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Piston Ring Cylinder Liner Tribological Review

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Ground System Performance Fluids
Force Projection Technology

Table of Contents

List of Equations

List of Figures

List of Tables

1. **Introduction**

Piston rings were invented for the intended use in the steam engine in 1852 by British engineer John Ramsbottom[1]. Ramsbottom reported reduced friction, less blow-by between chambers, and dramatic improvements to efficiency and power. Shortly thereafter, piston rings were adopted for the internal combustion engine, and were found early in development to yield the same efficiency and power benefits. Piston rings have thus been an integral part in improving the practicality, reliability and efficiency of the internal combustion engine. A modern internal combustion engine is generally characterized as either two stroke or four stroke.

Premix two stroke engines traditionally have an upper and lower compression ring without an oil control ring[2]. In lieu of an oil sump, lubrication is achieved by premixing oil and fuel. The oil therefore is sacrificial, and must be continually supplied just as fuel, in a ratio determined by manufacturer's specification. The oil and fuel mix is then supplied with air, which travels through the crankcase before being introduced into the combustion chamber during the intake stroke. Oil is deposited on the inside of the crankcase, crank components, piston skirt and cylinder walls during this process, effectively distributing lubricant to the system. During a combustion event, the heavier non-combustible hydrocarbons of the lubricant often leave soot behind. Thus, this type of engine generally has higher concentrations of particulate matter and unburnt hydrocarbons exhausted from the system.

Four strokes traditionally have three rings. In addition to the two compression rings of a two stroke engine, four strokes require an oil control ring. This is due to the crankcase also operating as an oil sump. An oil pump pressurizes and distributes the oil to tribological contacting bodies including the cylinder wall / piston ring interface. The top compression ring affords the bulk of the pressure generated by the combustion stroke. This ring withstands the highest pressure and operating temperature. The middle ring, also known as the second compression ring, scraper ring, or intermediate ring, provides some combustion pressure sealing, as well as oil management. The intermediate ring is sometimes tapered to encourage scraping of excess oil on the downward stroke such that only the necessary amount of oil film remains for the compression ring. The third ring, also known as the oil control ring, is normally a combination of two rails and an expander. The rails operate in boundary lubrication, managing the amount of oil at the piston / cylinder wall interface. The rails are held under tension and the separation is controlled via the expander.

2. **Project Background**

In 2014, Federal Mogul reported that in heavy duty diesel engine applications, 10% of fuel energy is wasted on mechanical losses attributed solely to friction[3]. Of these losses, a 25% contribution is from ring pack and cylinder friction. Thus, reducing these frictional losses can lead to improved engine efficiency, less fuel consumption and emissions. The scope of the report herein focuses on strategies to reduce piston ring frictional losses in heavy duty diesel, four stroke engines with a three ring pack, and methods to quantitatively characterize performance using benchtop tribological apparatuses.

Achieving improved fuel economy through the piston ring pack can be approached in several different hardware tactics such as proper piston ring tension and piston ring design. Proper ring tension can ensure minimal blow-by, whilst reducing the amount of force being applied to the cylinder walls, reducing the frictional force opposing reciprocating motion of the piston. Ring design can also aid in controlling this force as well as work to control the lubrication condition and entrainment of lubricant at the piston ring / cylinder wall interface. Hardware material selection is also critical in this system. Typically, compression rings and oil control rails are coated on the periphery with a wear resistant coating. Low friction, wear resilient coatings are currently being researched, as well as surface texturing to improve both wear and friction. Optimized lubricants for the specific operating conditions and hardware components can also greatly affect efficiency. Lower viscosity fluids can be pumped using less energy, and can be sheared easier. However, a lower viscosity fluid may not be sufficient to afford the severe operating conditions in boundary lubrication unless properly formulated. Thus, lubricant changes must be carefully monitored to ensure an optimal solution is selected and friction performance enhancement is not at the cost of hardware durability issues.

Though ring tension is critical in preventing blow-by in internal combustion engines, the oil control ring operates in boundary lubrication which can exhibit as much as twice the amount of friction of the top compression ring over a full cycle, the focus of this work will be in regards to the top compression ring. When a combustion event occurs, at the higher combustion pressures of diesel engines, the operating conditions of the top compression ring are the most severe and are under highly transient conditions when compared to the oil control ring. Thus, the friction and wear of the top compression ring are the highest of any ring under these short durations near top dead center (TDC). A tribological approach is necessary due to the paramount concern of both friction and wear performance.

The top compression ring / cylinder wall interface can be approached from a fundamental statics force balance. Figure 1 illustrates the position of a piston ring seated in a piston groove under combustion chamber pressure (Pc)[4]. The force exerted on the ring downward is due to the combustion chamber pressure multiplied by the surface area of the top of the ring. This force is opposed by the piston structure, the frictional force between the ring and cylinder wall, and a small portion of inner-ring pressure (Pir) multiplied by its limited surface area. The engine is reliant on proper seating of the compression ring in the piston groove to prevent blow-by. The propensity for blow-by is greatest when the combustion chamber gas pressure is at a maximum if not for the sealing force between the piston ring and piston structure, which is directly proportional to chamber pressure. Theoretically, if the frictional force opposes this sealing force, an engine could be less prone to blow-by by reducing the frictional force in this system. Inner-ring pressure also combats sealing, thus reducing either the pressure or area in which it acts could reduce blow-by. To understand the frictional force between the piston ring and cylinder wall, an understanding of the pressure exerted on the back of the ring exerting a force towards the cylinder wall must be considered. This force is opposed by a viscosity-pressure response from the lubricant, and a mechanical response from the cylinder wall structure. Depending on the geometry of the ring, the pressure response (*p*), will differ. The frictional force is a function of the normal force, ring tension, combustion chamber pressure, the shear-ability of the lubricant, and the surface to surface interaction between piston ring and cylinder wall. Thus, combustion chamber pressure works to seal and help prevent gas blow-by, while the friction force counters sealing. Greater combustion chamber pressure relates to greater frictional force. Under properly lubricated conditions, and in a properly functioning engine, the frictional force can be lessened. The disadvantage of a static force balance is that it does not take into account the highly transient conditions undergone by the system. It is useful to look at one moment when conditions are extreme, but the magnitude of the pressures and piston velocity will vary greatly with time or crank angle. Statics also lack mechanical dynamics, for example ring flex of a floating piston ring.

