Report Title: ***Lubricant Formulations to Enhance Engine Efficiency in Modern Internal Combustion Engines***

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DISCLAIMER

ABSTRACT

The research program presented aimed to investigate, develop, and demonstrate low-friction, environmentally-friendly and commercially-feasible lubricant formulations that would significantly improve the mechanical efficiency of modern engines without incurring increased wear, emissions or deterioration of the emission-aftertreatment system. A strategy was followed to identify and meet the variable requirements of lubricant formulations at various engine subsystems (Phase 1), develop the best composite lubricant formulation and implementation for the overall engine system (Phase 2), and then implement such a system to demonstrate improvements by actual engine testing (Phase 3).

Demonstration of a segregated, "dual loop", lubrication system, in light of modern formulation constraints, was investigated through modeling and experiment. Two dual loop prototypes were developed, incorporating independent oil systems for the engine valve train and power cylinder, decoupling many lubricant functional requirements. A combination of high viscosity lubricant in the valve train, with low viscosity in the power cylinder, increased fuel economy while maintaining wear protection in the head. Effective protection of subsystems from contamination and oil degradation, particularly the elimination of soot in the valve train, was also demonstrated. Improvements in mechanical efficiency of over 3.7% were shown.

First of their kind detailed friction and oil composition models were developed to further identify opportunities for friction and wear reduction. Novel techniques for investigating oil composition changes along the liner in modern friction models were developed, with differences in lubricant functional requirements along the liner identified. Model results indicated vaporization along the liner increases lubricant viscosity near piston top dead center, providing a potential wear reduction benefit.

The strategies developed in this study have potential for application in all modern reciprocating engines as they represent simple methods to improve fuel economy, durability, and emissions through modification of lubricant formulations and lubrication systems. The current program benefits future studies in many industries, including on and off road, locomotive, marine, and power generation. The progress made in this program has wide engine efficiency implications, and potential deployment of improved engine configurations or lubricants in the near term is possible.

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Lubricant Formulations to Enhance Engine Efficiency in Modern Internal Combustion Engines

Final Technical Report
Massachusetts Institute of Technology
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EXECUTIVE SUMMARY

This program aimed to investigate, develop, and demonstrate low-friction, environmentally-friendly and commercially-feasible lubricant formulations that would significantly improve the mechanical efficiency of modern engines by at least 10% (versus 2002 level) without incurring increased wear, emissions or deterioration of the emission-aftertreatment system. A strategy was followed to identify and meet the variable requirements of lubricant formulations at various engine subsystems using a unique engine experimental platform with the oil supply system segregated between the head and power cylinder subsystems. Detailed friction modeling was also conducted to investigate the impact of key lubricant properties on particular engine components. Implications for development of composite lubricant formulations, for implementation of more conventional engine systems were considered. Additional longer duration tests were conducted to experimentally investigate longer term benefits from a segregated lubrication system, such as those which may be encountered over an entire oil drain interval.

The program consisted of three phases, which included 12 tasks, as shown in the Milestones chart. Phase 1 investigates the ideal lubricant formulations tailored to each major engine component subsystem for the best performance. Phase 2 investigates composite lubricant formulations that retain most of the frictional benefits for the subsystems identified in Phase 1 and then identify the tradeoffs and compromises necessary for an optimal composite lubricant formulation for the combined subsystems. Phase 3 demonstrates the mechanical efficiency improvement for the best optimized lubricant formulation in an actual full-size engine over a range of operating conditions that both reflect those in standardized industry protocols and other driving conditions.

The team completed all assigned tasks. Major accomplishments were achieved for each phase. In Phase 1, the team modified a commercially available twin cylinder diesel engine to create a unique engine efficiency testing platform. Fitted with state of the art torsion meters to determine camshaft and brake torque, the engine is capable of operation with segregated lubrication loops between the head and power cylinder. The study confirmed reduction of base oil viscosity in the power cylinder subsystem, and an increase in viscosity in the valve train can significantly reduce mechanical friction. The team demonstrated mechanical efficiency improvements, using a segregated oil system, of up to 3.7% at a mid speed, mid load condition. Even greater improvements expected at lower load operating conditions. The study also provided, through experiment and modeling, additional data regarding the relationship between boundary and hydrodynamic friction in complicated regions of the engine, primarily the valve train and along the cylinder liner. A set of eight oils were specifically formulated for parametric study of base oil viscosity and additive effects in

particular engine subsystems. In Phase 2 the team developed a first of its kind oil composition and friction model detailing the effect of vaporization, fuel dilution, additive concentration, and soot entrainment along the cylinder liner. The effort identified opportunities for formulation of lubricants specifically aimed at reducing wear near piston top dead center, one of the harshest regions in the engine environment. The team also quantified the fuel economy benefits associated with operating an engine lubrication system in dual loop configuration by demonstrating opportunities for implementing commercially available multigrades as well as newer Group IV base oil technologies. In particular the configuration presents the opportunity for more cost effectively implementing the latter, more costly and advanced base lubricants. Efforts to segregate systems also highlighted opportunities for in situ control of lubricant parameters, specifically through temperature control in particular engine regions. These opportunities may be applied to conventional single lubrication loop systems, allowing frictional benefits identified through viscosity changes in the first phase to be implemented in more conventional engines. Phase 3 accomplishments provided additional insight into the emissions and oil drain interval implications of incorporated more advanced, dual lubrication loop systems in the field. In addition to the many practical aspects identified for such implementation, longer duration engine tests provided insight into oil aging in particular engine subsystems, providing information for more advanced engine zone friction composition models, as well as data quantifying the benefit of segregating oil systems. In particular, the team successfully field tested a dual loop lubricating system and demonstrated successful protection of particular engine subsystems from contamination, most notably the reduction of soot in the valve train.

During this program, the project team, which consisted of members from MIT as well as major US engine and lubricant formulation companies, participated in quarterly technical meetings to organize research activities. Regular quarterly reports, as well as a presentation at the 2014 DOE Annual Merit Review, were submitted to the program detailing progress at each stage. Student researchers published, and presented, two technical papers at the Society of Automotive Engineers 2014 Annual World Congress and Exhibition, and gave 5 presentations at the Society of Tribologists and Lubrication Engineers 2014 Annual Meeting. The delivery of three additional presentations, with another publication, is anticipated in the near future. The work presented in this final report is taken from the regular quarterly reports, as well as 4 academic theses completed in support of the project. The research played an important role in the development of several young engineers. Two research participants have continued as PhD candidates, two are employed by major US engine and auto manufacturers, and another is serving in a faculty position at a US university. The team interacted productively with industry and other universities in the lubrication community and look forward to continued opportunities in the future after the current program ends.

The strategies developed in this study have potential for application in all modern reciprocating engines as they represent simple methods to improve fuel economy, durability, and emissions. The current program benefits future studies in other industries as well, including transportation, and dual fuel engines for transportation and power generation. The progress made in this program has wide engine efficiency implications, and potential deployment of improved engine configurations or lubricants in the near term is possible.

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I. INTRODUCTION

A. Objectives

The overall program goal was to investigate, develop, and demonstrate low-friction, environmentally-friendly and commercially-feasible lubricant formulations that would significantly improve the mechanical efficiency of modern engines by at least 10% (versus 2002 level) without incurring increased wear, emissions or deterioration of the emission-aftertreatment system.

The specific project objectives include identifying the best lubricant formulations for individual engine subsystems, identifying the best composite lubricant formulation for the overall engine system, and demonstrating the mechanical efficiency improvement for the optimized lubricant formulation via engine testing.

Phase 1: Identify the best formulations for engine subsystems: To identify and develop the best lubricant formulations for major individual subsystems (viz. piston/ring/liner, valvetrain). The intermediate aims are to quantify the effects of lubricant properties (base oil and additives – traditional and non-traditional) on lubricant behavior and friction in major subsystems, such as the supply, depletion, degradation of oil and active species therein, thermal and mechanical conditions at lubricant/component interfaces, oil film thickness and surface layers, to achieve the best formulations.

Phase 2: Best composite formulations for the overall engine system: To identify the tradeoffs and compromises for an optimal composite lubricant formulation for various subsystems combined, considering lubricant additive interactions, mixing or non-mixing of the lubricant circulating among various engine subsystems. To investigate the potential of effecting strategic variable (time or spatial) lubricant properties via local conditioning, time release of additives, or mixing control. To determine time scales of lubricant degradation and compositional changes.

Phase 3: Proof of concept: To demonstrate the mechanical efficiency improvement for the best optimized lubricant formulation in an actual full-size engine over a range of operating conditions that both reflect those in standardized industry protocols and other driving conditions. Experiments on specialized test engine such as the floating-liner engine that measures liner friction contribution only will also be demonstrated.

Common to all phases, it is also an objective of the program to elucidate and demonstrate to the public at large the underlying mechanisms and processes that led to the development of the potential best formulations.

B. Scope of Work

A combination of modeling and experiments was pursued in a three-phase program with some overlap in the schedule of implementation. In Phase 1, we investigated the ideal lubricant formulations tailored to each major engine component subsystem for best performance. In Phase 2, we investigated composite lubricant formulations that retain most of the frictional benefits for the subsystems identified in Phase 1. While Phase 1 identifies the "theoretical" ideal limits, Phase 2 examined the synergistic or antagonistic interactions of combining different formulations to arrive at the most effective composite formulation in meeting the various and very different demands of the all subsystems. Phase 3 combined and demonstrated the cumulative results in Phase 1 and 2 in a system demonstration in a regular full-size engine.

In Modeling: For the power-cylinder system, we modeled the lubricant flow to the piston and rings, loss of volatile species in the oil, development of the oil film thicknesses, thus the lubrication regimes, as well as additive species concentrations, in the ring/liner and piston-skirt/liner interfaces, from which hydrodynamic and boundary lubrication friction can be calculated. In the valvetrain system, where boundary lubrication dominates, we focused on the absorption, desorption, and degradation rates of the active species (after decomposition) and wear products of the anti-wear and friction modifier agents.

Experimentally, a production engine was modified for independent lubricant circuits for the powertrain and valvetrain. We used a diesel engine in view of the increasing demand of lower CO2 emissions. By alternating changes in the lubricant for engine subsystem, incremental changes due to the impact of the lubricant on specific subsystems can be determined.

C. Tasks Performed

The project included the following phases and tasks:

Phase 1: Identify the best formulations that would apply to individual engine sub-systems, which operate in different lubrication regimes, without significant adverse effects such as increased oil consumption/emissions, wear, or S, P, ash by-products. Two major subsystems are the power-cylinder components and the valvetrain subsystem components. The crank bearings operate primarily in the hydrodynamic lubrication regime. Identify via modeling and experiments the flow rates, degradations, and local lubricant/additive species concentrations, and impact on friction, wear and emissions.

Phase 2: Investigate composite lubricant formulations that retain most of the frictional benefits for the subsystems identified in Phase 1 and then identify the tradeoffs and compromises necessary for an optimal composite lubricant formulation for the combined subsystems. Best compromises and tradeoffs for composite formulations for the overall engine system: Identify the tradeoffs and compromises in composite lubricant formulations that provide the best combined effects in friction and wear for all components. Results from Phase 1 will provide information on the sensitivity coefficients of each aspect of lubricant performance (friction, wear) on the lubrication formulation parameters, such as base-oil viscosity changes due to varying shear, temperature and particulate contamination, or wear rates changes due to anti-wear additive concentrations. Considerations will be given to additive interactions, as (boundary) friction modifiers function differently in different amounts and types of anti-wear additives. Conversely, certain anti-wear additives also affect friction. The "ideal" formulations for each subsystem serve as starting points from which an intricate process of formulation for an optimal composite lubricant that works for all components can be developed. The optimal formulation also depends on the degree and rates of mixing (or not at all) of lubricant from one subsystem to another. Novel concepts of controlling the lubricant properties – such as nano-particles on the lubricant temperature (through heat conductivity changes), in-situ lubricant conditioning (acid control) or time-release additive supplements in controlling local lubricant composition – could also be explored. While Phase 1 identifies the "theoretical" ideal limits, Phase 2 examines the synergistic or antagonistic interactions of combining different formulations to arrive at the most effective composite formulation in meeting the various demands of the all subsystems.

Phase 3: Demonstrate the mechanical efficiency improvement for the best optimized lubricant formulation in an actual full-size engine over a range of operating conditions that both reflect those in standardized industry protocols and other driving conditions. Demonstrate the candidate formulation in an actual operating engine and a floating liner engine system, as appropriate, to identify the friction improvements in the overall engine and in the various subsystems.

The designated phases included completion of the following tasks. Those marked with an asterisk were specifically added to the task list during the course of the project to better organize work efforts related to the planned phases.

Task 1.0 – Project Management and Plan
Task 2.0 – Modeling effect of lubricant parameters on friction/wear for subsystems
- Subtask 2.1 Analysis of lubricant effects on power-cylinder friction as a subsystem
- Subtask 2.2 Analysis of lubricant effects on valvetrain friction as a subsystem
- Subtask 2.3 Oil composition modeling*

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
- Subtask 4.1: Engine install and lubricant-supply circuit modification for subsystems
- Subtask 4.2: Engine lubricant diagnostics implementation
- Subtask 4.3: Parametric base oil & additive experiments, power-cylinder friction
- Subtask 4.4: Parametric base oil and additive experiments, valvetrain friction
- Subtask 4.5: Design and install of long term oil aging test engine*
- Subtask 4.6: Parametric experiments to determine subsystem composition/aging trends

Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 6.0 – Model lube formulations with regional variations
Task 7.0 – Test, optimize, composite oil formulations
Task 8.0 – Develop practical means to implement new formulations
Task 9.0 – Demonstrate, in an actual engine, quantitative improvements in mechanical efficiency of best formulations from study
Task 10.0 – Evaluations of the impact on emission-control systems
Task 11.0 – Technology transfer an interfacing with users and researchers
Task 12.0 – Reviews and Reports

D. Major Accomplishments

In this program, the major accomplishments include the following:

- Successfully built, tested, and quantified the benefits of a dual lubricating loop engine

The team modified a commercially available twin cylinder diesel engine to create a unique engine efficiency testing platform. Fitted with state of the art torsion meters to determine camshaft and brake torque, the engine is capable of operation with segregated lubrication loops between the head and power cylinder.

- Demonstrated fuel economy benefit through segregation of lubrication subsystems:

The study confirmed reduction of base oil viscosity in the power cylinder subsystem, and an increase in viscosity in the valve train, can significantly reduce mechanical friction. The team demonstrated mechanical efficiency improvements, using a segregated oil system, of up to 3.7% at a mid speed, mid load condition. Even greater improvements expected at lower load operating conditions. The study also provided, through experiment and modeling, additional data regarding the relationship between boundary and hydrodynamic friction in complicated regions of the engine, primarily the valve train and along the cylinder liner. A set of eight oils were specifically formulated for parametric study of base oil viscosity and additive effects in particular engine subsystems.

The team also quantified the fuel economy benefits associated with operating an engine lubrication system in dual loop configuration by demonstrating opportunities for implementing commercially available multigrades as well as newer Group IV base oil technologies. In particular the configuration presents the opportunity for more cost effectively implementing the latter, higher priced and advanced base lubricants.

Efforts to segregate systems also highlighted opportunities for in situ control of lubricant parameters, specifically through temperature control in particular engine regions. These opportunities may be applied to conventional single lubrication loop systems, allowing frictional benefits identified through viscosity changes in the first phase to be implemented in more conventional engines.

- Demonstrated oil drain benefit through segregation of lubrication subsystems:

In addition to the many practical aspects identified for such implementation, longer duration engine tests provided insight into oil aging in particular engine subsystems, providing information for more advanced engine zone friction composition models, as well as data quantifying the benefit of segregating oil systems. In particular, the team successfully field tested a dual loop lubricating system and demonstrated successful protection of particular engine subsystems from contamination, most notably the reduction of soot in the valve train.

- Demonstrated emissions benefit through segregation of lubrication subsystems:

Soot sampling confirmed the reduction in antiwear additives resulted in a reduction in Zn and P in the raw soot emitted from the engine. Introduction of a dual loop lubricating system provides are practical means to introduce existing low SAPS lubricants which are currently impractical due to valve train wear concerns. The use of a dual loop lubricant system allows for zero ZDDP lubricants in the power cylinder, while retaining formulations benefiting valve train performance, such as ZDDP or even higher sulfur and phosphorus containing extreme pressure anti-wear additives. The result is an immediate improvement in emissions performance with no loss, or even an increase in, valve train wear protection.

- Identified opportunities for improved fuel economy and wear performance along liner through novel composition based modeling approach

The team developed first of their kind oil composition and friction models detailing the effect of vaporization, fuel dilution, additive concentration, and soot entrainment along the cylinder liner. The effort identified opportunities for formulation of lubricants specifically aimed at reducing wear near piston top dead center, one of the harshest regions in the engine environment.

