In Situ Control of Lubricant Properties for Reduction of Power Cylinder Friction through Thermal Barrier Coating

By

Mark Allen Molewyk

B.S. Mechanical Engineering
Purdue University, 2012

SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

MASTER OF SCIENCE IN MECHANICAL ENGINEERING
AT THE
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

JUNE 2014

Signature redacted

Signature of Author...... ....
Department of Mechanical Engineering
May 9, 2014

Signature redacted

Certified by.......... ....
Victor W. Wong
Principal Research Scientist and Lecturer in Mechanical Engineering
Thesis Supervisor

Signature redacted

Accepted by.................................. ....
David E. Hardt
Chairman, Department Committee on Graduate Students

(Page intentionally left blank)

In Situ Control of Lubricant Properties for Reduction of Power Cylinder Friction through Thermal Barrier Coating

By

Mark Allen Molewyk

Submitted to the Department of Mechanical Engineering
on May 9, 2014 in Partial Fulfillment of the
Requirements for the Degree of Master of Science in
Mechanical Engineering

Abstract

Lowering lubricant viscosity to reduce friction generally carries a side effect of increased metal-metal contact in mixed or boundary lubrication, for example near top ring reversal along the engine cylinder liner. A strategy to reduce viscosity without increased metal-metal contact involves controlling the local viscosity away from top-ring-reversal locations. This paper discusses the implementation of insulation or thermal barrier coating (TBC) as a means of reducing local oil viscosity and power cylinder friction in internal combustion engines with minimal side effects of increased wear. TBC is selectively applied to the outside diameter of the cylinder liner to increase the local oil temperature along the liner. Due to the temperature dependence of oil viscosity, the increase in temperature from insulation results in a decrease in the local oil viscosity. The control of local viscosity through TBC targets areas of high hydrodynamic power losses mid-stroke while avoiding an increase in boundary friction near ring reversal. If temperatures near ring reversal remain unaltered, the expected result is the same oil viscosity, boundary friction, and wear rate near TDC as that of a non-insulated liner. In order to calculate the frictional benefit of insulating the cylinder liner, an in-cylinder heat transfer model predicts the temperatures along the liner. The local oil temperatures and engine performance parameters are then applied to a ring pack simulation to calculate the contributions to hydrodynamic and boundary friction power loss. The BsFC and wear rate results are then compared to baseline simulation data for TBC performance metrics. The results show the TBC insulated liner maintains adequate viscosity and film thickness near TDC for wear protection in the ring, while decreasing a significant portion of hydrodynamic for friction power loss in the mid-stroke. For the case studied, TBC offers a 0.7% BsFC improvement from the reduction in power cylinder friction with no increase in the wear rate of the ring pack.

Thesis Supervisor: Victor W. Wong
Title: Principal Research Scientist and Lecturer in Mechanical Engineering

(Page intentionally left blank)

Acknowledgements

I will never forget my time here at MIT. I have grown mentally and personally thanks to this wonderful university and the people in the community. Above all, it has been fun and unpredictable here in Boston. I have witnessed a community come together for both tragedy and success during my time in Boston. Many people have been pivotal to my growth and development here at MIT and I would like to take the time to thank them in this section.

My journey at MIT began with the guidance and grace of my advisor Dr. Victor Wong. Thanks to Dr. Wong, I was able to find a project with a perfect fit for my interests; and a project that stands to make an impact in the diesel engine industry. Thank you for the great opportunity to pursue my passion in diesel engine research.

This project would not have been possible without the support of my project sponsor, Detroit Diesel. Specifically, I would like to thank Kevin Sisken, Sandeep Singh, David Atherton, and Peter Attema for their support throughout the project. I would like to thank Jai Bansel, Maryann Devine, and Bob Salguiero at Infineum LLC for their direction in engine lubricant formulation. I would also like the thank Lloyd Kamo of Adiabatics Inc. for his expertise in thermal barrier coating insulations. Without these sponsors, the work described in this thesis would not be possible

Most importantly, I would like to thank my family. My parents' support, guidance, and encouragement throughout my life has allowed me to pursue a great education at MIT. My success is a direct reflection on their guidance in my life.

The journey would not have been as much fun if it were not for Levi Lentz, Pasquale Totaro, Tim Murray, Dave Bierman, Chris Baachman; thank you for the fun and exploring Boston with me.

I would also like to thank my fellow students and the faculty of the Sloan Automotive Lab. Sloan lab has provided the perfect environment for me to seek knowledge, get my hands dirty, and most of all form friendships. I would especially like to thank Mike Plumley and Tomas Vianna Martins. I have learned and grown as much from the people at MIT as I have from the classes at MIT. Mike and Tomas are prime examples of this. Mike, thank you for the mentorship and friendship. Tomas, thank you for continuing to push me as a student and a person. Finally, any list of Sloan Lab acknowledgements would not be complete without Janet Maslow. Janet, thank you for the help and guidance along the way. The lab runs smoother thanks to your efforts.

Special Acknowledgements

I would also like to thank the players of the Boston Red Sox. Attending MIT has given me the opportunity to live in Boston and watch the Red Sox under the lights at Fenway. Joining the Fenway Faithful has been a dream of mine ever since I had to part ways with my former dream of playing baseball competitively. In 2013, my baseball dream blossomed in the most unimaginable way possible. I witnessed the Red Sox win the World Series in Boston. Pandemonium ensued in the streets of Boston. Countless nights at Fenway from April to October; defeating the Yankees in 13 out of 19 games; The record setting 20-4 victory against the Tigers with 8 home runs; beholding the ALDS, ALCS, to World Series. All these memories were built first hand at Fenway. Thank you to the players for bringing charisma, passion, hard work, and dedication to the field at Fenway. It was a pleasure watching the Red Sox from 2012 to 2014 and even more unimaginable to watch the "band of bearded brothers" to win it all in 2013. Thank you for the memories.

Table of Contents

List of Figures

List of Tables

(Page intentionally left blank)

Nomenclature

(BDC)	Bottom Dead Center
(CAC)	Charge Air Cooler
(CO)	Carbon Monoxide
(DOC)	Diesel Oxidation Catalyst
(DoE)	Department of Energy
(DPF)	Diesel Particulate Filter
(EGR)	Exhaust Gas Recirculation
(EPA)	Environment Protection Agency
(FMEP)	Friction Mean Effective Pressure
(HC)	Hydrocarbon
(NOx)	Oxides of Nitrogen
(PAI)	Polyamide-imide
(PAO)	Polyalphaolefins
(PM)	Particulate Matter
(PPD)	Pour Point Depressants
(PSZ)	Plasma Spray Zirconia
(SCR)	Selective Catalytic Reduction
(TBC)	Thermal Barrier Coating
(TDC)	Top Dead Center
(VII)	Viscosity Index Improver
(VM)	Viscosity Modifier
(ZDDP)	Zinc dialkyldithiophosphates

(Page intentionally left blank)

1. Introduction

Vehicle fuel economy in diesel engines is becoming more important due to the approaching CO_2 emissions regulations. In order to achieve better fuel economy, engine manufacturers may recommend the use of lubricants with a thinner or less viscous base stock to reduce rubbing friction. These low viscosity lubricants result in a decrease in friction mean effective pressure (FMEP) [1] and utilize advanced additive packages to prevent an increase in wear due to the decrease in oil viscosity [2]. As fuel efficiency regulations require manufacturers to decrease rubbing friction with a possible expense of an increase in wear, this trend may not be acceptable in certain applications such as long haul diesels where fuel economy is important, yet the engine must be durable enough to meet the customer's needs. Lubricant manufacturers have made great progress in improving additive packages to reduce friction while protecting engine surfaces from increased metal on metal contact, but solutions that reduce friction while maintaining current wear levels may be found outside the regime of lubricant formulation. The local control of lubricant temperature and viscosity by insulating engine components with thermal barrier coating (TBC) is one solution that will offers a fuel economy improvement without increasing wear through component design rather than lubricant design. With a TBC insulated liner, the friction reduction of low viscous oil is achieved while maintaining the durability of 15W40 oil.

This paper will introduce the concept of in situ lubricant properties and how lubricant properties pertain to power cylinder friction. An understanding of in situ lubricant properties leads to the development of a TBC thermal management strategy in which friction is decreased, while wear remains untouched due to a metal on metal contact friction remaining constant. The models used to analyze the TBC insulated liner and piston friction are presented briefly to supplement the analysis of the recommended TBC coating. The TBC power cylinder investigations presented in this paper include the parametric studies that lead to a final TBC design. The paper concludes with an example recommendation of a TBC insulated cylinder liner, which maximizes local friction reduction, and compares the TBC insulated liners to reducing friction through low viscosity oils only.

1.1 Diesel Engine Emissions

Diesel engine emissions are byproducts of combustion taking place in the engine. Depending on the combustion recipe, air handling, and after-treatment system, the concentration of these emissions will change. Figure 1.1 presents the relative concentration of emissions in typical diesel engine combustion without an after treatment system. Carbon dioxide makes up about 10% of the overall emissions by concentration. Whereas pollutant emissions such as nitrogen oxides, particulate matter, hydrocarbons, and carbon monoxide create less than a percent of the overall emissions [3].

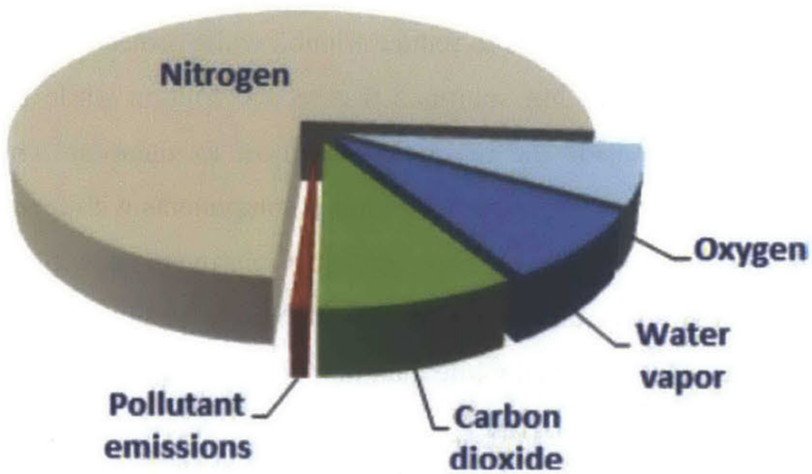

Figure 1.1 - Relative Concentration of Pollutant Emissions in Diesel Exhaust Gas [3]

1.1.1 Diesel Engine Pollutant Emissions Regulations

Diesel engine pollutant emissions are defined as nitrogen oxides (NOx), particulate matter (PM), hydrocarbons (HC), and carbon monoxide (CO). These emissions cause ground level ozone or smog in densely populated areas with heavy traffic. The fine particles of PM are small enough to lodge themselves deep within the respiratory system causing serious health issues. NOx emissions have been linked to health concerns with humans [4]. These serious health and environmental issues have created a focus on reducing these emissions in diesel engines.

Beginning in 1974, the United States Environmental Protection Agency (EPA) began regulating the pollutant emissions of diesel engines as shown in figure 1.2. The diesel engine emissions regulations focused on nitrogen oxides, particulate matter, hydrocarbons, and carbon monoxide. From 1994 to 2010, NOx and PM emissions were reduced 96% and 90% respectively.

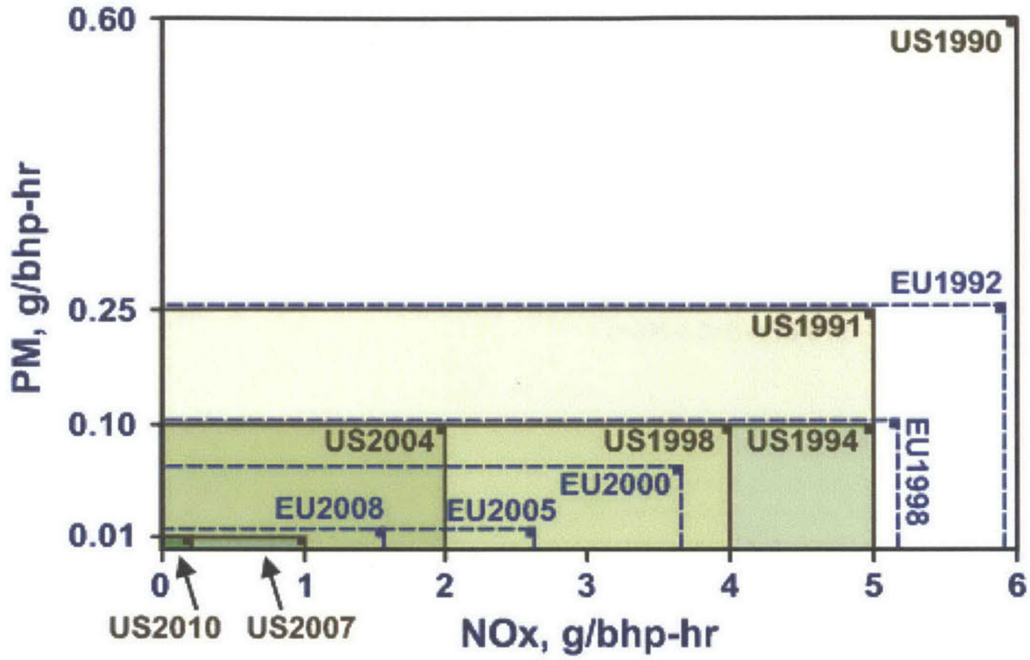

Figure 1.2 – United States and European Pollutant Emissions Regulations 1990 – 2010 [5]

1.1.1.1 Emission Control Strategies

The pollutant regulations drove change in diesel engine hardware. In the 1990's emissions control strategies focused on air handling and combustion solutions. Figure 1.3 illustrates the change in diesel engine technology to meet emissions regulations. NOx production rates during combustion depend on peak combustion temperatures. In order to lower temperatures, engine manufactures retarded timing and decreased intake manifold charge temperature. This required new fuel injector technology such as the common rail fuel system for control of injection timing and the charge air cooler (CAC) to reduce intake manifold temperatures after the turbocharger. The addition of the advanced fuel and air handling system led to more efficient and lower temperature combustion to combat HC and NOx emissions.

As regulations became more stringent, advanced combustion solutions continued to progress, and combustion hardware alone could not satisfy regulation of diesel engine pollutant emissions. Additional hardware was needed to move toward zero emissions vehicles. Exhaust gas recirculation (EGR) hardware was introduced to diesel engines to reduce NOx emissions even further. With EGR systems, a fraction of the exhaust gas flows from the exhaust manifold through a heat exchanger to cool the hot gas, and is introduced into the intake manifold charge. By introducing inert exhaust gas to the combustion chamber, the EGR gases simply absorb energy and do not take part in the combustion process. The addition of inert gases decreases peak temperatures and NOx emissions further.

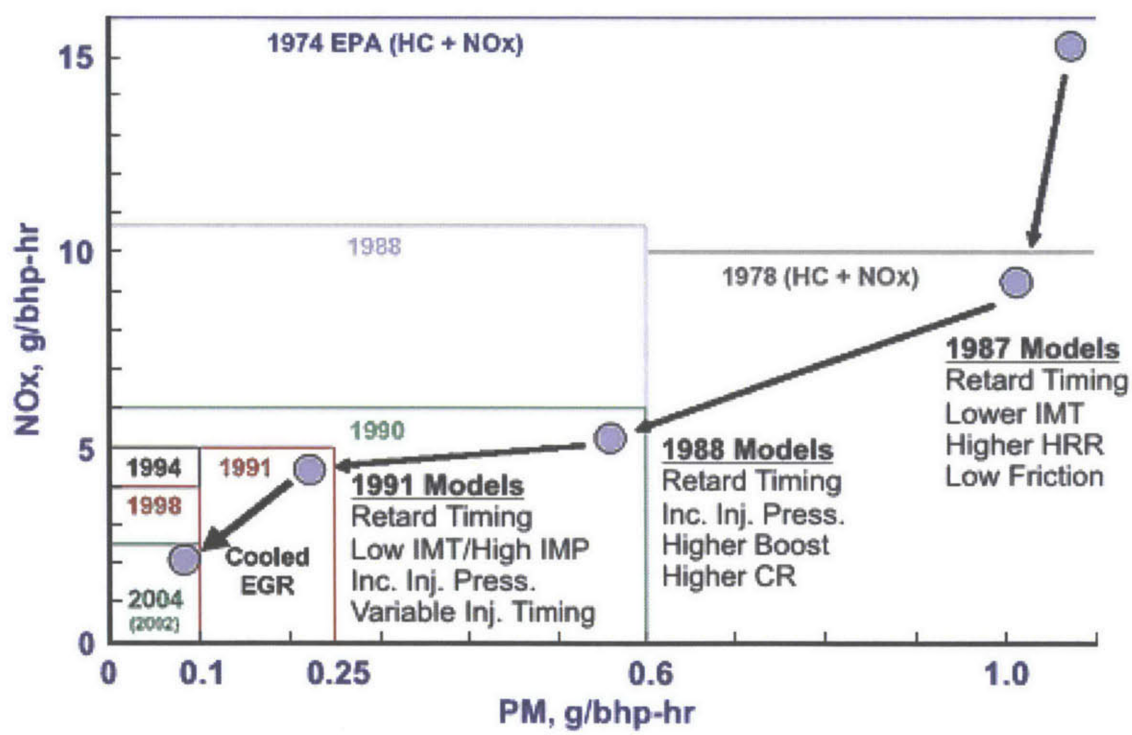

Figure 1.3 – Pollutant Emissions Regulations Impact on Engine Architecture [5]

Yet the time would come that advanced combustion and EGR would not be enough to meet pollutant emissions without additional hardware. Advanced after treatment systems would be added downstream from the turbocharger in order to reduce emissions. In order of sequence and chronology, the diesel oxidation catalyst (DOC), diesel particulate filter (DPF), and selective catalytic reduction (SCR) systems would be added to the diesel engine in order to progress towards a zero emissions vehicle. The diesel oxidation catalyst would further oxidize partially burned hydrocarbons to reduce HC emissions. Next, the DPF acted as a filter for PM emissions. Finally, the SCR was added to convert NOx emissions to N_2 and O_2 with the addition of ammonia injection and catalytic surfaces present in the SCR.

1.2 Carbon Dioxide Emission and Engine Efficiency Standards

In 2017, the emissions regulations will change directions to the reduction of greenhouse gas emissions, such as carbon dioxide. The current level of pollutant emissions will be maintained, while engine manufacturers must reduce the carbon dioxide output of the engine. Since carbon dioxide is a fundamental product of the combustion of hydrocarbons, the carbon dioxide regulations translate into a fuel efficiency regulation. Table 1-1 previews the 2017 regulations on CO_2 emissions. Table 1-1 also translates CO_2 emissions regulations into fuel economy standards.

Table 1-1 – 2017 Carbon Dioxide Emissions Standards [6]

	EPA Full Useful Life Emissions Standards (g CO2/ton-mile)	NHTSA Fuel Consumption Standards (gal/1,000 ton-mile)
Light Heavy Class 2b-5	373	36.7
Medium Heavy Class 6-7	225	22.1
Heavy Heavy Class 8	222	21.8

1.2.1 Diesel Engine Efficiency Improvement Efforts

Similar to how pollutant emissions have driven progress in diesel engine technology, the current CO2 emission regulations will drive research for new hardware that improves engine efficiency. The Department of Energy (DoE) SuperTruck program focuses on identifying these new technologies and demonstrating their effectiveness at improving engine efficiency. The goal of the DoE SuperTruck program is to demonstrate 50% brake thermal efficiency in a diesel engine test bed and 50% freight efficiency (measured in freight-ton-miles per gallon) on a class 8 tractor-trailer [7]. The improvements in engine efficiency focus on engine downsizing, component efficiency, waster heat recovery, and parasitic losses as shown in figure 1.4. The work completed in this paper focused on the reduction of parasitic losses. Similar to the addition of advanced fuel systems, air handling, EGR, and after treatment, carbon dioxide emissions standards will drive the addition of advanced systems being researched in the SuperTruck project for improvement of engine efficiency, such as the control of lubricant properties with thermal barrier coating which is presented in this paper.

