pressures of 100 kPa.

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[7] M. G. S. Naylor — Report (1992) ORNL/Sub/87-SA581/1
Development of Wear-Resistant Coatings for Diesel Engine Components

ring/liner	Level E - cylinder-on-flat (Cameron Plint machine)
225 N on 7.5 mm ring (30 N/mm)	typical max. heavy-duty diesel ring load at top-dead-center
Reciprocating, 5 mm stroke	reciprocating ring in bore
20 Hz	not given

Materials/surface treatments: various ring coatings on several types of ‘cylinder’ materials
Lubricant(s): fresh oil, 3.3% C sooty oil (2 kinds), PAO lubricant, and nonlubricated, and T = 200 – 450^o C
Correlation with actual components: used practical operating conditions
Other comments: Average wear rates for 4 tests using Cr-plated ring material on a pearlitic cast iron cylinder material at 350^o C in fresh 15W40 mineral oil:

Ring wear rate	1.12 x 10^{-7} mm3/N-m

Liner wear rate	8.39 x 10^{-6} mm^3/N-m
friction coefficient	0.12

There was a significant effect of used versus new oil.

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[8] M. P. Volarovich *Wear* (1958/9) Vol. **2**, pp. 203-216.
An Investigation of Piston-Cylinder and Shaft Bearing Friction at Low Temperatures
Piston/liner, shaft bearing Level D – piston in a piston sleeve
Apparatus designed to apply various shear forces Rationale: study lubricant properties
Reciprocating motion Simulate piston motion
Applied contact speed not given
Materials/surface treatments: standard ring and liner materials not described
Lubricant(s): variety with different viscosities Test temperatures down to – 60^{o} C
Correlation with actual components: not done

Other comments: This study was designed to examine the effects of lubricant thickening at cold temperatures on the starting friction of engines. The experimental conditions of the work were poorly documented.

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[9] E. Wacker <u>Metall. Aspects of Wear</u> (1979), pub. by DGM E.V., pp. 247-267.
The Use of a Testing Machine for Simulating Piston Ring Groove Wear
Ring in the piston's upper ring groove Level E – articulating specimen in multi-axis machine
Normal pressure 50-70 bar (0.725-1 ksi) Rationale for selection: truck diesel motor typ. pressure
Applied motion- complex multi-axis motions Simulate closely a ring's complex motions in its groove
Applied contact speed 1600-2650 rpm Rationale for selection: typical diesel engine speed
Materials: comparison of austenitic cast iron to Al-12%Si alloy piston materials
Lubricant(s): not given Test temperatures 200 – 250^{o} C
Correlation with actual components: (see Other comments, below)

Other comments:
(1) The ratio of the wear rates for the fired diesel engine: Al-Si: cast Fe was about a factor of 3.
In lab tests, the ratio of Al-Si wear to cast Fe wear could be adjusted by changing the frequency and contact pressure to approach the same ratio. But the wear rates per mm of sliding distance were much higher in the lab (50-100 X) compared with the actual engine.

(2) Background wear data from fired engines (Piston material = Niresist austenitic cast iron)

	4-stroke truck diesel engine	4-stroke car diesel engine
Piston material	Niresist austenitic cast iron	Al-12%Si alloy
Max. wear of upper face of the upper ring groove (mm/100 hr)	0.005 – 0.012	0.08 – 0.20
Max. wear of lower face of the upper ring groove (mm)	0.004 – 0.015	

(3) It was noted that the corrosion and abrasive effects were not well known and difficult to simulate. Studies indicated that the wear of the groove face was increased with dust in the oil (up to 200 ppm).

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[10] D. J. Patterson, S. H. Hill and S. C. Tung *Lubrication Engineering* (1993), February, pp. 89-95.
Bench Wear Testing of Engine Power Cylinder Components
Ring/liner simulation Level D/E – 3 bench methods compared

(1) Unidirectional sliding, LFW-1 block-on-ring machine, incremental loading 133-534 N, 197 rpm (0.36 m/s), lubricant SAE 20W oil at 90-93^{o}C, conditions selected for accelerated testing. (Level E)
(2) Reciprocating sliding, Cameron-Plint TE-77 machine, 80-120 N load corresponding to 900-1300 psi cylinder pressure, 10-40 Hz with 7.5 and 10 mm stroke, 60 or 120 hrs, SAE 5W-30 or 20W-20 oil at 150^{o} C. Used undersized ring to avoid edge contact (Level D)
(3) Reciprocating sliding EMA-LS9 machine (similar to Sloan machine see Ref. [1]), load 288 N, speed 500-700 cycles/min, AMOCO 300 oil at 149^{o} C delivered at 1 ml/hr, test duration 20 hrs typ.