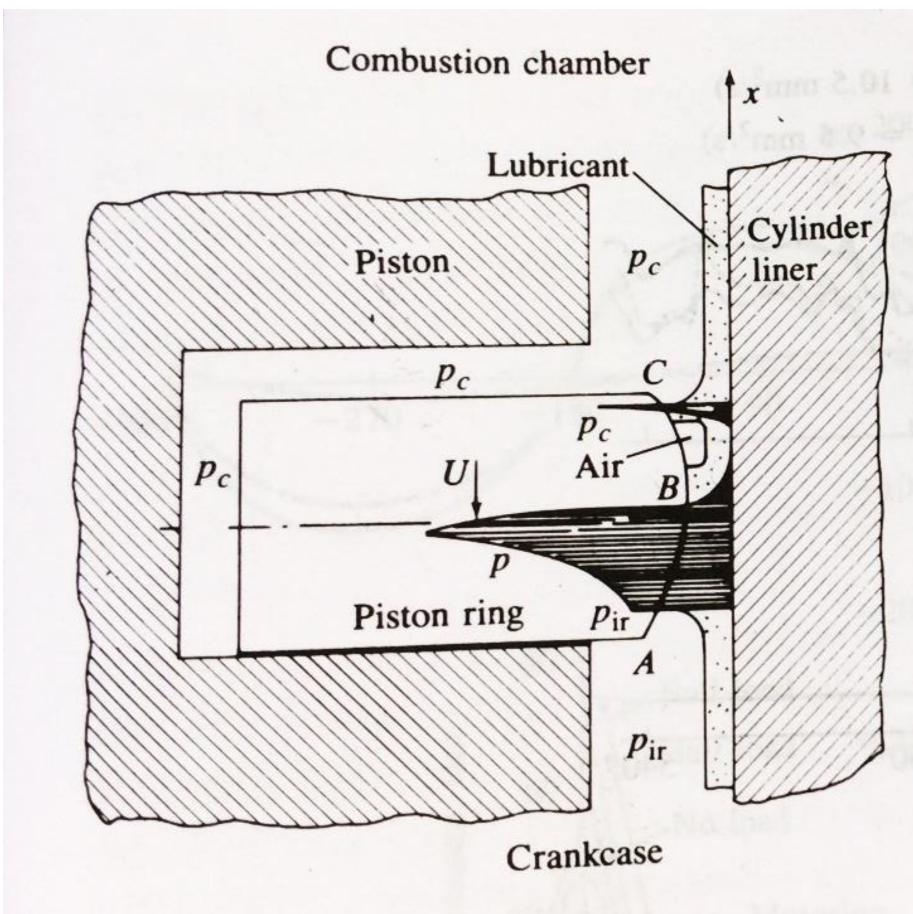

Figure 1: Top Compression Ring Cylinder Liner with Lubricant Film Schematic[4]

Tribological efforts can be employed to reduce the frictional force in this system. Tribology works to understand the lubricant interaction with the hardware/system, and accounts for operating conditions such as speed, load, and temperature of the system, as well as the material properties and design. Material surface interactions, as in the case of a piston ring and cylinder wall couple, can cause wear. Careful consideration of proper selection of wear couple materials will ensure proper mating of surfaces during break-in and sustainable wear over a system's lifetime. The lubricant is also an important component in the efficiency of the coupled materials. Antiwear additives contained in the oil must be compatible with coupled materials to generate critical antiwear lubricant films, preventing direct metal on metal contact. If the lubricant film is compromised, the antiwear film (i.e., tribofilm) is the last line of defense. Direct metal on metal contact can cause irreversible component wear, altering precise machining tolerances, and disrupting the performance of the system. If not controlled, this behavior can lead to scuffing, seizure, and failure.

The friction performance of a tribological system can be generalized using the Stribeck Curve (Figure 2)[5]. The Stribeck Curve is instrumental to characterize the lubrication regime of a tribological system. There are four different lubrication regimes; boundary, mixed, elasto-hydrodynamic, and hydrodynamic lubrication. Boundary lubrication operates in intimate surface to surface contact, with some lubricant support. Large amounts of wear and friction will occur during boundary lubrication and should be carefully monitored. Increasing the viscosity of the lubricant, velocity of the system, or reducing the load can help improve the lubricating condition. Fluid viscosity is inversely proportional to temperature, thus if higher viscosity is desired, lowering the operating temperature could be an option. However, frictional heating can locally reduce lubricant viscosity, which can cascade to degrade lubrication conditions, leading to more friction and wear. Mixed lubrication offers more lubricant support, but still has some surface to surface interaction of nanoscale asperities. The friction correspondingly is lower for this case as the lubricant can generally shear with less force than the force required to abrade, adhere and shear hard asperities. In elasto-hydrodynamic lubrication, lubricant separates the surfaces, however, significant elastic deformation occurs in the surfaces due to the applied load. Minimal wear occurs in elasto-hydrodynamic lubrication, but it is reliant on the material properties of the surfaces, such as the elastic modulus and hardness as fatigue wear issues can arise. Hydrodynamic lubrication refers to minimal elastic deformation of the surfaces, where the load is afforded predominantly by the lubricant film. Negligible wear occurs in these conditions and is preferred. However, the friction force due to shearing of fluid can be significant and, depending on how shear stable the lubricant is, can degrade the lubricant and affect its performance.

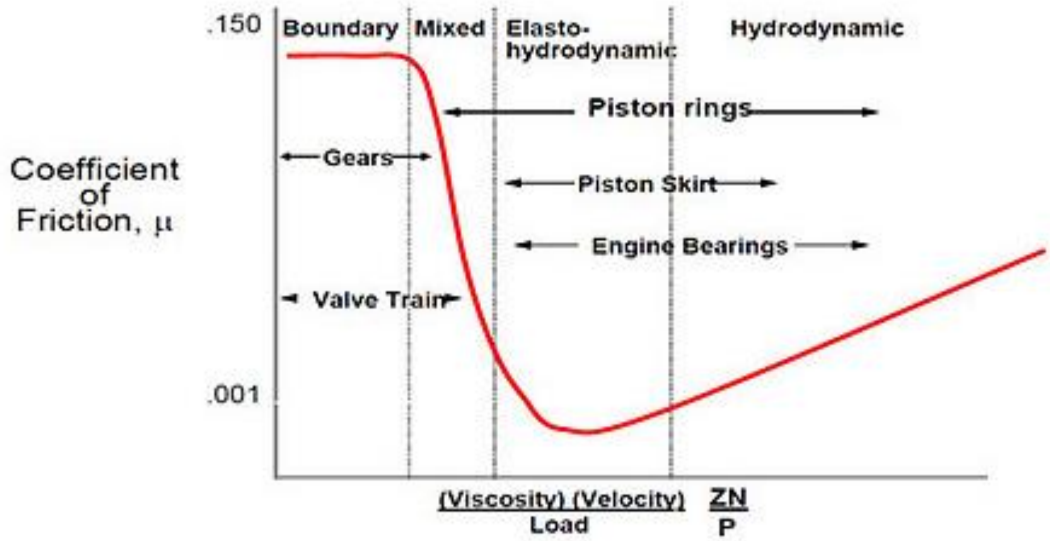

Figure 2: The Stribeck Curve[5]

By applying the Stribeck lubrication theory, the top compression ring leads to several areas of tribological interest. When the piston travels to TDC during the compression stroke, its velocity drops to zero before the piston changes direction on the down stroke expansion. The reciprocal nature of this process leads to changes in lubrication. Zero velocity also occurs at bottom dead center (BDC), however, BDC does not have the extreme pressures or temperatures as TDC during the transition from compression to expansion. Thus, the lubrication condition at TDC is most critical to control from a tribological perspective. Additionally, the piston ring / cylinder wall lubrication system will not directly switch to hydrodynamic after the change in direction. There will be a finite amount of time where it will operate in mixed and elasto-hydrodynamic lubrication.

Boundary and mixed lubrication can lead to significant mechanical wear modes. Specifically adhesive and abrasive wear can be attributed to these lubrication regimes. In a properly functioning engine, typically only these two wear modes will be prevalent when analyzing the surfaces of the piston ring and cylinder wall. It is not uncommon for step wear banding in cylinder liners near TDC and BDC due to these mechanical wear modes. Corrosion can also occur if there is local acidity from the combustion of fuel with high sulfur content, although this phenomena is normally addressed through corrosion inhibitor additives in the lubricant.