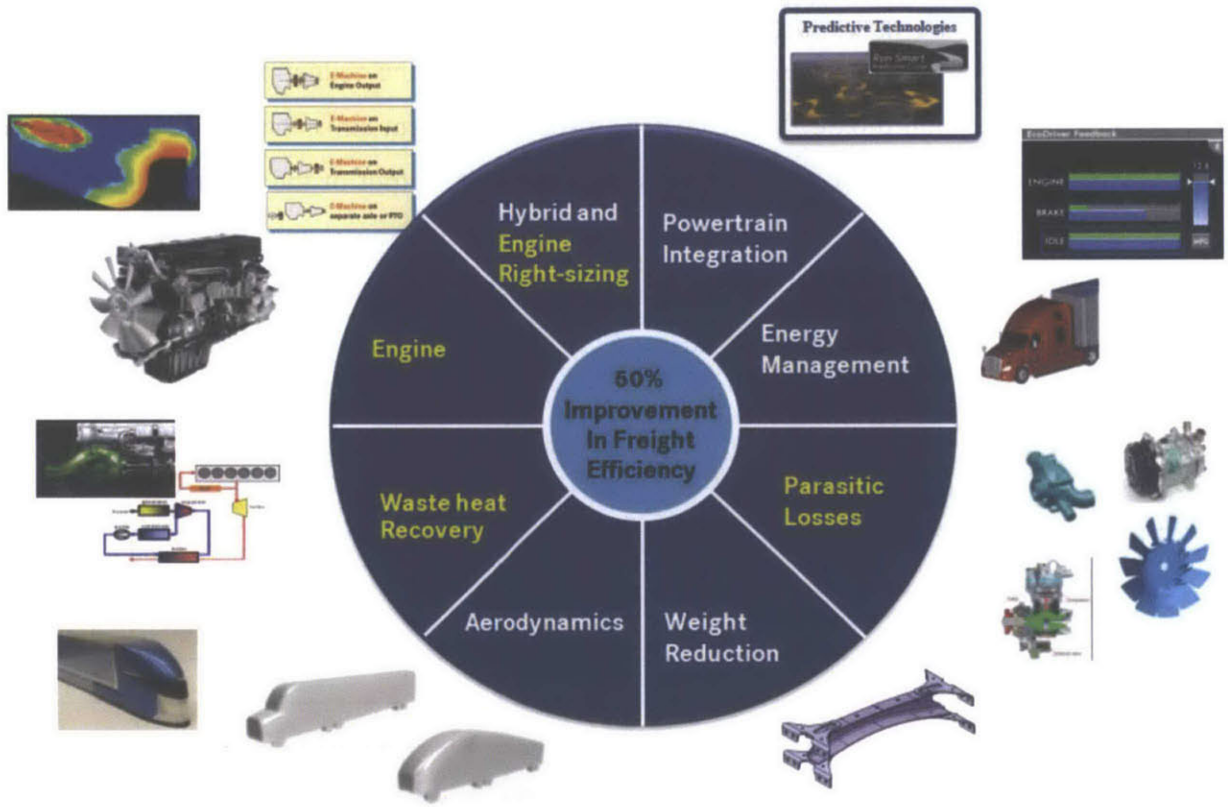

Figure 1.4 – Detroit Diesel SuperTruck Engineering Efforts [7]

1.2.2 Typical Friction Distribution in a HD Diesel Engine

MIT's contributions to the Detroit Diesel's SuperTruck program include the reduction of parasitic losses. Under Dr. Victor Wong's guidance, an expertise in engine lubrication has developed. The research completed in this thesis focuses on reducing lubrication losses in the power cylinder through a novel solution of insulating the cylinder liner. A close inspection of the energy losses in an engine, shown is figure 1.5, highlights why the power cylinder lubrication was targeted. Of the 100% of fuel energy injected into the combustion chamber, approximately

35% is converted to useful work and 10% is lost to mechanical friction. Breaking down the mechanical friction further, the power cylinder friction contributes the majority of mechanical losses in the engine a 40-50% [8].

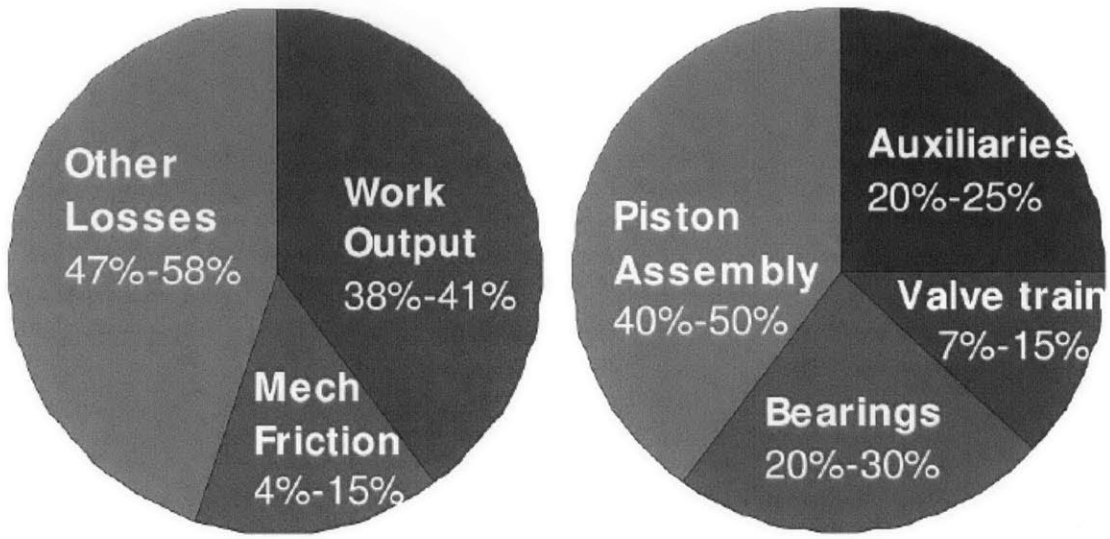

Figure 1.5 – Energy Losses in Typical Diesel Engine [8] [2]

1.3 Foreword

Now that an understanding of the carbon dioxide emissions and the contributions to parasitic losses has been developed, this paper will continue to explain the motivation behind implementing thermal barrier coating for a friction reduction. The motivation and basic concept necessary for the implementation of thermal barrier coating is explained in chapters 2 through 4, *Engine Oils, In Situ Control of Lubricant Properties, and Thermal Barrier Coating Insulation* respectively. In chapter 5, Power Cylinder Modelling, the analysis used to design the TBC insulated cylinder liner is presented for the reader's understanding. Finally, the results of various TBC investigations are presented in chapters 6 and 7.

(Page intentionally left blank)

2. Engine Oils

An understanding of engine oils and their formulation is essential when reducing friction in an engine. Figure 2.1 highlights the lubricated components' contribution to mechanical losses in a diesel engine [8]. If the viscosity of the specified engine oil was decreased, 75 percent of the engine's mechanical losses or 7.5 percent of the overall fuel energy would be affected [8]. The high contributions to engine losses from the lubricated components warrant a deeper understanding of the lubricant used in the engine. This chapter, entitled *Engine Oils,* will explain the meaning of engine oil grades, base oils, additive packages, and an ideal formulation strategy to reduce friction without a net increase in wear.

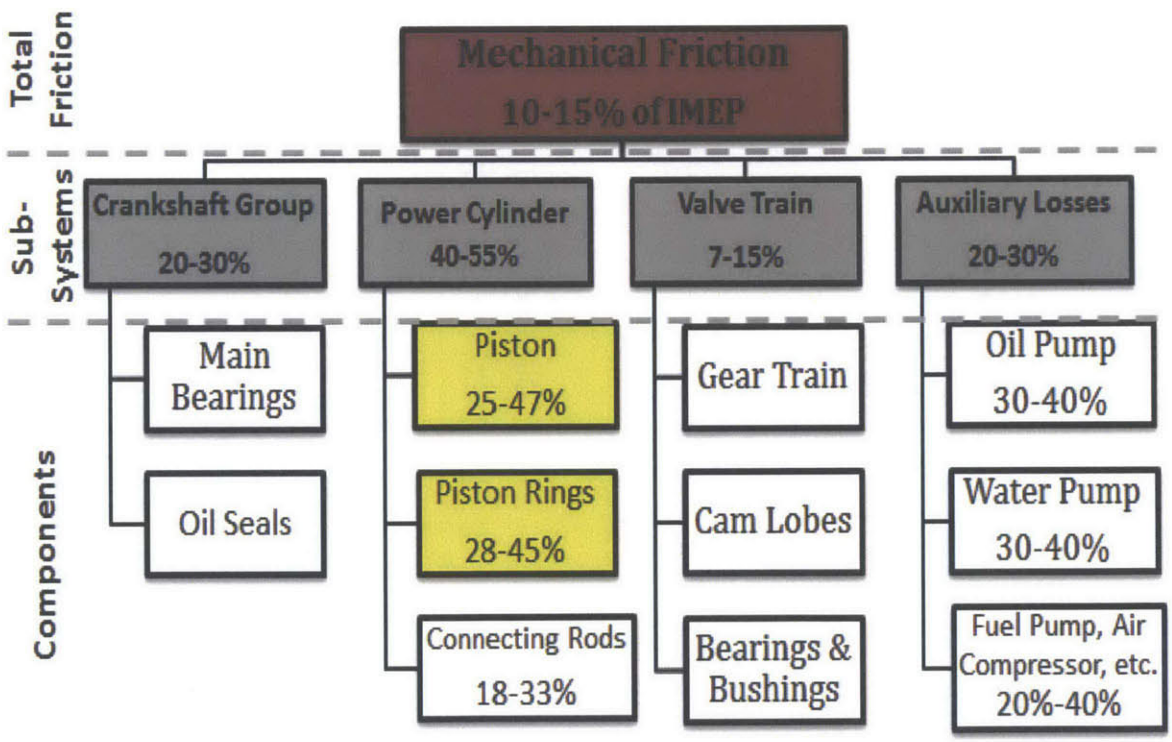

Figure 2.1 – Typical Mechanical Friction in a Diesel Engine [8]

2.1 Engine Oil Grades

Oil specifications are generally broken down into two categories of single and multi-grade oils. Although single grade oils are not generally used in engines, understanding the difference between a single and multi-grade oil helps understand the complexity of engine lubrication.

2.1.1 Single Grade Engine Oils

Single grade oils by definition are not allowed to contain viscosity modifiers. Viscosity modifiers, which are discussed in detail later in this chapter, improve the low temperature behavior of the oil by flattening out the temperature versus viscosity relationship. Figure 2.2 compares single grade versus multi-grade oils. The multi-grade 10w30 is less viscous than any single grade oil at low temperatures yet is comparable to SAE 30 oil near 100°C. This is due to the addition of viscosity modifiers.

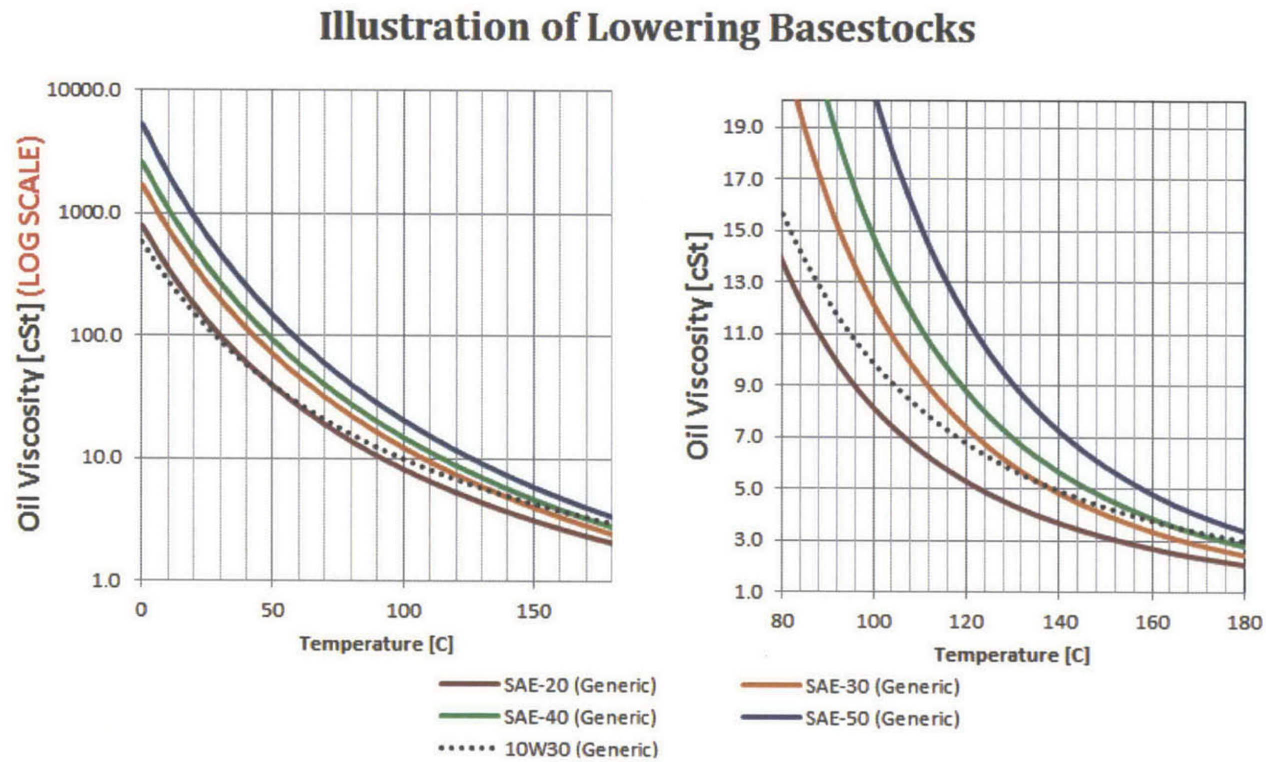

Figure 2.2 – Comparing Single-Grade and Multi-Grade Oils

2.1.2 Multi-grade Engine Oils

Multi-grade oils include viscosity modifiers in order to meet the cold and hot specifications of the oil. Cold specifications guaranty adequate lubrication conditions for engine startup. During start up, the engine must be able to pump the lubricant to the bearings before the engine seizes. The cold specifications are designed around this cold temperature pumping. The cold specifications are presented in table 2-2. By adding the viscosity modifier to the oil, the viscosity at high temperatures increases. This high temperature thickening is illustrated in figure 2.3. At high temperatures, the oil must meet two viscosity requirements. The first is that the oil falls within a given kinematic viscosity specification range at 100°C. The second requires the oil to have a given viscosity range at high temperature (150°C) and high shear rates. The high temperature specifications are listed in table 2-1.

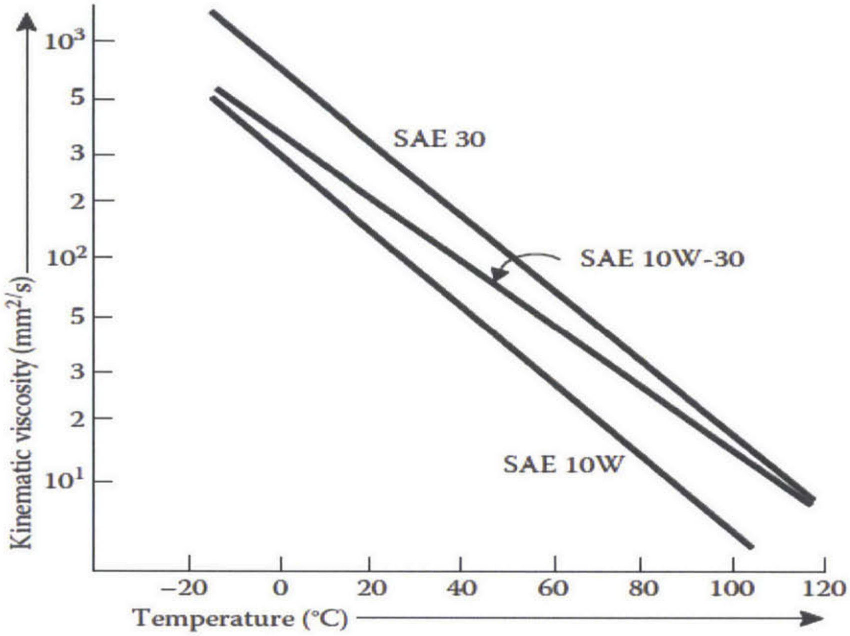

Figure 2.3 – Effect of Viscosity Modifiers and Multi-Grade Oils

Table 2-1 – SAE J300 High Temperature Specifications [9]

SAE Viscosity Chart (High Temp)				
SAE Viscosity	Kinematic Viscosity @ 100°C (cSt) Rating Minimum	Kinematic Viscosity @ 100°C (cSt) Rating Maximum	HTHS @ 150°C (cPa) Minimum	
20	5.6	9.3	2.6	
30	9.3	12.5	2.9	
40	12.5	16.3	2.9	10W or Less
40	12.5	16.3	3.7	15W or More
50	16.3	21.9	3.7	
60	21.9	26.1	3.7	

Table 2-2 – SAE J300 Cold Temperature Specifications [9]

SAE Winter Grades					
SAE Viscosity	Cranking Viscosity		Pumping Max (NYS)		Kinematic Viscosity @ 100°C (cSt)
	Spec. Temp. (°C)	Cranking Max (cP)	Spec. Temp. (°C)	Pumping Max (NYS)	
0W	-30	3250	-40	60000	3.8
5W	-25	3500	-35	60000	3.8
10W	-20	3500	-30	60000	4.1
15W	-15	3500	-25	60000	5.6
20W	-10	4500	-20	60000	5.6
25W	-5	6000	-15	60000	9.3

2.2 Base Oils

Base Oils make up 75-83% of the engine oil's composition [10]. The base oil is a mixture of different families and sizes of hydrocarbon molecules, which together form a soup of hydrocarbons that lubricate engine components and act as a carrier for lubricant additives [10]. The base oil is crucial to the formulation of finished engine oil, as it will set the lower limit of viscosity for any given oil. That is, the addition of the additive package and viscosity modifiers will only thicken or increase the viscosity of the oil. The cause of this thickening effect will be discussed in section 2.3.1 and is shown later in figures 2.6 and 2.7. Several practical lower limits exist when decreasing base oil viscosity. These lower limits include an increase in metal-on-metal contact, which leads to wear, volatility concerns, and vaporization of light hydrocarbon species [2].

Group 1 Base Oil – The organic hydrocarbons found in base oils group 1 through 3 are refined from crude oil. Group 1 base oils are the least refined of all five groups and therefore have higher sulfur content [11]. Group 1 uses less refinement processes than the other groups, which makes it the cheapest base oil.

Group 2 Base Oil – On top of the refinement processes of group 1 base oils, group 2 base oils go on to be fractionally distilled and hydro processed, which removes additional sulfur content [11]. The additional refinement also makes the group 2 base oils more accepting to additives and resistant to oxidation.

Group 3 Base Oil – Group 3 oils are organic hydrocarbons that have been refined to the highest level. The additional refinement creates the lowest sulfur content, highest additive susceptibility, and oxidation resistance available out of groups 1 through 3 [11].

Group 4 Base Oil – Group 4 oils include synthetic base oils such as polyalphaolefins (PAO). Synthetic base oils are attributed with great properties such as stability and low temperature properties, but struggle with susceptibility to additives [11]. To improve blending with additives, group 1 through 3 base oils may be added.