Correlation with actual components: Machine 1 and Machine 3 data for three different ring coatings were correlated with tests run on a engine dynamometer with both spark ignition and diesel engines.

	Correlation R^2 (%) Diesel Engine	Correlation R^2 (%) Spark Ignition
LFW –1	92	76
EMA-LS9	99	90

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[11] S. Venkatesh *Wear* (1973), Vol. **25**, pp. 65-71
Surface Treatments for Pistons and their Effect on Engine Performance
Diesel piston Level C – single-cylinder 4-stroke diesel
No stated contact pressure Dictated by operating conditions
Reciprocating motion Dictated by engine design
Applied contact speed: 1500 rpm Selected speed
Materials/surface treatments: untreated Al alloy piston, phosphated, various types of anodizing
Lubricant(s): not given Test temperature: maintained oil temperature at 80^{o} C

Other comments: Wear and friction not measured directly although there were observations on the appearance of the various surface treatments after exposure.

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[12] S. H. Hill, S. E. Hartfield-W∫nsch, and S. C. Tung *Tribology Trans*. (1996) Vol. **39** (4), 929-935.
Bench Wear Testing of Common Gasoline Engine Cylinder Bore Surface/Piston Ring Combinations
Ring/cylinder bore Level E – ring section on liner section (2 machines)
Test conditions for each machine:
(1) Reciprocating sliding, Cameron-Plint TE-77 machine, 80 N load, 10 Hz with 10 mm stroke, 40 hrs, SAE 5W-30 oil at 120^{o} C. Used undersized ring to avoid edge contact (Level D)
(2) Reciprocating sliding EMA-LS9 machine (similar to Sloan machine see Ref. [1]), load 72 N, speed 500 cycles/min with a 25 mm stroke length, SAE 5W-30 oil at 165^{o} C delivered at 1 ml/hr, test duration 30 hrs.

Materials/surface treatments: Rings: Cr-plated, Mo-plated, gas nitrided stainless steel; bores: cast iron, Al-Si C390 alloy, Nikasil plated liners

Correlation with actual components: not done
Other comments: Ring wear results varied more between the two machines than the bore segment wear. With ring wear, there was no apparent bias – that is some results were higher on one machine and others higher on the other. There seemed to be a better more consistent ranking of wear for the bore segments, even thought results were not the same quantitatively between the two machines.

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[13] V. D. N. Rao, D. M. Kabat, D. Yeager, and B. Lizotte Concepts in New Engine and Component Design, SAE Special Pub. SP-1225 (1997); also SAE Paper 970009,
Engine Studies of Solid Film Lubricant Coated Pistons
Piston skirt surfaces Levels D and C (with follow-up at level A)

Two testing machines:
1) Motored, single-cylinder test rig (level D)
Modified 4-stroke, using a 1.9L –CVH piston system motored rig with drop-in wet cylinder liners. Oil mist lubrication of the bore wall. Warm-up 1 hr, 1000 rpm, stabilize coolant at 80^{o}C and oil temperature at 65^o C, measure torque through the range 500-1500 rpm with 10 min increments, and then 1500-500 rpm. Durability tests = 20 hrs at 1500 rpm.
2) Fired engine in a dynamometer test cell (level C)
1992 model 1.9 l. CVH cast iron block, Al head, 5W30 oil. multi-stage break-in and running cycle.

Materials/surface treatments; various solid film lubricant coatings containing graphite, MoS_2, and BN.

Correlation with fired engines: Used machine 1 to screen coatings for machine 2. Those showing best results in the motored tests also performed well in the fired engine test. The leading film (epoxy based with MoS_2, graphite and hexagonal BN solid lubricant additives) was used in vehicle tests for up to 51,000 miles.