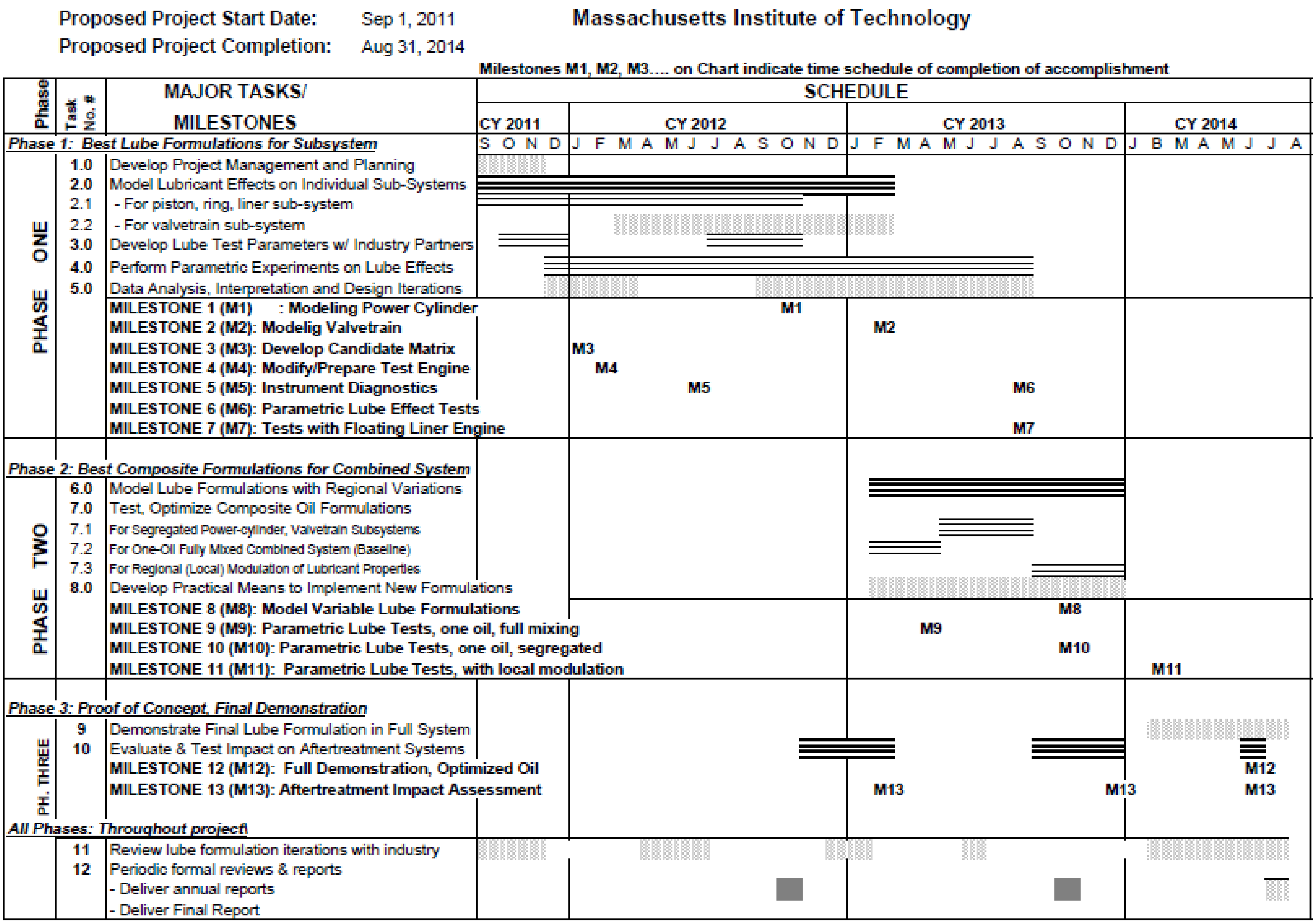


Figure 1 Project timeline.

II. RESULTS AND DISCUSSION

1. Background

1.1. Motivation and related studies

This study was motivated by the need for improved fuel economy and reduced harmful emissions from diesel internal combustion engines. Reducing wear and oil change intervals are related motivations. Options for improving fuel efficiency and emissions are many, and already fill volumes. This study focuses specifically on the engine lubrication system. Common to all internal combustion engines, the lubrication system serves two primary purposes, reducing friction and controlling engine wear and corrosion [1]. Friction reduction has a direct benefit on fuel economy, as mechanical efficiency is on the order of only 90% for today's engines. Migration of oil to the exhaust stream due to oil consumption has a significant impact on emissions, allowing for gains through proper formulation.

This report summarizes results contained in several works developed over the course of the research. Results and passages are specifically taken from several theses, primarily [2], [3], [4], and [5].

1.1.1. Fuel Economy

As reviewed in [2], estimates place diesel fuel costs at 25-35% of total haulage firm overheads [6]. In 2011 the first U.S. medium and heavy duty vehicle fuel efficiency standards were announced, affecting 2014-2018 model years, with a goal of efficiency improvements for combination tractors of 20%, and up to 15% for heavy duty pickups [7][8][9]. Recent research under the Supertruck program resulted in a 75% increase in class 8 fuel truck economy through a variety of means [10].

A new lubricant formulation category, PC-11, is under development. It is the first to have a "fuel economy" formulation requirement, which will be achieved by setting viscosity limits on HTHS. Industry expectations are that PC-11 will result in an HTHS 150 limit of 2.9-3.2 cP for the fuel economy specification [6]. A backward compatible lubricant, with a higher HTHS is also expected.

A large portion of engine energy losses are a result of mechanical friction. Richardson estimated mechanical friction accounted for 4-15% of total energy losses [11]. Reports from some OEM indicate fuel economy gains of up to 2% are possible in "line haul" mode as a result of the 33% friction reduction feasible from lubricant selection. At low speed, low load, these benefits increase to 3%, with even greater benefits at idling conditions [6].

Mechanical efficiency is a measure of that portion of the gross indicated power used to do useful work. It is the ratio of the brake power to the indicated power as defined in [1]. A thorough discussion is also provided in [2]. In this work, the primary interest is in those friction components effected by lubricants, so FMEP, as presented in the following chapters, is given as the difference in the NIMEP and BMEP, eliminating the effect of pumping losses, which, although significant, are not expected to be affected by engine lubricants.

1.1.2. Emissions

Emissions regulations influence lubricant development as a result of fuel sulfur reduction, motivated by reductions in sulfur dioxide, and additive limits, motivated by particular matter and nitrogen oxide (NO_x) reduction. Diesel engines oil consumption primarily contributes to engine out emissions of particulate matter (PM). Ash and volatile organic compound levels in the PM are largely due to oil consumption [12].

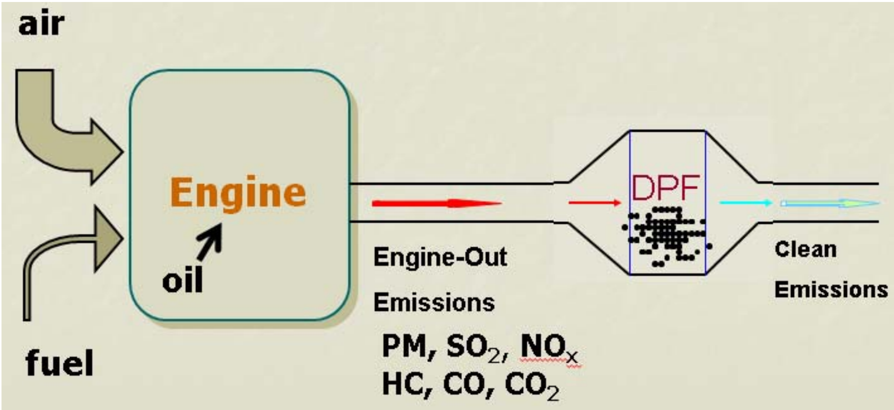


Figure 2 Diesel engine contributions to harmful emissions [2].

1.1.3. Oil Change Intervals

Harsh engine environments eventually degrade lubricants through complicated mechanisms. These place considerable burdens on lubricants. Drain intervals create increased hazardous waste, providing a secondary avenue for pollution from internal combustion engines. Increasing intervals is attractive, as it reduces maintenance requirements and the need for hazardous waste disposal. Passenger car change intervals typically vary from 3,000 miles for short trip and severe service to over 7500 miles for less severe service [13][14].

1.1.4. Engine Durability and Failure

Engine durability is directly related to lubricant composition and aging. When lubrication problems occur it tends to be due to the entrainment of combustion products and lubricant degradation. While lowering viscosity is of great benefit for fuel economy, failures may result from increased wear due to metal to metal contact. The valve train is often considered the most vulnerable subsystem for such contact; however the power cylinder may also suffer from significant wear. Excessive soot is known to cause wear and subsequent failure in valve trains. Recent studies show that soot plays a significant role in wear in low viscosity applications [6]. Such failure concerns are the motivation for many valve train durability tests in oil API category certification. Valve failure may also occur due to corrosion of the valve from the combustion or head side of the valve. In a dual loop lubrication system protection from valve stem corrosion may be achieved if acid levels in the lubricant are reduced. Valve train parts not exposed to the combustion chamber should receive greater protection. Deposit buildup and, or, corrosion on the valve stem or tip may also develop from degraded lubricants. Wear, pitting, and scuffing are of concern for cam lobe and follower wear as well.

1.2.Lubrication System Design

In this study the term ‘conventional system’ is used to describe a system with one pump delivering lubricant to all subsystems as is typical of current engine designs for which a common system serves the valve train and crankcase subsystems. While a useful hardware configuration, it creates lubricant design tradeoffs. Recent implementation of emissions aftertreatment systems increased the impact of these tradeoffs [12]. Splitting the lubrication system may decouple some of these requirements as discussed in [2] in terms of axiomatic design principals. This provides opportunities for reduced friction, emissions, and overall oil dependency. The term ‘dual’ or ‘split’ system is intended to describe an engine lubrication of atypical design for which the valve train and power cylinder subsystems have separate lubrication loops which do not interact. The term ‘split’ or ‘segregated’ may also be used more broadly to describe any configurations for which one part of the engine’s lubrication system is segregated (Figure 3).

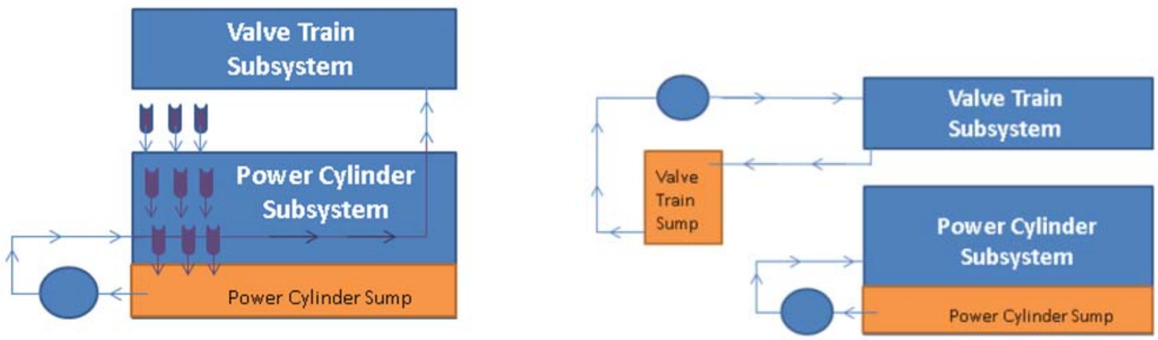


Figure 3 Conventional (left) and dual loop (right) lubrication system configurations [2].

The experimental studies in this work relate to a particular small diesel engine with a crankshaft driven mechanical oil pump which draws oil from the main engine sump and delivers it to the main bearings and the valve train camshaft journal bearings and rocker arms by way of a single oil filter. The oil lubricates other components in the engine by splashing. Oil is delivered to camshaft journals and rocker arms by pressurized passages. Other valve train components are lubricated by splash. A schematic of the system, modified from the Kohler shop manual, is given in Figure 4. The subject system differs from many automotive engines in that it has a belt driven camshaft. Many engines employ a chain driven camshaft, or a gear box, with lubrication draining back to the main sump.

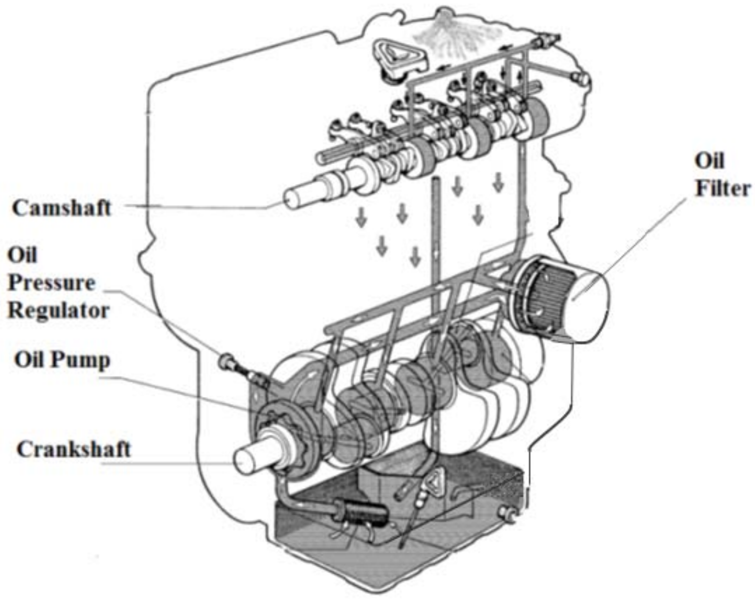


Figure 4 Lubricant system for 3 cylinder Kohler KDW 1003 (turbocharger removed) [15][3]. A similar configuration, the KDW 702, a twin cylinder naturally aspirated diesel engine, was used in this study [2].

1.3.Lubricant Formulation

Typical automotive lubricants consist of base oil and an additive package. The base oil accounts for roughly 75-90% of an engine oil formulation by mass. Viscosity modifiers, if used, account for approximately 10% of the total formulation. The rest of the additive formulation consists of the detergent inhibitor (DI) package and other additives, with dispersants accounting for 6%, antiwear 1.5%, and detergents 3.5% [16]. Additive impacts, other than the viscosity changes from viscosity modifiers, may have a limited effect on overall fuel economy as compared to viscosity changes [17]. This is likely due to the significantly greater portion of friction that is attributed to hydrodynamic losses in the power cylinder system as will be discussed. Additive optimization is expected to have greater impact on valve train friction [16]. A more thorough discussion of base stocks and additive packages is presented in [2].

2. Experimental Apparatus

2.1. Introduction

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 3.0 - Develop lubricant test parameters w/ partners from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.1: Engine install and lubricant-supply circuit modification for subsystems
Subtask 4.2: Engine lubricant diagnostics implementation
Subtask 4.5: Design and install of long term oil aging test engine*
Task 8.0 – Develop practical means to implement new formulations
Task 11.0 – Technology transfer an interfacing with users and researchers

Various test platforms were constructed for the purpose of collecting data. The engine chosen for the study is the Kohler KDW702, an EPA tier 4, twin cylinder, 686 cc, naturally aspirated compression ignition engine rated at 12.5 kW (16.8 hp) at 3600 rpm and 29.9 ft. lbs at 2000 rpm. It has overhead cams and indirect fuel injection. The crankcase is cast iron and the head is aluminum. The engine employs a belt driven overhead camshaft. This facilitated the instrumentation of a torque sensor for determining valve train friction to optimizing subsystem lubricants and alleviated the need to seal oil passages in the presence of an internal chain drive or gear box. The two cylinder model was chosen over the three cylinder model to reduce the uncertainty in IMEP estimation due to the use of multiple cylinder pressure transducers. In addition less fuel was required to run tests on a small engine which was convenient for oil aging and other extended duration tests.

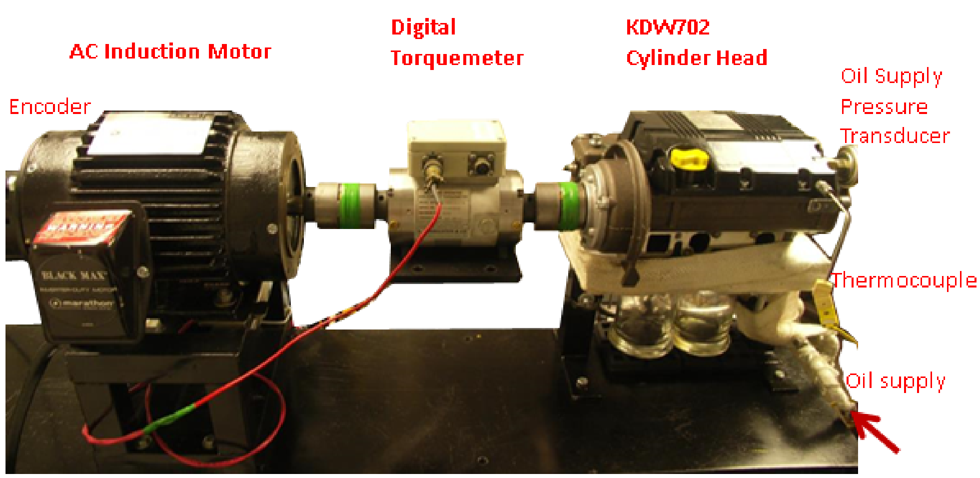


Figure 5 Cylinder head bench test rig shown without insulation or shaft guards [2].

2.2.Cylinder Head Bench Rig

The cylinder head test rig is described in detail in [3]. The rig was designed and constructed primarily to validate valve train torque sensor sizing for the main test cell engine. It was used to validate engine test results, as well as study the effects of different lubricants on specific valve train components. The unit was also used to size the valve train oil sump for the oil aging test rig.

Figure 5 shows the test rig with the 2 hp AC induction motor and 56 Nm Himmelstein torque meter. Instantaneous camshaft torque was recorded by integrating the sensors with National Instruments hardware and a LabVIEW virtual instrument developed for the study. A heated oil sump and bronze gear pump provided oil supply.

Figure 6 Fore and aft views of engine test cell rig [2].

2.3.Engine Test Cell Rig

The ‘engine test cell rig’ was used for engine friction tests throughout the study. A detailed discussion of the test cell design and features is given in [3]. Engine start and experimental motoring were achieved using a 15 hp Marathon AC induction motor. Figure 6 shows the instrumented engine in the test cell.