Group 5 Base Oil – Group 5 base oils include oils that do not qualify for categories 1 through 4. One such group 5 base oil is ester base oil. Ester's temperature and viscosity properties could allow for the formulation of ultra-low viscosity oil, but may have compatibility issues with components such as oil seals and aluminum.

2.2.1 Example of Base Oils

Figure 2.4 presents seven different PAO base oils along with four single grade oils. Figure 2.4 allows for a comparison of base oil viscosity to finished product oils. It should be noted that PAO 2 through PAO 6 have lower viscosity than SAE 20 oil. To formulate SAE 20 oil, PAO 2 through PAO 6 would need to be present with the additive. The reason for the selection of these base oils for SAE 20 oil is that the additive package will only increase the viscosity from the base oil viscosity.

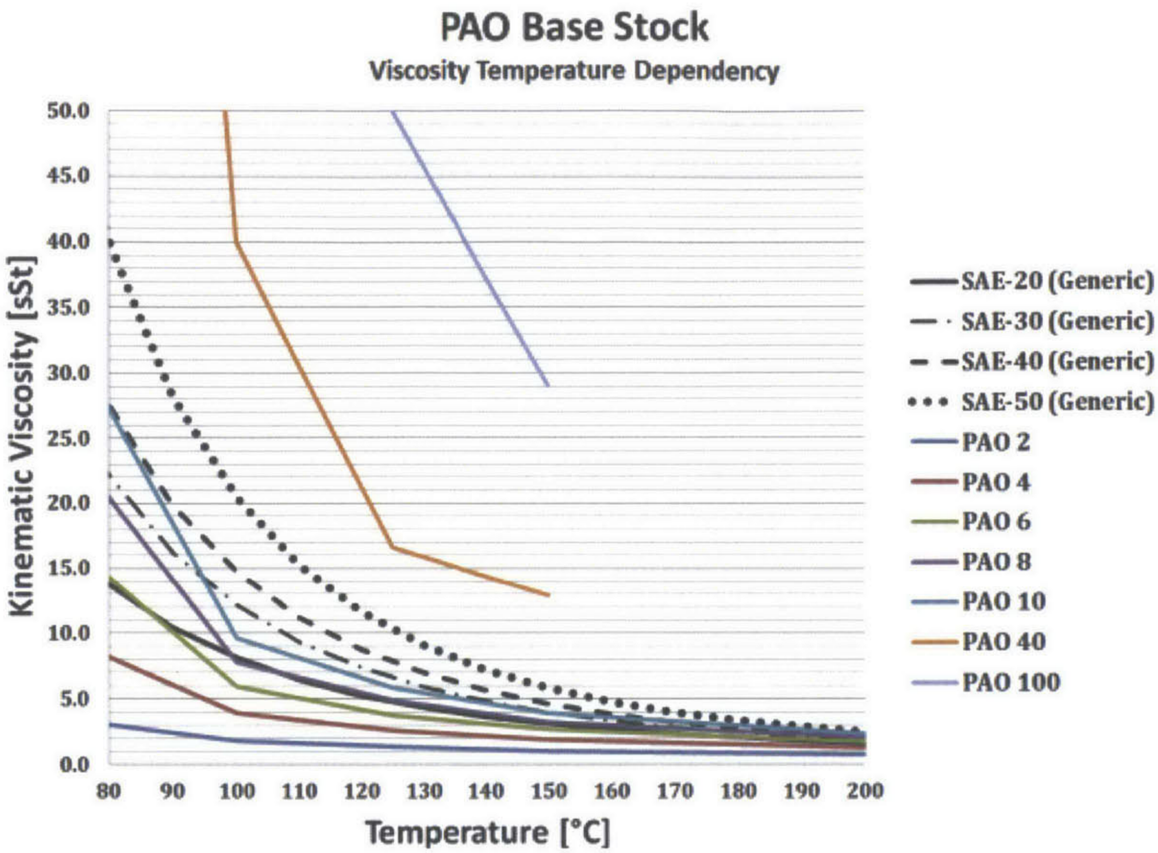

Figure 2.4 – Example of PAO Base Stocks [12]

2.3 Additives

Base oil alone is not suitable for normal engine operation. Before the oil is ready to be placed in the crankcase, several additives are introduced to the base oil to enhance properties of the base oil or add new properties that improve the life and performance of the oil. Figure 2.5 shows a typical distribution of each type of additive before the package is introduced into the base oil. The roles of these additives are presented in the sections that follow.

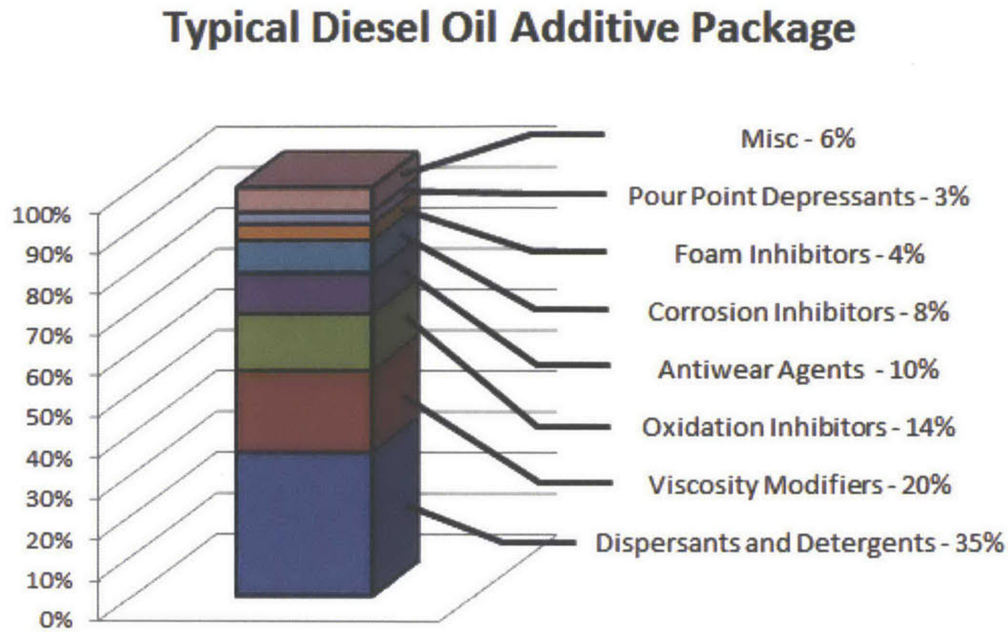

Figure 2.5 Estimated Oil Additive Package [10]

2.3.1 Viscosity modifiers

Viscosity modifiers (VM) or Viscosity Index Improvers (VII) alter the base oil's temperature dependence viscosity. Viscosity modifiers increase the overall viscosity of the oil and flatten the viscosity and temperature curve making the oil less sensitive to temperature changes. The overall change in the viscosity and temperature curve due to the addition of VM is shown in figure 2.6. The thickening effect of the VM can be attributed to the structure of the VM molecule. At low temperatures, the long tangled chain of the VM molecule shrinks and becomes insoluble to the base oil. Thus, the VM molecule does not increase the viscosity significantly at lower temperatures. At high temperatures, the VM chain expands and becomes soluble with the base oil, which increases the viscosity of the oil as the temperature increases. Figure 2.7 presents the solubility and structure effects of the VM molecule.

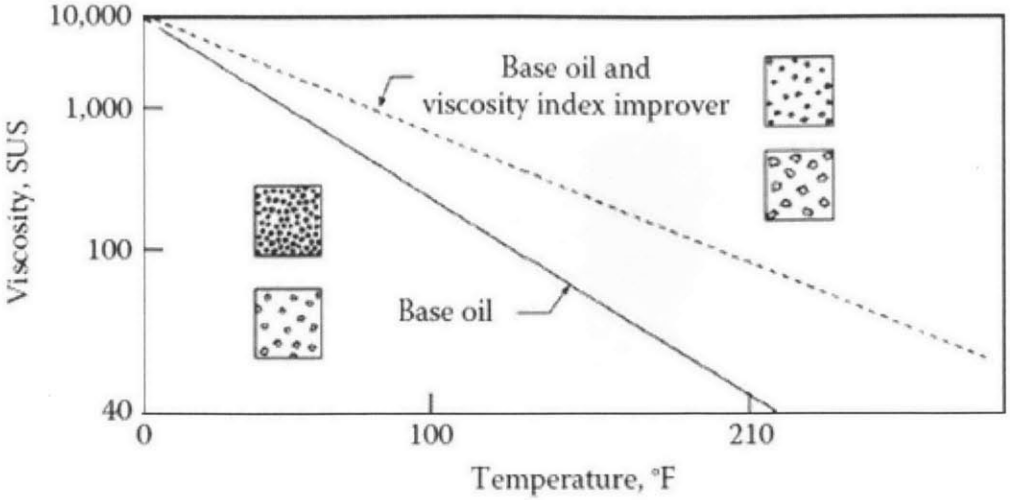

Figure 2.6 – Effect of Viscosity Modifier on Base Oil [13]

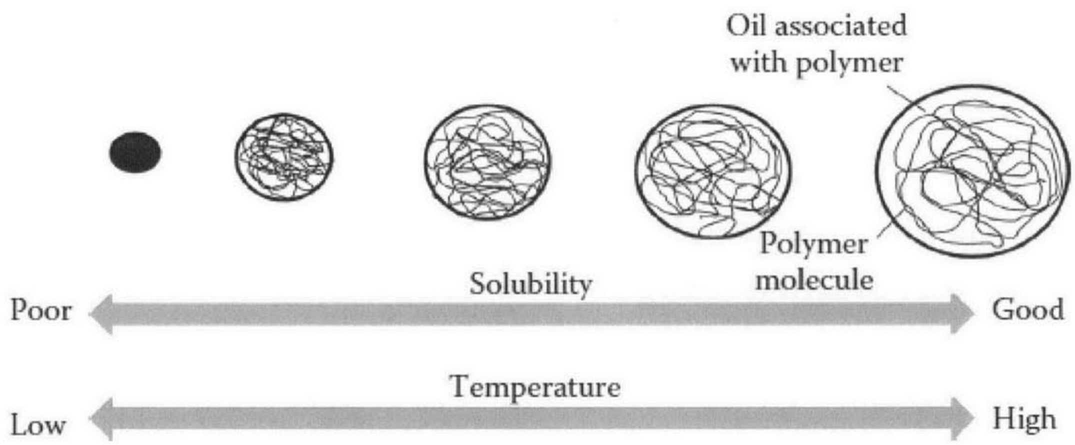

Figure 2.7 – Mechanisms of Viscosity Modifier on Oil Viscosity Increase [13]

In addition to the temperature effects of viscosity modifier, the VM also introduces a shear dependent portion of viscosity. As the long chain VM molecule shears during component lubrication, the long chains align rather than stay in their globular form. This alignment decreases the viscosity of the oil from the VM. Figure 2.8 presents the shear dependent portion of viscosity known as temporary shear loss. If the VM molecule shears beyond its limit, the molecule will break apart and the oil will begin to lose overall viscosity due to permanent shear loss.

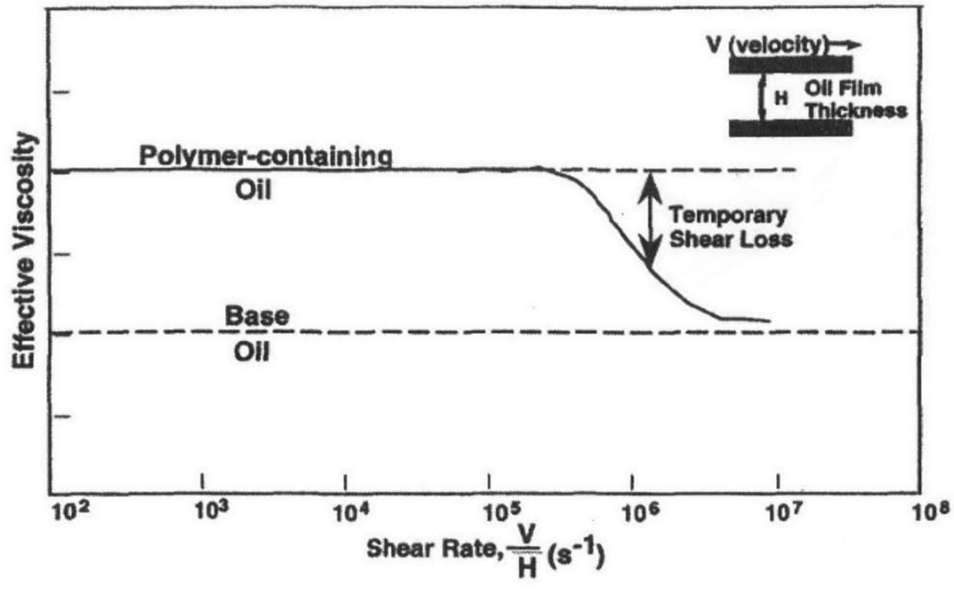

Figure 2.8 – Shear Dependent Viscosity Due to Viscosity Modifier [14]

2.3.2 Friction modifiers

Friction modifiers form films on surfaces in the engine. Once a friction modifier film forms, the coefficient of friction for metal-on-metal contact or boundary friction is reduced due to the newly formed film. Friction modifier attaches onto the metal surface due to their polar head, which physically or chemically attaches the friction modifier molecule to the surface [14]. Figure 2.9illustrates the reduction in power loss due to the application of friction modifiers. As temperature of the lubricant increases, the viscosity of the lubricant decreases. When the lubricant becomes too thin, boundary friction increases and the power losses within the engine will increase. With a friction modifier, this increase in friction due to boundary friction dominance can be delayed as shown in figure 2.9.

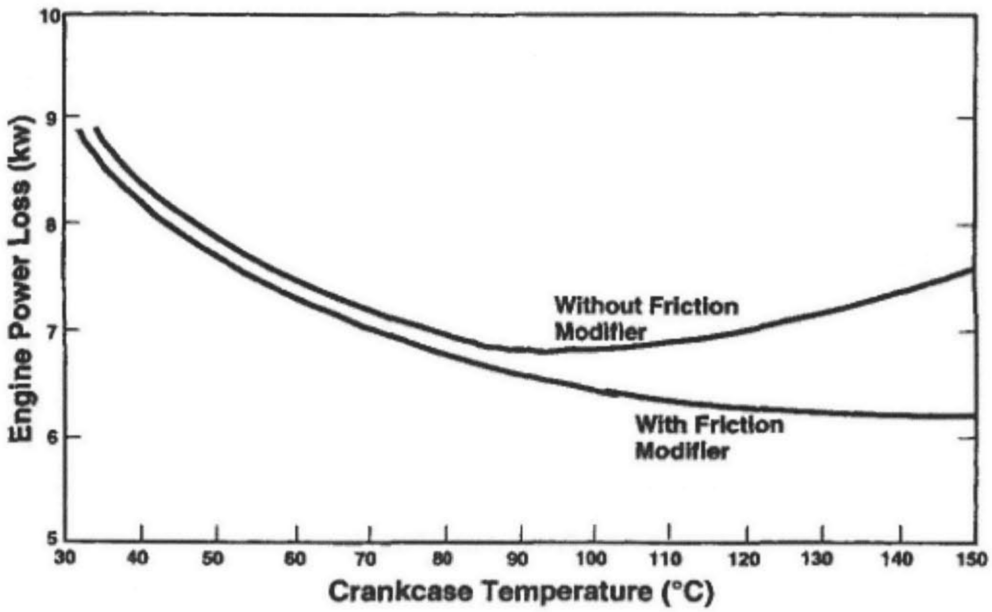

Figure 2.9 – Effect of Friction Modifiers on Engine Power Loss [14]

2.3.3 Dispersants

Dispersants and detergents both perform similar roles of cleaning the oil from unwanted foreign material. There are three key differences between detergents and dispersants. First, dispersants are metal-free, while detergents are not [14]. Metal content is crucial in oil formulation because it will lead to ash, which clogs DPF's. Detergents are basic while dispersants do not control oil acidity [14]. Dispersants are much larger than detergents [14]. The size of the molecule is important because dispersants suspend foreign material so it does not react with the rest of the oil. By suspending foreign material, dispersants prevent deposit build up on engine components.

2.3.4 Detergents

Detergents neutralize the acid products of combustion that would otherwise attack metal surfaces and cause corrosive wear [15]. Detergents can suspend unwanted combustion by-products in the oil similar to dispersants, but due to the smaller molecular size, detergents are not as effective at suspending unwanted molecules. In addition to controlling the acidity of the oil, detergents attack radical combustion products that if left untouched would oxidize the oil.

2.3.5 Oxidation inhibitors

Oxidation produces harmful by-products that deteriorate hydrocarbon molecules in the oil. This shortens the useful life of the oil. Oxidation reactions take place between radical hydrocarbon molecules and oxygen. Oxidation reactions accelerate with high temperatures typically seen in normal engine operation. Oxidation inhibitors or antioxidants generally contain sulfur, phosphorus, copper, and boron. A common oxidation inhibitor is zinc dialkyldithiophosphates (ZDDP).

2.3.6 Rust and corrosion inhibitors

Rust and corrosion inhibitors form protection films on metal surfaces to prevent the reactions that form rust and corrosion. By forming a thin film on the metallic surface, water cannot interact with the metal to form rust.

2.3.7 Foam inhibitors

Foam forms in the oil during the transport and lubrication of components. If too much foam remains in the oil, the oil will be harder to pump and transport to the lubricated components, and will have a lower viscosity. Foam inhibitors reduce foam by reducing the surface tension of bubbles. This reduction of surface tension will help the air bubbles leave the oil.

2.3.8 Pour point depressants

Pour point depressants (PPD) are added to the oil to help with flow and storage at low temperatures. Without PPDs, the oil would become waxy or gel-like and have poor flow characteristics during engine start up.

(Page intentionally left blank)

3. In Situ Lubricant Properties

A careful inspection of lubricant properties, friction, and wear versus cylinder liner positioner, an in situ point of view, provides insight into the ever-changing conditions that the ring pack and skirt experience during engine operation. A TBC insulated cylinder liner alters the temperature profile along the cylinder liner, which, in turn, increases the local oil film temperature. By designing or controlling the temperature profile along the cylinder liner, mid-stroke friction losses within the power cylinder can be targeted by altering in situ lubricant properties. The fundamentals of applying TBC insulation for reduction of power cylinder friction are based on the concept that local lubricant properties affect ring pack and skirt friction due to the large variation in viscosity from TDC to BDC. These studies show that the variations in lubricant properties along the cylinder liner are important to understanding the friction and lubrication regime along the cylinder liner [2]. For the in situ analyses presented in this paper, friction, viscosity, and temperature are analyzed versus cylinder liner position rather than versus crank angle due to the significant variations in local temperatures and viscosity. The in situ friction plots allow conclusions to be drawn on the local lubrication conditions that decrease or increase power cylinder friction. Understanding what parameters affect power cylinder friction locally will help engineers target power cylinder friction with TBC insulation.

3.1 Power Cylinder Oil Environment

Unlike the bearings or valve train which operates predominately in hydrodynamic or boundary friction respectively, the power cylinder's lubrication environment varies between both hydrodynamic and boundary friction depending on the location of the piston with respect to the stroke. Temperature, viscosity, shear rates, and film thicknesses differ greatly along the piston's stroke due to the discrete processes taking place during the intake, compression, expansion, and exhaust strokes. In addition, the component speed is continuously changing as the piston moves up and down the cylinder liner. These position and process dependent variations cause the ring pack and skirt to experience boundary, mixed, and hydrodynamic lubrication regimes during operation. The Stribeck curve, figure 3.1, illustrates this variance in lubrication regime for each engine component. Note the wide range of operation for the piston rings and skirt when compared to the bearings and valve train as seen in figure 3.1.