Adhesive wear, shown in Figure 3, occurs due to localized asperity contact between counter-faces[6]. The localized frictional heating and extreme pressure can cause micro-welding, and brittle fracture of the softer material. Material can be transferred from the softer material to the harder material, and can also create third body abrasive particulate as well. If both surfaces are equally hard, micro-welding can occur, and the amount of force needed to shear the weld is greater, causing higher friction and frictional heating. The transfer of material is uncontrolled, and unpredictable in nature. Thus, proper wear couple selection is needed to ensure controlled and minimized adhesive wear in boundary and mixed lubrication.

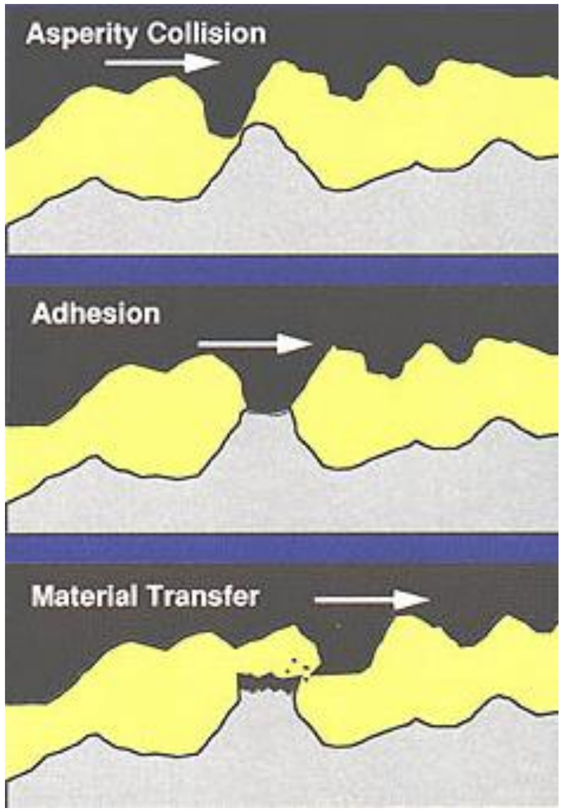

Figure 3: Adhesive Wear Process[6]

Similarly, abrasive wear acts predominately in boundary and mixed lubrication regimes as illustrated in Figure 4[6]. Hard asperities can plastically deform softer materials, displacing material at the leading edge or to either side. The plastic flow of material can cause third body abrasive particulate. Generation of third body abrasives can be correlated to the plastic flow-ability of the material, or lack thereof. Brittle materials have a higher propensity to generate third body abrasives due to the limited conditions of plastic deformation without complete failure. Markings along the sliding direction are the signature of abrasive wear. Worn/used piston rings will generally have abrasive wear markings parallel to the width of the ring along the sliding direction which can be seen with the naked eye. Surface topography analysis of the surface can quantify the wear occurred over the part lifetime.

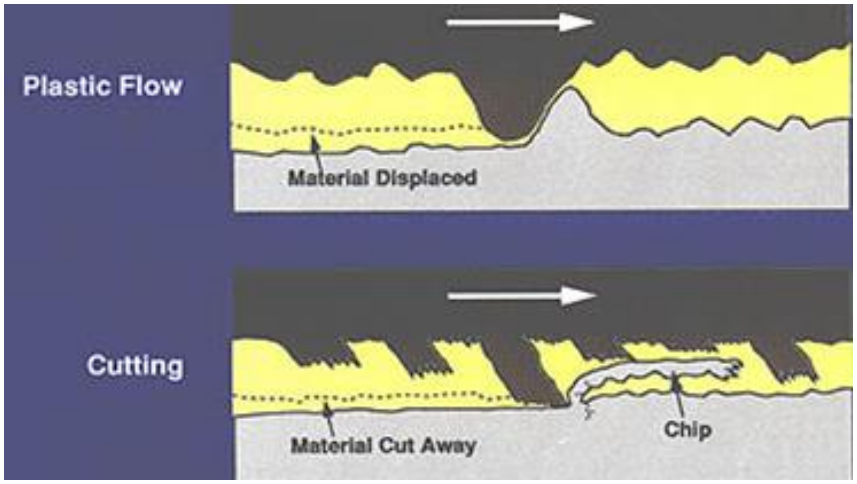

Figure 4: Abrasive Wear Mechanisms[6]

Third body abrasive wear as shown in Figure 5, behaves similarly to that of abrasive wear[6]. Unique to this case is the presence of a hard particle either generated from the mechanical wear of the part, or entrained from the lubricant via alternative processes. In heavy duty diesel engines there is a higher chance for particulate matter and soot generation, which can act as third body abrasives if entrained by the lubricant. Thus, there is a higher probability for heavy duty diesel engines to have issues with third body abrasive particulate wear on the piston rings and cylinder wall. The engine oil and filter can be analyzed to quantify particulate matter generated. Size, geometry, mechanical properties such as hardness, chemical composition amongst others can have an effect on how an abrasive particle interacts in a lubricated condition.

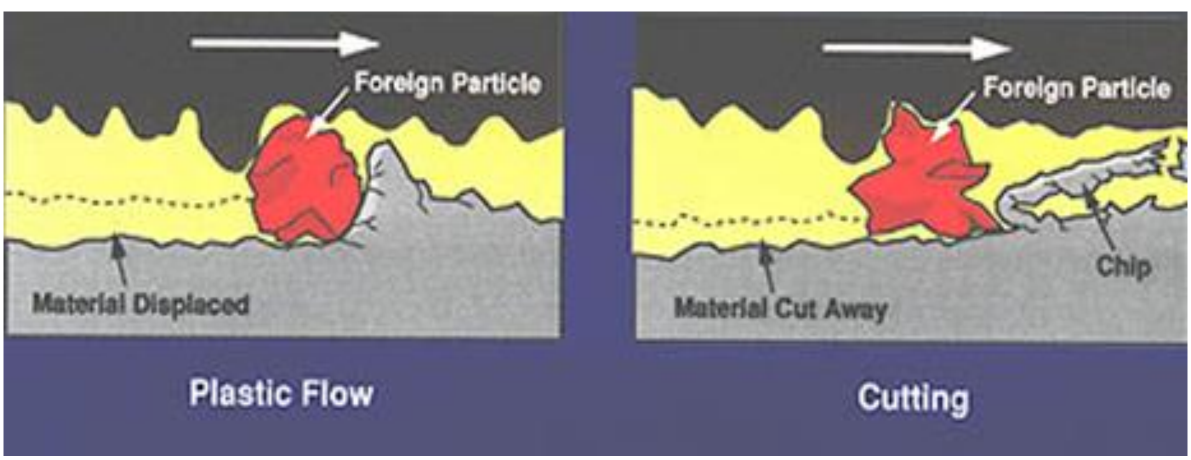

Figure 5: Third Body Abrasive Wear Mechanisms[6]