The test cell engine modified lubrication system allowed for segregated operation. Technical guidance was available from Kohler Engine to assist with modification. To date this is one of few engines instrumented with a separated valve train lubricant circuit.   . Segregation was achieved by plugging oil drain lines to the main sump from the cylinder head as well as the oil supply line from the installed oil pump as described in [3] and [2]. Typical valve train oil pressures were kept at 4 bar for most tests. The system design is depicted in Figure 7. Fittings were installed (not shown) to convert to a conventional system using external hoses from a sandwich adapter on the power cylinder oil filter housing. The crankcase breather was routed as separate tube through sump. Block temperatures were controlled through coolant temperature regulation. Typical coolant temperatures for were set at 80°C, consistent with the engine thermostat setting, as the thermostat was removed during modification. SAE 15W40 oil is recommended for the engine. Commercially available ultra-low sulfur diesel (ULSD) was used exclusively for tests in this study

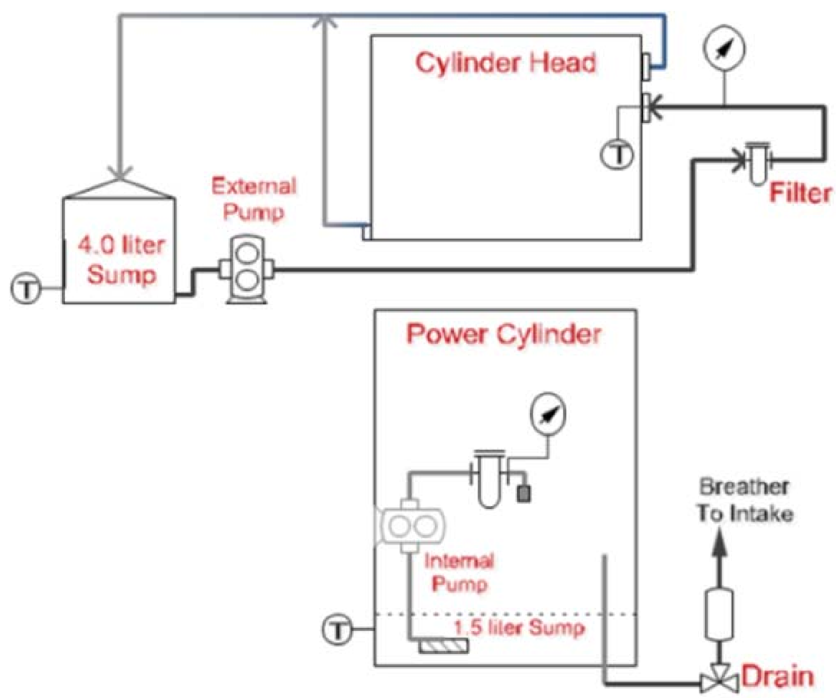


Figure 7 Dual loop lubrication system for test cell engine in split configuration [2].

The data acquisition system consisted of National Instruments (NI) hardware components with a virtual instrument interface developed in LabVIEW as described in [3] and [2]. An example of the front panel's main tab is shown in Figure 8. The tab allows continuous monitoring of engine parameters as well as user executed report writing functionality. Other tabs (not shown) allow for monitoring of parameter vs. time and parameter vs. crank angle information.

In the case of the valve train, all work is considered a loss to the auxiliary systems and friction. The average torque over a cycle represents the work loss during that cycle, and therefore the valve train FMEP is given by the average torque on the camshaft over the cycle. For a four stroke engine, with torque T:

$$MEP = \frac{T 4\pi}{V_d} \quad (1)$$

For the KDW 702 valve train this includes camshaft and rocker losses, as well as pumping losses for the fuel lift pump, and the two fuel injectors. A custom camshaft timing pulley, as described in [3], was used to determine instantaneous valve train torque.

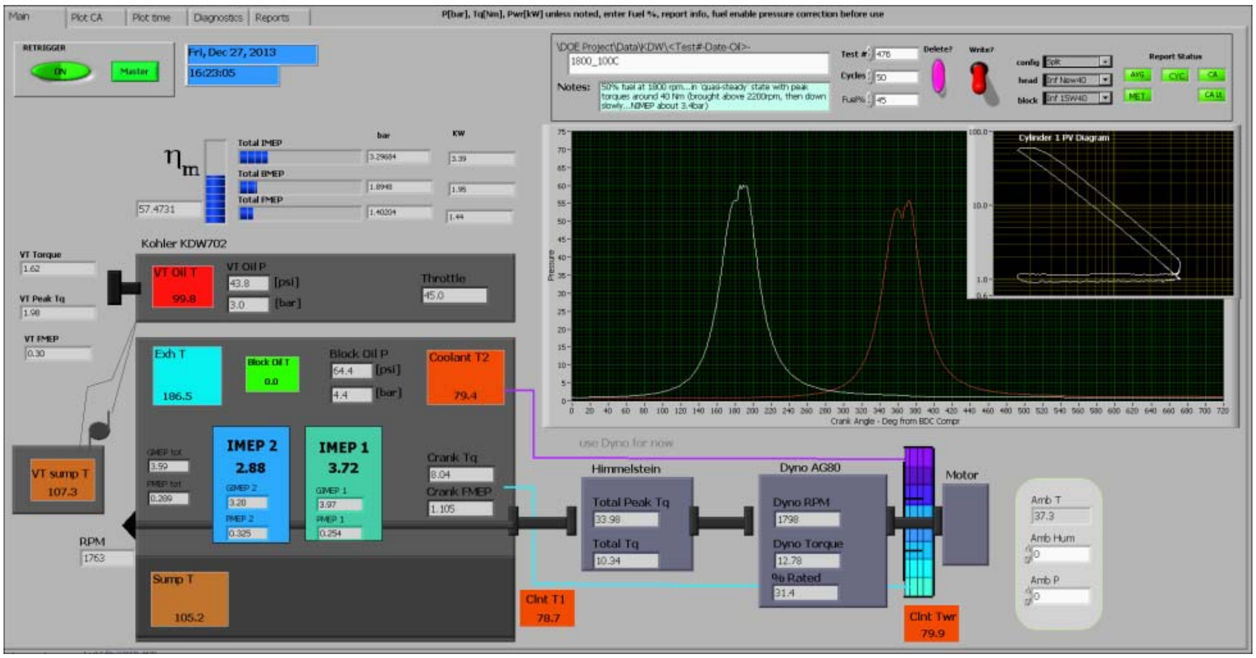


Figure 8 Front panel, main tab, for test cell engine LabVIEW user interface [2].

The net indicated mean effective pressure (NIMEP) and gross IMEP (GIMEP) were determined from the pressure trace using two Kistler 6052C transducers. In the data acquisition system pressure was measured per crank angle. The change in volume from one crank angle to the next was determined from the cylinder geometry and is therefore assumed, not measured. NIMEP was calculated using the trapezoidal rule in the virtual instrument as given by the following, where *i* represents a given crank angle during the cycle.

$$NIMEP = \frac{\Sigma\left(\frac{P_i + P_{i-1}}{2}\right)(V_i - V_{i-1})}{V_d} \quad (2)$$

The determination of NIMEP is highly sensitive to changes in the assumed volume. Determination of the values, along with uncertainty considerations, is described in detail in [2].

Brake mean effective pressure was determined by integration of the instantaneous torque over a cycle as described by the following equation. The primary sensor was a Himmelstein MCRT 49704V with a range of 0 - 564 Nm and repeatability of 0.03%. This represented 0.4% of the maximum brake torque of the engine studied. A 107 hp Froude Consine AG 80 eddy current dynamometer was used to place a load on the engine. The dynamometer was fitted with a load cell for which data was also recorded for later comparison. With brake torque given by T_b, the BMEP becomes:

$$BMEP = \frac{T_b 4\pi}{V_d} \tag{3}$$

Experimental uncertainty was significant with respect to the data collected as discussed in [2]. Average torque values for valve train tests varied from 1.0 to 2.0 Nm over the duration of tests, with an overall uncertainty of 0.04 Nm estimated. For the subject engine this corresponds to a valve train FMEP of 0.007 bar, corresponding to a 2%-4% uncertainty over the range of values measured. To allow for comparison tests were conducted with the same lubricant, a 15W40, in the head. The high sensor range of the crankshaft torque sensor was necessary due to the high torque experienced on the crankshaft due to the twin cylinder configuration. As a result, an uncertainty of 0.17 Nm, or a BMEP of 0.03 bar, was realized for the subject engine.

The most stable operating condition was at 2400 rpm and approximately 50% fuel rack position. Stable operating points were also available at 1800 rpm with loading corresponding to a fuel rack of less than 45%. To minimize steady state errors, efforts were made to maintain the same operating condition for each test. Steady state was achieved by warming the engine up at a constant fuel rack position of 50% then waiting for temperatures to stabilize. Load was varied from the steady state condition for periods of less than 2 minutes, and then returned to the 50% fuel setting, to maintain the same oil temperatures throughout testing. The general operating conditions for each test condition are summarized in Table 1, with the approximate standard deviation observed during testing.

Table 1 Engine operating conditions at steady state [2].

Operating Parameter	2400 rpm condition	1800 rpm condition
Speed (rpm)	2397 +/-1.4	1798 +/- 2.3
Coolant Temp (C)	78.7 +/- 2.1	78.4 +/- 2.0
Ambient Temp (C)	39.7 +/- 0.9	38.6 +/- 1.2
Sump Temp (C)	113 +/- 1.3	98 +/-1.7
Block oil pressure (bar)	4.45 +/- 0.16	4.29 +/- 0.11
Valve Train Oil Temp (C)	100 +/- 1.0	100 +/- 0.8
Valve Train oil pressure (bar)	3.07 +/-0.09	3.08 +/-0.09

Encoder alignment procedures, including analysis of pressure trace data and consideration of ambient pressure changes, are described in [2]. Each experimental data point consists of 50 cycles collected and analyzed with the LabVIEW virtual instrument. Standard deviation for each point was calculated and recorded to eliminate data sets that may contain significant errors, including misfires or triggering issues. The average standard deviation in NIMEP for a set of 50 cycles was 0.018 bar. The average standard deviation for total FMEP results was 0.012 bar. The average for valve train FMEP was 4.99 x 10^{-4} bar.

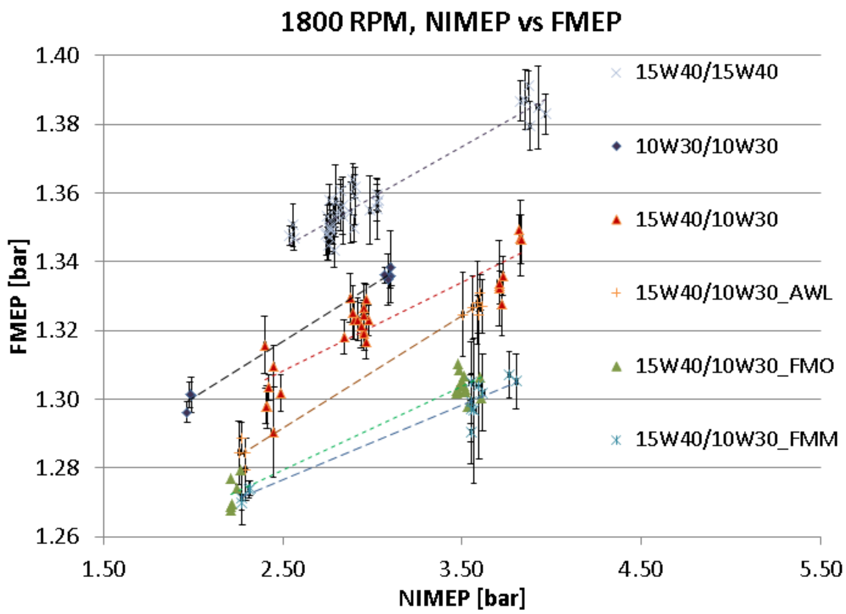


Figure 9 Detailed NIMEP vs. FMEP data with uncertainty bars for precision error [2].

A sample of data is given in Figure 9. Uncertainty bars are based on a 95% confidence interval. Bias error was not included. NIMEP uncertainty is based on the standard deviation of calculated NIMEP for 50 cycles and a 95% confidence interval as detailed in [2]. FMEP uncertainty is the result of the NIMEP and BMEP estimates. The plotted data represents the average NIMEP over 50 cycles vs. the average FMEP over 50 cycles, taken by subtracting the average BMEP from the average NIMEP. Precision was lower than the bias errors discussed previously. The governing bias error was the BMEP calculation, with torque sensor nonrepeatability errors of 0.03 bar. To allow for comparison of the behavior of one lubricant to another at a particular NIMEP value, a best fit line was developed using Excel Analysis of Variance (ANOVA) tools as detailed in [2]. A sample result is given in Figure 10 for tests conducted to compare various power cylinder lubricants.

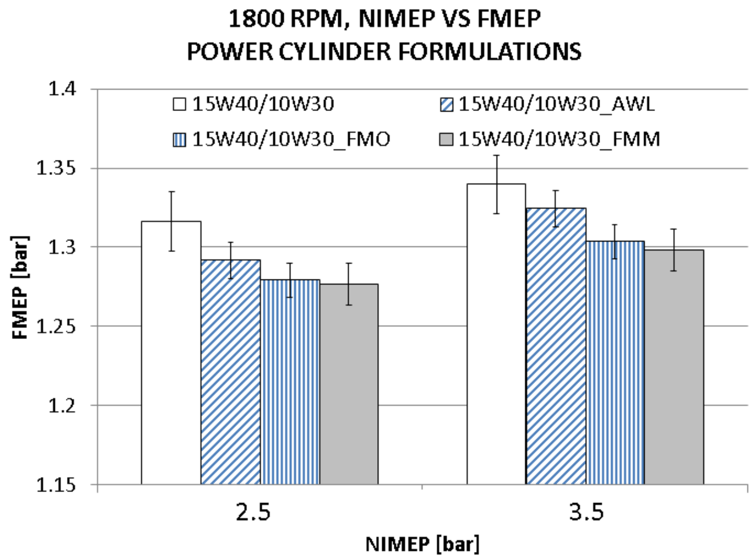


Figure 10 Comparison of NIMEP and FMEP for power cylinder lubricant additive parametric studies [2].

Due to the difficulty in determining small values, such as FMEP, from two large experimental values, particularly in a system as noisy as a fired internal combustion engine, results are subject to considerable uncertainty. Steady state behavior of the system was assessed by running identical lubricants on back to back days to assess the potential impact of fluctuations. Daily and monthly changes are discussed in detail in [2].

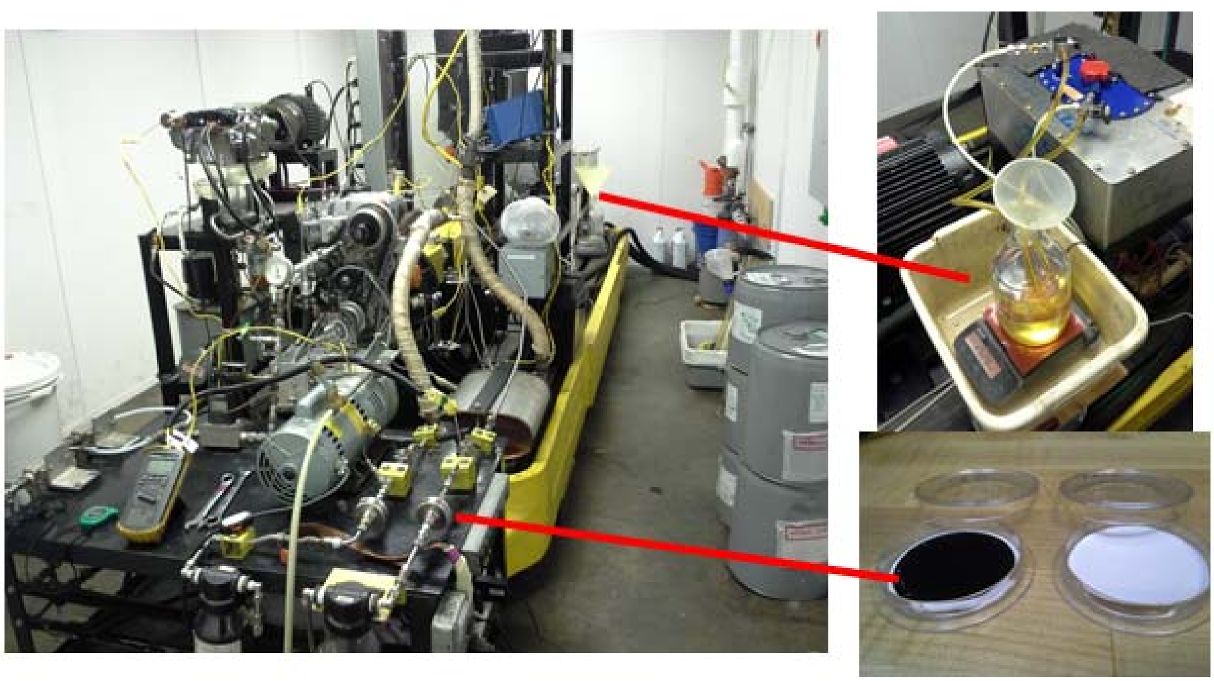


Figure 11 A soot sampling cart was developed and used to collect raw particulate matter for elemental testing. Fuel consumption was estimated using a scale as depicted on the right.

An interest in emissions sampling for Task 10 motivated the modification of the engine system to support soot sampling. It was similar to that described in [12], and is shown in Figure 11.