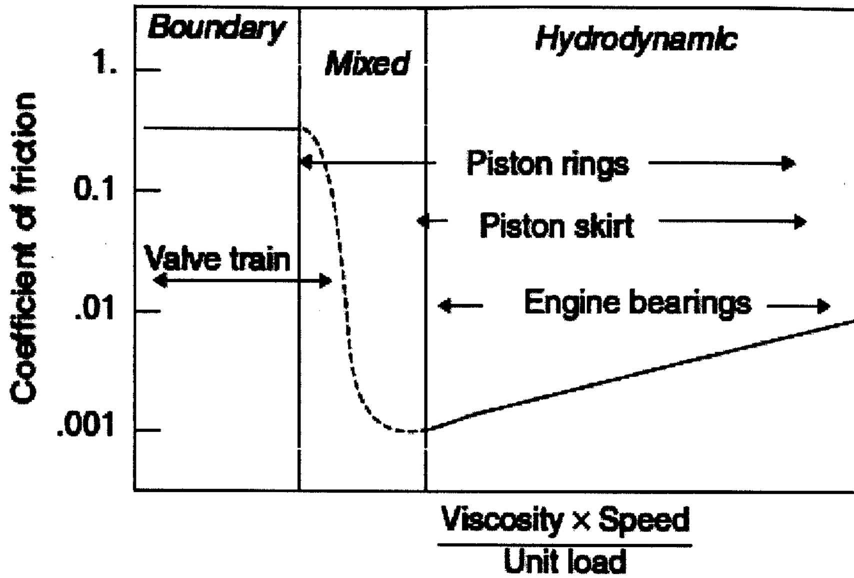

Figure 3.1 - Typical Stribeck Curve [16]

Spatial variations in temperature play an important role in power cylinder friction due to the temperature dependency of oil viscosity [2]. Figure 3.2 shows the temperature variation along the cylinder liner in a typical heavy-duty engine near peak torque operating conditions. Applying the viscometric temperature relationships for various oil grades, the local viscosity along the cylinder liner can be predicted as shown in figure 3.3. It should be noted that the viscosity at TDC is much lower than the viscosity at BDC, or the oil is thicker at BDC, due to the high temperatures at top dead center (TDC) and relatively lower temperature at bottom dead center (BDC). Local viscosity can then be used to predict the oil film thickness along the cylinder liner as shown in figure 3.5. The presentation of figures 3.2 through 3.4 provide an example of the variation in temperature, viscosity, and film thickness versus cylinder liner position for three common oils and illustrate the need for an in situ power cylinder analysis. Chapter 5, *Power Cylinder Model*, will discuss the models used to generate these figures, but the figures are currently presented as a fundamental example.

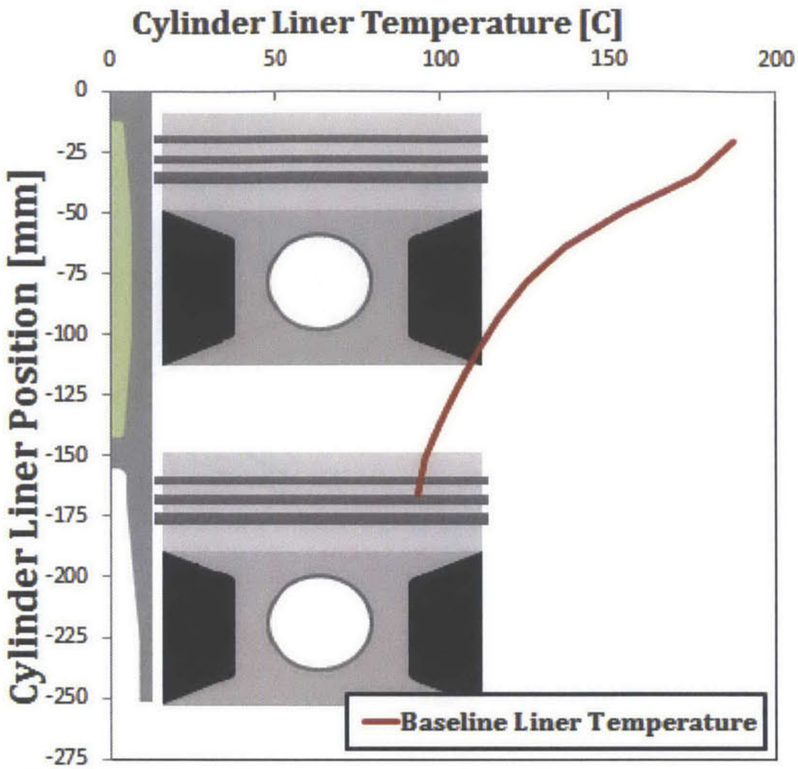

Figure 3.2 - Cylinder Liner Temperature at Peak Torque Operation Conditions

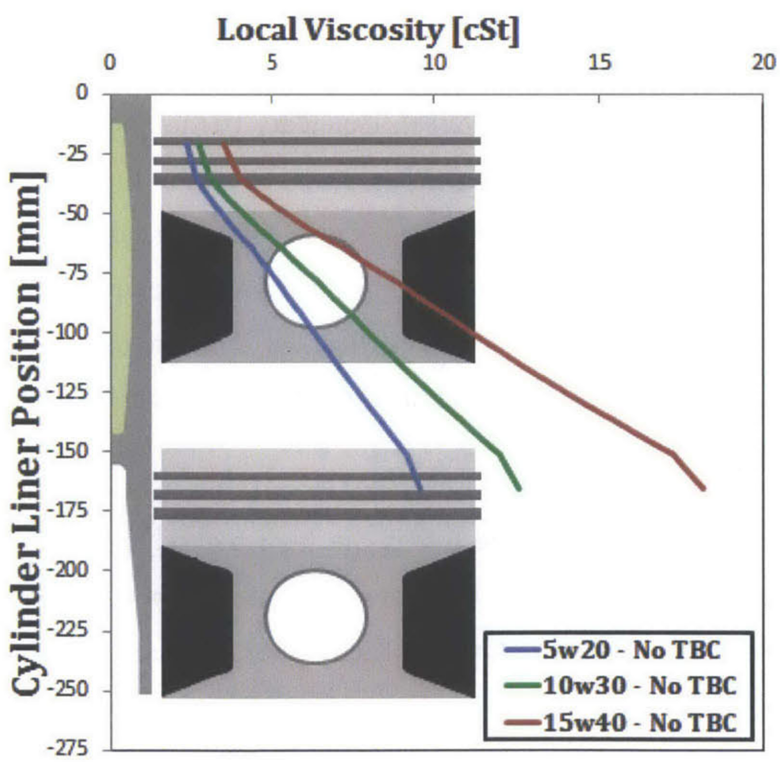

Figure 3.3 – In Situ Oil Viscosity at Peak Torque Operation Conditions

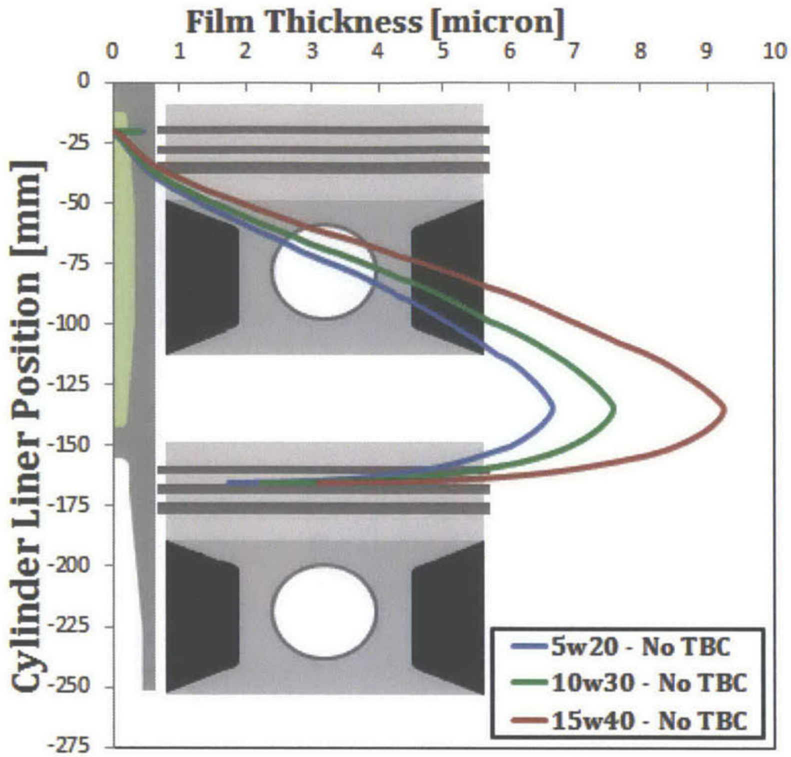

Figure 3.4 – Top Ring Oil Film Thickness at Peak Torque Operation Conditions

Figure 3.5 highlights two major concerns with power cylinder friction. The first concern is ring wear due to boundary friction at ring reversal. Boundary friction occurs near TDC during ring reversal occurs due to decreasing oil film thicknesses as shown in figure 3.4. Near TDC, the low viscosity, high cylinder pressures, and low speeds give rise to boundary friction and wear at the top of the stroke. Wear near TDC is unavoidable, but proper temperature profile design with TBC insulation will maintain and not affect wear. If TDC remains uninsulated and TDC temperatures are maintained, then approximately the same wear is expected, which will be shown later.

The second concern presented in figure 3.5 is the high hydrodynamic friction in mid-stroke. During the mid-stroke, the majority of power loss occurs due to the presence of significant levels of hydrodynamic friction. High hydrodynamic friction occurs due to high local oil viscosity and lower temperatures, as shown in figure 3.3. Figure 3.4 shows film thickness is near a maximum and boundary friction is an afterthought during mid-stroke due to this large film thickness.

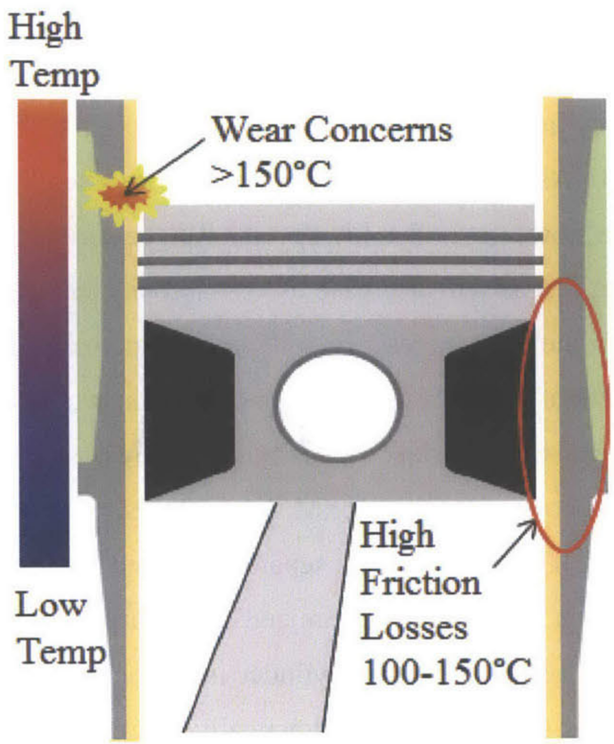

Figure 3.5 – Lubrication Concerns of Power Cylinder

Varying lubrication regimes illustrated by both high boundary friction at ring reversal and high power loss during mid-stroke illustrate the difficulty of solving power cylinder friction with lubricant formulation alone. Without an advanced additive package or viscometric profile design, the friction and wear needs of the power cylinder cannot concurrently be met. The wide range of lubrication regimes are cause by the diverse temperature, viscosity, and loads experienced as a function of crank angle or liner position. An in situ or local view of the power cylinder's lubrication needs is required to understand how engine component design can alter or control these conditions. Through lubricant property control, component design, and lubricant formula, the lubrication needs of the power cylinder can be met without affecting the engine's durability.

Local lubricant properties are of interest due to their influence on the lubrication regimes as the ring pack and skirt move up and down the liner. The knowledge of how lubricant properties affect the onset of different lubrication regimes will help target hydrodynamic friction and maintain boundary friction.

3.1.1 Ideal Engine Oil Needs

The needs of the power cylinder kit cannot be met when the bulk viscosity is decreased or the SAE oil grade is lowered. As bulk oil viscosity decreases, a tradeoff between friction reduction and wear increase in the power cylinder kit occurs. Ring pack wear occurs due to boundary friction near ring reversal. Wear will decrease as oil viscosity increases; while the majority of power losses further down the liner increase as oil viscosity increases. Depending on the selected engine, the boundary friction at ring reversal can be the main wear concern for power cylinder kit life and wear [2]. Decreasing the bulk viscosity will decrease the mid-stroke friction losses, but it will also decrease viscosity at ring reversal, which leads to a wear increase. To combat the friction and wear trade off, TBC applications separate the ring reversal and mid-stroke into two different lubrication regimes to create an optimized temperature profile/oil viscosity design. A proper application of TBC insulation on the cylinder liner does not affect the temperatures near ring reversal while the mid-stroke viscosity drastically decreases due to temperature increase near the TBC insulation.

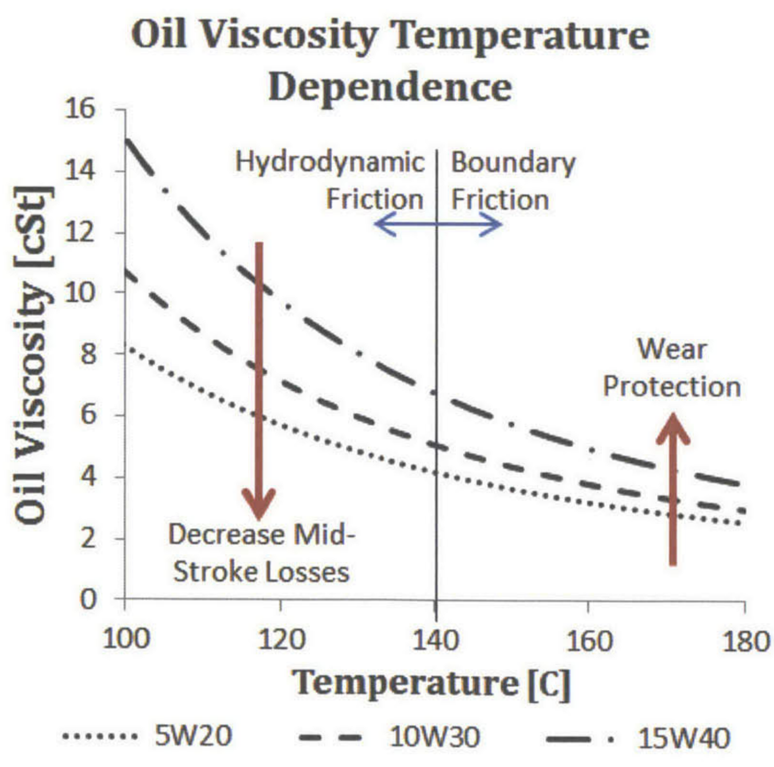

Figure 3.6 - Temperature Dependence of Oil Viscosity and the Ideal Power Cylinder Oil

Considering the local temperatures of seen by each of the power cylinder surfaces aids in the design of an ideal temperature profile. Figure 3.7 breaks down the temperature ranges experienced by the top ring versus the piston skirt. According to figure 3.7, temperatures above 140°C affect ring reversal while temperatures below 140°C affect hydrodynamic friction and the piston skirt. To avoid the tradeoff between friction and wear due to leveraging oil formulation alone, the oil viscosity near 100°C needs to decrease while increasing the oil viscosity near ring reversal temperatures such as 140°C. Figure 3.6 translates these ideal oil needs into an oil temperature and viscosity curve for viscometric design. The trends in friction and wear would suggest a flattening of the viscosity versus temperature curve or moving towards a completely temperature independent oil would be ideal for the power cylinder. Such oil does not exist, but insulating the cylinder liner can decrease local oil viscosity only in mid-stroke while maintaining the same TDC temperatures. Insulating the cylinder liner can meet the ideal needs of the power cylinder.

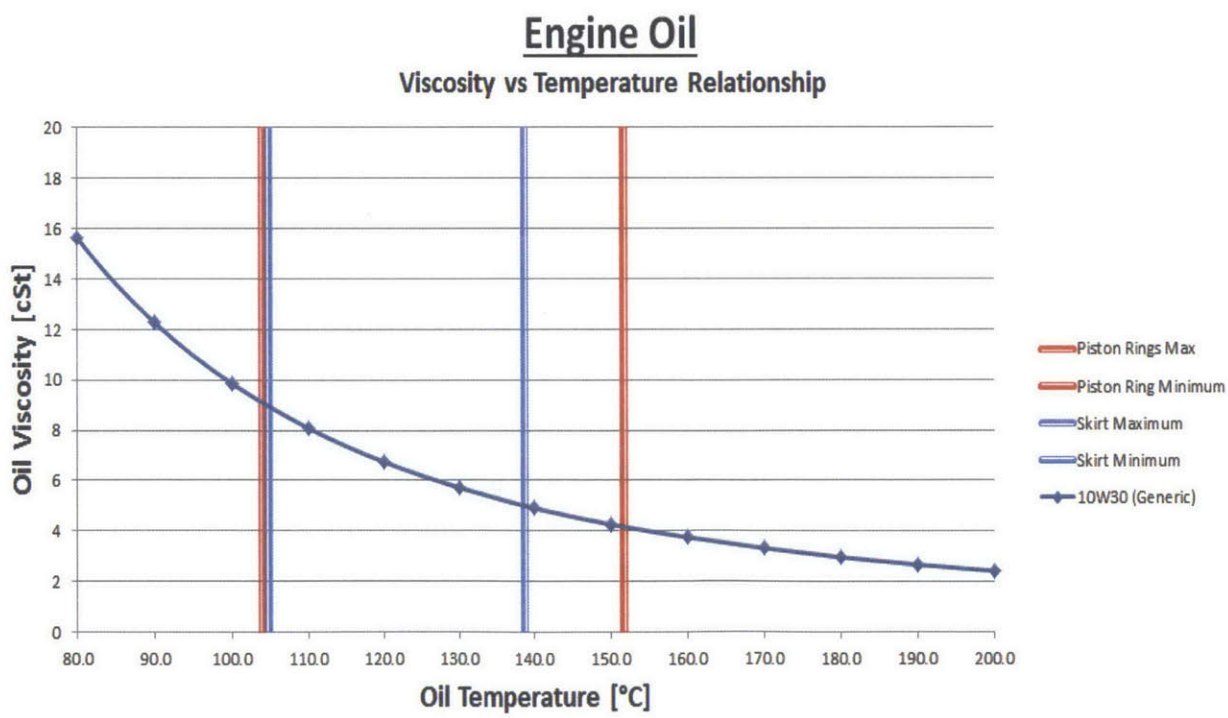

Figure 3.7 – Temperature Environment of Top Ring and Piston Skirt

Figure 3.8 presents a viscometric solution that would meet both the friction and wear needs of the power cylinder. Near ring reversal (above 140°C), the oil viscosity begins to thicken, and wear protection is improved. In the fuel economy region (100°C to 140°C), the viscosity is at a minimum to target hydrodynamic friction. This theoretical oil viscosity profile presents a possible solution to the ideal needs of the power cylinder. Yet, this theoretical oil viscometric profile is far from reality with current oil technology. This oil design strategy is presented merely to illustrate the challenges of meeting the engine's needs through oil formulation alone.