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[14] V. D. N. Rao, D. M. Kabat, R. Rose, D. Yeager, R. Brandt, and D. Y. Leong SAE Paper (1997) SAE 970008
Performance of Plasmaspray Coated Bore 4.6L-V8 Aluminum Block Engines in Dynamometer and Fleet vehicle Durability Tests
Cylinder bores Level of Simulation (C and A)
Break-in: 850 rpm idle (1 hr), 1500 rpm (2 hr), 2000 rpm wide open throttle (1) hr; followed by 100 hr test.

Materials/surface treatments: Aluminum cylinder bores and Al sleeves, plasma-sprayed coatings of iron oxide base and with BN additives

Correlation of results with actual components: Both test cell and fleet tests were conducted.
Comments: Complex combinations of test conditions made it impossible to directly compare results of test cell runs with fleet tests. The primary emphasis was in down-selecting coatings in the dynamometer tests and using fleet tests to verify results, albeit with different metrics.

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[15] V. D. N. Rao, D. M. Kabat, H. A. Cikanek, C. A. Fuinari, and G. Wuest SAE Paper (1997) SAE 970023 (in Applications for Aluminum in Vehicle Design, SAE Spec Pub-1251)
Material Systems for Cylinder Bore Applications – Plasma Spray Technology
Cylinder bore Level D
Test conditions (see Ref {13], Machine 1)
Materials/surface treatments; 14 different plasma sprayed coatings

Other comments or discussion: Results were also compared with scuffing tests using the LS9 (Sloan) machine. The LS-9 data were said to produce scuffing effects due to dominant boundary regime lubrication, but the motored rig was said to produce not only boundary lubrication, but because of its longer stroke, mixed film as well.

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[16] P. I. Lacey and R. T. Stockwell *Tribology Trans*actions, Vol. **42**(1), 192-201
Development of a Methodology to Predict Cylinder Liner Scuffing in the 6V92TA Engine Lubricant Test
Cylinder bore Level E – (with correlation to Level C)
Test conditions: Two laboratory tests were compared with engine results and lubricant volatility tests
1) 4-ball lubricant test – in accordance with ASTM D-4172 (for wear resistance of a lubricant) and ASTM D-2783 (load-carrying capacity of a lubricant). Wear resistance is given as average wear scar diameter on the 3 fixed balls.

2) Block-on-ring sliding wear - (as described in ASTM D-2714, "LFW-1" machine) Modified spindle speed for 1200 rpm (2.20 m/s), normal load 270 kgf, wear scar width used as a measure of wear.

Lubricants: 25 commercial crankcase lubricants were evaluate in type 6V92TA or 6V53T engines.

Other comments or discussion: Most commercial diesels operate with a top ring reversal point temperature of 180-200^{o} C. Typical steady state specific wear rates for liners should be in the range of 10^{-10} to 10^{-13} mm^3/N-**mm** (i.e., equivalent to 10^{-7} to 10^{-10} mm^3/N-**m**). Minimum calculated oil film thickness at 2300 rpm was ~ 0.5 µm. Correlation of basic LFW-1 results with scuffing on actual liner 6V92TA was R^2 = 54%. Inadequate to distinguish between best and worst oils. New test involves a parameter that combines LFW-1 results with lubricant boiling fraction to predict % scuffing observed on an engine test. Ring and block surface finishes are designed to simulate a run-in condition, 600 rpm (1.1 m/s), 85^{o} C, 20 min, 90, 120, and 150 kg contact load, SAE 01 block and SAE 4620 steel ring. This improved the correlation R^2 to 70%.

3.0 Conclusions and Recommendations

3.1 *Conclusions*

There is clear evidence that most of the engine cylinder and ring wear occur when an engine is started up from a cold condition, and at the location on the stroke where the lubricant film is least well-established, at the hot end of the cylinder (top-dead-center). Despite this observation, investigators usually attempt to simulate steady-state (hot oil) running conditions in laboratory-scale tests. It seems prudent then to design accelerated ring/cylinder friction and wear tests under conditions that simulate cold, start-up conditions. Furthermore, the condition of the oil also affects component wear and friction. Therefore, even though the test conditions might be run 'cold,' the oil condition should also be degraded, as it would be after exposure to hot running and with increased abrasivity due to the degradation of anti-wear additives and soot).

