

A Study of the Friction of Oil Control Rings Using the Floating Liner Engine

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ABSTRACT

The oil control ring (OCR) controls the supply of lubricating oil to the top two rings of the piston ring pack and has a significant contribution to friction of the system. This study investigates the two most prevalent types of OCR in the automotive market: the twin land oil control ring (TLOCR) and three piece oil control ring (TPOCR). First, the basis for TLOCR friction on varying liner roughness is established. Then the effect of changing the land width and spring tension on different liner surfaces for the TLOCR is investigated, and distinct trends are identified. A comparison is then done between the TLOCR and TPOCR on different liner surfaces. Results showed the TPOCR displayed different patterns of friction compared the TLOCR in certain cases.

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INTRODUCTION

The recent worldwide quest to increase the fuel efficiency of internal combustion engines has led to great effects to reduce friction of many of the components in these engines. One area of major concern pertaining to this area is the friction of the piston/ring pack assembly. Because of the importance and necessity of this component to the internal combustion engine, any improvements can have relatively large implications for friction reduction. Of the components of the piston assembly, the oil control ring (OCR) has one of the largest influences on friction [1].

Piston Assembly

The standard piston ring pack assembly features a series of three rings, shown in Figure 1, which are set in circumferential grooves around the piston.

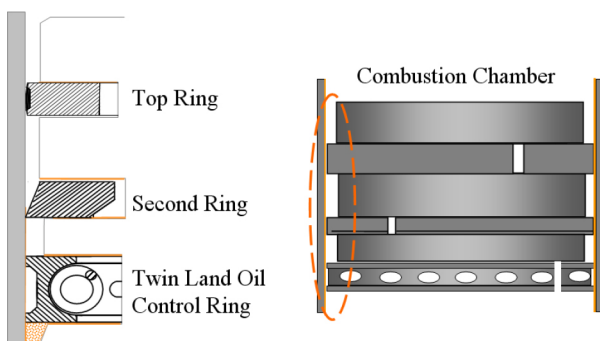


Figure 1. Piston Rings on the Piston Assembly [2]

The top ring serves to seal off the combustion gasses, while the second ring provides secondary control of oil flows on the liner and sealing. Primary control of oil flow is accomplished by the third ring, known as

the oil control ring or OCR. This ring controls the amount of oil that is available to the top two rings. The dilemma in designing the system lies in the need for the rings to seal off the combustion chamber and minimize the amount of oil that passes by into the combustion chamber (a process known as oil consumption), all while minimizing the friction of the system as much as possible. Increasing ring tension may decrease oil consumption, but can also increase friction. Any oil that passes by can burn during combustion, which increases oil consumption and emission levels, as well as poisons the exhaust after-treatment with inorganic components of additives in the oil [3, 4].

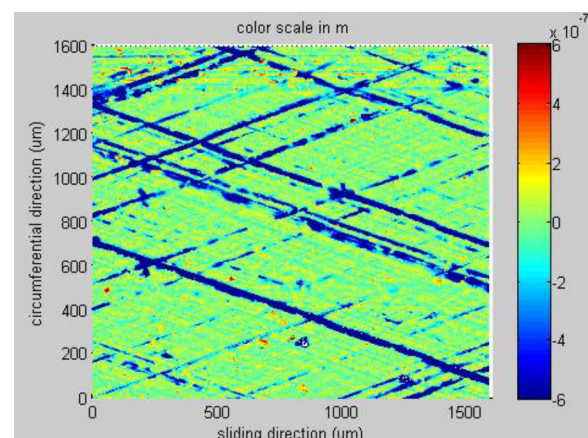


Figure 2. Confocal Microscopy Measurement Showing Characteristic Roughness of Typical Cylinder Liner [6]

In addition to the piston and rings, the cylinder liner has certain characteristics which play a critical role in the quandary between oil consumption and friction. The surface is machined with a characteristic roughness that typically takes the form of smoother plateau areas and

deep valleys, in a cross-hatched pattern [2]. In recent years, manufacturers have developed processes to hone the cylinder liners in such a way as to achieve a desired characteristic roughness [5].