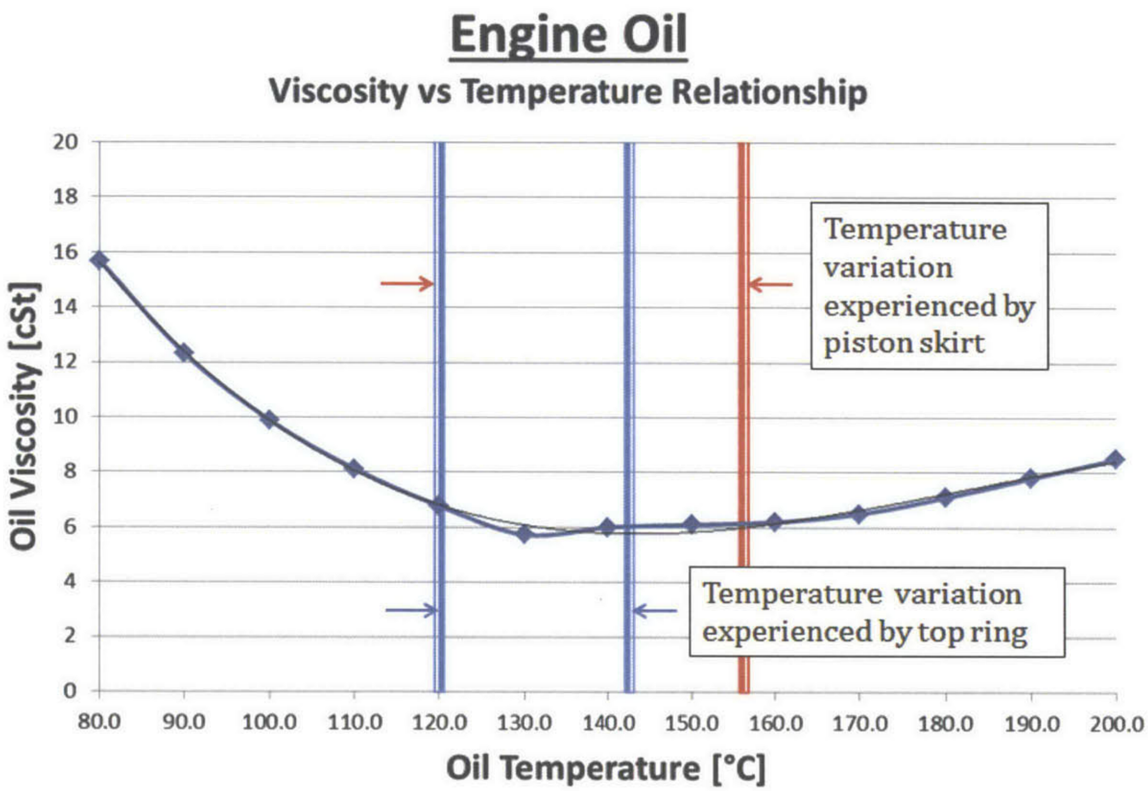

Figure 3.8 – Ideal "Durability" Oil Formulation

4. Thermal Barrier Coating Insulation

Thermal barrier coatings (TBC) are insulating materials, which are designed to cover metallic surfaces at high temperatures at which the metal would melt. In order to protect the metal surface from the high temperatures, TBC possess excellent insulation properties. Insulating components exposed to high temperatures creates a large delta in temperature between the combustion gases at high temperatures and the metal/TBC interface. TBC coatings have been applied in turbines to prevent the high post combustion temperatures and in diesel engines in hopes of producing a lower heat rejection engine.

4.1 TBC in Low Heat Rejection Engines

The application of thermal barrier coatings (TBC) is not new to the internal combustion engine. Previous studies have used thermal barrier coatings to create a low heat rejection engine. Of the energy input into the cylinder as fuel, approximately 60% of that fuel energy is sent out to the exhaust manifold or rejected to the coolant in the form of heat loss. In a low heat rejection engine, the efficiency increases by decreasing the heat rejection or energy loss to the coolant. This allows the piston to extract more of the useful energy released by combustion. Caution must be used when insulating the cylinder liner. If adequate adjustments are not made to the combustion inputs, peak cylinder temperatures and NOx emissions will increase. Prior art of low heat rejection engines using TBC insulation finds an increase in thermal efficiency of 5 to 7% [17].

4.2 Application of TBC for Friction Reduction

In this paper, thermal barrier coating is applied on the outside of the liner to raise local temperatures to reduce oil viscosity rather than modifying the heat rejection engine. Applying TBC insulation to reduce rubbing friction has been considered in previous TBC studies, but at the time, these studies were focused on developing a low heat rejection engine. Wong et al separates engine friction contributions into three categories a temperature dependent, surface dependent, and independent or constant friction. Twenty five percent of the total FMEP is from the temperature or viscosity dependent contribution [17]. By considering the oil temperature to be close to the surface temperature, Wong et al used scaling arguments to predict the benefit of an increased surface temperature assuming the oil does not degrade significantly at high temperatures and the oil fill thickness remains constant. The analysis in this paper will build on the concept of temperature dependent friction and apply TBC insulation to target the hydrodynamic friction in the power cylinder. Instead of using scaling arguments, a ring pack simulation is applied to find the friction and film thickness through application of the in situ lubricant properties.

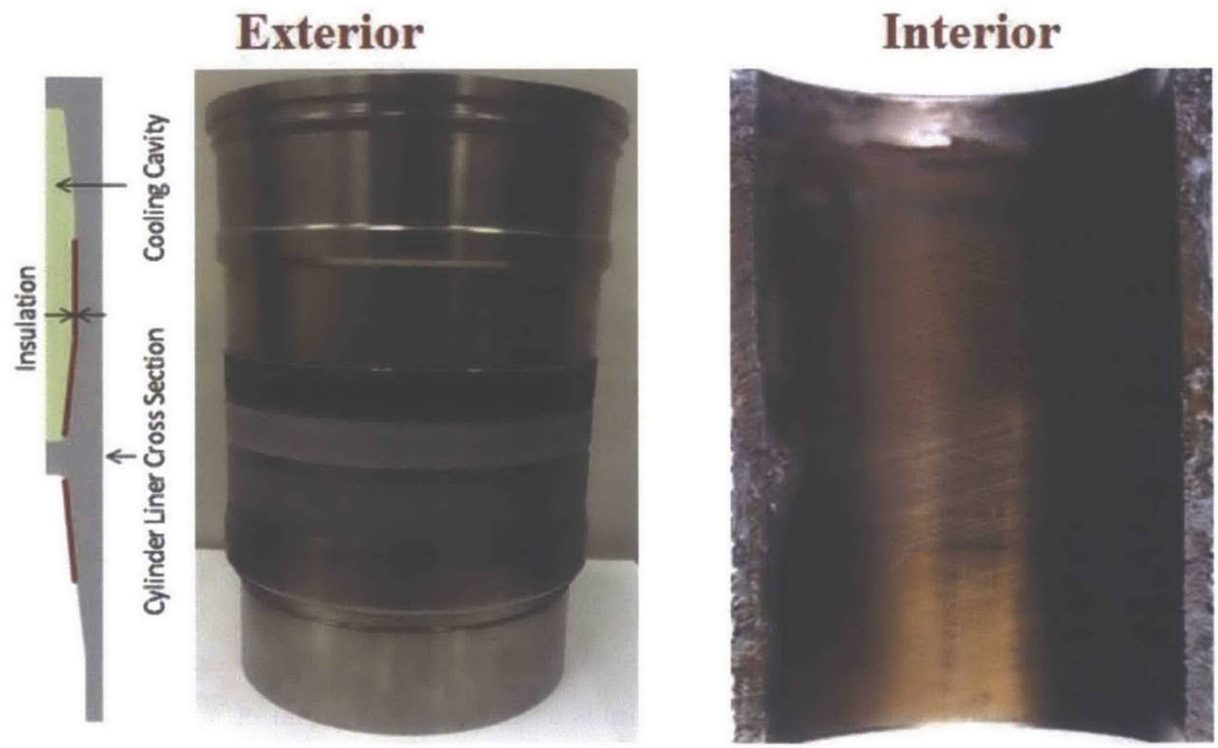

Figure 4.1 – Example of Thermal Barrier Coating on Cylinder Liner

4.3 Examples of Thermal Barrier Coatings

The required thermal barrier coating depends on the engine geometry and performance, which varies from application to application. This sub-section presents examples of thermal barrier coating materials used in during the TBC liner design. Discussions after this section will not necessarily select a specific coating since the coating selection will depend on the engine selected. A wide variety of TBC insulation materials exist. For this reason, this paper will focus on the resulting temperature profiles rather than the TBC profile used to produce a specific temperature profile. The reader can obtain similar temperature profiles with various coatings and thicknesses. The objective of this paper is to show the benefit of designing for a temperature profile along the cylinder liner and not to offer a design recommendation. Only the final solution will present a schematic of TBC thickness along the liner.

4.3.1 Plasma Spray Zirconia

Plasma spray zirconia (PSZ) is a coating commonly used in the insulation of gas turbine blades and historically used in low heat rejection applications. PSZ offers a low conductivity of 1.3 W/m-K, which makes it a desirable insulator. To apply PSZ coating, zirconia ceramic is heated until the ceramic melts and is sprayed on the metal surfaces in a thin layer. With current processes, the maximum thickness of PSZ is approximately 1 mm, so the presented solutions must remain under this thickness. Due to the high temperatures at the application surface, PSZ may warp a finished cylinder liner. If the cylinder liner warps, the liner manufacturer may consider honing the liner after the exterior of the liner is coated.

4.3.2 Ballistic or Syntactic Foam

Ballistic foam is a product supplied by Adiabatics Inc. of Columbus, Indiana. The epoxy-like mixture was developed to protect the underside of military vehicles from explosions, due to the high bond strength and high thermal resistance of coatings. The bond strength of ballistic foam is 70 MPa, and it can resist constant surface temperatures of 180°C at the surface. The small spheres with air are added to the epoxy base to increase the thermal resistance. Typical conductivity values for ballistic foam range from 0.25 to 0.4 W/m-K. Ballistic foam can be machined when applied on the outside of the cylinder liner. Figure 4.2 shows an example of the ballistic foam epoxy applied to the outside of an aluminum pipe with varying amounts of air spheres.

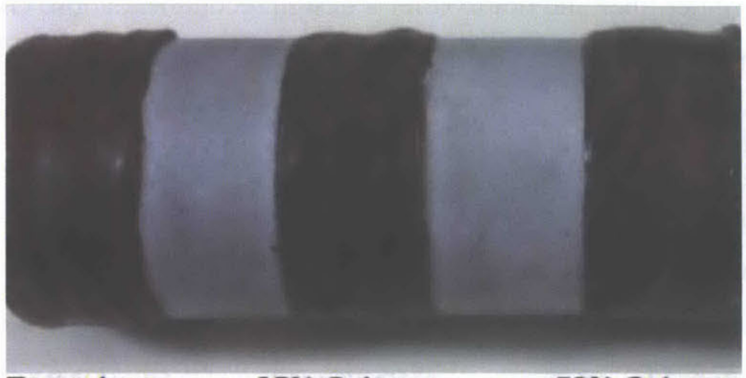

Figure 4.2 – Example of Ballistic Foam on Aluminum Pipe

4.3.3 Aluminum-Silica Sheet

Aluminum blanket or alumina-silica sheet offers the best thermal resistance of the three TBC materials at 0.2 W/m-K conductivity. Similar to traditional insulation, the alumina blanket consists of a fibrous woven aluminum silicate. Yet this insulation must withstand the conditions present in the cooling jacket so a polyamide-imide (PAI) sealant is coated on the outside of the fibrous blanket. After further engine tests, the aluminum blanket and PAI sealant do not withstand the acidity of coolant and break down quickly. An example of the aluminum blanket with PAI sealant is presented in figure 4.3.

Figure 4.3 - Example of Aluminum Blanket on Aluminum Pipe

4.4 Effectiveness of Thermal Barrier Coatings

Figure 4.4 shows the relative thermal resistance of PSZ. The application of 0.23 mm of PSZ insulation on the outside of the cylinder liner has the same thermal resistance as 10 mm of cast iron. This example of PSZ's relatively high thermal resistance illustrates why TBC insulation was chosen to change the local heat transfer rather than changing other components of the cooling jacket.

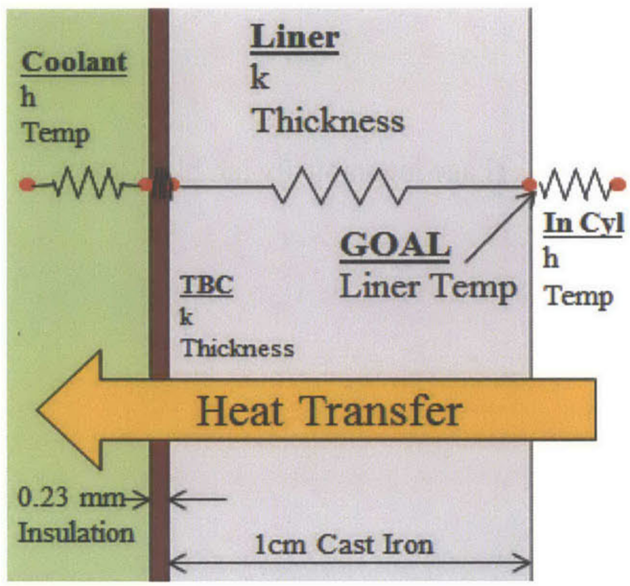

Figure 4.4 – Plasma Spray Zirconia TBC - Thermal Resistance Comparison

(Page intentionally left blank)

5. Power Cylinder Modelling

The goal of insulating the cylinder liner with TBC insulation is to decrease friction without an increase in wear of power cylinder components. For this reason, the design of the TBC insulated liner requires an analysis that can assess both friction and wear of the piston assembly. The power cylinder analysis is comprised of several simulations that translate how insulation affects local temperatures, viscosity, and friction. This section will cover the workflow and basic concepts of the models used to predict a TBC insulated cylinder liner's friction reduction. The work flow chart in figure 5.1 provides an overview of the analyses completed to design the TBC insulated cylinder liners. First, the desired temperature profile on the inside of the cylinder liner is calculated using an in-cylinder heat transfer model. The temperature profile determines the local viscosity along the liner for the power cylinder lubrication analysis. The results of the in-cylinder heat transfer model also provide the wall temperatures of the piston crown and cylinder head temperatures. Once the local viscosity along the liner is known, the ring pack model finds the power cylinder's contributions to both hydrodynamic friction and boundary friction. The boundary friction component is used to calculate the power cylinder wear, while both hydrodynamic and boundary friction are applied to find friction reduction. Film thickness, local viscosity, and shear rate are also obtained from the ring pack program to provide supporting evidence for the hydrodynamic and boundary friction calculations.

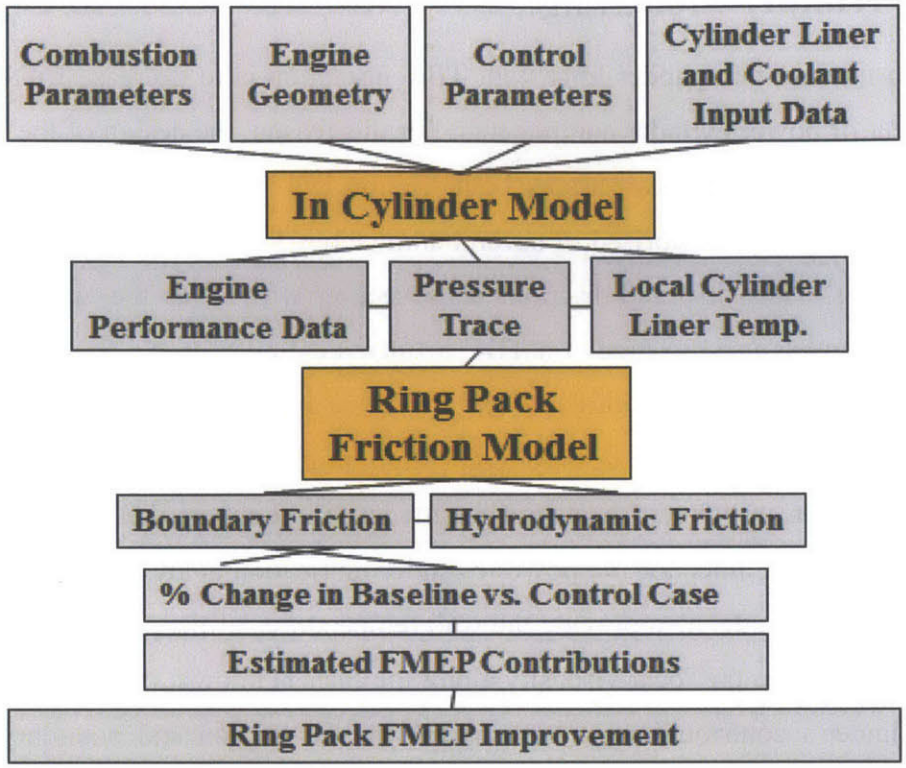

Figure 5.1 - Work Flow Diagram of Power Cylinder Simulation

5.1 In Cylinder Heat Transfer Simulation

The in-cylinder heat transfer model consists of a simple mathematical model for cylinder pressure and temperature. A schematic of the in-cylinder heat transfer model is presented in figure 5.2. The heat transfer rate at every crank angle is calculated using the combustion gas temperature and applying the theory of thermal circuits for the gas, piston, cylinder head, and cylinder liner. The in-cylinder model used in this study [18] divides the portion of the cylinder liner exposed to combustion gases into 10 segments to find the surface temperatures and heat transfer rates along the cylinder liner. Each cylinder liner segment is comprised of a base metal and insulation as shown in figure 5.3. The TBC can be place on the inside or outside of the base metal. For the analysis presented in this paper, the TBC is always applied to the outside of the cylinder liner. By applying material to the outside of the liner, the cross hatching on the inside of the cylinder liner is maintained.