Wear volumes and wear rates occurring via mechanical wear modes such as abrasive and adhesive wear in boundary and mixed lubrication regimes are quantified by Archard's Wear Equation (Eq. 1)[7]. Wear volume is normalized by the sliding length, but is a function of the load and hardness of the softer material multiplied by a wear coefficient. The wear coefficient (*K*) can be empirically quantified to predict wear rates in other systems, where *V* is the wear volume, *L* is the sliding length, *P* is the load, and *H* is the hardness of the softer metal.

$$\frac{V}{L} = K \times \frac{P}{H} \qquad \text{(Eq. 1)}$$

To counter wear, antiwear lubrication additives work to generate a nanoscale protective film on material surfaces called tribofilms. Lubricant base stocks alone do not have an inherent ability to generate robust tribofilms, instead they are reliant on antiwear additives. Antiwear additives, having a chemical affinity for the surfaces, undergo chemical processes under temperature, load and shear to form protective tribofilms. Without antiwear additives, lubrication films can be generated from the degradation of the base stock, but will not exhibit the material properties desired to prevent wear. Tribofilms, therefore, work to prevent direct metal on metal contact, mitigate wear and failure modes, and act as a sacrificial boundary when the lubrication film is compromised. The most effective antiwear additive used over the past several decades is zinc dialkyl dithiophosphate (ZDDP) and a representative tribofilm is shown in Figure 6[8]. The zinc and phosphorous content in the compound generate a pseudo-glass on the surface. Stress augmented diffusion can occur between the iron substrate media and the zinc phosphate glass, creating a tribofilm with desirable mechanical properties to prevent wear. The composition of the film varies as a function of depth due to a stress augmented diffusion mechanism, with the highest percentages of iron near substrate.

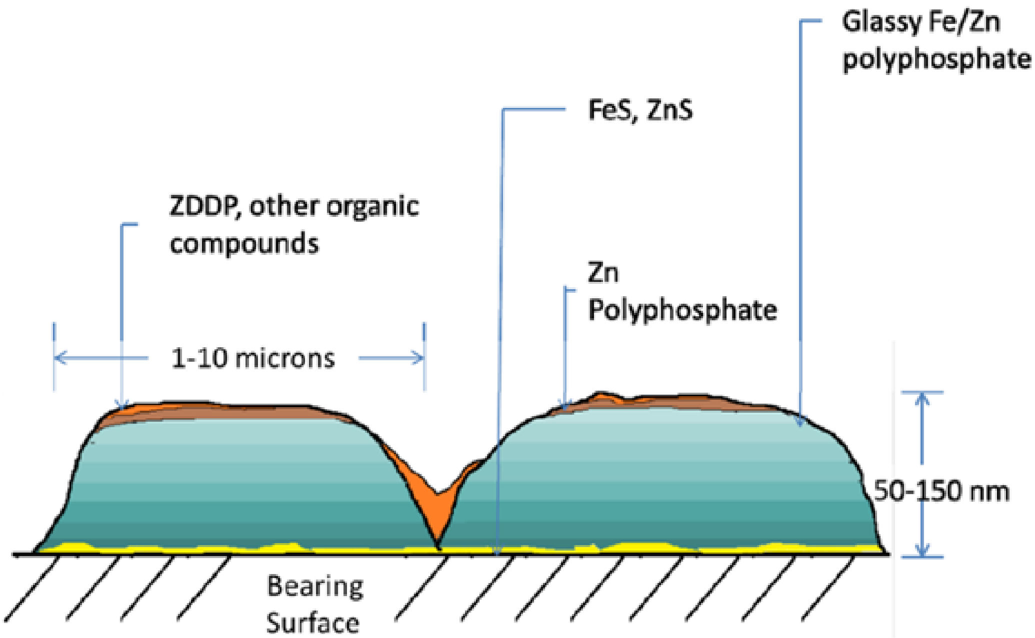

Figure 6: Tribofilm Composition Diagram[8]

3. **Approach**

The top compression ring in a heavy duty diesel engine application was selected due to the higher operating pressures and compression ratios causing larger backpressure on the top compression ring. Due to the highly transient operating conditions of the ring and liner, friction and wear were taken into account in understanding the complex tribological system.

The relation of engine friction and wear data to that of a benchtop apparatus can aid in screening lubricants, wear couples, material selection, and design changes when done correctly. Benchtop studies can be used to make empirical correlations. Selection of proper operating conditions, such as oil supply rate, temperature, load, and reciprocating frequency can greatly affect the correlation. The simplest of benchtop tribological instrument for ring and liner testing consist of a ball on flat high frequency reciprocation setup. These instruments operate in pure sliding. The area of contact is single point; therefore, the effective pressure applied is highly concentrated and changes drastically over the course of a test depending on how much wear occurs. Slightly more feasible are block on ring and cylinder on flat reciprocating motion to simulate line contact under pure sliding. However, there are a small subset of tribometers capable of affixing a segment of a ring opposed to a cylinder liner, such as the Schwingung (oscillating), Reibung (friction), and Verschleisse (wear) (SRV) tribometer. The SRV (Figure 7) will give an area of contact closer to that of application[9]. A literature review was conducted on the piston ring / cylinder liner contact tribological testing conducted in recent years, which is subsequently discussed.

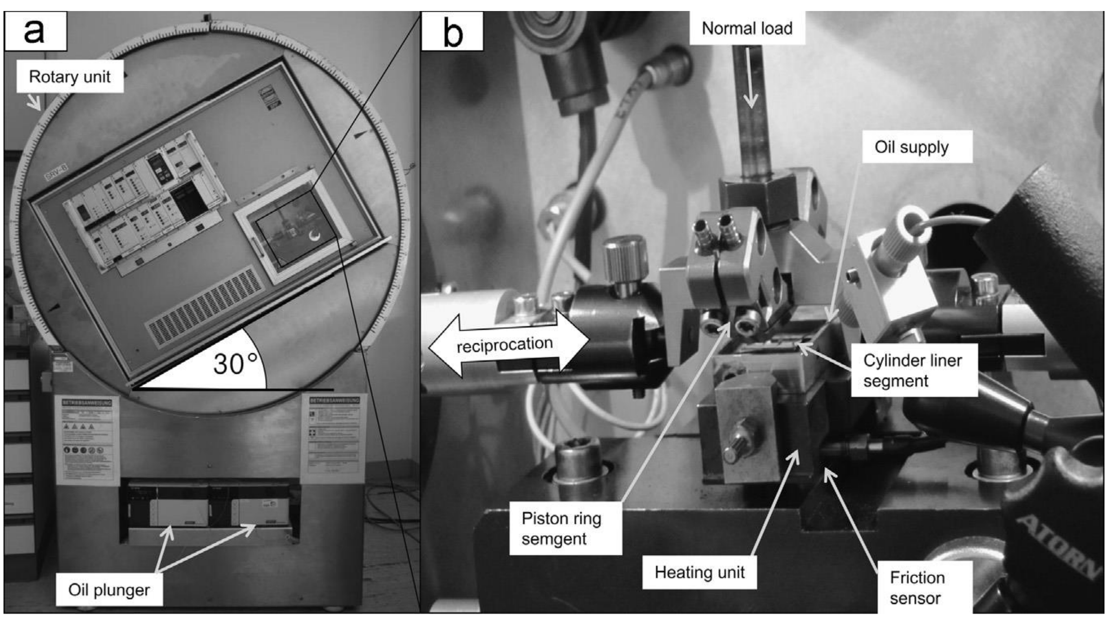

Figure 7: High Frequency Reciprocating Piston Ring on Liner Tribological Apparatus[9]

4. **Literature Review**

4.1. Study One: The influence of oil supply and cylinder liner temperature on friction, wear and scuffing behavior of piston ring cylinder liner contacts – A new model test[9]

The first study of interest investigated oil supply and cylinder liner temperature in an SRV piston ring on cylinder liner tribometer. Material selections were made based on typical materials used in light duty diesel passenger cars. The test materials were chromium plated embedded with aluminum oxide piston rings opposed to gray cast iron cylinder liners segmented from production hardware. Segmented hardware specimens were changed and fresh oil was used for each test.