Lubrication Theory in the Piston Assembly

When discussing the contribution of friction from the piston assembly, there are some fundamentals which must first be understood. The friction between the piston rings and cylinder liner is governed by the principles of lubrication theory, which were originally developed for bearing applications [7]. The friction coefficient is typically plotted on a Stribeck Curve as a function of oil dynamic viscosity and rotational speed divided by the load applied. In the piston assembly application, the friction coefficient is plotted as a function of the dynamic viscosity of the oil, μ ; the instantaneous piston linear speed, U ; the ring tension in tangential direction, F_t and the diameter of the engine cylinder bore [8]. The Y axis of the curve is calculated from the friction data at each crank angle.

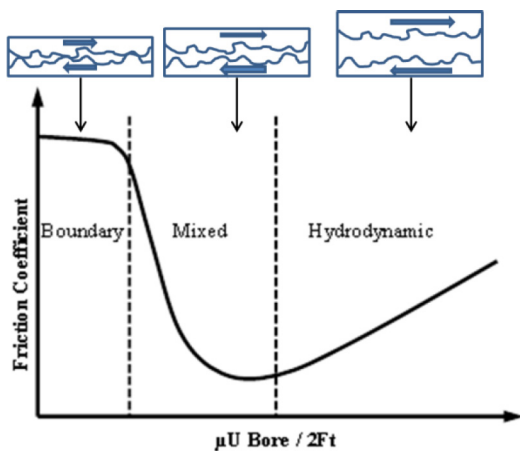


Figure 3. Instantaneous Stribeck Curve showing Lubrication Regimes [1]

When discussing friction between two sliding surfaces, the friction coefficient is the sum of the sliding contact coefficient between two dry, solid surfaces (f_b) and the hydrodynamic coefficient of friction, f_h .

$$F_c = \alpha f_b + (1 - \alpha) f_h \quad (1)$$

In this case, α represents the solid-to-solid contact constant and is a value ranging from 0 to 1, with 1 being complete solid-to-solid contact [1]. There are three regimes that describe the nature of the contact, as shown in Figure 3. The first regime, called the boundary region, occurs when α approaches 1, and there is significant contact between the asperities of the two sliding surfaces. This region is characterized by a high overall coefficient of friction and high material wear.

The second region, on the right side of the chart, is the hydrodynamic region. In this region, where α approaches 0, hydrodynamic pressure in the lubricant trapped between the two sliding surfaces causes them to separate, so that no asperity contact occurs. The friction that arises in this region comes from the shearing of the fluid, and is much lower than in the boundary region [1]. Typically, little to no wear occurs in the hydrodynamic regime, as long as the lubricant maintains its integrity.

The third (and middle) region is the mixed region, named for the fact that it includes a mixture of the other two regimes. While some asperity contact still occurs, there is also a degree of separation caused by hydrodynamic pressure in the lubrication [1]. As seen on the Stribeck curve, the minimum friction coefficient occurs in this region.

In addition to the instantaneous Stribeck curve, there is also an average Stribeck curve. This difference in this approach is that the friction coefficient is calculated from the measured FMEP for a cycle, rather than the frictional force at each crank angle. This term, known as the average friction coefficient, makes up the Y axis and is a function of the FMEP per cycle, the displaced volume of the cylinder, V_d , the distance traveled by the piston in one cycle (4 times the stroke length, l), and the tangential force of the OCR ring, F_t .

$$Avg. F_c = \frac{FMEP * V_d}{4 * l * 2\pi F_t} \quad (2)$$

The X axis of the average Stribeck curve differs from the instantaneous version only by the velocity term. Since the Y axis is a function of the average friction work per cycle (FMEP), only a characteristic velocity is needed. Due to its simplicity of calculation, the maximum piston speed, U_{max} , was chosen. The complete X axis term is shown in Equation 3.

$$\frac{\mu * U_{max} * Bore}{2F_t} \quad (3)$$

A unique feature of the average Stribeck curve is that ring tension appears in both the X and Y axis, meaning it will cancel out and the resulting graph is normalized for ring tension, allowing for comparisons of different OCR tensions at the same time.