Details regarding collection and measurement are given in [2]. Diesel particulate matter is defined by the EPA under US 40CFR86 [18] as matter collected on a fiber filter which has been diluted and cooled to 52ºC. For the purposes of this work “soot” or “raw PM” refers to particulate collected from the exhaust at temperatures above 52ºC. The fuel chosen for the study was ultra-low sulfur diesel (ULSD). Sample density and flow rate were used to determine the fraction of exhaust sample captured with the sampling system. Brake torque was used to determine engine load for determination of brake specific fuel consumption (bsfc) and brake specific soot emission. Soot was collected with a separate stainless steel sampling line used to draw raw exhaust through the filter holder.

2.4.Mobile Oil Aging Rig

A more robust dual lubrication loop prototype was designed for longer duration field testing and oil sampling as described in [2]. The KDW 702 was also chosen for this unit. A commercial light tower provided an enclosure and load for the modified engine. A Magnum Pro model MLT 3060 was used as shown in Figure 12. The aging rig incorporated a dual loop lubrication system. Since oil aging is dependent on the sump size, a smaller valve train sump of 1200 ml, with integrated heating element, was chosen.

Figure 12 A full scale field test was conducted over a period of 4 weeks. The engine was operated for 250 hours to provide data on oil aging in each engine subsystem. Samples were sent to an independent lab for analysis. Results are expected next quarter.

Details regarding valve train oil pump selection and flow control are also described in [2]. An initial prototype incorporating a 12VDC electric oil pump was replaced by a mechanical hydraulic pump mounted on the camshaft power take off as shown in Figure 13 and Figure 14. A small sampling valve was installed downstream of the relief valve for oil aging samples.

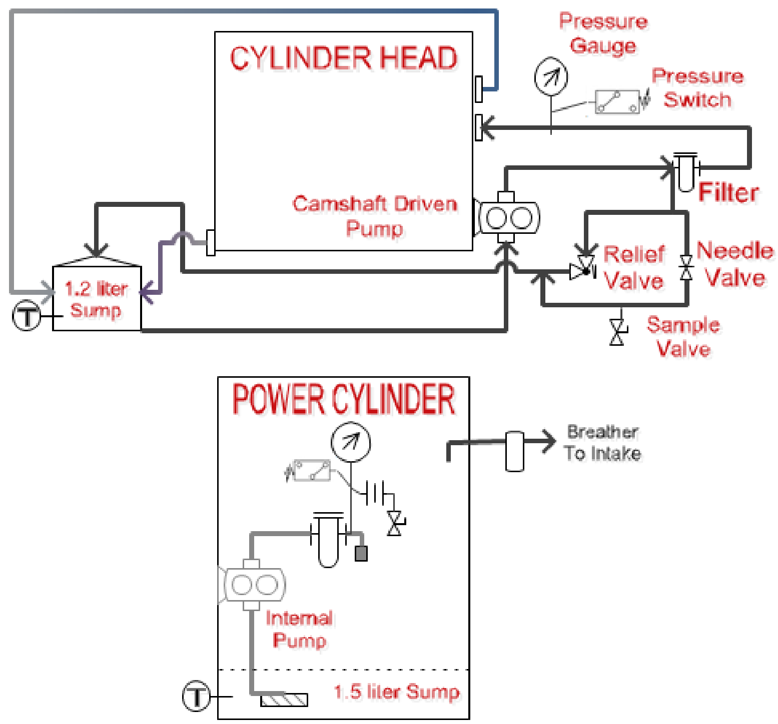


Figure 13 Dual loop lubrication system for light tower engine [2].

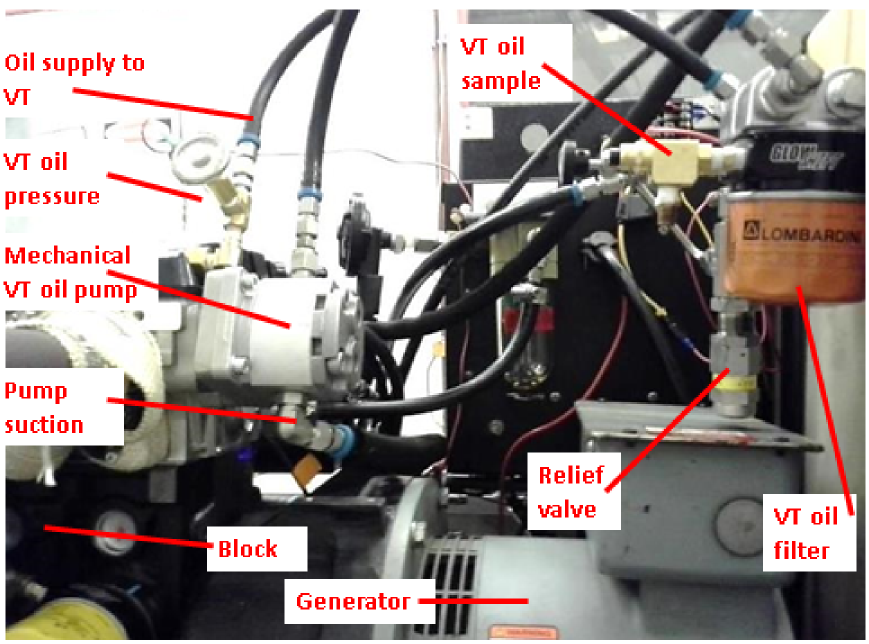


Figure 14 Mechanical valve train oil pump mounted on camshaft power take off (PTO) [2].

2.5.Lubricants used in this study

A set of lubricants was formulated to allow investigation of base oil and additive effects in fired engine tests. The initial set consisted of commercial grade lubricants from the CJ-4 family. Additive packages were identical, and base oil composition was varied to achieve a desired kinematic viscosity profile. The lubricants each consisted of a Group II/III base formulation with the same viscosity modifier in different concentration. The purpose for the formulation was to characterize the engine subsystem response to changes in viscosity, without regard to additive effects. Experimental results for the formulated oils are given in the following table. These lubricants were used for experimental studies described in following chapters. The experimental viscosity and density values were used in the related modeling studies.

Table 2 Lubricant specifications for base oil comparison [2].

Designation/ Grade	KV40 (cSt)	KV100 (cSt)	Density (kg/m3)	HTHS 150 (cP)
5W20	45.3	8.24	869	3.06
10W30	71.6	10.72	873	3.23
15W40	115.1	15.07	879	3.76*
NEW40	117.3	15.20	863	4.68

*An initial estimate of this value was 4.07 cP, later measured again at 3.76 cP.

The fourth lubricant was formulated with a polyalphaolefin (PAO). The lubricant design sought to provide the same kinematic viscosity near 100°C as the 15W40 multigrade formulated. The KV40 and KV100 values were greater than those for the 15W40 lubricant by 1 to 2 percent respectively (Table 2), hence when viscosity is extrapolated using Walther's correlation the NEW40 curve traces that of the 15W40 closely. Since this is an extrapolation the data can not conclusively reveal whether kinematic viscosities are in fact the same at values over 100°C.

Table 3 Lubricant specifications for base oil comparison with different shear rates [2].

Designation/ Grade	HTHS 150 10^6 s^{-1}(cP)	HTHS 150 10^7 s^{-1} (cP)	HTHS 100 10^6 s^{-1} (cP)	HTHS 100 10^7 s^{-1} (cP)	HTHS 75 10^6 s^{-1} (cP)	HTHS' 75 10^7 s^{-1} (cP)
5W20	3.06	2.36	5.83	5.27	10.72	8.77
10W30	3.23	2.69	7.27	6.05	13.72	10.60**
15W40	3.76*	3.16	9.35	7.30	18.01	12.98**
NEW40	4.68	4.36	12.72	11.26	25.18	22.82

*An initial estimate of this value was 4.07 cP, later measured again at 3.76 cP.

** Data only available up to $7x10^6s^{-1}$, values extrapolated for $1x10^7s^{-1}$.

Of particular interest in the case of the NEW40 is the shear thinning behavior. Laboratory test results of the shear thinning behavior of the lubricant are described in [2]. The 15W40 exhibited

20% lower viscosity than the NEW40 at 10^6 s^{-1}, increasing to 30% less at 10^7 s^{-1}. The NEW40 lubricant did exhibit some thinning, with the 10^6 s^{-1} viscosity being 10% lower than the non-sheared viscosity, which may be compared to the 28% drop for the 15W40. Similar behavior is exhibited at 100°C, although the NEW40 exhibits less shear thinning, with the high shear viscosity at 10^6 s^{-1} only 4% lower than the non-sheared viscosity. Results are given in Table 3 for three different temperatures and shear rates. The designation HTHS' is intended to distinguish the viscosity at the higher shear rate from that typically defined as HTHS.

Table 4 Lubricant specifications for power cylinder additive effect comparison trials [2].

Designation	Description	KV40 (cSt)	KV100 (cSt)	HTHS 150 (cP)
10W30	Baseline	71.6	10.72	3.23
10W30_FMM	Increased nonorganic friction modifier	70.7	10.8	3.21
10W30_FMO	Increased organic friction modifier	70.0	10.6	3.18
10W30_AWL	Reduced antiwear	71.5	10.9	3.24

The final set of lubricants was developed to investigate the potential opportunities for additive variation in the power cylinder subsystem. The 10W30 CJ-4 formulation was chosen as the baseline. Since wear was not quantitatively monitored lower viscosity base oil was not used, even though 5W20 was shown to provide even greater overall friction benefit. The levels of additive were altered from the baseline 10W30, and included lubricants with additional molybdenum and organic friction modifiers, designated FMM and FMO respectively, as well as a reduced anti-wear (AWL) version. Sulfated ash levels remained the same for the first two lubricants, with a slight reduction for the third lubricant.

3. Engine Power Cylinder Efficiency Modeling

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

- Task 2.0 – Modeling effect of lubricant parameters on friction/wear for subsystems
  - Subtask 2.1 Analysis of lubricant effects on power-cylinder friction as a subsystem
- Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.

A background on friction regimes and modeling methods is provided in detail in [2] and [4]. Much of the initial work covered in this Section, including some passages, is taken from [19] and [2] with permission.

Models previously developed at MIT and in the literature were used to assess the impact of different lubrication strategies on the power cylinder subsystem. Top ring friction was modeled using the modified Reynolds equation proposed by Patir and Cheng and later developed by Tian [20][21]. Contact friction was estimated with Greenwood and Tripp's asperity contact model, as used by Hu [22]. Parameters such as ring profile, thickness, and roughness were held constant. Skirt effects were analyzed using a model developed by Mansouri and Wong [23]. Additional background on these models may be found in [24] and [25].

Table 5 MM11 engine parameters used in modeling study [19].

Displacement	11.0 L
Bore	125.0 mm
Stroke	150.0 mm
Ring Thickness	2.6 mm
Ring Tension	32.0 N
Ring roughness	0.5 µm

In a similar manner to that given in [26], a factor was used as a measure of relative wear as given by Equation (2). Using ring velocity, U, and the normal load under the ring, the integral of contact pressure, the 'wear factor', with units of Watts, is:

$$Q_{wear} = U \int P_{contact} \qquad (2)$$

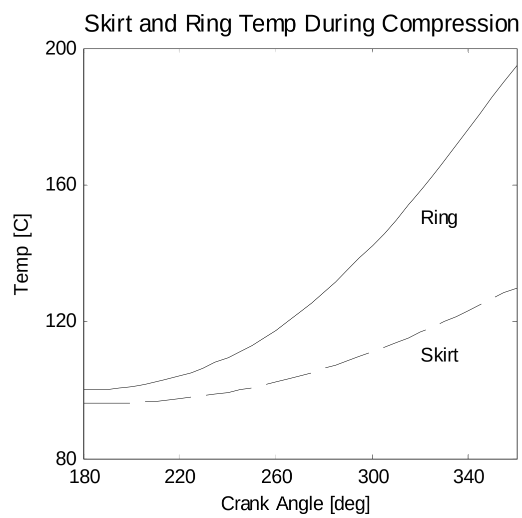


Figure 15 Comparison of temperatures under power cylinder components over the compression stroke [19].

The use of peak wear may be a better measure of wear risk, since failure at a particular location may be the first consequence of excessive wear. Of course these modeling factors will not predict the allowable limit for the engine or application which are traditionally determined through detailed wear studies.

As a reference for this work, typical parameters for a modern turbocharged 11.0 liter diesel were used as shown in Table 5. The engine parameters do not include proprietary data from any single manufacturer. The development of the model is discussed in [5].

Details on temperature estimates along the liner are described in [2]. Values used for the A100 engine operating condition are given in Figure 15.

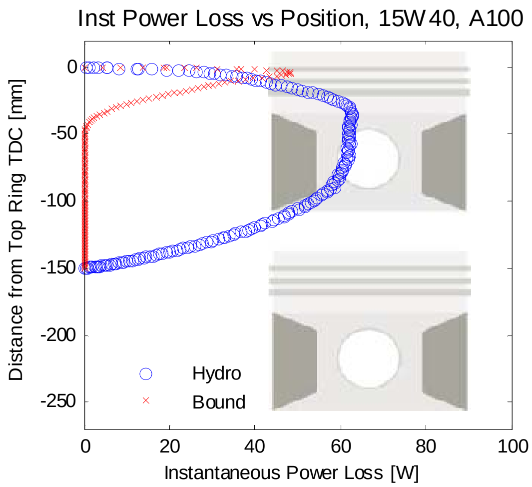


Figure 16 Hydrodynamic and boundary losses under top ring during A100 condition [19].

Insight into the relationship between boundary and hydrodynamic friction impacts were obtained by considering the losses in the position domain as shown in Figure 16, where the local power loss during a cycle is given as a function of position along the liner. The majority of friction losses occur during mid-stroke.

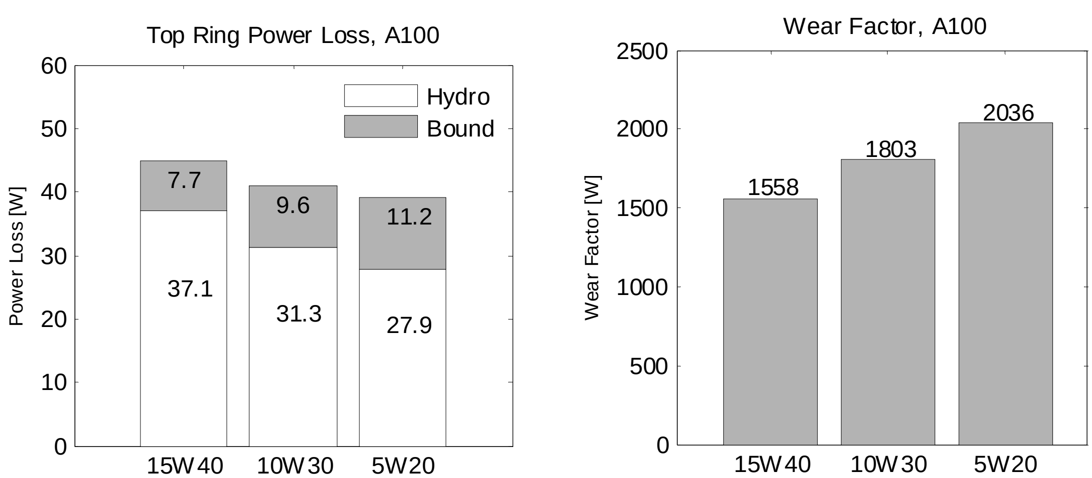


Figure 17 Hydrodynamic and boundary friction losses under top ring for A100 condition (left) and associated peak wear factor (right) [19].

As multigrades with lower viscosity are used, a decrease in hydrodynamic friction is realized at the cost of an increase in boundary friction as shown in modeling results shown in Figure 17. Boundary friction is experienced at low velocities, so it has a relatively low impact on the overall friction response. As a result, when changing from the 15W40 to the 5W20 oil the boundary friction losses increase 45% while hydrodynamic losses decrease 25% as expected from the roughly 40% decrease in viscosity. Total losses decrease by 13% for the given condition. The increase in boundary friction is accompanied by an increased wear concern. Modeling efforts quantified the magnitude of wear factor for different lubricants as described in [2].

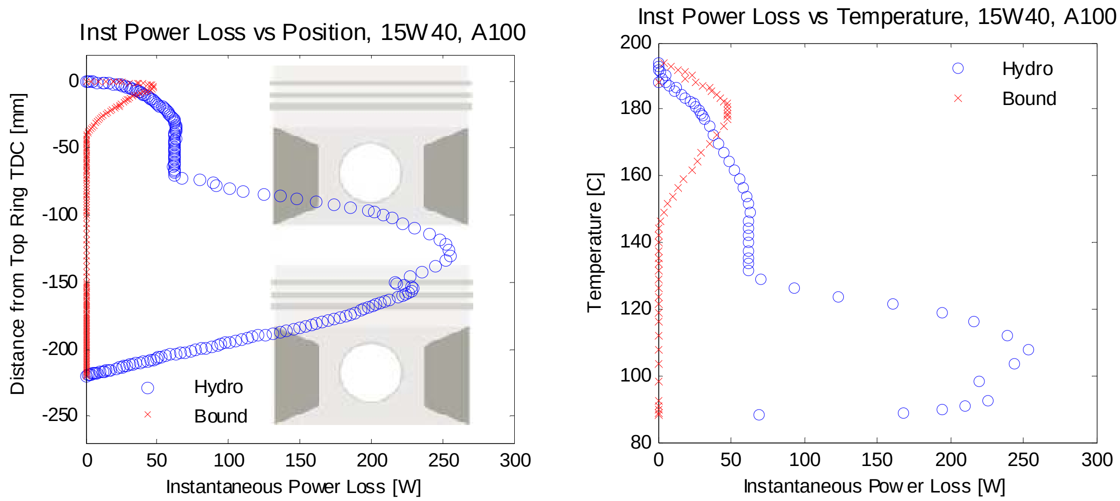


Figure 18 Power losses for top ring and skirt with respect to position and temperature along liner [19].