A review of the technical literature was conducted in an attempt to establish the most promising approaches to simulating the friction and wear of diesel engine rings and cylinder bore friction and wear using laboratory-scale test methods. Correlations between bench-scale tests and full sized engine tests were rare or the descriptions incomplete. Sometimes qualitative observations were used instead of quantitative friction and wear measurements.

Laboratory-scale studies of materials and lubricants for piston ring lubrication have been conducted for decades, but the conditions are extremely varied and there is no evidence in the literature to suggest that serious attempts at standardizing such tests were made. Standardization requires significant attention to specifying test conditions, understanding the variability inherent in the test method, and establishing formats for reporting the data that enable a useful comparison. The two commercial testing machines that appear most often in such studies are the Sloan LS-9 machine and the Plint TE-77. ORNL has led the development of ASTM standard test method G-133 that uses the Plint TE-77 under lubricated conditions in hot oil (ball-on-flat). Modification of that method to use ring and liner segments is therefore recommended.

Lacking close similarity in bench-testing procedures and methods, it is practically impossible to conduct a meaningful quantitative comparison of friction and wear data throughout the tribology literature. Among the most useful were studies that conducted several types of tests and compared their results. The most meaningful comparisons were possible only in terms of relative rankings or observations that some materials and surface treatments perform relatively well in laboratory tests and engine tests. Even then, the materials usually differed in preparation and surface condition.

3.2 *Recommendations*

Based on a literature review and consideration of issues raised in discussions with colleagues, there seems to be an opportunity to aid the diesel and automotive engine development community in the area of testing standardization and the use of degraded oils in laboratory tests.

(1) Develop a new standard test procedure based on the current ASTM G-133 reciprocating test, but that uses ring and liner segments with degraded (or simulated degraded) oil under repeated cold start conditions.

(2) Formulate a basic lubricant composition that produces the effects of degraded oil when using standardized testing procedures. This may require such approaches as ‘cooking the oil’ and adding abrasives to simulate the effects of carbon soot on the removal of anti-wear additives.

Appendix A

Factors Affecting Ring/Liner Tribology

Appendix A.

Factors Affecting Ring/Liner Tribology

[Summary based on: (1) M. G. Naylor, P. Kodali, and J. C. Wang (2001) "Diesel Engine Wear," Chap. 33 in Modern Tribology Handbook, ed. B. Bhushan, CRC Press. and (2) D. E. Richardson (2000) "Review of Power Cylinder Friction for Diesel Engines", *J. of Engineering for Gas Turbines and Power*, Vol. 122, pp. 506-519.]

Key: **C** = controllable variable, **E** = primary influence of this aspect

Aspect	**Design**	**Assembly and Fabric.**	**Operating Conditions**	**Lubrication Conditions**	**Materials in Contact**	**Description of Effect(s)**
bore size	**C**					minimal effect on ring/liner wear; contact pressures tend to remain similar for different sizes; friction power of rings and rods is proportional to the bore size
bore distortion		**C**		**E**	**E**	excessive bore distortion can increase ring/liner wear, also effects the lubricant film thickness and its ability to support pressure
stroke	**C**		**E**	**E**		affects piston travel velocity and lubrication regime; friction power of rings and rods is proportional to $(\text{stroke})^2$
connecting rod length	**C**		**E**	**E**		affects piston travel velocity and lubrication regime
connecting rod motion	**C**		**E**		**E**	rods constrained from moving along the axis of the engine with the piston rather than between crank throws; reduces friction
cylinder pressure	**C**		**C**	**E**	**E**	higher pressure produces more engine load and increased temperature which reduces oil viscosity and film thickness, and leads to more contact and more wear; high pressure can increase radial pressure of ring on liner if the ring width is relatively large (see "ring width" below); gas pressure can also affect the retention of lubricant between the ring and liner
piston skirt design	**C**	**C**		**E**	**E**	minimize skirt to bore contact to reduce friction
piston speed			**C**	**E**	**E**	higher piston speed translates to more contact cycles per unit time, but may also help reduce wear because it facilitates hydrodynamic lubrication; friction power of rings and rods proportional to $(\text{rpm})^2$, or as alternately derived, to $(\text{piston velocity})^{1.5}$
piston mass	**C**		**E**			less inertia and lower friction due to surface contact force reduction
piston surface treatments	**C**				**C/E**	select surface treatments to enhance lubricity and reduce wear; different approaches apply depending on the type of wear and the location on the piston
bore surface treatments	**C**		**E**	**E**	**C/E**	can reduce skirt and oil ring friction contributions in the lower end of the bore