Each test run consisted of bringing the system up to the desired temperature based on the test. No load or relative motion was applied during this initial equilibration step, as a cold start could cause undesired wear or scuffing. Oil was then supplied at a rate such that the area of contact was flooded, and excess would be expelled from the system to ensure a properly lubricated condition at start of test. The oil supply rate was gradually reduced to the desired rate over a 12 minute time period. A normal load was then applied and the piston ring segment was allowed time to reach the test temperature before a stroke of 3 millimeters (mm) at a frequency of 20 hertz (Hz) was initiated. The maximum sliding speed for this scenario given the operating conditions would equate to 0.18 m/s. This speed is adequate for mimicking the desired TDC or BDC conditions where the piston slows, stops, and starts, but is over an order of magnitude slower than needed for mid-stroke conditions.

Oil supply rate and liner temperature can both have a large effect on load carrying capacity (LCC). The definition of LCC can vary based on the application. In this case, LCC is intended to describe the load for which, if a tribological system exceeds, failure is imminent. Failure modes could be attributed to loss of lubrication, excessive wear, scuffing and seizure. For this investigation, the load was set and, if a failure was not present, new segmented specimens were used and the test load was increased in 50 Newton (N) increments. Once a failure was found, the load would be reduced until three passes occurred in a row. That was deemed as the LCC of the system.

Figure 8 details an LCC map dependent on liner temperature and oil supply rate[9]. The maximum liner temperature of 230 °C was set based on the highest temperature observed in a previous motored engine study of a light-weight diesel engine. A loss of lubricant scenario (Oil supply rate of zero micro liters per minute) yields a very low LCC. At the start of test, the contact area is initially flooded, but without continuous oil supply, oil will be squeezed/pushed out of the area of contact and LCC suffers greatly. Liner temperatures of 230 °C produced similar LCCs. Temperatures that high can cause coking and degrade the oil, leaving residual hydrocarbons and poor tribological conditions. A sharp increase in LCC was noted for a 10 °C reduction in liner temperature off of the maximum tested. Yet to be answered is the ability to extend the temperature range with the use of an oil with a higher temperature. Oil supply rate has a similar trend, as rates of at least 0.1 microliters per minute offer much superior LCC.

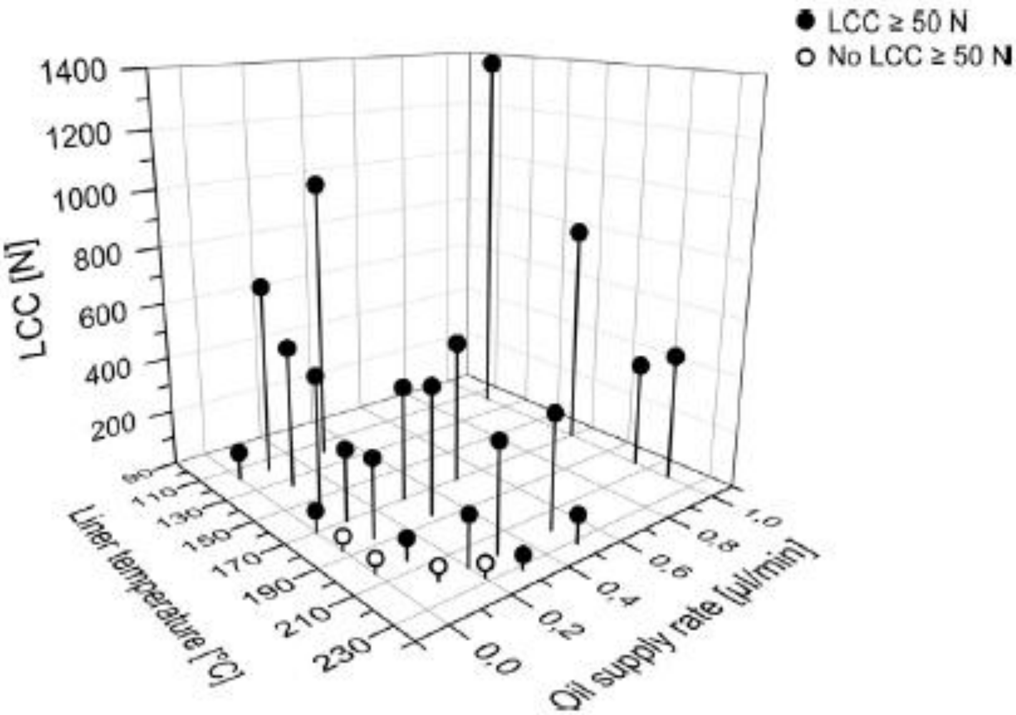

Figure 8: LCC map and scuffing limit for the 0W30 engine oil A dependent from liner temperature and oil supply rate[9]

The LCC study dependent on liner temperature was repeated for 3 different oils shown in Figure 9[9]. Two 5W-30 oils were selected and one 0W-20. A constant oil supply rate of 0.1 microliters per minute was pumped into the contact and the scuffing limit was set to 100 N. The 0W-30 and two 5W-30s performed similarly across the full range of temperatures, whereas the 0W-20 offered slightly less LCC. Viscosity has a direct effect on lubrication condition according to Stribeck lubrication theory. Thus, the results from the selection of a lower viscosity grade 0W-20 oil would offer slightly less LCC performance when compared to 30 grades. All seem to follow linear trends with large experimental variability. The advantage of using a lower viscosity oil would be in mid-stroke under hydrodynamic lubrication as less shear force would be required to work the fluid.