The piston skirt experiences a significantly lower temperature range than the upper ring. The design allows for greater surface area exposed to the oil film. As a result, nearly all losses from the skirt in this condition are hydrodynamic and occur at a lower region along the liner. The center of the skirt passes from 70 mm to 220 mm below TDC of the top ring. Plotting the top ring and skirt along the liner, as in Figure 18, indicates the greatest boundary losses are confined to the upper liner where higher temperatures occur.

Temperature is of great interest to the oil formulator. Figure 18 presents the losses in the temperature domain for the case of 15W40 at the A100 condition for the skirt and top ring, providing insight into the temperatures at which losses and wear occur.

A detailed study of bearing friction for the proposed engine model is not included, however bearing friction will tend to increase low temperature hydrodynamic losses.

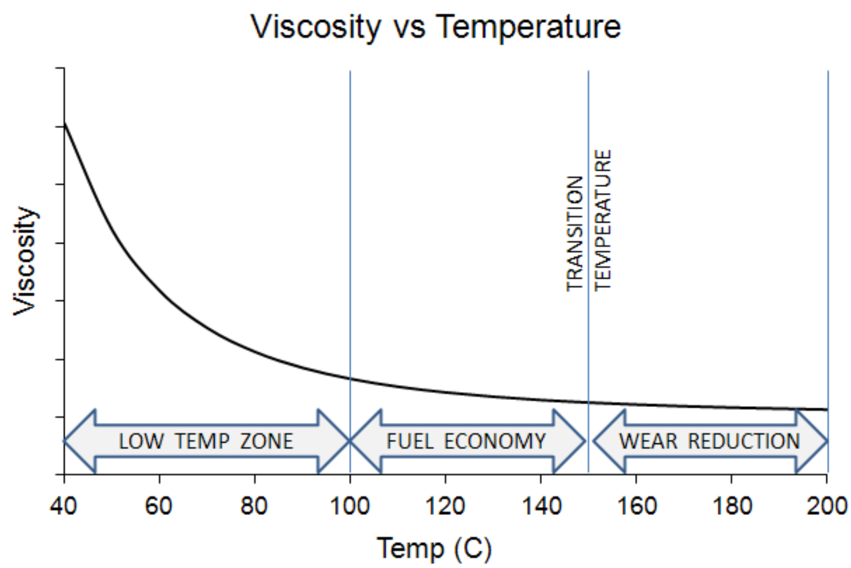


Figure 19 Viscosity vs. temperature profile with designated fuel economy and wear reduction regions [19].

Figure 19 shows a typical oil temperature profile considering the temperature ranges of interest to the formulator. In the previous results it is evident that hydrodynamic losses are greatest at mid-stroke temperatures, while regions with higher boundary friction and wear occur at higher temperatures, above 150°C for this case. Reducing viscosity below 150°C, and increasing it above 150°C, should result in hydrodynamic friction and wear reductions. In the given case the conclusion may be drawn that high friction losses are associated with lower temperatures along the liner. The second and oil control rings (OCR) are not included in the previous results. They would likely contribute to a combination of boundary and hydrodynamic losses at temperatures between 100°C and 180°C. The reduced pressure behind these rings is in contrast to the relatively high pressures under the top ring, which contribute to that rings high boundary friction.

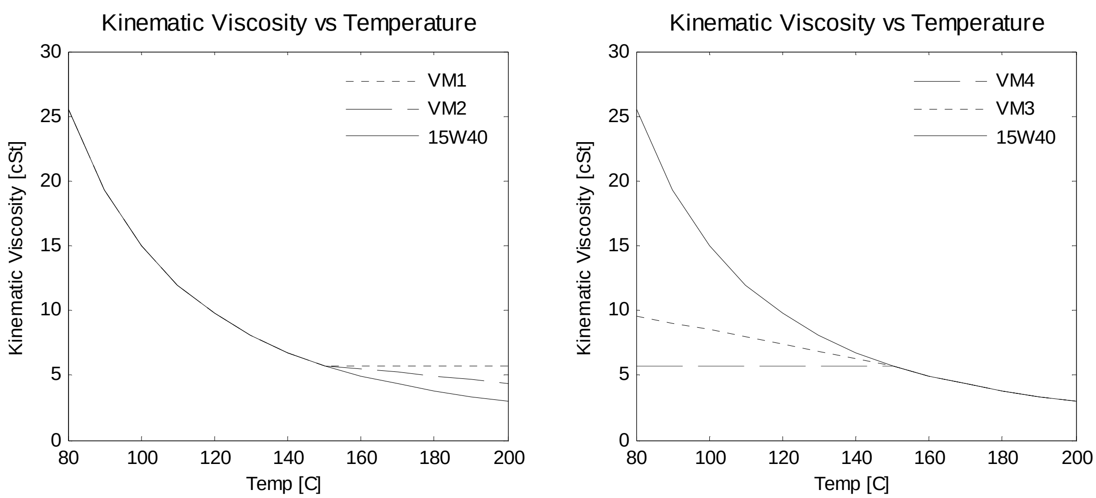


Figure 20 Viscosity temperature profiles with varied wear reduction index [19].

A set of general descriptors was proposed in [19] to frame discussions regarding the suitability of formulations based on their temperature dependence, including the *transition temperature*, *fuel economy index,* and *wear reduction index*.

Theoretical viscosity profiles were used to show the limits of potential benefits which may be achieved from variation of the fuel economy and wear reduction indices as discussed in [19]. Figure 20 shows a typical 15W40 lubricant with variations in the wear reduction and fuel economy indices. The resulting friction power loss and wear factor resulting from the variation of the respective indices is shown in Figure 21. The optimal case is that of VM5, for which reduced low temperature viscosity and increased high temperature viscosity result in a 20% reduction in peak wear factor and a 15% decrease in friction losses. This may set the bounds of those savings obtainable through viscosity temperature profile manipulation by the formulator with regard to base oils.

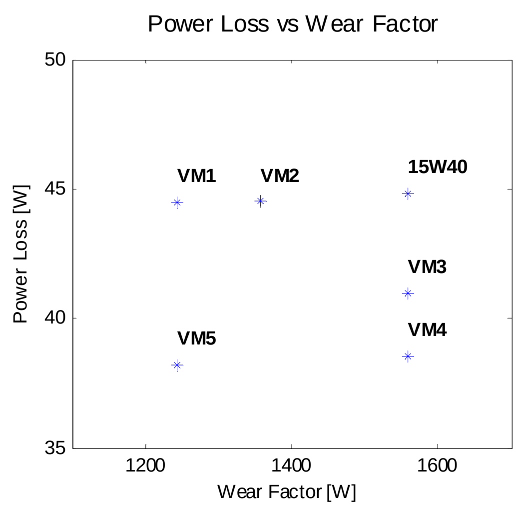


Figure 21 Friction and wear effect from changing wear reduction and fuel economy indices [19].

If low temperature viscosity could be lowered further, to the limit of the maximum wear value along the liner, the hydrodynamic viscosity may be further reduced. While difficult from a formulation standpoint, lower viscosities may be further achieved through in situ techniques, as discussed in [27].

Modeling efforts confirmed the majority of power cylinder friction is hydrodynamic. Temporary shear thinning was also shown to provide a benefit. Different functional requirements along the liner, resulting from piston speed and liner temperature, were also identified which informed later composition based modeling efforts in this region.

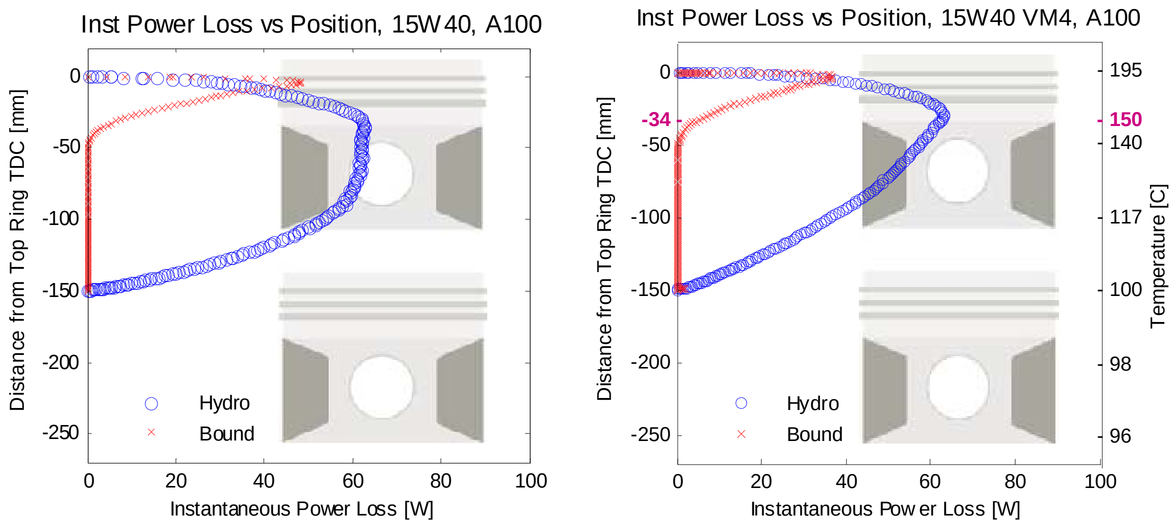


Figure 22 Top ring power losses for the 15W40 and VM4 viscosity profile cases [19].

4. Power Cylinder Efficiency Modeling with composition effects - vaporization

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 2.0 – Modeling effect of lubricant parameters on friction/wear for subsystems
Subtask 2.3 Oil composition modeling
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 6.0 – Model lube formulations with regional variations

Oil composition is known to vary along the liner. A portion of the modeling efforts in this study were devoted to a novel approach to friction studies in this region. They resulted in development of a first of its kind oil composition sensitive friction model. Specifically, variations in composition along the liner due to vaporization, and fuel and soot contamination, were specifically investigated with the goal of developing a better understanding of lubricant influence on friction in this region. The effect of changing oil composition on friction along the liner has not been widely studied in terms of modeling. Efforts discussed in this section are described in greater detail in [2] and [4].

Modeling efforts, previously completed at MIT by Audette [28], were developed for integration into existing friction models using rheological models developed to estimate viscosity as a function of lubricant species composition. Specifically, molecular weight was used to estimate viscosity as described in [19] and [4].

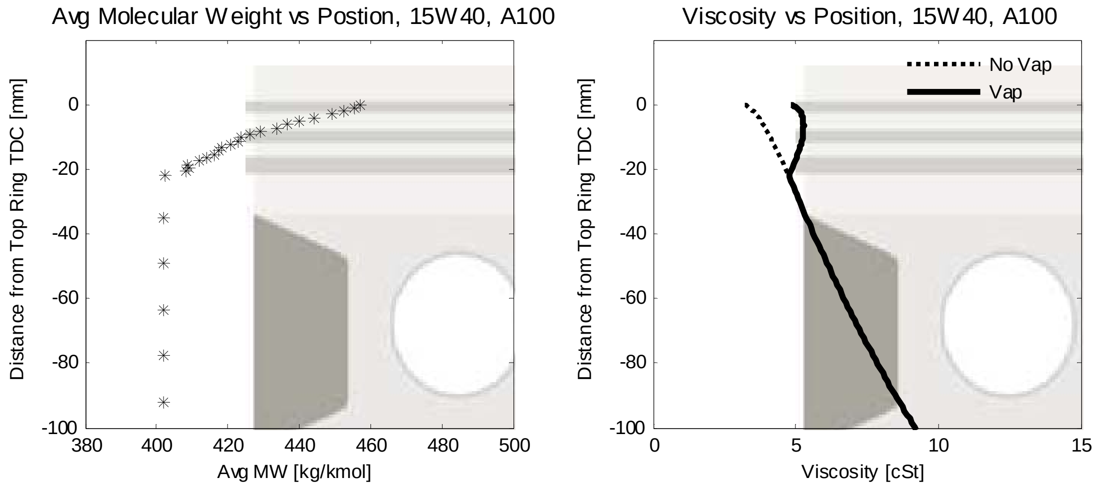


Figure 23 Average molecular weight and viscosity along liner for assumed 15W40 lubricant composition [19].

Sample results, showing molecular weight as a function of position due to vaporization, followed by the resulting viscosity estimate, are given in the following figures. The friction results developed with the vaporization corrected viscosity are compared to the original 15W40 results developed for the A100 condition. The change resulted in a 1.1% decrease in total top ring friction due to a 15% decrease in boundary friction and a 1.6% increase in hydrodynamic friction. The wear factor decreased 17%, with peak wear still occurring 5 mm below top ring TDC during the power stroke [2].

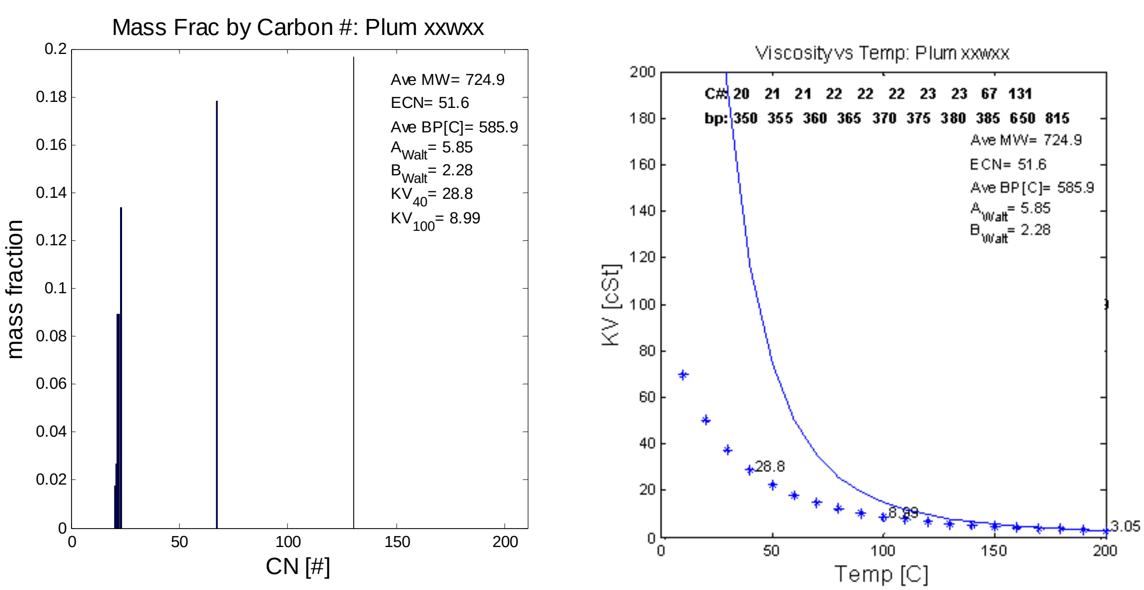


Figure 24 Proposed base oil formulation with light and heavy species. Assumed viscosity temperature profile of each species shown on right [2].

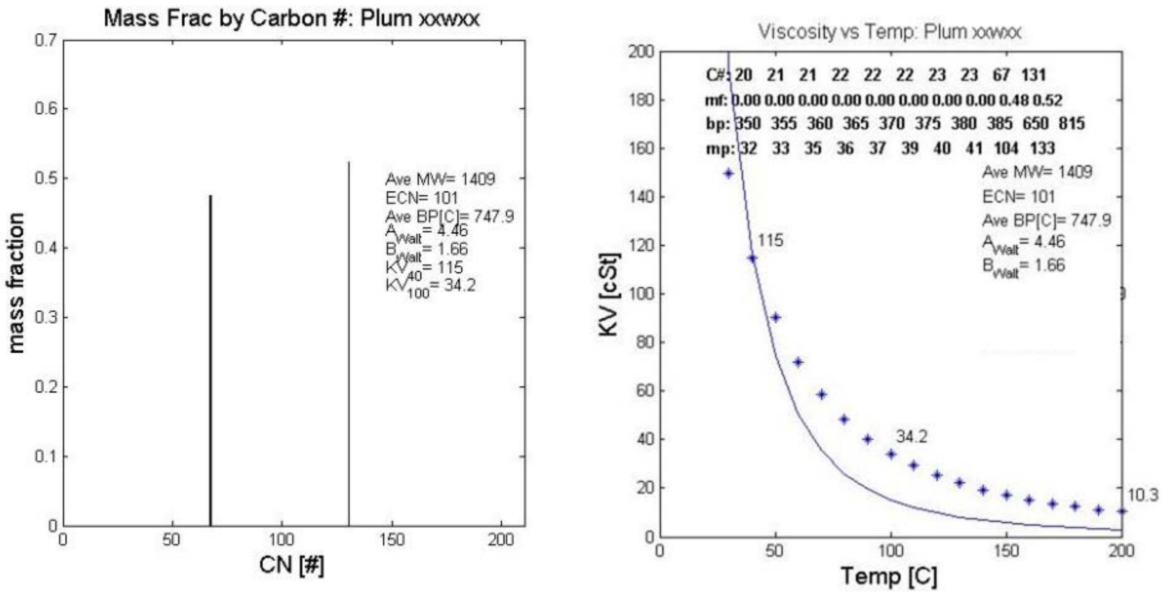


Figure 25 Mass composition and viscosity temperature profile of proposed formulation following vaporization depletion at TDC [2].