Test Methods

The main tool used to acquire experimental data in this study is the Floating Liner Engine (FLE). The FLE is a specialized internal combustion engine that is used to measure friction and wear related to the power cylinder system [9, 10, 11, 12, 13, 14, 15, 16, 17].

Table 1. Floating Liner Engine Specifications

Displaced volume	0.496L
Stroke	92.8 mm
Bore	82.51 mm
Compression ratio	10:1
Maximum BMEP	0.7MPa
Maximum engine speed	3000RPM

The advantage of this system is it allows for very accurate measurements of friction for a system that is quite complicated and unsteady. The system is most useful when used to identify trends by isolating certain components, such as the effect of only increasing OCR tension. While these trends can help to better understand the broad picture of friction from the piston assembly, they should not be taken as an ultimatum, such as “a 10 N OCR is the best”. The FLE is also helpful when used in conjunction with modeling efforts.

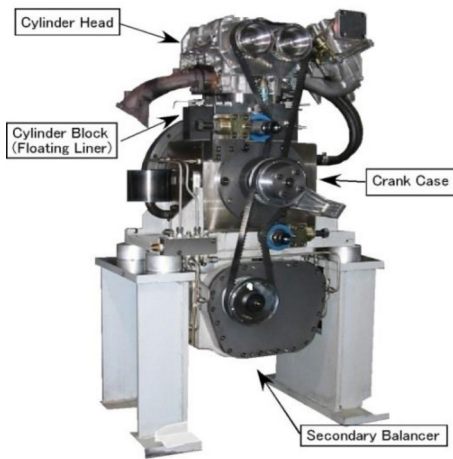


Figure 4. Floating Liner Engine

This particular FLE was provided by Professor Masaki Takiguchi at Tokyo City University (formerly Musashi Institute of Technology) in 2009. It utilizes a “free floating design”, in which the cylinder sleeve is connected to the cylinder via two load sensors (Figure 5). This setup allows accurate measurement of friction contributions from the piston assembly while the engine is in operation.

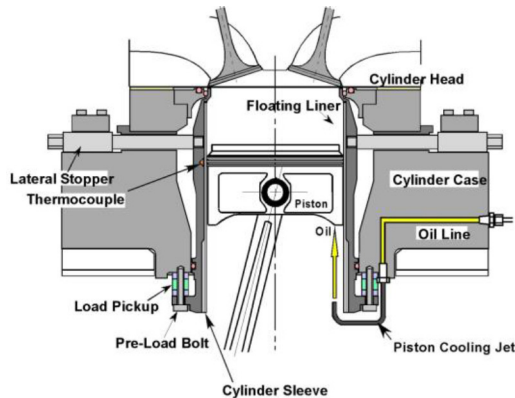


Figure 5. Cross-Sectional View of FLE Cylinder

There are two types of standard test procedure that can be accomplished with the FLE. Operating the engine under fired conditions (referred to as a Fired or Firing Test) involves operating the engine in the conventional sense, using standard 87 octane gasoline. Cylinder pressures are controlled at both 10 and 20 bar peak pressures, which correlates to 2 bar and 4 bar IMEP respectively.

	Fired Test Conditions		Motored Test Conditions
Load (IMEP)	2 bar	4 bar	N/A
RPM Tested	1000-2000		100-1000
RPM Increment	500		100
Oil Temp	85 C $\pm$ 1		40, 60, 80, 100 C $\pm$ 0.5
Liner Temp	100 C $\pm$ 1		40, 60, 80, 100 C $\pm$ 0.5
Piston Diameter	82.455 mm		82.382 mm
Skirt Coating Thickness	15 μ m (each side)		

Figure 6. FLE Standard Test Conditions

The procedure involves first warming up the engine, then collecting data “points” at 1000, 1500, and 2000 RPM at 2 bar IMEP cylinder pressure. For fired tests, each data collection “point” represents the average of data collected for 90 engine cycles. Next, the cylinder pressure is increased to 4 bar IMEP, and the data is again recorded at the same engine speeds. This complete procedure, referred to as a “sweep”, is repeated 5 times to complete a test “run”. The speed and pressure is limited by the design of the FLE, which must maintain a delicate pressure balance for adequate measurement. For this reason, specific temperature, viscosity, OCR design, and speed combinations were chosen to target the “sweet spot” of the Stribeck curve, which is where the friction regime transitions from boundary to hydrodynamic. This goal is to identify trends and principles that can be applied to other applications as well as validate modeling.