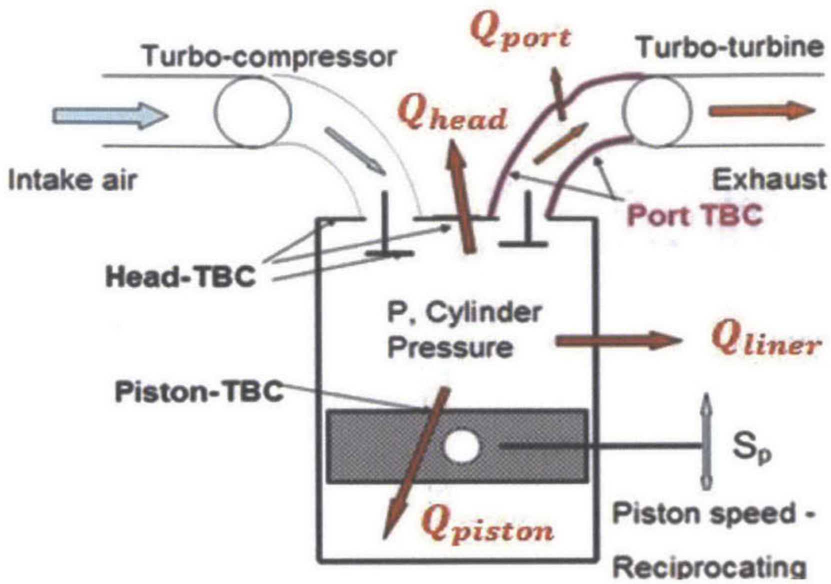

Figure 5.2 – In Cylinder Model - Overview

5.1.1 In Cylinder Model - Governing Equations

The in-cylinder heat transfer model is governed by the First Law of Thermodynamics, equation (5.1), applied to the combustion chamber. The basic equation for the first law can be expanded by defining the different components of work and heat transfer. The first term is the energy stored within the control volume. The energy stored relates to the pressure rise within the cylinder. The heat transfer has two components that include the heat released by combustion and the heat rejected to the coolant, oil, and block. The work extracted by the piston is related to the expansion of the combustion chamber and is included in the term with the expansion rate of the control volume. Applying the ideal gas law simplifies these equations so the gas constant and constant specific heats can be applied. From this equation, the cylinder pressure and pressure rise is calculated.

$$\frac{dE_{cv}}{dt} = \frac{dQ_{cv}}{dt} - \frac{dW_{cv}}{dt} + \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} \tag{5.1}$$

Energy Storage **Heat Transfer** **Work Transfer**

$$\left(\frac{C_v}{R}\right) V \frac{dp}{d\theta} = \left(\frac{dQ_{ch}}{d\theta} - \frac{dQ_{ht}}{d\theta}\right) - \left(\frac{C_v}{R+1}\right) p \frac{dV}{d\theta} \tag{5.2}$$

Equation (5.2) is used to calculate the cylinder pressure versus time where:

- p is the cylinder pressure
- V is the cylinder volume
- θ is the crank angle
- C_v is the specific heat at constant volume
- R is the gas constant
- Q_{ch} is the heat release from combustion
- Q_{ht} is the heat rejection to the surroundings

5.1.2 In Cylinder Model – Combustion Calculations

To calculate energy released by combustion, a curve fit based on experimental inputs is applied. Equation (5.3) is a Wiebe function that models the amount of fuel burned based on the crank angle θ relative to the start of combustion θ_{soc} and the 90% burn duration θ_d. Start of combustion is known based on the engine's designated injection timing plus ignition delay, which depends on the in-cylinder conditions. The burn duration can be analytically obtained or experimentally measured. For this paper, the burn duration is measured off a heavy-duty diesel engine. Once the burn rate of the fuel is known, the heat release rate from combustion is applied to equation (5.2)

$$x_b(\theta) = 1 - exp\left[-a\left(\frac{\theta - \theta_S}{\theta_d}\right)^n\right] \tag{5.3}$$

5.1.3 In Cylinder Model – Heat Transfer

The total heat rejection needed for equation (5.2) is comprised of the cylinder liner, cylinder head, and piston heat transfer. The in-cylinder model breaks down these paths into 12 distinct paths for heat transfer, which include the piston crown, cylinder head, and 10 segments on the cylinder liner. The cylinder liner is broken into 10 segments axially up and down the cylinder liner for the length of the stroke as shown in figure 5.3. Each liner segment can communicate axially and radially to create 2D mesh of thermal circuits along the cylinder liner.

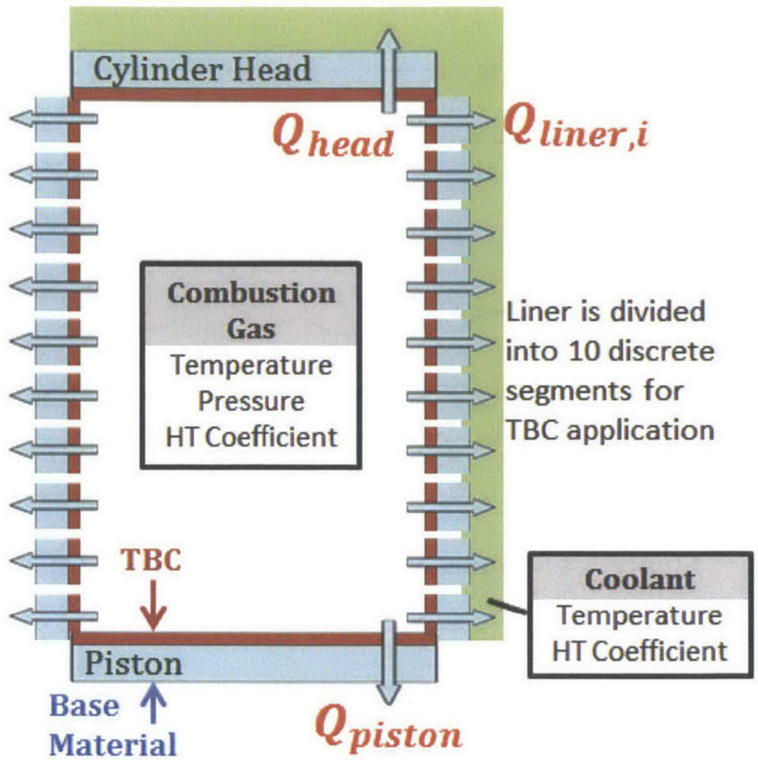

Figure 5.3- In-Cylinder Model - Heat Transfer

5.2 Piston Ring Pack Friction Model

The ring pack model utilizes a modified Reynolds' lubrication equation applied to the ring and liner interface shown in figure 5.4. The modified Reynolds' lubrication equation was developed by Patir and Cheng and later used by Tian [19]. Figure 5.4 illustrates the interface modeled in the Takata et al ring pack model. Metal on metal contact pressure was estimated by applying curve fits from Greenwood and Tripp's asperity contact model, as used by Hu [20].

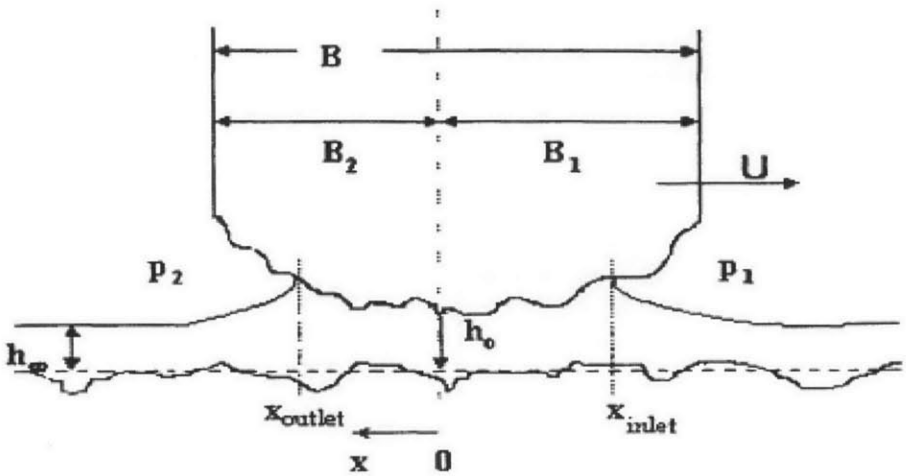

Figure 5.4 – Model of Ring Pack and Liner [21]

5.2.1 Hydrodynamic Friction – Reynold's Equation

To calculate the hydrodynamics friction losses, the Reynold's equation is applied to the ring sliding over the oil film on the cylinder liner. The Reynold's equation only applies in the hydrodynamic and mix lubrication regimes, because there must be oil flowing underneath the ring for the equation to apply. Equation (5.4) is the Reynold's lubrication equation. The pressure profile under the ring and film thickness is solved from equation (5.4). Once the pressure profile under the ring is known, the normal and friction force acting on the ring can be solved.

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right) = -6U\frac{\partial h}{\partial x} + 12\frac{\partial h}{\partial t} \qquad (5.4)$$

Where:
P is the local pressure under the ring
U is the surface or piston velocity
h is the film thickness height between the ring and liner
μ is the oil viscosity

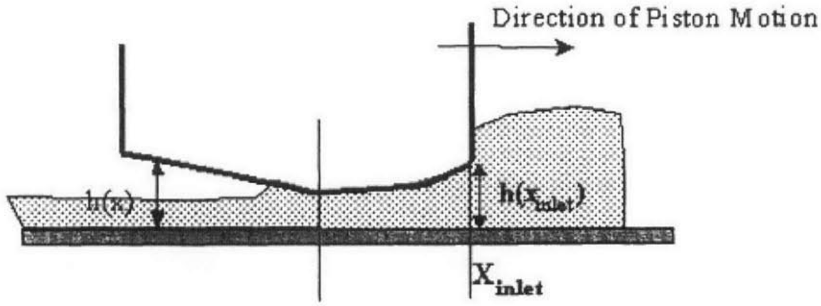

Figure 5.5 – Illustration of Fully-Flooded Inlet Condition [21]

[21] In order to solve the Reynold's equation, several boundary conditions must be satisfied. The first two boundary conditions state the pressure under the ring at the inlet and outlet must be the same as the gas pressure at the inlet and outlet respectively. For the top ring, this would mean the combustion chamber pressure is the inlet pressure and the pressure at the first land of the piston is the outlet pressure on the up stroke. The third boundary condition states the mass flow rate at any point under the ring must be equal to the oil flow rate entering the inlet of the ring. Boundary condition #3 satisfies the conservation of mass. Assuming a fully flooded ring or adequate oil supply is present, the oil film thickness at the leading edge of the ring is high enough to meet the front of the ring's barrel shape profile as shown in figure 5.5.

$$\boldsymbol{BC\ \#1\!:}\ \ \boldsymbol{P(x|_{inlet}) = P_1} \tag{5.5}$$

$$\boldsymbol{BC\ \#2\!:}\ \ \boldsymbol{P(x|_{outlet}) = P_2} \tag{5.6}$$

$$\boldsymbol{BC\ \#3\!:}\ \ \boldsymbol{Q(x|_{inlet}) = U \cdot h(x|_{inlet})} \tag{5.7}$$

$$\boldsymbol{BC\ \#4a\!:}\ \ \boldsymbol{dP/dx|_{outlet} = 0} \tag{5.8}$$

$$\textbf{BC \#4b:}\ \ \boldsymbol{Q(x|_{inlet}) = a = h(x|_{outlet}) \cdot \frac{dx}{dt}} \tag{5.9}$$

The final boundary condition depends on the location of the piston [21]. In the middle of the stroke, a classic Reynold's boundary condition applies to the outlet of the ring. The Reynold's boundary condition states that pressure does not change from the ring outlet to the combustion gas, or that the slope of the pressure under the ring in zero. When the ring approaches TDC or

BDC and boundary friction is present, a non-separation boundary condition is applied [22]. According to Takata et al, the non-separation boundary condition "assumes that all of the oil exiting the ring/liner interface at the outlet stays attached to the ring, where it accumulates."

5.2.2 Boundary Friction - Asperity Contact Model

If the film thickness under the ring is too small, then the ring and liner surfaces will contact and boundary lubrication will occur. The onset of boundary friction occurs when the asperities of two surfaces come in contact. Since the profiles of the metallic surface and asperities is random and depends on the manufacturing processes, a statistical approach is used to correlate when contact will occur for two surfaces with a measured composite roughness. Greenwood and Tripp's model is applied to find the contact pressure based on the distance between two surfaces. This contact pressure provides the normal force between the ring and the liner for friction force calculations.

$$P_c = \begin{cases} K'E'A\left(\Omega - \frac{h}{\sigma}\right)^z & if: \frac{h}{\sigma} \leq \Omega \quad \text{(Boundary Lubrication)} \\ 0 & if: \frac{h}{\sigma} > \Omega \quad \text{(Hydrodynamic Lubrication)} \end{cases} \tag{5.10}$$

5.3 Engine Description and Operating Points

A generic 11.0L heavy-duty turbo diesel was applied to the power cylinder and heat models previously mentioned. This engine represents current technology in the heavy-duty industry, but is not specific to any OEM to prevent issues with confidential data. Specifications of the engine are listed in table 5-1. The torque curve for the engine is presented in figure 5.6. The models were run at peak torque, which produces the hottest liner temperatures. The peak liner temperatures produce the upper design limit of the TBC liner due to temperature limits of the components, TBC material, and engine oil.

Table 5-1 - Heavy Duty Turbo Diesel Specifications

Description	Value	Units
Displacement	11.0	L
Cylinders	Inline 6	#
Bore	125	mm
Stroke	150	mm
Con Rod Length	240	mm
Peak Torque	1350 ft-lbs @ 1100 RPM	
Rated Power	350 hp @ 2000 RPM	

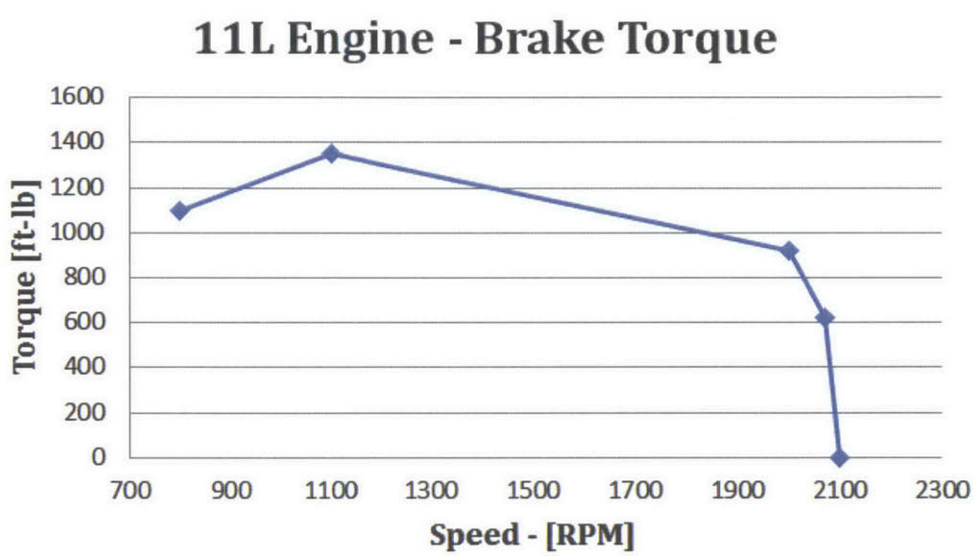

Figure 5.6 – Torque Curve for Selected 11L Engine

5.4 Thermal Barrier Coating Analysis

The thermal barrier coating analyses completed presents a TBC profile that is optimized for friction reduction and wear concern. The engine speed and load were set at peak torque. The analyses present two parametric studies that were completed to find the optimal coating location along the cylinder liner. The criterions used to ensure an optimized coating location include FMEP and a wear parameter to guarantee the TBC liner can reduce friction while maintaining wear superior to a low viscous oil. These criterions are introduced in the following paragraphs.

5.4.1 Criterion to Maintain Power Cylinder / Liner Wear

As bulk oil viscosity decreases, boundary friction within the power cylinder increases. This tradeoff in friction and wear due to bulk viscosity is unavoidable through means of viscometrics alone. While oil manufacturers can avoid this problem with oil additives such as tribofilm forming friction modifiers [2], the TBC insulated liner does not change the oil composition. To capture this tradeoff, the criterions used in these analyses ensure the optimal friction reduction design captures both a friction reduction and wear. If implemented correctly, the TBC insulated liner will not increase wear.

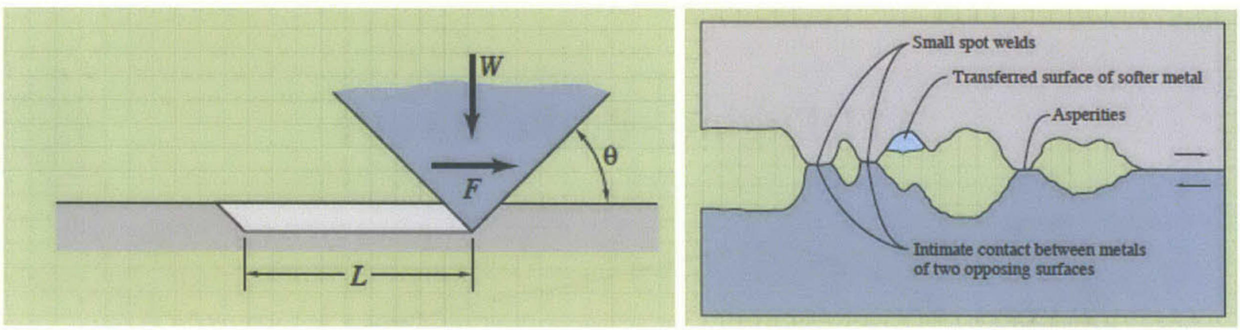

Figure 5.7 – Archard Wear Model [23]

To capture the possible increase in wear due to higher temperatures and lower viscosity near TDC, a wear parameter for metal-on-metal contact, specifically adhesive wear, is applied in the analyses to follow. Figure 5.7 illustrates the mechanism and a simplified view of adhesive wear, which is applied in the Archard wear model. The wear rate due to adhesive wear is captured by the Archard wear equation shown in equation 1. Instead of directly using the Archard wear equation and calculating wear volume and component life, a ratio of the magnitude of maximum boundary friction power loss for the TBC insulated liner and the baseline case is considered. Equation 2 captures the wear criterion that will be referred to as ~~the~~ wear factor. This ratio captures the increase in wear for the TBC case versus the baseline engine. This method was selected over methods such as an average film thickness near TDC [21] because wear is a local phenomenon. A failure due to excessive wear occurs only at the maximum point. It does not matter what the wear rate is in the surrounding area, but only at the first location to fail. Furthermore, the conclusions drawn from this equation can only comment on the metal-on-metal rubbing between the face of the rings/skirt and the cylinder liner. Other forms of wear between additional surfaces, such as the ring and the ring groove will not be considered, because the analyses completed do not calculate the forces at the ring / ring groove interface.

Equation 1 - Archard Equation for Adhesive Wear

$$Q = \frac{KWL}{H} \qquad or \qquad \dot{Q} = \frac{KWV_{pist}}{H}$$

Q is wear volume	W is load
H is hardness of soft material	L is sliding distance
V_{pist} is piston velocity	K is wear constant

Equation 2 - Wear Factor

$$\%Increase(Wear) = \frac{Local\ Power\ Loss\ Due\ to\ Boundary\ Friction\ Only}{MAX(Power\ Loss\ Due\ to\ Boundary\ Friction\ for\ Baseline)}$$

5.4.2 Criterion to Decrease Rubbing Friction

The purpose of the in situ lubricant analysis is to capture a friction reduction in the power cylinder components. To quantify the friction reduction benefit for each case, the FMEP for the piston rings and skirt was calculated. An improvement in BsFC is calculated using equation 3. By assuming fixed contributions from the ring pack, skirt, and power cylinder to total mechanical friction, the BsFC improvement scales with the calculated power cylinder improvement. The assumed contributions of engine friction are based off ranges from Richardson et al and presented in figure 5.8.