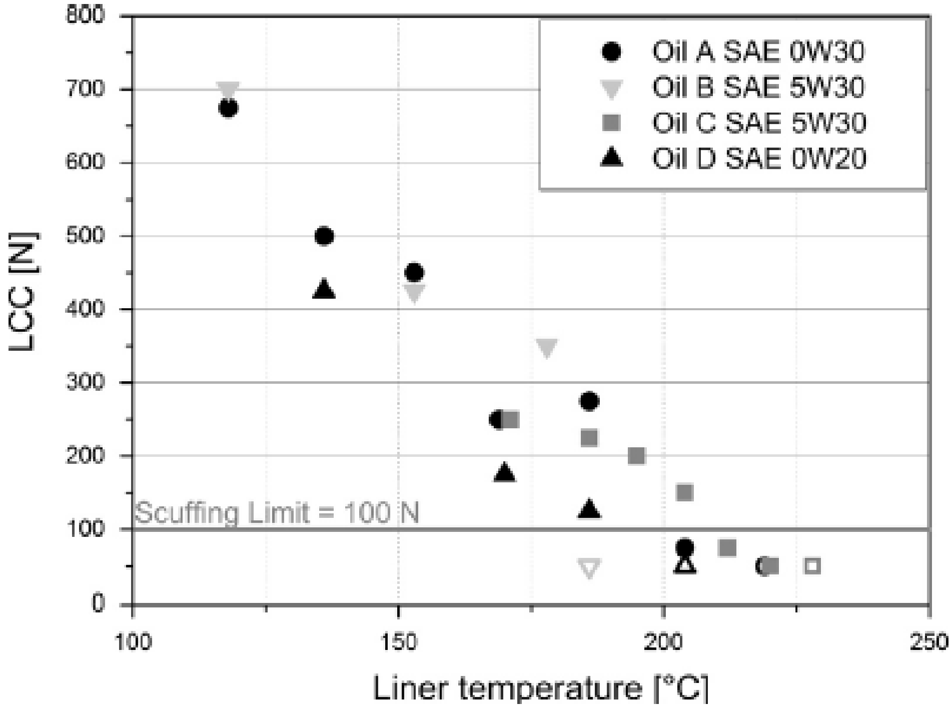

Figure 9: Load carrying capacity of fully formulated engine oils[9]

4.2. Study Two: Syntheses and Tribological Property of CrMoN/MoS2 Multilayer Films on Piston Rings of Heavy Vehicle Engine[10]

The second study investigated an experimental multilayer piston ring containing chromium molybdenum and nitrogen layer with a molybdenum disulfide layer. Molybdenum disulfide is a commonly used solid lubricant, having excellent tribological properties in compression and shear due to its lamellar structure. Chromium and molybdenum have been historically used as wear resistant coatings for piston rings. Piston rings have undergone processes called nitriding to boost hardness and wear resistance as well. The combination in a multilayered approach increases the risk of surface to surface delamination if not properly applied and adhered. Production costs are also greatly increased with such multilayered strategies, but can yield beneficial tribological performance. The chromium, molybdenum, nitrogen coating was applied via magnetron sputtering at a controlled film thickness of 5 micrometers to piston rings. Amorphous molybdenum disulfide was then applied to the 5 micron thick film. The coated piston rings were segmented for tribological testing, unmodified cylinder liners were used as counter-faces.

All tribological tests conducted in this study used an SRV piston ring on cylinder liner benchtop tribometer. The load was cycled between 290 and 400 N with 120 second (s) time intervals. Tests were run at a frequency of 15 Hz at 100 °C with a 4 mm stroke for 7200 s. The resulting frictional trace for an unmodified chromium piston ring and a multilayer piston ring are shown below in Figure 10[10]. For the unmodified piston ring, a steady state friction was observed until ~2400 s. Large fluctuations from a baseline friction trace allude to breakdown of a lubricant film, severe wear, or an early indication of a precursor to scuffing and failure. The fiction trace would continue to fluctuate until scuffing and failure shortly after 4000 s. Under the same conditions, the multilayer piston ring performed consistent throughout the full duration of test, with a slight upward trend, but not alarming. A step function can be noted at the two different loads as well for the case of the multilayer ring.

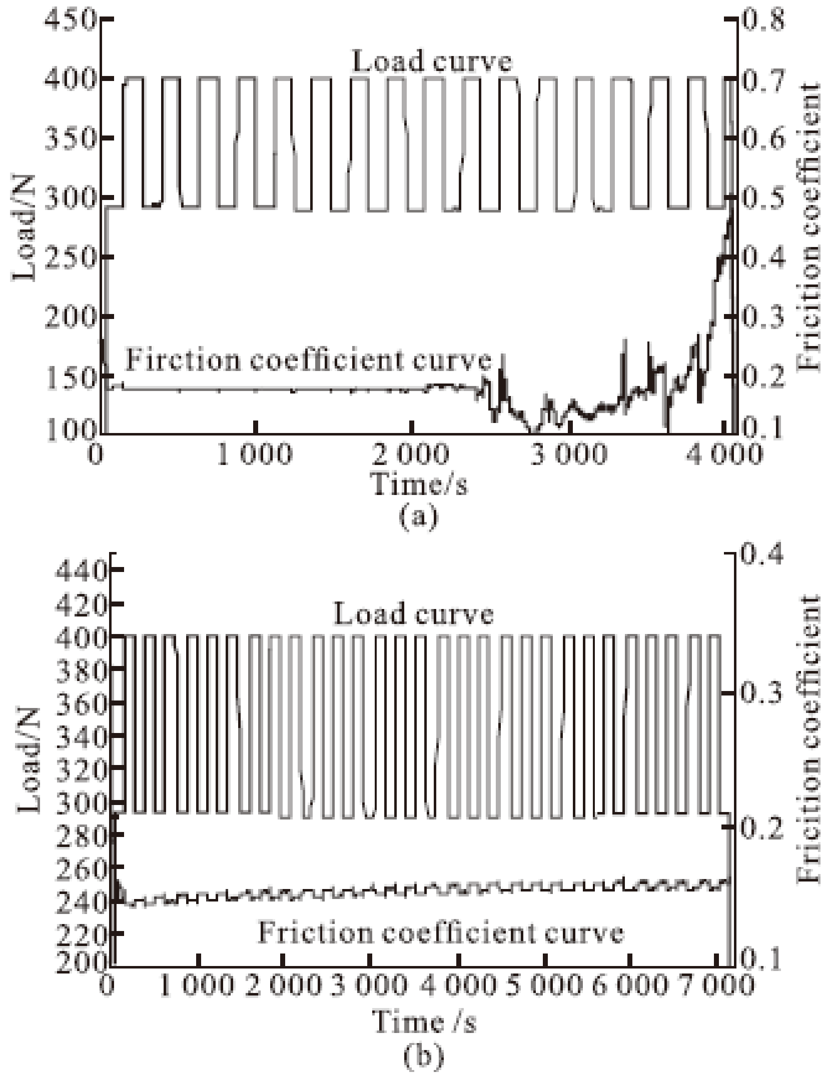

Figure 10: Friction coefficient curve of different piston ring-cylinder liner friction pairs under dynamic load (a) Cr plated piston ring-cylinder liner friction pair; (b) CrMoN/MoS2 piston ring-cylinder liner friction pair[10]

The segmented piston ring and liner test specimens were weighed before and after testing to determine a wear rate over the course of the test shown in Figure 11[10]. The multilayer piston ring and liner test exhibited a wear rate of roughly 33% that of the production chromium piston ring. However, due to the short duration and scuffing/failure of the chromium ring cylinder liner test, the wear rate would not be expected to be linear throughout the test duration. The amount of wear during a scuffing/failure event is normally substantially large and could skew the data. A fair contrast would be to run the chromium piston ring cylinder liner test for ~2000 s, before failure/scuffing, for a more accurate wear rate.