The FLE can also collect data in an alternate method known as motored testing (referred to as motored tests). This procedure, developed for this engine by K. Liao [8], utilizes a dynamometer to power the engine, rather than combustion. The cylinder head is removed and the crank case is opened to alleviate pressure imbalances. In addition, a smaller diameter piston is used, so as to minimize friction contributions from the piston skirt and further isolate OCR friction. Using the FLE in this manner allows consistent analysis of factors with small friction traces (such as the OCR), without the unsteady effects of combustion. The standard procedure for testing under motored conditions is to collect a data point (average of 30 engine cycles) at 100 RPM intervals from 100 to 1000 RPM. This “sweep” is conducted five times with both the oil and thrust side of the liner temperature being held at 40 degrees C $\pm$ 0.5 degrees. The procedure is then repeated at oil and liner temperatures of 60, 80, and 100 degrees C $\pm$ 0.5 degrees C to complete the entire test “run” [8].

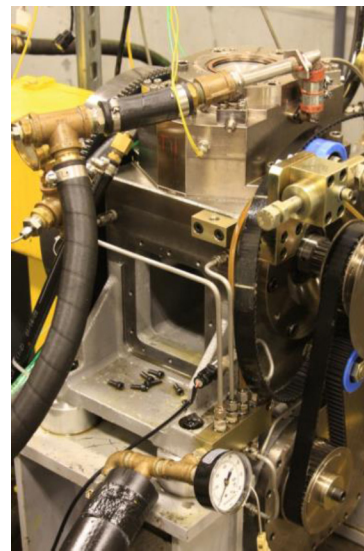


Figure 7. Example of the FLE in Motored Test Configuration

In his PhD thesis work, K. Liao developed the methods and test procedures to acquire data under motoring conditions, to include identifying the engine speeds and liner temperatures that display all lubrication regimes: boundary, mixed, and hydrodynamic. K. Liao further discussed proper methods to measure and select components such as piston rings, liners, and oil. Another motivation of the

research work was documenting and analyzing the friction change of cylinder liners throughout the break-in process. Finally, Liao developed and discussed ways to match existing models to experimental results, with great success [8].

FRICITION OF THE OIL CONTROL RINGS

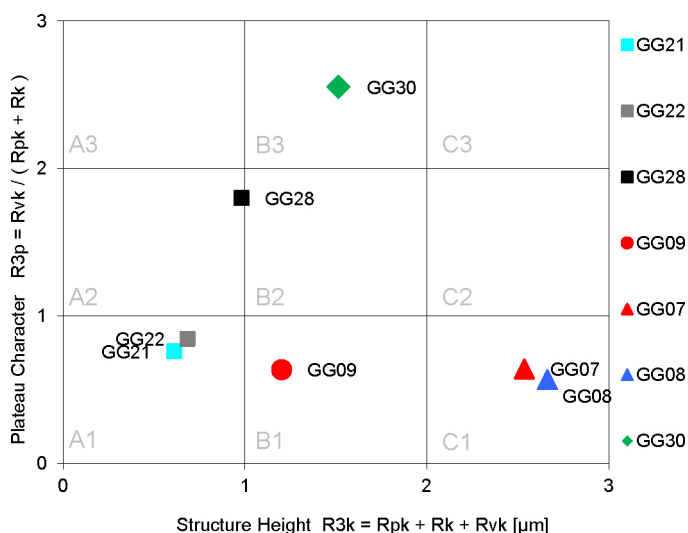
Since the tension of the OCR can surpass the tension of the other two rings combined, there is a great interest in focusing on friction trends associated with different parameters and types of OCRs. Furthermore, the OCR controls the amount of oil supplied to the other two rings, and therefore has a direct effect on their friction as well. A key feature of this ring is how it interacts with the liner surface.