liner finish	**C**	**C**		**E**	**E**	honing pattern and finish effects; discontinuous honing marks or ‘folded metal’ can increase wear
ring – width	**C**			**E**	**E**	affects pressure exerted radially on the liner; wider implies higher pressure on the wall; friction power loss is proportional to (ring width)$^{1/2}$
ring – profile	**C**		**E**	**E**	**E**	affects net gas pressure and lubrication regime, must be optimized for given engine to decrease wear; ring profile effects enter into calculations as adjustments to empirical constants; some advantages in friction using barrel or skewed barrel profiles; barrel profiles can also shift the position of the friction peak in the friction versus crank-angle plot
ring – cross-section	**C**			**E**	**E**	reduction in cross-section can reduce friction loss by reducing the peak friction force on the top ring
ring – tension	**C**					lower tension can make ring more conformable and reduce friction
ring - materials				**E**	**C/E**	can increase or decrease wear of the bore; can affect running-in; can affect conformity of sliding contact and oil film thickness
piston ring land clearance	**C**		**E**	**E**	**E**	affects flow of gases through the ring pack which affects ring forces and ultimately wear; too much clearance can allow carbon deposits to form and liner bore polishing to occur
piston ring groove clearance	**C**		**E**	**E**	**E**	large clearances may lead to ring breakage, but small clearances may lead to scuffing and sticking (see also “piston ring land clearance”)
number of cylinders	**C**				**E**	friction power (loss) is linearly proportional to the number of cylinders
temperature	**C**		**C**	**E**	**E**	affected by cooling system design and engine operating variables; affects oil viscosity and film thickness, affects lubricant stability/degradation; affects deposit formation; most of these effects lead to increased liner/ring wear; lower temperatures can lead to more sulphuric and sulphurous acid condensation and hence, corrosive attack, in higher S fuels
fuel – sulfiur content	**C**		**C**	**E**	**E**	high S can lead to acid formation and attack of liner surfaces; reducing S in fuels suppresses the corrosive attack; reducing S also reduces fuel lubricity which creates wear problems.
fuel – combustion			**C**	**E**	**E**	combustion products, like soot, can greatly influence wear and friction; fuel combustion products are a major contributor to engine deposits, with lesser contribution from lubricant (see also “deposits and soot”)
deposits / soot	**C**	**C**	**C**	**E**	**E**	deposits forming on different parts of the piston and liner have different compositions and effects; carbon deposits in particular can affect bore polishing, scuffing, and sticking; it has been proposed that soot can remove anti-wear lubricant additives, like ZDDP, and so increase wear

lubricant – composition			C/E	C/E	E	lubricant additives affect both friction and wear; exposure to operating conditions can alter composition which, in turn, changes the operating conditions; studied showed potential increases in frictional ring-pack loss of up to 40% due to oil degradation; polymer additives can benefit friction in certain lubrication regimes (hydrodynamic), but not much in boundary lubrication (see also “lubricant – regime”)
lubricant – cleanliness		C	C	E	E	presence of contaminants can cause abrasive wear; contaminants include soot, machining chips, sand from casting, road dust, wear particles, additive precipitates, fuel combustion products, ash (see also “filtration”)
lubricant – regime	C	C	C	E	E	boundary, mixed film, and hydrodynamic lubrication can occur at different parts of the piston stroke; surface separation, friction and wear depend on the operating regime
lubricant - viscosity			C	C/E	E	lubricant viscosity affects the flow and film formation of lubricants; too high and the lubricant cannot flow where it is needed; too low and the film will not support the bearing pressure causing solid/solid contact; affected by temperature, chemistry, and time of exposure to operating conditions
lubricant – windage	C		C/E			drag of crankshaft can affect parasitic losses; oil level in the crankcase has an effect on this
filtration	C		C	E	E	affects the ability to remove contaminants from lubricants and thus affects wear and viscosity
liner bore materials				E	C/E	affects the conformity of the components and the changes in surface finish with time; run-in ability affects longer term performance; wear particles can enter the lubricant; wear resistance increases engine life and durability

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