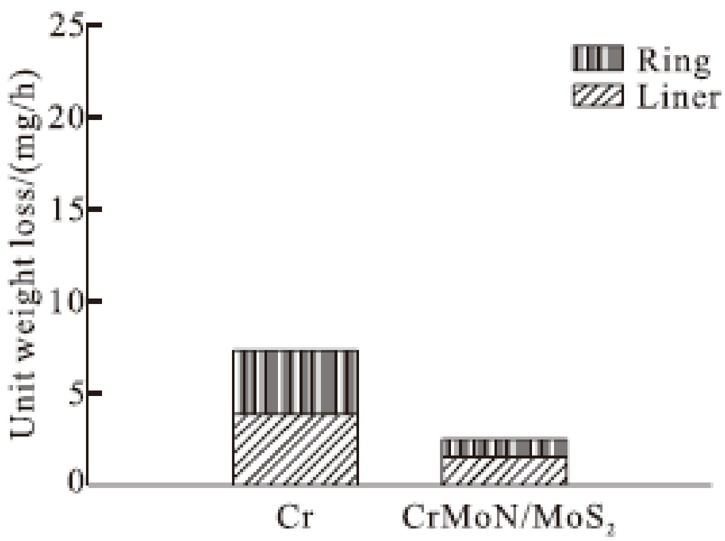

Figure 11: Unit weight loss of different piston ring-cylinder liner friction pairs under dynamic load[10]

Figure 12 shows the resultant wear on both cylinder liner and piston ring for the tests via scanning electron microscopy (SEM)[10]. The chromium plated piston ring, which scuffed (failed), showed large amounts of adhesive wear with cracking of the chromium layer. Abrasive wear is also present in areas where adhesion was not present. Severe adhesive wear and plastic deformation of material is indicative of microwelding and tearing of metal, which leads to large amounts of friction and wear. Tearing of microwelds can result in cracks evident in the figure as well. A shorter duration test should be run, before scuffing/failure, to understand the magnitude of adhesive wear present during a steady coefficient of friction. As presented, Figure 10 showed failure of test, and Figures 11 and 12 corroborate that information. The cylinder liner in Figure 12 was observed to be in good condition, though some material was observed to be removed via adhesion and abrasive wear in the sliding direction, occurring either from third body abrasive wear generated through the duration of the test, or two body abrasiveness of the piston ring. The multilayer piston ring and liner exhibited less extreme wear, though some abrasive wear can be observed along the sliding direction with minimal adhesive wear.

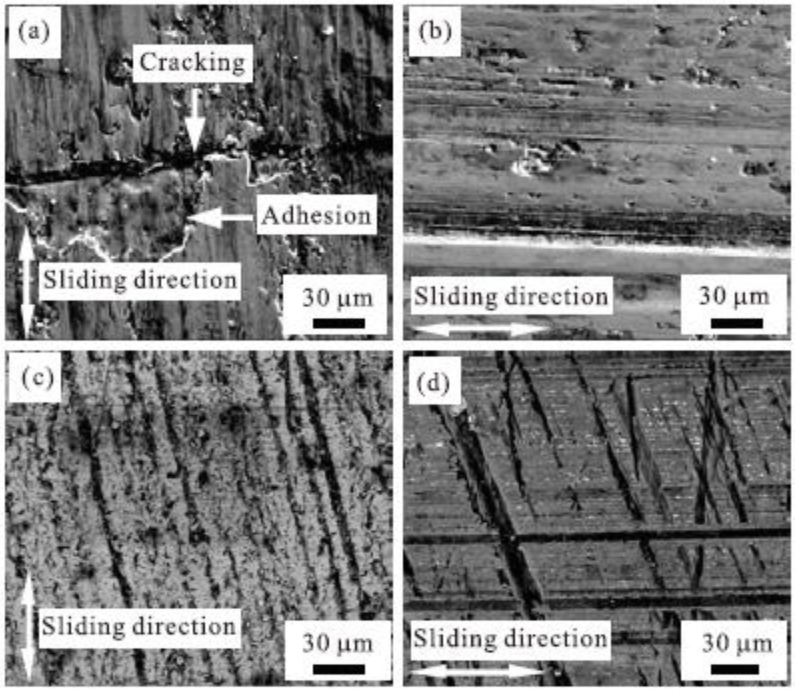

Figure 12: SEM morphologies and EDS analysis of wear scars of different friction pairs: (a) Cr plated piston ring; (b) cylinder liner; (c) CrMoN/MoS2 piston ring; (d) cylinder liner[10]

4.3. Study Three: The impact of microstructural alterations at spray coated cylinder running surfaces of diesel engines – Findings from motor and laboratory benchmark tests[11]

The third study investigated a variety of piston ring and liner wear couples in an SRV tribometer. Cylinder liners were coated via twin wire arc sprayed (TWAS) and plasma transfer wire arc (PTWA) methods. Cylinder liners were coated with a nickel silicon carbon composite film, TWAS with a carbon containing iron feedstock, FeC0.9, PTWA of a carbon containing iron feedstock, FeC0.3, and a PTWA composite feedstock containing iron, chromium, carbon and boron FeCr9C0.4B2. The numerical values following the element represent intended percent by weight concentration in the feedstock material to be coated on the cylinder liner. Piston ring materials included nitrided steel and chromium ceramic (CKS). Film thickness and morphology were verified via microscopy and transmission electron microscopy (TEM).

A load of 200 N, frequency of 40 Hz, stroke of 4 mm and a temperature of 190 °C was applied to each couple. Average coefficients of friction and wear rates are shown below in Table 1[11]. Wear rates were determined via 3D confocal white light interferometry. A coefficient of friction around 0.1 for boundary lubrication is a typical value. Lower than 0.1 would be observed as being beneficial, having less energy lost to friction. Low friction is not always accompanied by low wear values, so both must be taken into account when determining the best tribological pairing. The nickel silicon carbide cylinder liner against a nitrided steel piston ring yielded average results, having a coefficient of friction at 0.10 and an intermediate wear rate. However, changing the piston ring to ceramic chromium exacerbated the friction and wear, performing worse in both parameters. Carbon containing iron TWAS coated cylinder liner opposed to a nitrided steel piston ring showed greatly reduced wear with some frictional performance enhancement as well. This effect was furthered by changing the piston ring material to ceramic chromium. Thus, an antagonistic relationship was observed for CKS in the case of a nickel silicon carbide liner, whereas a synergistic relationship was observed for carbon containing iron TWAS. Finally, a carbon containing iron PTWA coating cylinder liner opposed to a nitrided steel piston ring yielded inferior results when compared to TWAS.

Table 1: Average coefficient of friction and calculated linear wear rates[11]

Tribological pairing	SRV© test	
	COF μ_{SRV}	Wear rate Wl/t [nm/h]
NiSiC vs. steel nitrided	0.10	169.7
NiSiC vs. CKS	0.15	271.4
TWAS FeC0.9 vs. steel nitrided	0.08	55.4
TWAS FeC0.9 vs. CKS	0.05	32.7
PTWA FeC 0.3 vs. steel nitrided	0.13	211.2
PTWA FeC0.3 vs. CKS	–	–
PTWA FeCr9C0.4B2	–	45.2
PTWA FeCr9C0.4B2	–	72.1

Near and subsurface microscopy techniques were employed to elucidate the underlying mechanisms for wear of the PTWA coated piston rings to evaluate performance observed to be below that of expected. Figure 13a shows a 50 nanometer layer thickness of ferrite with a very small grain size of less than 20 nanometers[11]. The presence of ferrite and cementite was observed (Figure 13b) as well as, numerous areas of dislocations were formed which allude to the higher than expected wear rates of the material. The subsurface zone also showed ferrite in a different crystallinity and orientation. The total combination of oxide layers had a thickness ranging from 200 to 500 nanometers.