Liner Surfaces

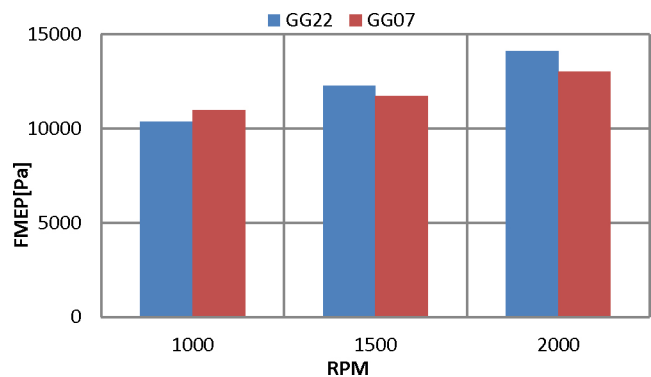
Every component discussed in this study interacts in some way with the liner surface. Just a few functions of the liner surface are: provide a friction coefficient due to asperity contact, promote oil retention, and transport of worn debris. When referring to friction, the liner surface is critical to achieving mixed friction (and the lowest friction coefficient on the Stribeck curve) as the piston assembly slides back and forth. The exact nature of this roughness is an area of concern which is easily tested in the FLE.

A variety of liner roughnesses in the classical cross-hatched honing were tested in the FLE. These liners are made of grey cast iron, and are classified according to structure height and plateau character, shown below in Table 2. Smoother liners are located to the left of the graph, such as the GG21 and GG22, while rougher liners such as the GG07 fall to the right. Likewise, liners with larger plateaus are located at the top of the chart, while those with narrower, more peak-like features are located at the bottom.

Table 2. Roughness of Liners Tested with Unit Pressure Notation [18]



analyze two of these, one from either end of the roughness spectrum. The GG22 represents a smooth liner while the GG07 represents a rough one. These two have nearly the same R3p value. The FMEP comparison under fired conditions, displayed in Figure 8 below, shows that FMEP at low speeds where the friction is predominantly contact friction is lower for smoother liners. As the speed of the engine increases, and the regime becomes more and more hydrodynamic, the rougher liner shows increasingly lower FMEP when compared to the smooth.



This effect is also shown by a comparison of the friction traces, as seen in Figure 9. At the same engine speed, the rougher GG07 liner exhibits higher friction in the boundary friction areas of TDC and BDC, while showing lower friction at the hydrodynamic mid-stroke.

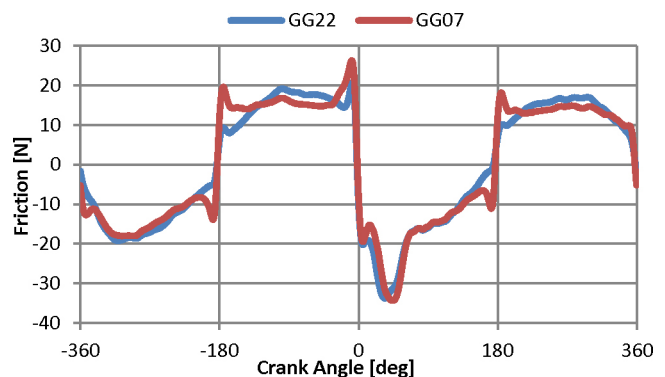


Figure 9. Comparison of Smooth and Rough Liners at 1500 RPM 2 bar IMEP

Data collected under motored conditions reinforces the findings under fired. Different temperatures, lubricants, and tensions were plotted on an average Stribeck curve. The friction regime could be changed by changing oil viscosity, through either temperature or actual oil formulation. Figure 10 shows three different roughness liners compared using the lower (HTHS 1.4) viscosity oil, in which the friction regime is dominated by mixed and boundary friction. In this regime the smoother liners (GG22 and GG28) show a lower friction coefficient. Figure 11 increases viscosity to the HTHS 2.9 oil, so that the friction regime stays mainly hydrodynamic. This leads to the rougher liners having a progressively lower friction coefficient.