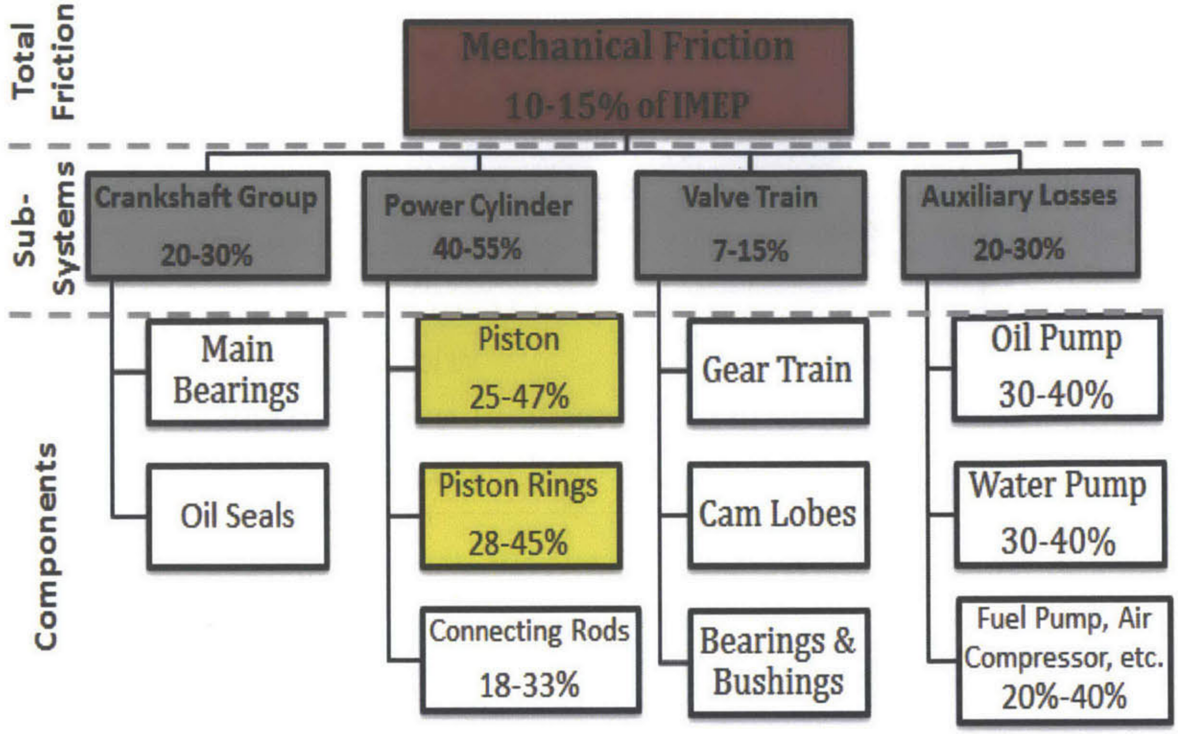

Figure 5.8 - Typical Contributions to Mechanical Friction in Diesel Engine [8]

Equation 3 - BsFC Improvement

$$\%\ BsFC\ Improvement = (\%Impr.RP)\left(\frac{RP\ Friction}{PC\ Friction}\right)\left(\frac{PC\ Friction}{Total\ Friction}\right)(1-\eta_{mech})$$

Where:

$(\%Impr.RP) =$ Calculated improvement in ring pack friction

$\left(\frac{RP\ Friction}{PC\ Friction}\right) =$ Assumed contribution of ring pack friction to power cylinder friction

$\left(\frac{PC\ Friction}{Total\ Friction}\right) =$ Assumed contribution of power cylinder friction to total mechanical friction

$\eta_{mech} =$ Mechanical Efficiency

6. TBC Insulated Liner – Constant Insulation Thickness [24]

6.1 Parametric Study #1 – Varying TBC Insulation Length

Two parametric studies were completed to develop conclusions on guidelines for coating the cylinder liner with TBC insulation. The first parametric study investigates the effect of the insulation length down the cylinder liner while holding the starting location of the insulation constant. This investigation develops conclusions on the ending location of the TBC insulation. Parametric study #1 attempts to prove the length of TBC insulation down the cylinder liner has a maximum BsFC benefit, or a local maximum exists when the insulation spans a given starting location down to BDC. The engine performance and component limits were also considered in this study by observing the changes in heat rejection to the coolant and piston temperature.

To simplify the insulation length parametric study, a fixed insulation starting point was chosen on the outside of the cylinder liner. The insulation length down the cylinder liner was increased and a BsFC improvement due to insulation was calculated. It is important to note that as the insulation length increased, the maximum liner temperature of 220°C was maintained by changing the insulation thickness. This ensures the design limits for liner temperature and oil temperature are not exceeded.

Results from Parametric Study #1 – Varying TBC Insulation Length

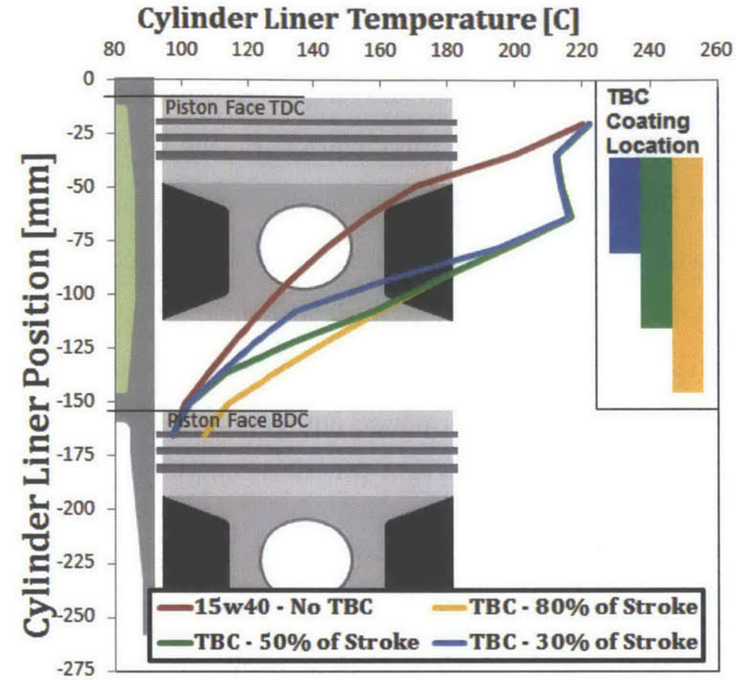

Figure 6.1 - TBC Insulation Length Study – TBC and Baseline Liner Temperature Profiles

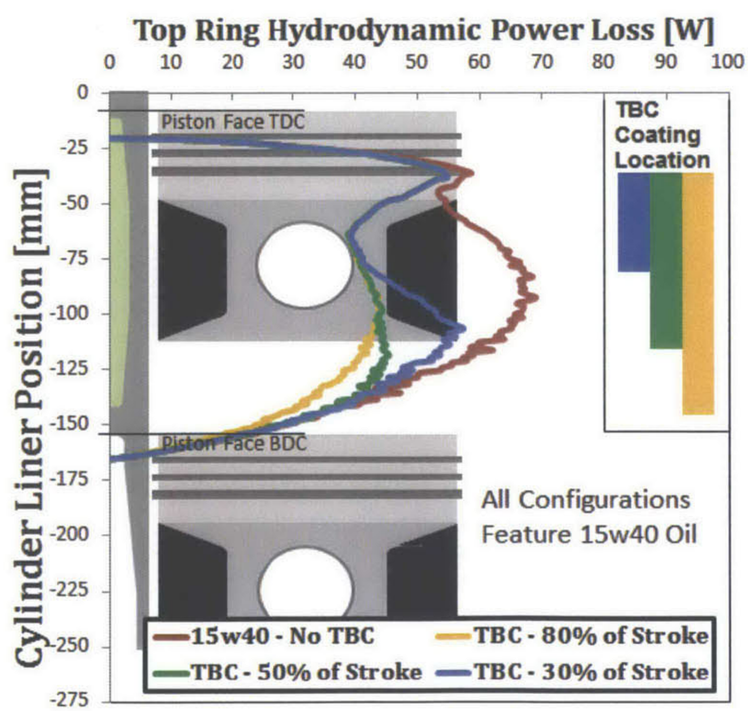

Figure 6.2 – TBC Insulation Length Study – Top Ring Hydrodynamic Friction (Only Expansion Stroke is shown)

Figure 6.1 shows the cylinder liner temperature profile for varying the length of insulation. The change in temperature from the baseline cylinder liner temperature profile affects the local viscosity so that the mid-stroke viscosity is well below viscosity oil at mid-stroke even with 15w40 oil combined with the insulated cylinder liner. The significantly lower oil viscosity decreases friction along the cylinder liner as shown in figure 6.2. It should also be noted that the TBC insulated liners decrease the mid stroke power losses significantly from the 15w40 baseline case. According to figure 6.3, the ring pack friction improves from 12.5% for the shallowest case, to 21.7% for the longest TBC insulation length. The reductions in ring pack FMEP correspond to a 0.25%, 0.35%, and 0.43% BsFC improvement respectively.

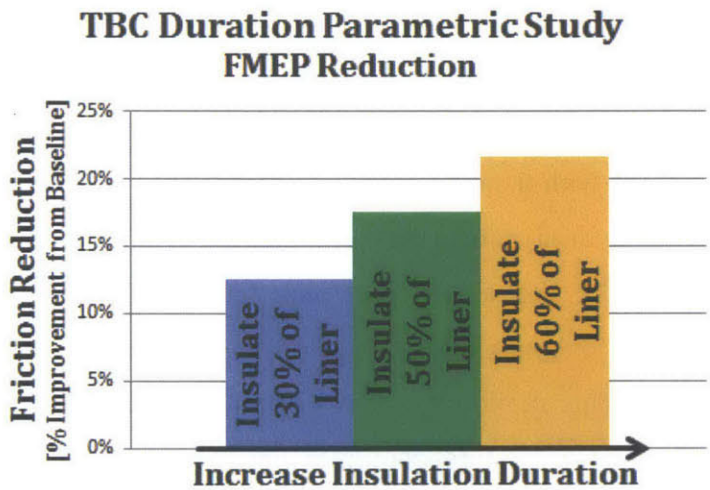

Figure 6.3 – TBC Insulation Length Study – FMEP Improvement

6.1.1 Conclusions from Parametric Study #1 – Varying TBC Insulation Length

The results of the TBC insulation length parametric study show that as the coating length down the cylinder liner increases, the ring pack friction decreases. This trend leads to the conclusion that the full length of the cylinder liner should be coated with TBC insulation for the maximum fuel economy improvement. Applying this conclusion, the remainder of the TBC investigations will coat the TBC insulation the specified starting location to the end of the stroke to ensure maximum BsFC improvement as found in parametric study #1.

6.2 Parametric Study #2 – Varying TBC Insulation starting location

Continuing with the conclusions from the first parametric study, the distance from the cylinder head at which the TBC insulation is first applied is varied. Per the conclusion of parametric study #1, the length of TBC coating extends to the end of the stroke for all cases in the second parametric study. The expected outcome of this parametric study is that a maximum benefit of TBC will exist due to the location of maximum temperature coinciding with the location of maximum power loss

Results from Parametric Study #2– Varying TBC Insulation starting location

The results of the TBC starting point parametric study are shown in figures 6.4 through 6.6. Figure 6.4 confirms delaying the TBC insulation starting point also delays the maximum temperature further down the cylinder liner. For maximum friction reduction, the maximum temperature should coincide with the maximum instantaneous hydrodynamic friction. Figure 6.5 presents the instantaneous hydrodynamic friction along the cylinder liner. If the TBC insulation coats the bottom 50% of the stroke, the maximum hydrodynamic friction reduction can be achieved as seen in figure 6.5. A maximum benefit by changing the starting location of the TBC is observed in figure 6.6. This maximum occurs due to the diminishing benefit of the TBC insulation if the maximum temperature continues past the point of maximum instantaneous hydrodynamic friction.

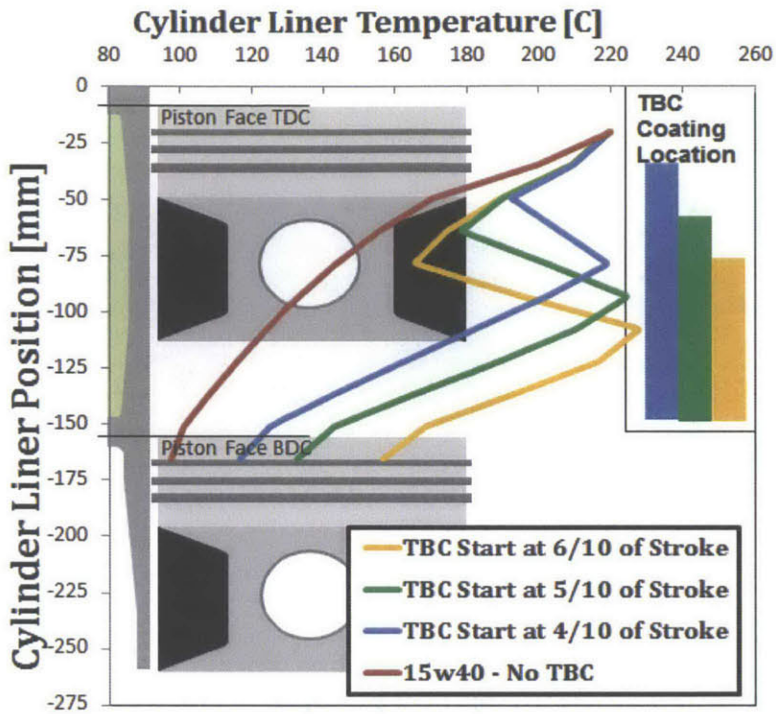

Figure 6.4 – Varying TBC Insulation Starting Position – TBC and Baseline Liner Temperature Profiles

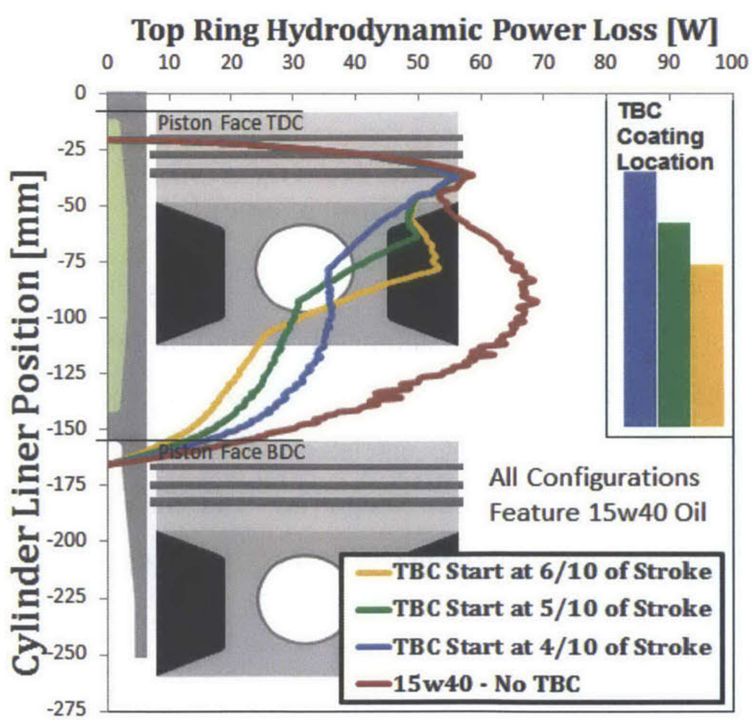

Figure 6.5 – Varying TBC Insulation Starting Position – Top Ring Hydrodynamic Friction (Only Expansion Stroke is shown)

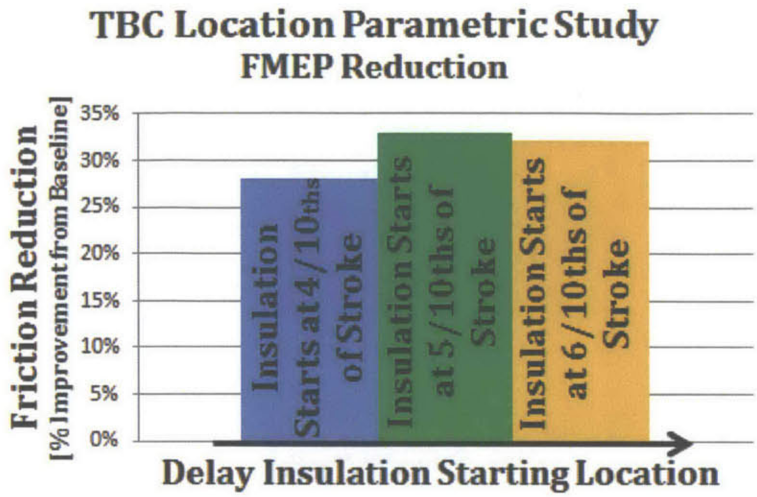

Figure 6.6 – Varying TBC Insulation Starting Position – FMEP Improvement

6.2.1 Conclusions from Parametric Study #2– Varying TBC Insulation starting location

By varying the starting position of the TBC insulation, it can be observed that there is a point of maximum BsFC improvement as shown in figure 6.6. If the application zones of the TBC are too high or too low, the maximum TBC benefit may not be obtained. The TBC insulation should be positioned so the maximum temperature occurs at the point of maximum hydrodynamic friction for the best BsFC improvement.

6.3 Optimized TBC Design FOR BsFC improvement

The optimized TBC insulated cylinder liner was designed using the conclusions from the two previous parametric studies in which the optimal TBC starting position and insulation length were determined. This section presents the optimal design recommended from the two parametric studies and compares the TBC insulated cylinder liner to low viscosity oils for a relative comparison of friction and wear. Supporting data for the film thickness and local viscosity are also presented, along with the in situ friction loss plots.

6.3.1 Optimal TBC Cylinder Liner Design – FMEP Benefit

The first goal of the TBC insulated cylinder liner is to significant reduction in power cylinder friction. To evaluate the effectiveness of TBC insulation at reducing friction, the estimated BsFC improvement is used. The final selection of a TBC insulated cylinder liner is plotted along with 15w40 and 5w20 for comparison. Figure 6.7 compares the capability of the TBC insulated cylinder liner and 5w20 oil to reduce hydrodynamic friction. A closer look at viscosity in figure 6.8 shows the local viscosity of a TBC insulated liner is much less than 5w20 oil in mid-stroke. For this reason, the TBC offers superior friction reduction to 5w20 oil. The TBC insulated liner offers 17.78 W/cylinder average power losses, while 15w40 and 5w20 offer 26.55 W/cylinder and 22.96 W/cylinder average power losses respectively. That corresponds to a 33.0% improvement for the TBC insulated liner. The predicted BsFC benefit from the TBC is 0.7% BsFC at peak torque. The comparison of BsFC improvement for a low viscosity oil and TBC insulated liner is presented in figure 6.9.