Developing on the approximations made in the procedures described in [19] a new procedure was developed to estimate viscosity as a function of composition. Using viscosity and molecular weight relationships for low carbon number paraffins estimates were made for higher molecular weight compositions as described in [2]. Using the correlation, a theoretical composition profile was developed, consisting of a lubricant consisting of a very heavy species and a very light species. The composition is graphically depicted in Figure 24. The resulting viscosity profile, modeled on the left, is similar to that of a typical 15W40, represented by the solid line in the Temp-KV plot. As expected, vaporization near TDC results in evaporation of all light species, leaving the heavy constituents behind and resulting in a significant viscosity increase near TDC as shown in Figure 25. The result, is a 2% overall decrease in top ring friction for the vaporized case. More importantly, boundary friction was reduced 50%, resulting in a comparable decrease in wear factor.

The results suggest an opportunity for optimal formulation of lubricants for condition along the liner. The presence of heavy molecules, perhaps even polymers currently employed as viscosity modifiers, could lead to reduced wear near TDC. Procedures for expanding this modeling approach are detailed in [2] and [4]. Future work in this area may be of great interest.

An alternative model for mixture rheologies was also developed with a more detailed emphasis on vaporization coupled with oil transport along the liner as discussed in [4]. Increased viscosity near TDC as a result of vaporization was confirmed. In this model vaporization was investigated along the entire liner, as opposed to just that portion over TDC of the oil control ring (OCR). The result in increased viscosity along the entire length after several cycles.

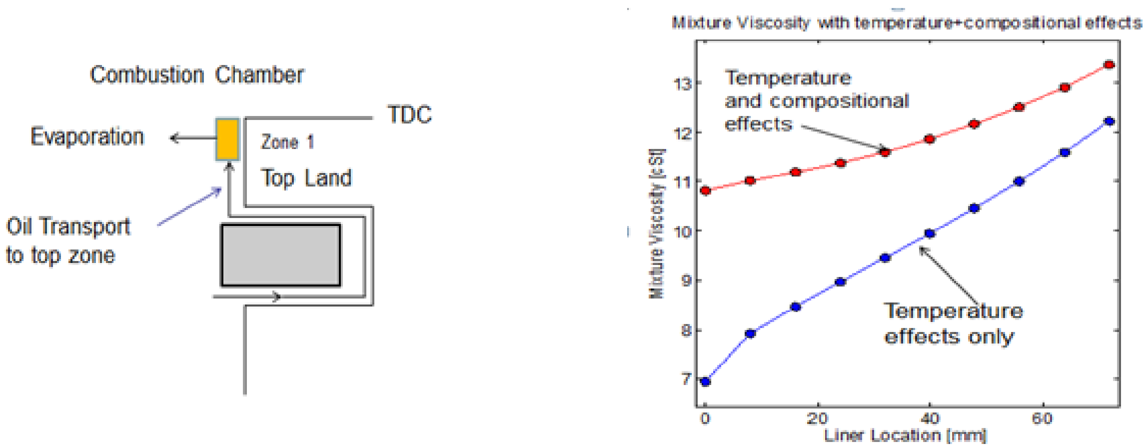


Figure 26 Schematic of oil transport with vaporization and compositional temperature effect on viscosity along liner.

5. Power Cylinder Efficiency Modeling with composition effects – Fuel, Additives and Soot

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 2.0 – Modeling effect of lubricant parameters on friction/wear for subsystems
Subtask 2.3 Oil composition modeling
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 6.0 – Model lube formulations with regional variations

5.1.Composition effects – Additives

Additive concentrations were also considered along the liner based on variation due to vaporization. A detailed discussion is presented in [4]. Specifically, zinc phosphate and calcium sulfate components found in ZDDP and detergents were investigated. A mixture of 90% hydrocarbons and 5% zinc phosphate and 5% calcium sulfate was assumed for an initial composition matrix in the oil composition model. Calcium and zinc are known to be less volatile than other constituents. Due to the vaporization of light hydrocarbons, retention of additives occurs where vaporization rate for light hydrocarbons are highest. The cylinder liner was analyzed by splitting it into ten zones and determining the composition of the additives in each zone. Experimental results from [29] were used to determine an enrichment factor for each additive. Enrichment factor is defined as ring zone composition divided by sump concentration. Figure 27 shows the comparison between experimental data and simulation data. The simulation overestimated the enrichment factor in the topmost zone. An average of the top three zone enrichment factors is in better agreement with Watson's data.

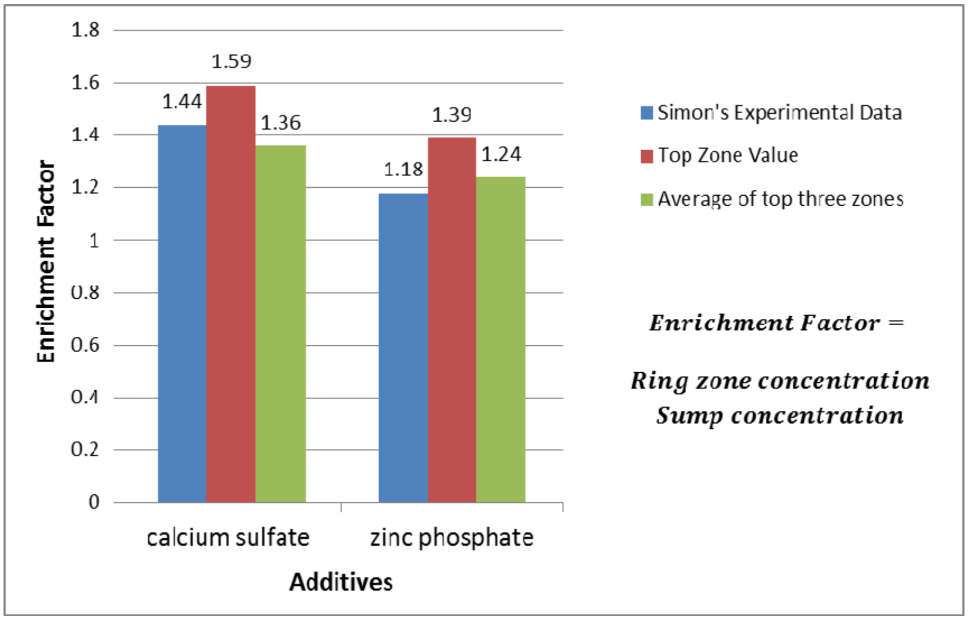


Figure 27 Comparison of experimental data (leftmost column) to simulation data (right columns). [4]

5.2.Composition effects – Fuel dilution

Fuel dilution effects were also investigated along the liner. Using parameters for the KDW 702 and the blending equation used by Smith as given in Equation (4), estimates were made of the viscosity change resulting from an assumed fuel dilution in the top 25% of the liner.

$$ln v_m = ln x_f v_f + \ln v_o v_o \qquad (4)$$

The goal of the estimate was to determine the impact of significant dilution in this region. The effort may be extended to more sophisticated blending correlations along the length of the liner. The diesel fuel was assumed to have a kinematic viscosity of 7.9 cSt at 40ºC and 2.0 cSt at 100ºC. Complete saturation of fuel results in a viscosity decreases of 75% compared to a 10W30 lubricant as indicated in Figure 28. 20% fuel dilution in the upper liner results in a 25% increase in boundary friction and wear factor. The related drop in hydrodynamic friction was not sufficient to make up the difference in power loss either, resulting in an overall increase in total friction. Complete saturation with fuel resulted in a 300% increase in boundary friction and wear factor. At these dilution levels wear will be the primary concern. In this region high temperatures would lead to significant vaporization of fuel due to high volatility.

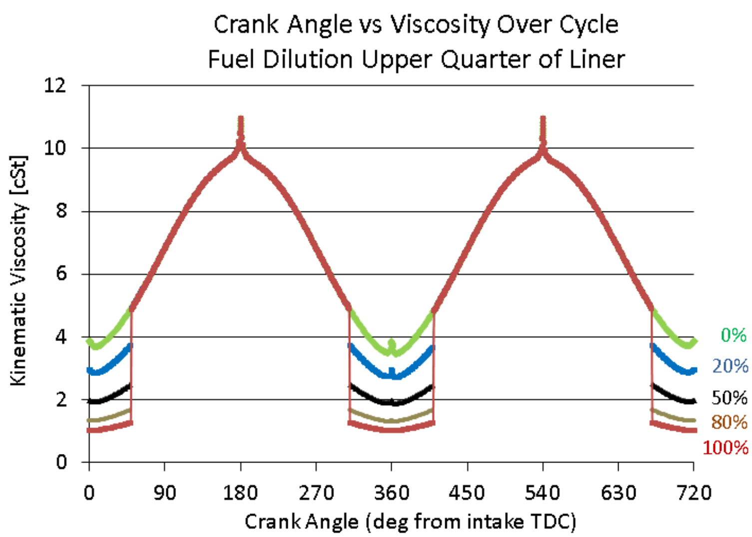


Figure 28 Effect of fuel dilution on upper quarter of liner on viscosity over one engine cycle. Mass fraction of fuel denoted on the right [2].

A more sophisticated model was subsequently developed using the simulation algorithm presented for additives. This model accounted for oil transport and vaporization, with fuel impingement along the liner as shown below. To model fuel impingement on the cylinder liner, a fuel dilution rate of 5% was used. Fuel component of similar molecular weight and boiling point as $C_{25}H_{52}$ was used. Its effect on viscosity is shown in Figure 28. Fuel impingement causes a decrease in oil viscosity along the cylinder liner. With a higher fuel dilution rate, a higher decrease in oil viscosity is expected.

The model was finally modified to account for soot transport along the liner. A detailed discussion of viscosity thickening as a result of soot entrainment is given in [4].

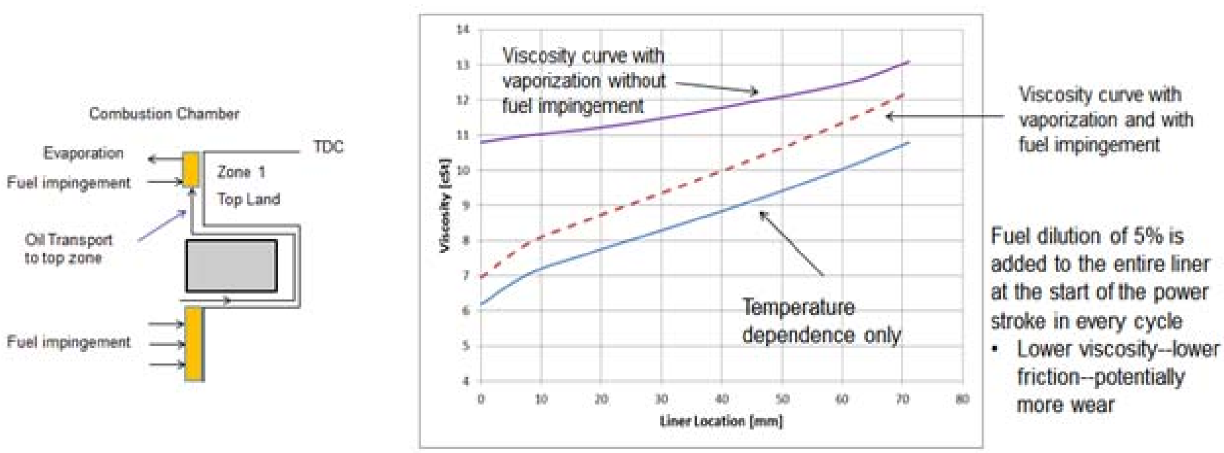


Figure 29 Fuel impingement and resulting viscosity changes along liner with 5% fuel dilution. [4]

6. Engine Power Cylinder Efficiency Modeling with composition effects – Viscosity Modifier Contributions

Lube formulation modeling was extended by considering the effects of viscosity modifiers (VMs) on base stock rheological properties. Specifically, the team sought to develop a model for the resulting viscosity-temperature behavior following mixing of a base stock with a given VM. Industry partners were consulted, along with partners from other MIT research consortiums, to develop a better understanding of viscosity modifier behavior in the practical environment such as the internal combustion engine. The goal was a model reflecting mixture compositions with varying mass fraction, similar to that discussed in the last section. Development of such a model is, at this point, highly ambitious, but could present additional opportunities for formulation similar to those determined using vaporization modeling techniques.

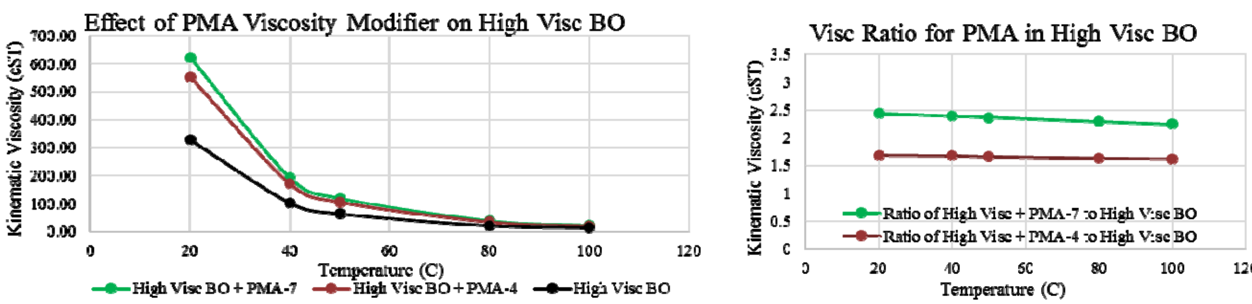


Figure 30 Thickening efficiency for base oil viscosity modifier blends.

To develop the current model, thickening efficiency was assessed using data from actual lubricants tested at a partner lab. The following figure indicates this phenomenon, wherein the viscosity modifier and base oil composition (VM+BO) viscosities are normalized by the BO viscosity. An approach to oil composition modeling along the liner, using this concept, is discussed in [2].

7. Engine Valve Train Efficiency Modeling

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 2.0 – Modeling effect of lubricant parameters on friction/wear for subsystems
Subtask 2.2 Analysis of lubricant effects on valvetrain friction as a subsystem
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.

A detailed valve train model was produced in GT SUITE, a commercial software for engine investigation. Details of the model are provided in [2]. Model results, given in Table 6, indicated the majority of losses were boundary in nature. Modeling and experimental results also suggested high pressures in the valve train may contribute to different base oil behavior [2]. Shear thinning effects were also investigated using the model.

Table 6 Valve train branch component power losses [2].

	Valve guides	Valve contact	Rocker Arm	Follower
Magnitude (W)	7.0	11.0	66.0	2.1
% of total subsystem friction	6.6	10.5	62.4	1.0
% hydrodynamic	10.0	1.0	0.0	7.0
% boundary	90.0	99	0.0.0	93.0
% mixed	0.0	0.0	100	0.0

The valve train branch, those components between the camshaft lobes and valves, was the main source of boundary and mixed lubrication behavior according to modeling results. Experimental results confirmed this, with journal bearings behaving in the hydrodynamic lubrication regime for the entire speed range of the camshaft as shown in Figure 31.

Several experiments were run on the cylinder head bench and the main test cell engine in motored and fired configurations to characterize the contribution of fuel injection, fuel pumping, and other losses to the valve train. These were then compared to previously developed valve train model results. The engine valve train exhibited similar behavior to the cylinder head bench, demonstrating mixed boundary and hydrodynamic behavior. The results were reviewed with lubricant formulation partners who noted that heavy duty diesel engines tend to exhibit predominantly hydrodynamic lubrication behavior in the valve train. As a result, where the subject engine in this study may benefit from higher viscosity lubricants in the head, other engines may not. This highlights the importance of understanding the specific tribological characteristics of engines for lubrication development.

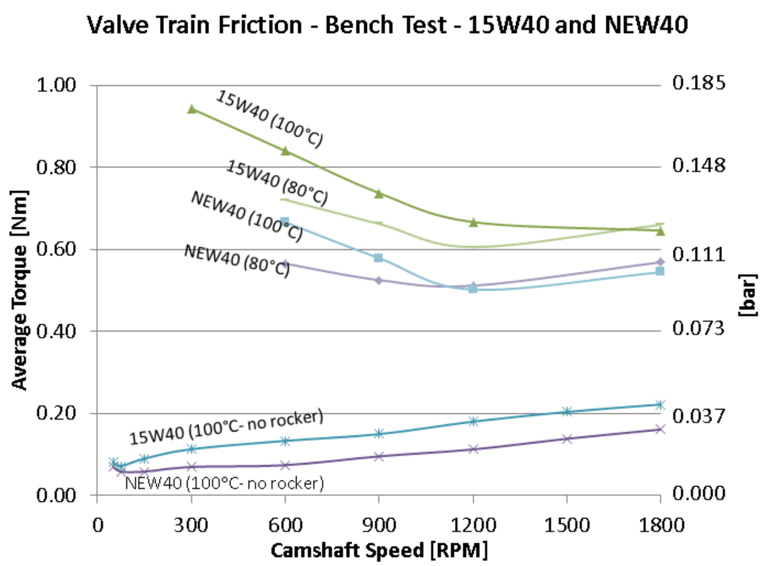


Figure 31 Valve train friction benefit from Newtonian 40 vs. 15W40 [2].

8. Engine Power Cylinder Efficiency Experimental Studies

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.3: Parametric base oil & additive experiments, power-cylinder friction
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 7.0 – Test, optimize, composite oil formulations
Task 10.0 – Evaluations of the impact on emission-control systems

Test cell engine studies were conducted with the engine dual loop lubricating system in segregated configuration, with separate lubricants in the head and the power cylinder. For cases where comparison of subsystem performance is of particular interest the lubricant in the other subsystem was kept constant to allow for comparison of total engine and specific subsystem data.

Figure 32 shows sample data in the form of power cylinder friction loads for various lubricant combinations in the head and power cylinder subsystem.