Honing

Tests show that friction change from changing honing, and therefore the roughness of the contact surface, is dependent on the speed of the engine. Although several liners were tested, it is only necessary to

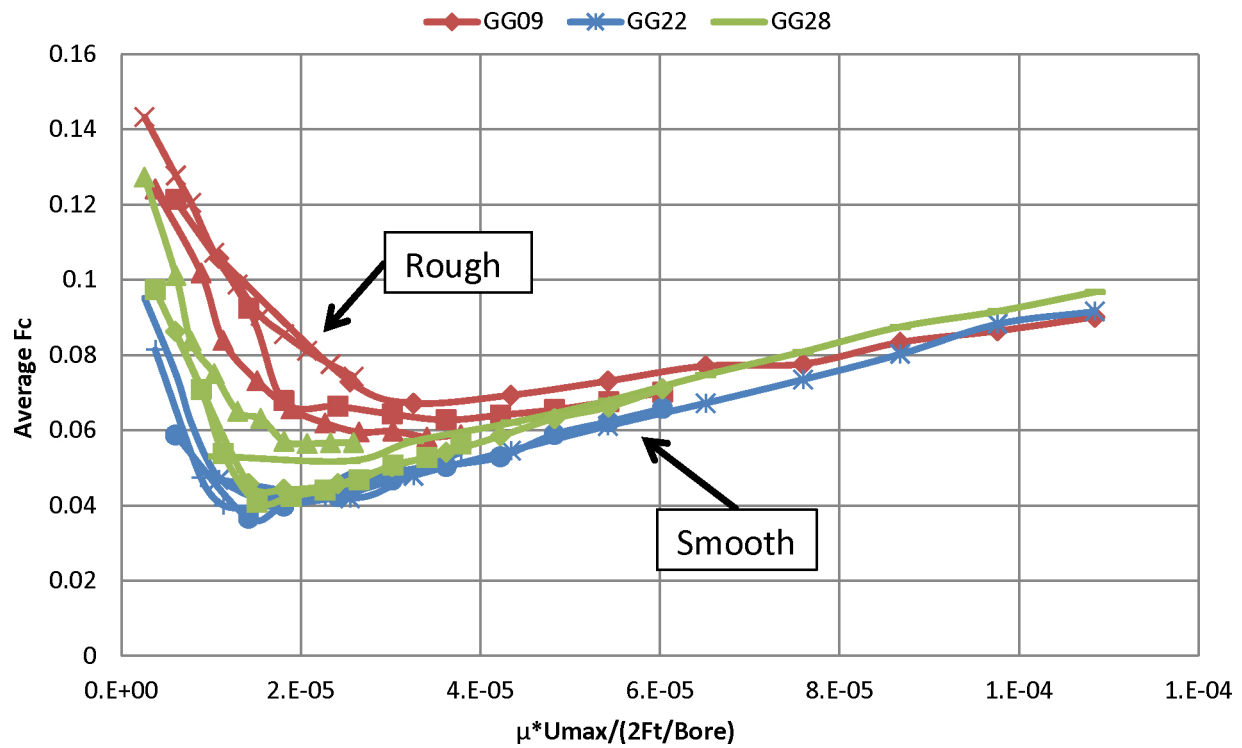


Figure 10. Average Stribeck Curve Comparing Liner Roughness in Mixed Friction Regime