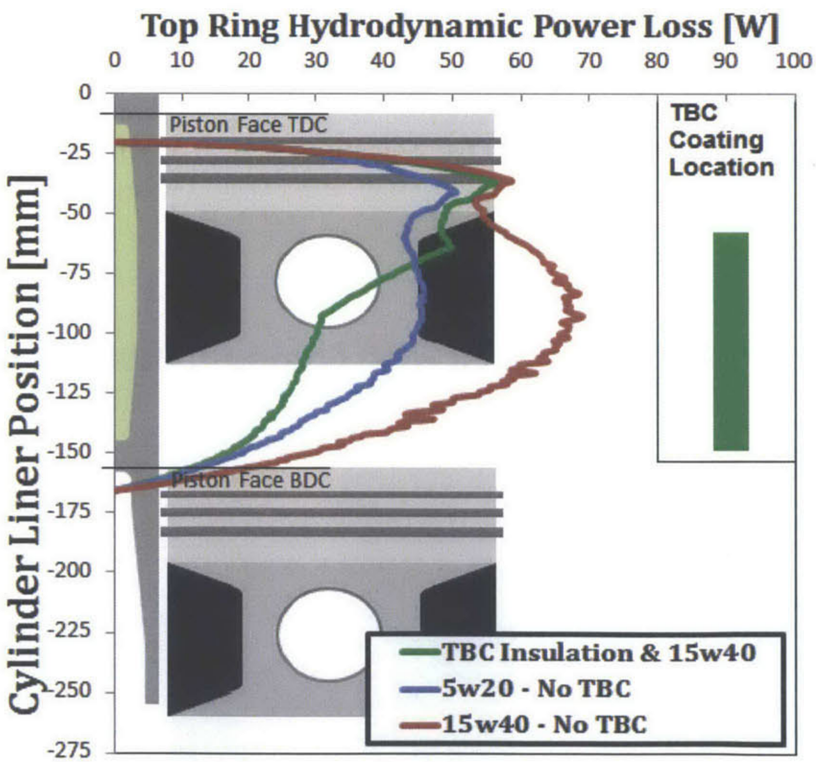

Figure 6.7 – Final TBC Selection – Top Ring Hydrodynamic Friction

(Only Expansion Stroke is shown)

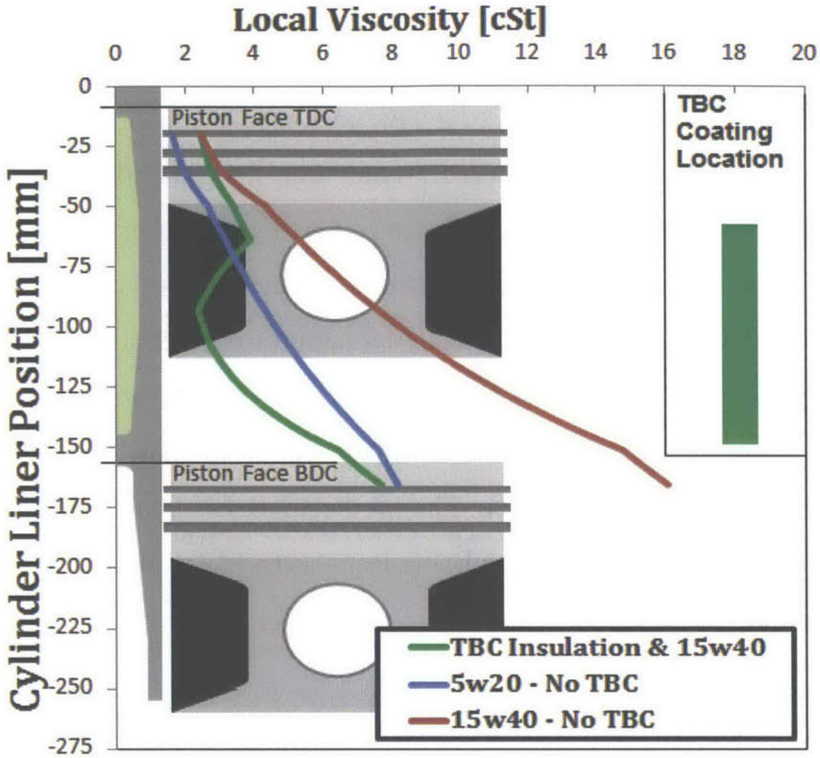

Figure 6.8 - Final TBC Selection – Local Viscosity

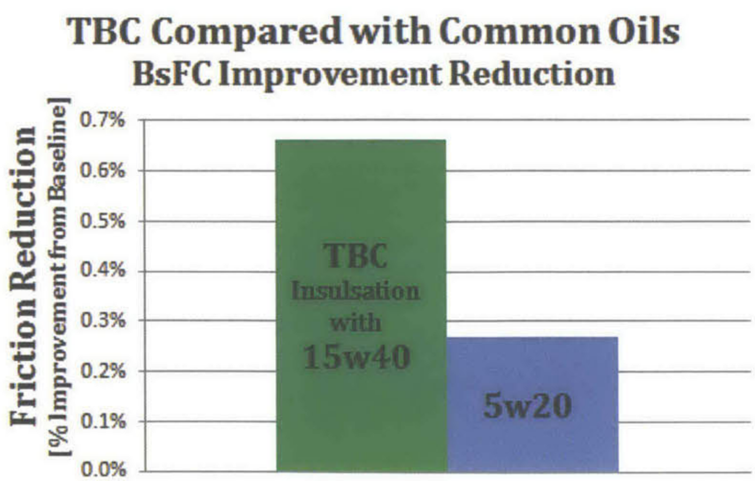

Figure 6.9 - Final TBC Selection – FMEP

6.3.2 Optimal TBC Cylinder Liner Design – Wear

The TBC insulated cylinder liner cannot only reduce friction. The TBC insulation must also offer a superior solution with neutral wear impact. To evaluate the power cylinder wear, compare the boundary friction scaling of the Archard wear equation for the baseline and TBC cases. Figure 6.10 presents the boundary friction for the three cases of engine oils and the TBC liner. A close inspection of figure 6.10 shows the boundary friction for the TBC case and the 15w40 case almost constant. Figure 6.11 compares the increase in wear from boundary friction (figure 6.10) according to Archard's wear equation. The TBC insulated cylinder liner exhibits similar wear to the 15w40 with a 1.3% increase from 15w40 to TBC insulation. The wear for the 5w20 oil increases by 34.5% for the same condition assuming the oil additive package does not change. The TBC insulated liner achieves 0.7% BsFC improvement with constant wear and superior ability to decreasing bulk oil viscosity. Figure 6.12 explains the origins of these benefits. Near TDC, the film thickness for 15w40 and TBC insulation are equal and offer similar wear protection. The film thickness of 5w20 is slightly lower, which causes an increase in boundary friction. In mid-stroke, the film thickness is significantly decreased, which corresponds to the decrease in hydrodynamic friction.

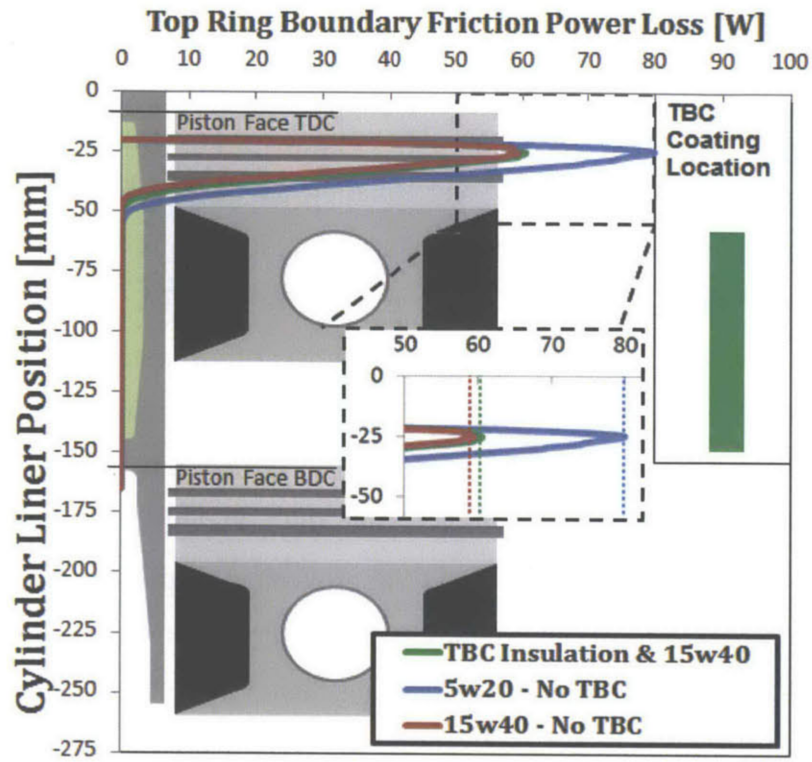

Figure 6.10 – Final TBC Selection – Top Ring Boundary Friction (Only Expansion Stroke is shown)

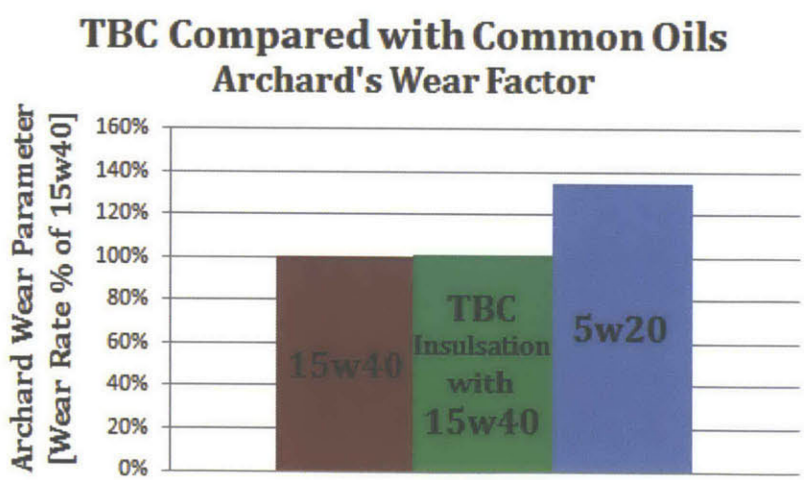

Figure 6.11 – Final TBC Selection – Archard Wear Analysis

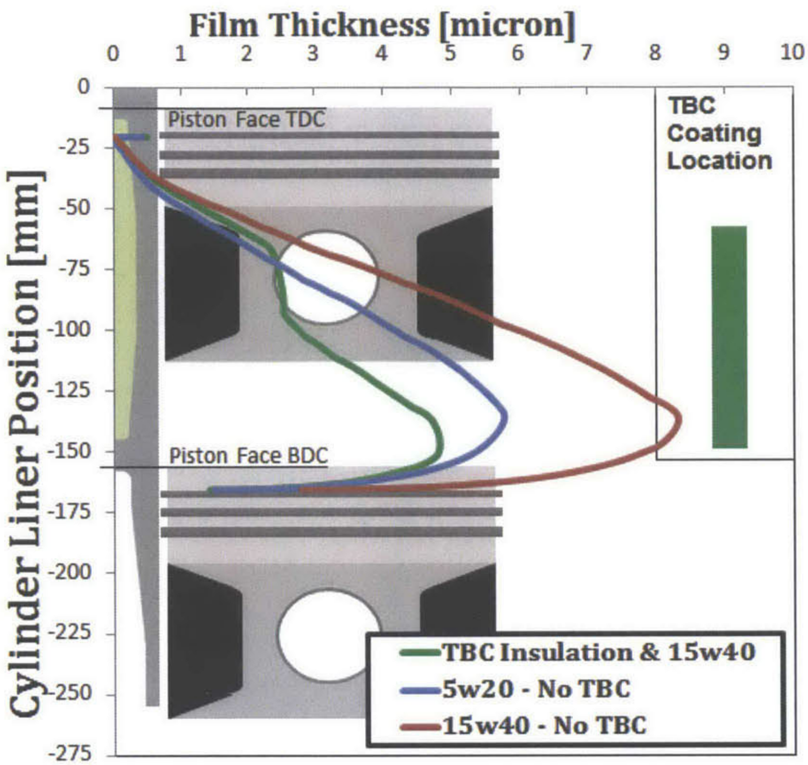

Figure 6.12 - Final TBC Selection – Film Thickness

6.4 Final TBC Design

The selected TBC insulated cylinder liner discussed in the previous section is presented in figure 13. The figure provides an example of one TBC insulation solution and the small space claim the application requires. This design utilizes a plasma spray zirconia that is 0.5 mm thick. The PSZ is applied to the outside of the cylinder liner in the cooling jacket. The existing engine architecture does not need to change to fit TBC insulation. The predicted BsFC improvement is 0.7% BsFC. This is achieved while offering superior wear protection when compared to 5w20 oil.

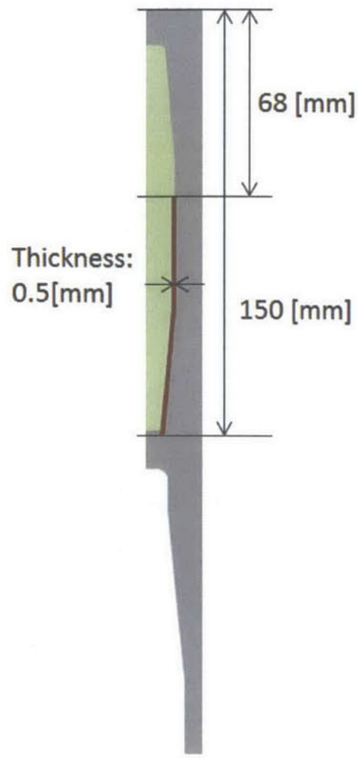

Figure 6.13 - Final TBC Design

7. Conclusions

The in situ control of lubricant properties or the design of the cylinder liner temperature profiles for friction reduction is beneficial for both power cylinder friction and wear. By insulating the cylinder liner with TBC, the mid-stroke hydrodynamic power losses are be targeted and significantly decreased without affecting TDC lubrication conditions, which increase wear if altered. The insulation increases local temperatures along the inside of cylinder liner where the oil film is present. The increase in temperature from insulation decreases local oil viscosity. Applying the insulation to the mid-stroke results in local viscosity well below that of low viscosity oil in normal engine operation (no insulation). A lack of insulation near TDC, where metal on metal contact of the rings and liner is a concern, allows baseline temperatures to be maintained and viscosity near top ring reversal is left unaffected. The maintenance of TDC temperatures allows the high viscosity and wear benefits of a 15w40 oil to be achieved while the resulting FMEP is reduced beyond that of low viscosity oils. The studies presented in this paper demonstrate TBC insulation applied to the outside of the liner improve power cylinder FMEP by 33.0% which corresponds to a 0.7% BsFC improvement while maintaining a wear rate similar to a 15w40 oil.

When designing a cylinder liner with TBC insulation two trends should be kept in mind. The first trend indicates insulation should be applied from a designated starting location to the end of the stroke. Adding TBC towards BDC only decreases FMEP further. The second observed trend finds the maximum friction benefit is obtained if the maximum liner temperature coincides with the point of maximum power loss along the cylinder liner. The maximum liner temperature can be moved by changing the starting location of TBC insulation. A parametric study such as the one completed in this paper can be completed to find that optimized TBC benefit.

6.5 Continuing Work

At the close of this paper, TBC has finished the proof of concept phase in the concept's research and development cycle. The benefits of thermal barrier coating have been measured on an engine test bed and are in the same magnitude as those presented in this paper. The merit of local control of lubricant properties in the power cylinder has been proven yet additions tests must be completed to ready this concept for production. These next steps in analysis and testing include the following:

- **Emissions Impact** – An increase in combustion temperatures results in an increase in NOx emissions. By insulating the cylinder liner with TBC, the combustion gas temperatures may have risen with the cylinder liner surface temperatures. The in-cylinder model applied in this analysis focused on the in cylinder heat transfer rather than combustion. Further research into the effects of TBC on combustion and emissions should be completed to validate TBC is an emissions neutral solution.
- **Bore Distortion** – By increasing the cylinder liner surface temperatures, bore distortion due to increased temperature might occur. Bore distortion depends not only on the surface temperature increase, but also the cylinder liner and block assembly geometry and clearances. Further simulations are needed to predict the impact of TBC on cylinder liner bore distortion.
- **Oil Consumption** – If bore distortion becomes an issue with TBC insulated cylinder liners, an increase in oil consumption within the power cylinder might occur. Engine tests that confirm oil consumption is not an issue for TBC insulated cylinder liners need to be completed. If oil consumption does increase with TBC, engine tests may offer alterations to current ring pack configurations to prevent increase oil consumptions.
- **Oil Deterioration** – Oil oxidation is highly temperature depended. Increasing oil temperatures will increase oil oxidation and accelerate the deterioration of the oil. Oil sampling during engine validation will confirm or deny the increase in oil deterioration due to the increase in mid-stroke temperatures.
- **Piston Deposits** – Piston deposits form near TDC when the oils' temperature limits are exceeded. By increasing surface temperatures in the power cylinder, piston deposits may form. During testing and validation of TBC, the power cylinder components should be checked for the formation of piston deposits

Coating Durability – The coatings selected in this paper ran on an engine for limited hours without issue. Yet the coating must last the full life of the engine or the cylinder liners will need to be replaced. Additional durability tests will be needed to validate the life of TBC insulated cylinder liners.

8. Works Cited

1. J. Heywood, Internal Combustion Engine Fundamentals, New York: McGraw-Hill, 1988.

2. Plumley, Michael, Victor Wong, and Mark Molewyk. "Optimizing Base Oil Viscosity Temperature Dependence For Power Cylinder Friction Reduction ." *SAE International.* (2014): n. page. Print.

3. Majewski, Addy W. "What Are Diesel Emissions." *DieselNet Technology Guide.* Ecopoint Inc. August 2012. https://www.dieselnet.com/tech/emi_intro.php.

4. United States Environmental Protection Agency, "Nitrogen Dioxide," 1 November 2011. [Online]. Available: http://www.epa.gov/air/nitrogenoxides/. [Accessed 22 3 2012].

5. Majewski, Addy W., and Hannu Jääskeläinen. "Engine Design for Low Emissions." *DieselNet Technology Guide*, Ecopoint Inc. September 2009. https://www.dieselnet.com/tech/engine_design.php#fe.

6. U.S. Environmental Protection Agency. *EPA and NHTSA Adopt First-Ever Program to Reduce Greenhouse Gas Emissions and Improve Fuel Efficiency of Medium-and Heavy-Duty Vehicles*. Office of Transportation and Air Quality. Available at: http://www.epa.gov/otaq/climate/documents/420f11031.pdf.

7. Sisken, Kevin. "SuperTruck Program: Engine Project Review" Department of Energy: Energy Efficiency and Renewable Energy, May 2013, Detroit MI. Ed. Department of Energy. 2013. Available at:http://www4.eere.energy.gov/vehiclesandfuels/resources/merit-review/sites/default/files/ace058_sisken_2013_o.pdf

8. D. Richardson, "Review of Power Cylinder Friction for Diesel Engines," *Journal of Engineering for Gas Turbines and Power-Transactions of the ASME,* vol. 122, no. 4, pp. 506-519, 2000.

9. Engine Oil Viscosity Classification, SAE-J300, April 4, 2013.

10. T. Boschert, "The Lubricant Contribution to Future Low Emission Engine Design," Diesel Particulate and NOx Emissions Course (University of Leeds), Ann Arbor, MI, 2002.

11. Bruce, Robert W., and Joseph M. Perez. "Chapter 24: Base Oils." In *Handbook of Lubrication and Tribology*. Second ed. Boca Raton, FL: CRC Press, 2012.

12. Booser, E. Richard, and Ronald L. Shubkin. "Chapter 3: Polyalphaolefins." In *Tribology data handbook*. Boca Raton, Fla.: CRC Press, 1997.

13. Yamaguchi, Elaine S., et al. "Chapter 21: Additive Technology." In *Handbook of Lubrication and Tribology*. Second ed. Boca Raton, FL: CRC Press, 2012.

14. Totten, George E., and Syed Q. A. Rizvi. "Additives and Additive Chemistry." In *Fuels and lubricants handbook technology, properties, performance, and testing*. West Conshohocken, PA: ASTM International, 2003.

15. Rudnick, Leslie R., and Syed Q. A. Rizvi. "Chapter 4: Detergents." In *Lubricant additives chemistry and applications*. 2nd ed. Boca Raton, FL: CRC Press, 2009.

16. Simon , Tung, Michael McMillan, et al. "Handbook of Lubrication and Tribology: Application and Maintenance ." *Chapter 1: Automotive Engine Oil*. Ed. George E. Totten. 2. Boca Raton, FL: Taylor & Francis Group LLC, 2006. Print.

17. Wong, V., Bauer, W., Kamo, R., Bryzik, W. et al., "Assessment of Thin Thermal Barrier Coatings for I.C. Engines," SAE Technical Paper 950980, 1995, doi:10.4271/950980.

18. James, Chistopher J. *Analysis of Parasitic Losses in Heavy Duty Diesel Engines*. Massachusetts Institute of Technology , 2012. Print.

19. T. Tian, "Modeling the Performance of the Piston Ring-Pack in Internal Combustion Engines," *PhD Thesis, Massachusetts Institute of Technology,* 1997.

20. Hu, Y., et. al., "Numerical Simulation of Piston Ring in Mixed Lubrication - A Non-Axisymmetrical Analysis," ASME Journal of Tribology, 1993.

21. R. Takata, "Effects of Lubricant Viscosity and Surface Texturing on Ring-pack Performance in Internal Combustion Engines," *MIT SM Thesis,* 2006.

22. Tian, T., et.al., "A Piston Ring-Pack Film Thickness and Friction Model for Multigrade Oils and Rough Surfaces," SAE paper 962032, 1996.

23. Hamrock, Bernard J., and Bo O. Jacobson. "Chapter 8: Lubrication, Friction and Wear." In Fundamentals of machine elements. Boston: WCB/McGraw-Hill, 1999.

24. Molewyk, M., Wong, V., and James, C., "In Situ Control of Lubricant Properties for Reduction of Power Cylinder Friction through Thermal Barrier Coating," SAE Technical Paper 2014-01-1659, 2014, doi:10.4271/2014-01-1659.