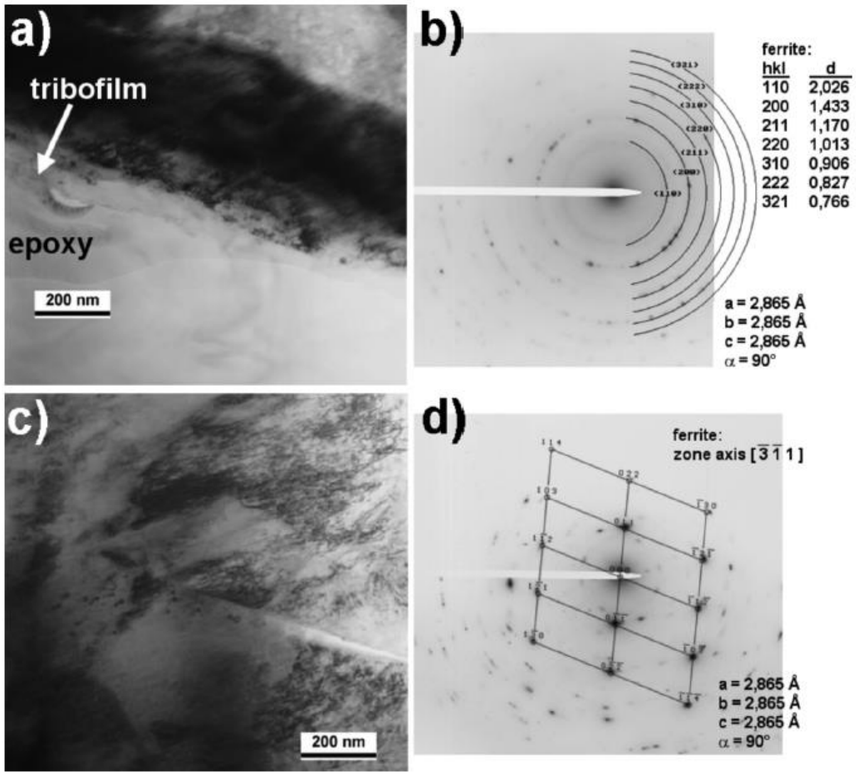

Figure 13: TEM micrographs of the surface (a) and subsurface zone (c) of a PTWA FeC0.3 liner specimen after SRV test. The ring pattern of the surface (b) shows a nanocrystalline microstructure consisting of α-Fe. The diffraction for the subsurface zone in (d) indicates a μc α-Fe structure[11]

5. **Conclusions and Recommendations**

Friction accounts for a substantial amount of wasted energy in passenger cars and the heavy duty diesel trucking industry. Reducing the amount of mechanical losses due to friction can allow for better fuel economy, lower operating temperatures (due to less frictional heating), and longer hardware lifetime/maintenance intervals. This challenge has been approached with design changes, material selection changes and lubricant additive/formulation improvements.

Top compression ring against cylinder liner performance must be understood from a tribological approach. Benchtop instrumentation can be used as a screening tool for wear couples or tribological pairs. However, simple ball on flat reciprocating testing is lacking. A greater chance to gather enough experimental data to develop an empirical correlation from benchtop tribological apparatus to motored/fired testing requires the use of actual hardware components. Apparatuses like the SRV enable the use of segmented hardware, which gives an advantage. The applied maximum load of 2000 N is ample to mimic the force applied by the top compression ring of a heavy duty diesel engine near TDC after a combustion event. The maximum temperature of 290 °C is also over that of what could be expected under extreme conditions in a heavy duty diesel engine. The frequency maximum of 50 Hz or 3000 rpm is ample for diesel application as well. The oil pump control in microliter per minute resolution allows for precise lubrication condition control, but a good understanding of how much oil is supplied to the top compression ring must be understood before a setting should be selected to give a good representation of a fired engine and the amount of oil regulated by the oil control ring and intermediate ring. The apparatus can be tilted to mimic a V-block, upright or opposed piston engine. However, the piston ring is affixed in the top assembly of the apparatus and does not allow for the complex dynamics of a piston ring in a groove interacting with a cylinder liner under combustion chamber back-pressure. The pressure in a motored engine varies dramatically in sub-crank angle resolution, which cannot be accounted for in the SRV. A better option, which does not currently exist, would be to pneumatically apply a backpressure to a floating ring to allow for the dynamics of the piston ring in a groove with controlled clearances. The max stoke length of 5 mm is over an order of magnitude lower than that of engines, and does not allow for the correct mimicry of mid-stroke hydrodynamic lubrication conditions, thus is limited to locations near TDC and BDC under boundary lubrication. The max bore size of 100 mm for the upper specimen holder also limits the application for heavy duty diesel engine application, commonly having bore sizes exceeding well over 120 mm.

In conclusion, the SRV is a promising new instrument for TARDEC to use as a screening tool for the top compression ring cylinder liner contact at conditions near TDC or BDC. Some of the complex dynamics are lost, but an empirical correlation could be developed if enough experimental data is gathered, which would be specific to the engine operating conditions. Additionally, TARDEC could utilize the instrument to rapidly and inexpensively screen new and emerging technologies. This would provide superior testing fidelity compared to standard benchtop tribometers using pure sliding configurations (Ball-on-ball, ball-on-flat, block-on-ring, etc.). While not a substitute for fired and unfired rig, dynamometer, or vehicle testing, technology down selection via this benchtop testing could significantly reduce overall testing cost. This is particularly useful for projects with several technology approaches or a technology with unknown performance or operating conditions. Segmented hardware components could be provided for an engine of interest while evaluating multiple lubricant technologies to compare the friction and wear performance. Alternatively, a standard lubricant could be used to test new hardware designs, materials or coatings. Testing could rapidly be conducted over a wide range of test conditions to cover the breadth of operational conditions relevant to a military vehicle duty cycle. Both friction and wear should be monitored during testing as low friction does not necessarily correlate to superior hardware durability. Careful post-tribological test analysis need to be conducted to investigate the wear mechanisms for a full understanding of the tribological performance of the system. Sufficient data collection could allow for empirical correlations to be developed relating benchtop technology performance to engine friction, fuel economy, wear, and durability improvements.

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List of Symbols, Abbreviations, and Acronyms

°C	Degrees Celsius
μ	Micro
BDC	Bottom Dead Center
C	Carbon
CKS	Chromium-ceramic
Cr	Chromium
EDS	Energy Dispersive Spectroscopy
h	Hour
Hz	Hertz
Fe	Iron
L	Liter
LCC	Load Carrying Capacity
m/s	Meter Per Second
Mo	Molybdenum
mm	Millimeter
N	Newton
Ni	Nickel
nm	Nanometer
PTWA	Plasma Transfer Wire Arc
rpm	Rotations Per Minute
s	Second
S	Sulfur
SAE	Society of Automotive Engineers
SEM	Scanning Electron Microscopy
Si	Silicon
SRV	Schwingung Reibung Verschleisse
TEM	Transmission Electron Microscopy
TDC	Top Dead Center
TWAS	Twin Wire Arc Spray
ZDDP	Zinc Dialkyl Dithiophosphate
Zn	Zinc