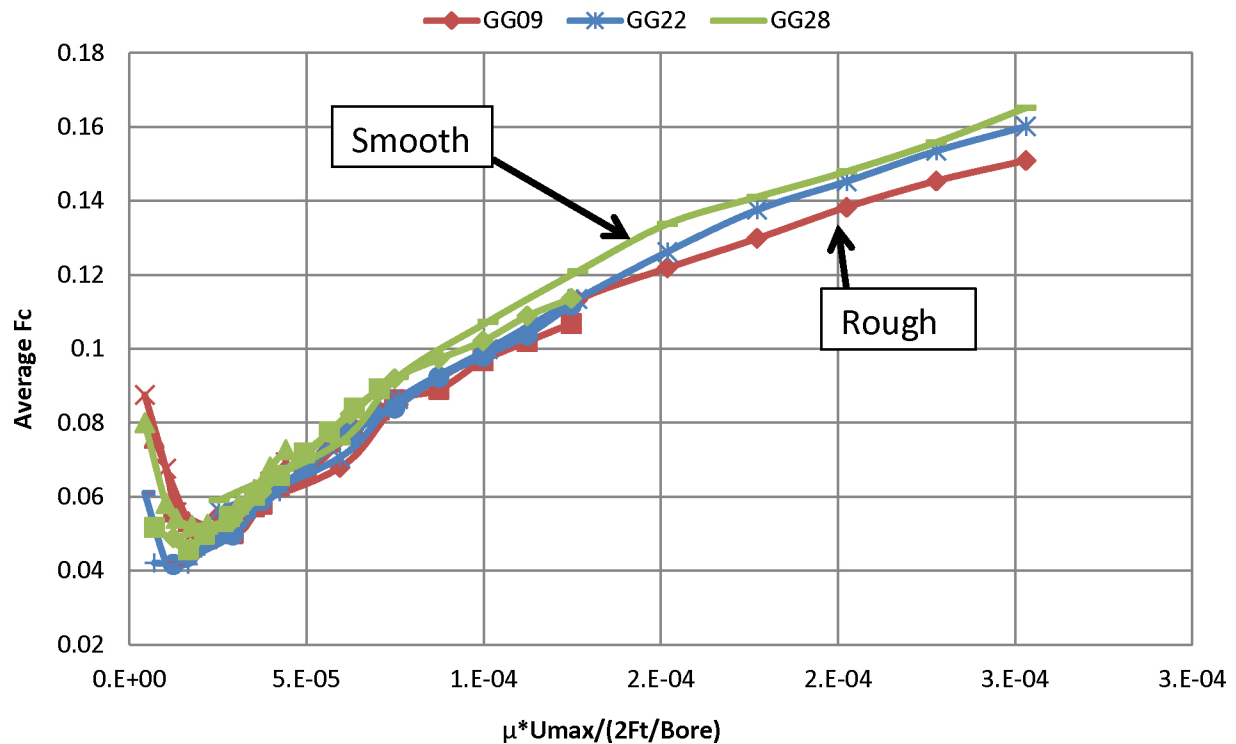


Figure 11. Average Stribeck Curve Comparing Liner Roughness in Hydrodynamic Friction Regime

The overall conclusion is when the regime is dominated by mixed or boundary friction, the smoother liner surfaces perform better. This principle can be explained as follows. Equation 4 shows how the friction between two sliding surfaces (in this case the liner and piston assembly) is dependent on the viscosity of the oil, the speed of the sliding contact, and the separation between the surfaces. However,

this relationship is only true in purely hydrodynamic situations, when the separation of the surfaces is large enough so that no contact is made. In this case friction comes from the shear stress in the lubricant.

$$\text{Pure Hydrodynamic: } F_c \sim f_h \sim \frac{\mu U}{h}$$



When either speed or viscosity is decreased enough so that the separation is comparable to the size of the asperities, solid on solid contact begins to occur. At this point, another term is introduced into the friction equation, which accounts for this contact. This new equation, which represents the friction in the mixed/boundary region, is shown below as Equation 5.

$$\text{Mixed: } F_c \sim f_b + f_h \sim \frac{\mu U}{h} + \text{boundary friction} \quad (5)$$

When in the mixed/boundary friction regime, the additional term due to increased solid on solid contact between sliding surfaces can influence overall friction. For smoother liners, the smaller asperities mean that the contact stops sooner, so that at the same speed the smoother liner has less contact than a rougher liner and therefore lower overall friction. This is particularly true at TDC and BDC when slower sliding speeds decrease hydrodynamic pressure.

When friction is predominantly hydrodynamic, the rougher liners show less friction than the smoother liners. The reasons for this can be explained by graphing the hydrodynamic pressure between the ring and liner (P_h) versus the clearance, h , on a log-log scale, with a constant μU value of 0.015 N/m.