At 1800 rpm with an NIMEP of 3.5 bar a 7.1% reduction in friction is realized when comparing the case of 15W40/15W40 in the head/power cylinder and 15W40/5W20. Mechanical efficiency increased from 60.3% to 64.0%. The efficiency values are based on the reported FMEP, which does not include PMEP. The 5W20 HTHS 150 value was 81% of that of the 15W40, so a hydrodynamic friction reduction of 10% is expected. As discussed in [2], results may also be interpreted to indicate the HTHS 150 value is still a better indicator of overall engine friction than the HTHS 100.

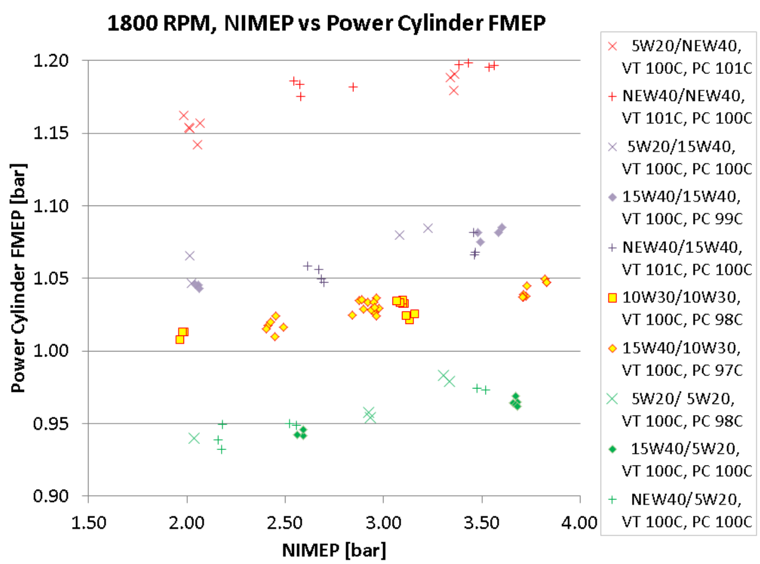


Figure 32 Power cylinder NIMEP vs. FMEP results at 1800 rpm [2].

The effects of shear thinning were investigated in the power cylinder using a multigrade 15W40 and a PAO based SAE 40 (“NEW40”) lubricant. Figure 33 compares the resulting friction losses for the power cylinder subsystem with the NEW40 and 15W40 for the 2400 rpm condition. The resulting friction increase was approximately 8% at 50% load. The difference climbs to 9% at 90% load, although at these higher loads engine operation experienced greater vibration which may have an impact on results. At higher loads local temperatures and shear rates are expected to increase, which should account for a greater impact of shear thinning on the final result. The findings are consistent with literature reports that the friction response correlates to HTHS 150, and is to be expected given higher sump temperatures at this condition.

At 1800 rpm a 9% difference is realized as indicated in Figure 32. The mechanical efficiency dropped by 3.6% when using the NEW40 in the power cylinder. Results at 1800 rpm were also used to show that friction response correlates well to the HTHS 150 value as described in [2].

Potential benefits which may be achieved by increasing the friction modifier, or decreasing the anti-wear, in the power cylinder subsystem were also studied. A common lubricant, a 15W40, was used in the head to allow for comparison of total engine friction to reduce one uncertainty factor. The head lubricant was kept constant to isolate the effects of valve train friction changes. The results indicate that increased levels of friction modifier have a small effect on the lubricant performance, an approximatley 2.5% reduction for the organic and non organic modifiers. The small change is anticipated considering the magnitude of hydrodynamic lubrication in this subsystem. If a lower viscosity lubricant were used the gains from the friction modifier may be evident due to the anticipated increased boundary friction. The measured HTHS 150 values may account for a portion of the gains realized in the tests. Some benefit was observed from the reducing the antiwear in the additive package. A 2% reduction in friction was realized at the 1800 rpm, low load condition. HTHS value comparisons indicate the gain is not due to viscosity differences and may be from reduced impact of the tribofilm and its effect on boundary friction.

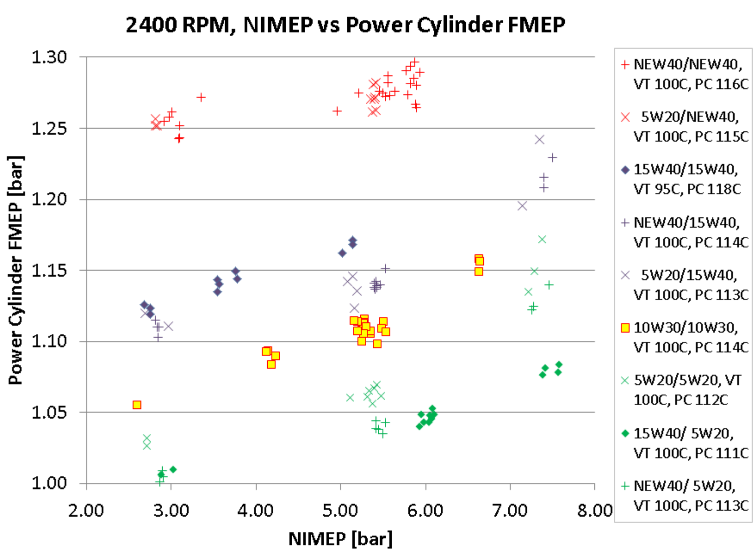


Figure 33 Power cylinder NIMEP vs. FMEP results at 2400 rpm [2].

The differences in performance fall within the range of experimental uncertainty for the test apparatus, so definitive conclusions may not be drawn as described in [2]. Research partners indicated that small benefits, of a few percent, have been realized by reducing anti-wear additives where appropriate.

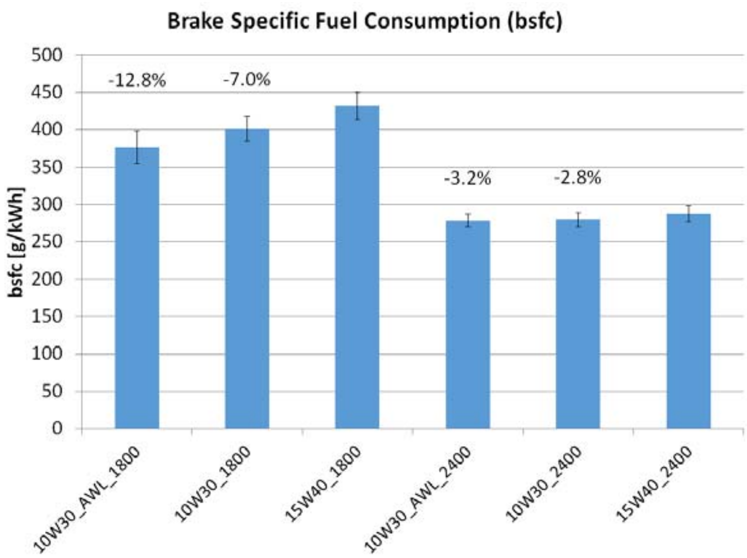


Figure 34 Brake specific fuel consumption with various lubricants [2].

During emissions testing fuel consumption was also measured to determine changes in bsfc directly. Results are given in Figure 34. Results for bsfc trended as expected, with 15W40 consumption higher than the 10W30 as explained in [2]. The measured bsfc improvement may be subject to considerable experimental uncertainty. While the trends are as expected, greater sensor resolution would improve the confidence in the result.

Experimental results supported the finding that the power cylinder subsystem is predominantly hydrodynamic. Temporary shear thinning provides an important friction benefit as well. Increased friction modifier in the power cylinder lubricant further reduced friction losses, although to a much more modest extent, and within the experimental uncertainty of the test apparatus. Reduced antiwear decreased friction by two percent at the low speed low load condition, although this was also within the bounds of uncertainty of the test apparatus. Other studies have shown that reduced antiwear, and increased friction modifier, may provide efficiency benefits.

9. Engine Valve Train Efficiency Experimental Studies

This section includes results related specifically to the following tasks listed in Section I, Part C, “Tasks Performed”:

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry

Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.4: Parametric base oil and additive experiments, valvetrain friction
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 7.0 – Test, optimize, composite oil formulations

Valve train experimental studies were conducted with the cylinder head bench and test cell engine. The subject valve train exhibited predominantly mixed lubrication behavior. Friction decreased with increased speed or viscosity at low speed operating conditions, and increasing with higher speed or viscosity at the higher end of the engine speed range as described in [3]. Viscosity had limited impact, supporting findings that, due to boundary friction, additives such as friction modifiers may have a greater effect [16].

Increased viscosity, whether by higher viscosity multigrade or decreased temperature, improved efficiency for the valve train as given in [3] and [2], particularly at low speeds for the single overhead camshaft configuration in the subject engine.

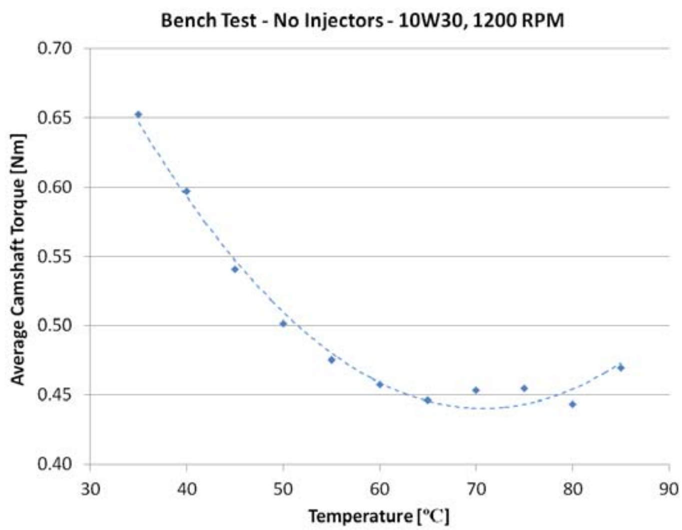


Figure 35 Valve train friction vs. temperature, 10W30 lubricant, 1200 camshaft rpm, based on data in [3].

PAO base SAE 40 lubricant reduced valve train friction up to 25% in bench tests when compared to a multigrade of the same kinematic viscosity as depicted in Figure 36 [2]. Benefits were less than 10% in the fired test engine. The effect may be a result of higher viscosity at high shear rate, or more sophisticated response of the base oil under high pressures.

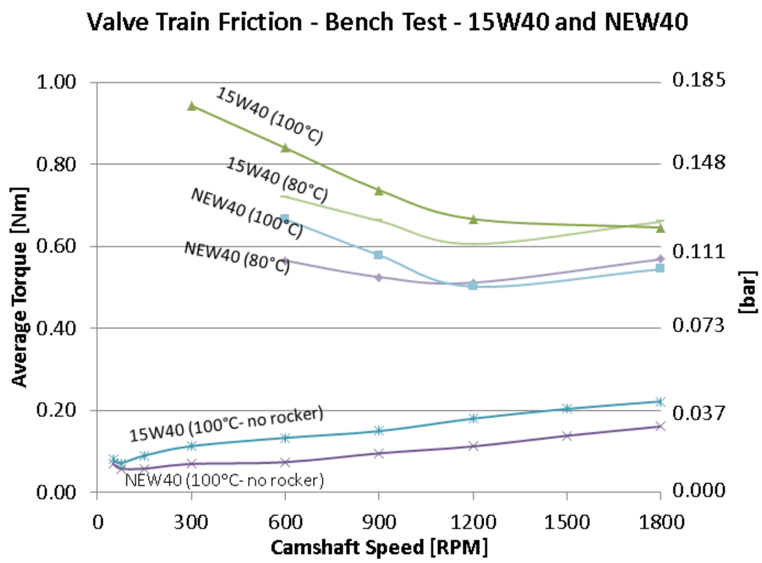


Figure 36 Valve train friction benefit from Newtonian 40 vs. 15W40 [2].

Doubling lubricant flow rate, through greater feed pressures, reduced valve train friction by approximately 10% as shown in Figure 37 [2]. The savings are on the same order as the increase in pumping loss, so more in depth study is warranted to optimize the lubricant flow rate for the given system.

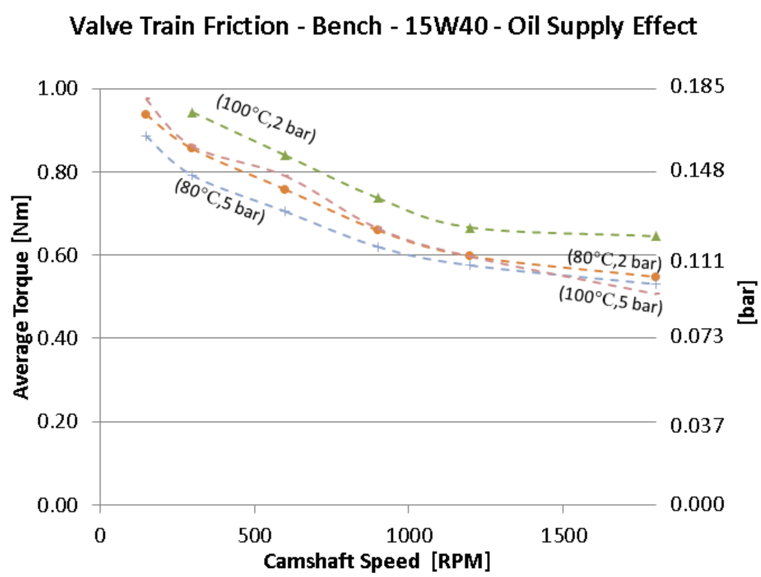


Figure 37 Effect of oil supply pressure on camshaft torque [2].

10. Engine Dual Loop Efficiency Experimental Studies

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.6: Parametric experiments to determine subsystem composition and aging trends
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 6.0 – Model lube formulations with regional variations
Task 7.0 – Test, optimize, composite oil formulations
Task 8.0 – Develop practical means to implement new formulations
Task 11.0 – Technology transfer an interfacing with users and researchers

Experimental studies for segregated lubricant systems were carried out on the test cell engine. Results at the two primary engine operating conditions are given in Figure 38 and Table 7. The dual loop lubricating system facilitated an improvement in mechanical efficiency by allowing the use of PAO base oils in the valve train, allowing a total engine friction reduction of 2% is possible, which would result in a 0.4% bsfc improvement at 80% efficiency. Under real driving conditions, with lower total mechanical efficiencies, higher gains are likely.

A dual loop engine facilitates the gain from using a PAO in three ways. First, it allows the use of the heavier lubricant without the expense of heavy hydrodynamic losses in the power cylinder subsystem. In this manner the system also allowed for gains using regular multigrade lubricants. Second, the potential for longer oil change intervals in the valve train, through use of a dual lubricant system, may make the use of Group IV base oils more attractive from a cost perspective. Finally, the system allows for more refined temperature control [2].

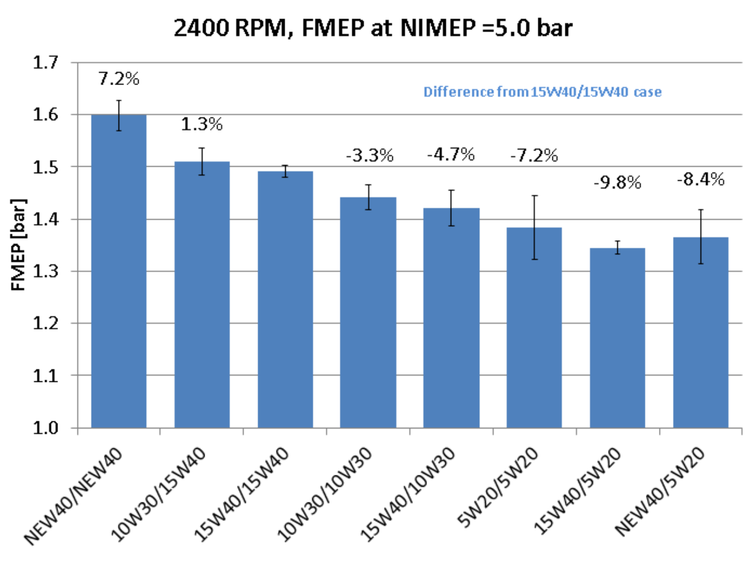


Figure 38 Dual loop friction results, fired test engine 5W20 to NEW40 lubricants at 2400 engine rpm and NIMEP 5.0 bar [2].

Experimental results in this study and others quantify the gains which may be achieved by using a lower viscosity lubricant in the power cylinder. A 5W20 lubricant, in lieu of a 15W40, provided an 8% friction reduction, corresponding to a mechanical efficiency benefit of 3.7%, and a potential fuel economy benefit of 1.5% for the low speed low load test condition studied. This is consistent with recent reports in the literature. Splitting the lubrication system may allow for the use of lower viscosity lubricants in the power cylinder subsystem. As with the valve train, it may afford greater opportunity for in situ control of temperature as well, since control changes in the power cylinder lubricant should not adversely affect valve train performance.

The segregated system allowed for improved mechanical efficiency by providing greater opportunity for in situ rheological property control, as in the control of viscosity through temperature variation in the head.

Dual loop lubricating systems allow increased opportunities to moderate lubricant flow rate in particular engine subsystems at different engine operating conditions. Optimization of lubricant flow rate may reduce engine friction losses in the subject test engine by approximately 10% when auxiliary systems are taken into account [2]. The use of a 5W20 in the power cylinder subsystem for the test engine would result in a 25% reduction in oil pumping power requirements. This may account for a significant portion of the gains measured from the 5W20 lubricant use in the experimental studies.

Table 7 FMEP results for 1800 rpm condition with relative difference for common vs. different lubricants [2].