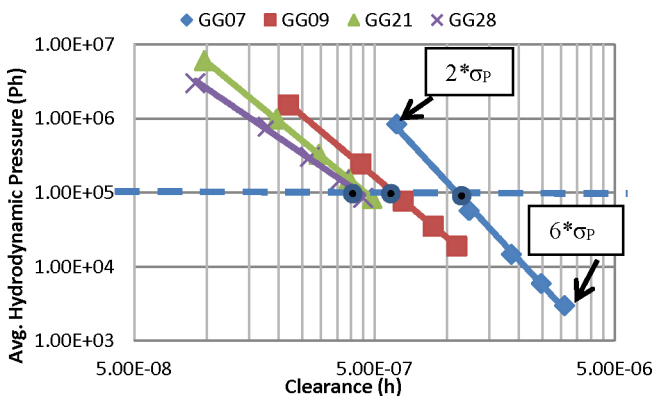


Figure 12. Modeled Relationship between Hydrodynamic Pressure and Film Thickness for TLOCs [19]

The key to the different liners graphed in Figure 12 is their position along the x and y-axis. Notice how each graphed line contains five data points each. For each line, the upper-left most point represents a clearance of $2 \cdot \sigma_P$. σ_P is the standard deviation of the characteristic heights of the peaks and valleys (R3k that make up the plateau liner roughness). Therefore, based on the principles of a Gaussian distribution, 99.7% of the asperity heights lie within three standard deviations or $3 \cdot \sigma_P$. This means that any position on the graphed lines above and left of the second data point ($3 \cdot \sigma_P$) will have some degree of asperity contact, as the clearance is within the height of the asperities. Any point below or to the right of the second data point will have virtually no asperity contact, and will be purely hydrodynamic.

For a specific unit pressure, say 1 bar (shown on the graph by the blue dashed line), the different liner finishes have different levels of clearance. For example, the GG07 has a clearance just below its $3 \cdot \sigma_P$ point, meaning it will still be close to the mixed region, and closer to the minimum friction coefficient point on the Stribeck Curve. Furthermore, notice that this point lies further along the x-axis, meaning means the GG07 will have a higher film thickness to support the same load. Since film thickness is inversely proportional to hydrodynamic friction coefficient, the rougher GG07 liner will exhibit lower friction in the hydrodynamic regime.

It is important to note that all data collected in the previous section was with the TLOC. The trends discussed should be limited to TLOC applications. Similar trends observed for the TPOCR have not been fully explored at this time, and could possibly occur for other reasons.

TLOC Parameter Comparison

The Twin Land Oil Control Ring, or TLOC, is a design that utilizes two pieces to perform the functions needed: a twin land ring and tension spring.

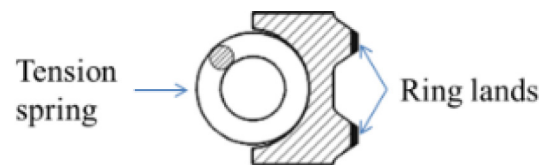


Figure 13. Twin Land Oil Control Ring Cross-Section

The two ring lands, highlighted in Figure 13, provide the unit pressure radially outward against the cylinder liner to control the amount of oil on down-stroke. Also, the ring features slots in the horizontal direction (which can be observed in Figure 1) which allow oil to flow from the contact area, and thereby reduce pressure buildup. The two parameters of TLOC design that were studied were the width of the two lands and the tension of the spring exerting a tangential force.

Ring Parameters

The outward radial pressure exerted by the OCR controls the thickness of the oil film on the liner during the down-stroke of the piston. By altering the width of the rings' normal surface to the liner (increasing surface area) and/or the normal force applied by the ring (by changing the spring tension), the unit pressure can be altered and in turn change the amount of oil supplied. This means the friction of the top two rings is highly reliant on the OCR [20]. Equation 6 shows how changing the land width changes the unit pressure of the OCR on the liner, with F_t being the ring tension, L_w being the land width, and B being the cylinder bore diameter.

$$\text{Unit Pressure} = \frac{F_t}{L_w B} \quad (6)$$