Configuration	FMEP 3.5 bar	FMEP 3.5 bar uncert	% FMEP difference 15W40/ 15W40	% FMEP difference common	Mech Efficiency (w/out PMEP)
5W20/NEW40	1.515	0.028	9.1%	0.1%	56.7%
NEW40/NEW40	1.513	0.019	9.0%	0.0%	56.8%
5W20/15W40	1.390	0.080	0.1%	0.1%	60.3%
15W40/15W40	1.388	0.032	0.0%	0.0%	60.3%
NEW40/15W40	1.378	0.052	-0.8%	-0.8%	60.6%
15W40/10W30	1.334	0.019	-3.9%	-1.2%	61.9%
10W30/10W30	1.350	0.009	-2.8%	0.0%	61.4%
5W20/5W20	1.290	0.046	-7.1%	0.0%	63.2%
15W40/5W20	1.259	0.017	-9.3%	-2.4%	64.0%
NEW40/5W20	1.267	0.017	-8.8%	-1.8%	63.8%

Additive optimization, possible as a result of decoupling of functional requirements in a dual loop lubricating system, may provide additional opportunity for improved mechanical efficiency. This is discussed in further detail in [2].

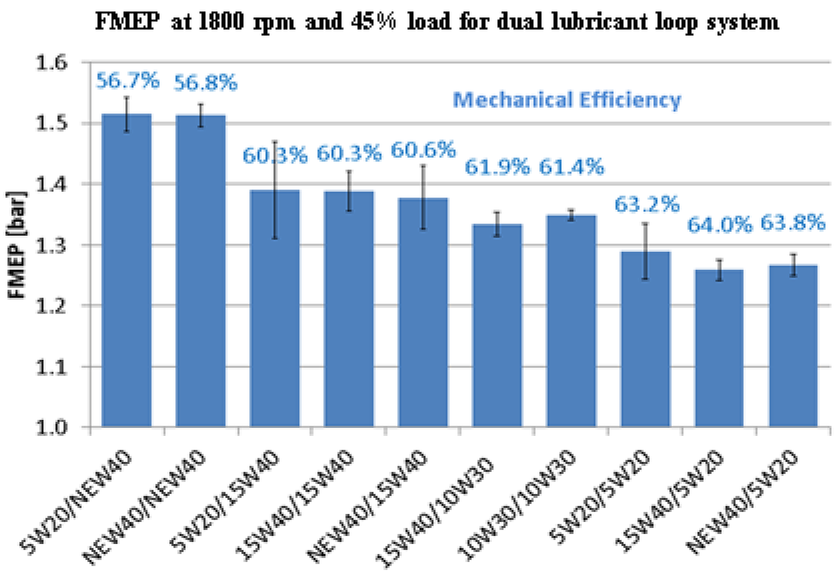


Figure 39 Parametric test results from test cell engine for varied base oils in the valve train/power cylinder subsystem [2].

11. Engine Oil Aging Studies

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.5: Design and install of long term oil aging test engine*
Subtask 4.6: Parametric experiments to determine subsystem composition and aging trends
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 6.0 – Model lube formulations with regional variations
Task 8.0 – Develop practical means to implement new formulations
Task 9.0 – Demonstrate, in an actual engine, quantitative improvements in mechanical efficiency of best formulations from study
Task 11.0 – Technology transfer an interfacing with users and researchers

While not a main objective of the present work, the role of a segregated oil system in extending oil drain intervals was investigated to determine opportunities for benefits as well as inform tasks directed at modeling oil composition in different subsystems. The effort, conducted primarily through experiment study with the mobile oil aging rig, resulted in improved knowledge of lubricant properties in each subsystem, as well as quantification of the potential benefits of operating with a segregated lubrication system. A long duration test, consisting of one regular oil drain interval for the subject engine, was conducted with regular lubricant samples drawn and sent to laboratory for analysis. The test conditions are described in the following table.

Table 8 Test condition range for 250 hour oil aging test with mobile aging rig [2].

Ambient Temperature	16-36°C
Ambient relative humidity	40-80%
Valve train oil sump	64-73°C
Valve train oil pressure	85-93 psi
Load	2.1 kW (32%)
Main oil sump	95-101°C

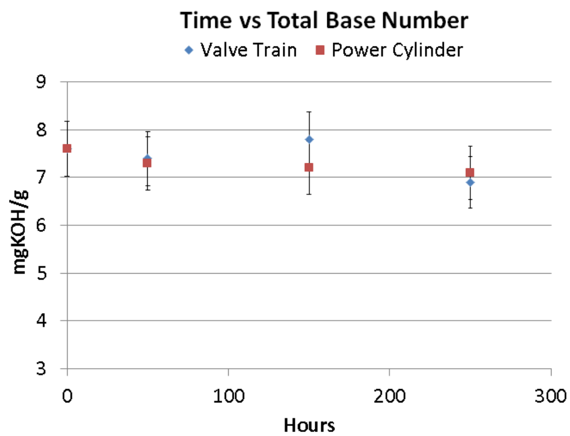
Details of the oils analysis procedures are given in [2]. Results for viscosity tests are summarized in the following table. Samples were drawn from the power cylinder and valve train subsystems. KV100 denotes kinematic viscosity at 100ºC. DV denotes dynamic viscosity.

The dual lubricating system protected components in the power cylinder subsystem from fuel contamination, as can be seen in the results of Table 9. The issue raises a concern for dual loop lubricating systems, in that the valve train lubricant, not exposed to the high crank case and cylinder temperatures, does not benefit from volatilization of lighter fuel species. Periodic heating of the valve train sump may provide a balance between long term oxidation concerns and the desire to minimize volatiles such as fuel and water.

Table 9 Viscosity data from 250 hour dual lubrication loop field test including fuel dilution [2].

Location	Time (hr)	KV40 (cSt)	KV100 (cSt)	KV100, % change	Density 40°C (g/ml)	Density 100°C (g/ml)	DV100 (cP)	HTHS 100 (cP)	μ_∞/μ_0	Fuel Dilution (mass %)
New Oil (15W40)	0	112.6	14.73	0	0.8627	0.825	12.2	9.4	0.77	0.0
Valve Train	50	101.8	13.81	9.6	0.8628	0.825	11.4			1.34
	150	89.76	12.77	20.3	0.8624	0.825	10.5			4.08
	250	81.73	12.01	27.4	0.8615	0.824	9.9	8.05	0.81	5.30
Power Cylinder	50	109.3	14.42	2.9	0.8634	0.826	11.9			<0.01
	150	110.7	14.52	1.7	0.8635	0.826	12.0			<0.01
	250	111.1	14.63	1.4	0.8642	0.826	12.1	9.73	0.80	<0.01

Additive depletion was also investigated, along with changes in total base and acid number, and oxidation behavior as detailed in the following figures. Water levels remained low in each subsystem for the given operating conditions despite significantly lower oil sump temperatures in the valve train. Condensation may be of concern in different operating environments. Oxidation levels in the valve train subsystem were high, possibly due to the fuel dilution experience due to an issue with the fuel rail seals in the valve train.



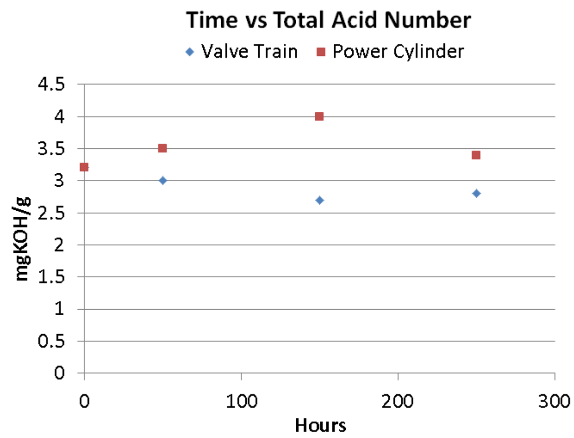


Figure 40 Total base number and total acid number for valve train and power cylinder lubricants [2].

Dual loop systems are expected to maintain lower TAN in the valve train, justifying reduction of initial TBN in valve train lubricants. In this study the 250 hour test, combined with the use of ULSD per new fuel standards, was insufficient to show a significant change in TBN. The results are given in Figure 40. Slight increases in TAN in the power cylinder, relative to the valve train, were observed during tests as expected. The use of a lower TBN lubricant may be possible in the valve train given the lack of exposure to high temperatures. This could potentially save on detergent and additive cost.

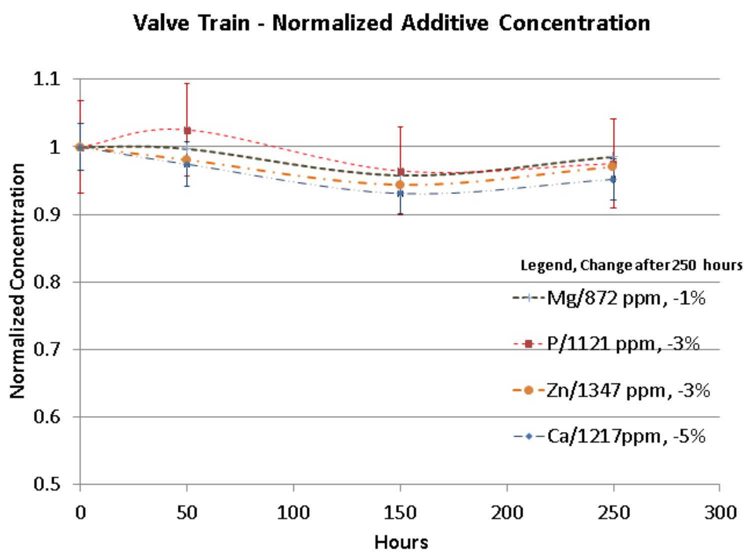


Figure 41 Additive concentrations for valve train lubricant. Change at 250 hours shown in legend. Error bars for P and Ca given based on ASTM D5185 repeatability [2].

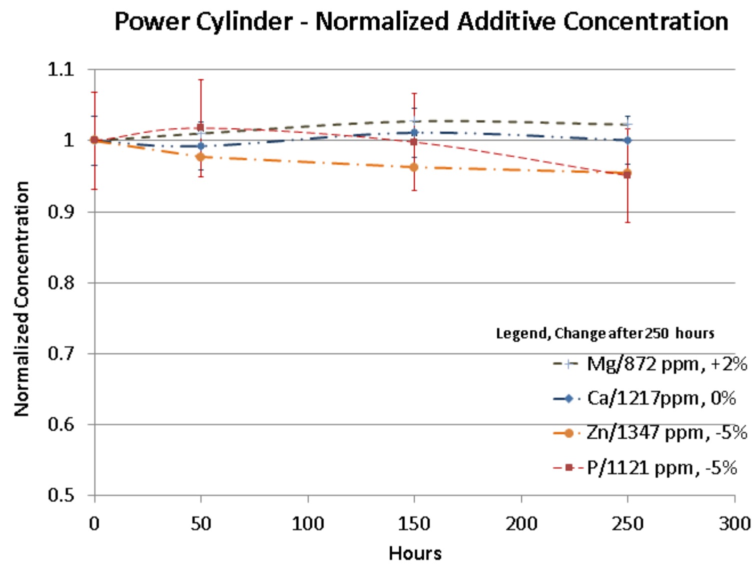


Figure 42 Additive concentrations for power cylinder lubricant [2].

Visual analysis, as indicated in Figure 43, indicates the valve train was protected from soot contamination. Soot results from FTIR analysis confirm this per Table 10.

Table 10 FTIR results for valve train and power cylinder field test [2].

Source	Time (hr)	Nitration (A/cm)	Oxidation (A/cm)	Soot (A/cm)
New	0	0.32	0.63	0.00
Valve Train	250	0.87	6.53	0.13
Power Cylinder	250	4.79	4.63	3.15

The aging study suggests the use of a dual loop lubricating system offers significant opportunities for improved oil performance over time, which can lead to reduced viscosity changes and wear, as well as improved oil drain intervals.

Figure 43 Oil samples at 50 and 250 hours. New oil is on the left of each photo. The valve train sample is in the center, and power cylinder lubricant sample on the right [2].

12. Engine Aftertreatment impact considerations

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 3.0 - Develop experimental/analytical lubricant test parameters in consultation with team participant(s) from lubricant/additive industry
Task 4.0 – Develop parametric experiments, lubricant & additive effects on subsystems
Subtask 4.3: Parametric base oil & additive experiments, power-cylinder friction
Task 5.0 – Data analysis, interpretation, and iteration between modeling and testing.
Task 10.0 – Evaluations of the impact on emission-control systems

Experiments in the study considered the emissions impact of reducing antiwear additives in the crank case to investigate potential fuel economy benefits as well as opportunities for optimizing formulations with respect to aftertreatment systems. Emission samples were taken while the test cell engine was operated using 15W40 lubricant in the head and three different lubricants in the power cylinder, 15W40, 10W30, and reduced antiwear 10W30. Samples were taken at the two experimental engine operating conditions.

Lubricant formulators did not disclose the elemental composition of the lubricants used, and only indicated the 10W30_AWL had 'reduced anti wear'. The results of soot concentration, based on ICP-AES analysis by an independent lab, are given in Figure 44. Samples were managed as discussed in [2]. Trends for the 2400 rpm case are as expected. P and Zn concentrations were lower for the 10W30_AWL as opposed to the 10W30. Repeatability for the tests is not known. Only one or two samples were submitted for analysis for each case, so a suitable assessment of precision may not be made. Repeatability may be further suspect given the results of the 1800 rpm test results. In those tests Al and Fe levels were higher for the 15W40 than the 10W30 and 10W30_AWL. The 10W30 lubricant still exhibited higher levels of P, Zn, and Ca concentration in the soot than the 10W30_AWL and 15W40.

Soot emission results are given in the following figure. The result differences are within the uncertainty of the measurements, so general conclusions are difficult to draw. Oil contributions will likely be below 25%. ULSD fuel was used. Ash contributions will be an order of magnitude lower, so differences may not be resolved from soot estimates alone. Ash concentrations may be determined through other means as discussed with the ICP-AES results. Decreased viscosity has been shown to result in increased soot [30][31].

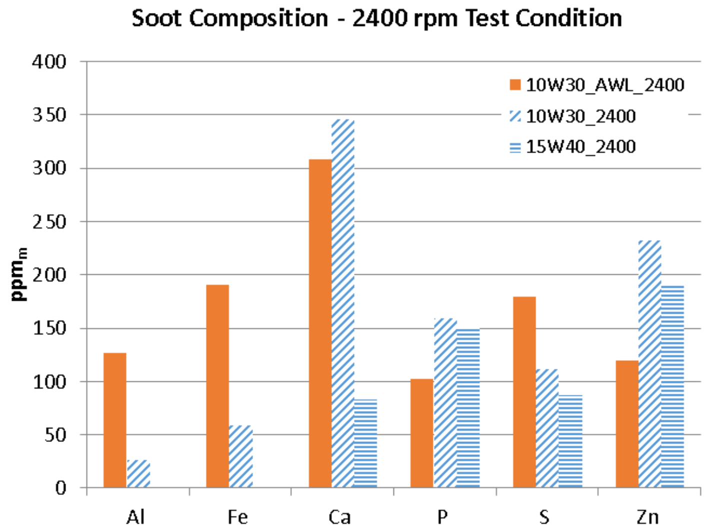


Figure 44 Elemental composition of soot samples taken under 2400 rpm operating condition using ICP-AES [2].

10W30_AWL has lower levels of ZDDP, and lower concentrations of these additives in the soot for the 15W40 case may indicate the 15W40 exhibits lower oil consumption. The absolute level of ZDDP is difficult to determine from emissions alone. The 15W40 is expected to have lower consumption due to lower volatility and increased viscosity as indicated in other studies [2] [29]. Volatility is limited to 15% for 10W30, and 13% for 15W40 per CJ-4 specifications. Formulators indicated the lubricants had the same additive packages, although the elemental composition is not known.

Some 10W30 samples were taken with and without the crankcase breather installed for comparison. Soot sampling was conducted with the crankcase breather attached and disconnected. The modified split system engine included a crankcase breathing tube routed straight from the crankcase to the intake. No significant differences in additive concentration or soot emission were detected from these tests. The average soot emission at 1800 rpm with the breather out was higher than with the breather in, which was not expected. The differences were within a standard deviation of the gravimetric measurements. They are therefore well within the uncertainty of the measurement if precision error of the brake power is taken into account. Increased sampling may improve the precision of the measurement, however large differences are not expected.

13. Opportunities for training and professional development

13.1. General opportunities

This section includes results related specifically to the following tasks listed in Section I, Part C, "Tasks Performed":

Task 1.0 – Project Management and Plan
Task 11.0 – Technology transfer an interfacing with users and researchers
Task 12.0 – Reviews and Reports

The project provided significant opportunities for the training and professional development of graduate students. Five graduate students participated in significant professional development in the form of experimental design, engine modification, literature review, and presentation over the course of the study. Four completed masters' degrees, one completed a doctorate. Professional development and training of all participants included experimental design, development of analytical tools, and work with partners, suppliers, co-workers, and superiors in technology sharing and procurement procedures. The team held half day quarterly reviews with industry partners to share knowledge and effectively plan future efforts. One graduate student was named to, and served on, the STLE Engine and Drivetrain committee, chairing peer review sessions at the 2014 and 2015 annual meetings.

Industry partners included:

Infineum, LLP –Linden, New Jersey – Additive manufacturer collaborating in lubricant formulation, defining test parameters, and furnishing lubricant/lubricant-additives

Kohler Engine Inc. – Kohler, Wisconsin –Advisory role in engine configuration details. Provided engines at cost, donated cylinder head.

Cummins Filtration Inc – Cookeville, TN - Provide feedback and assistance with practical lubricant enabling technologies that will make the implementation of the optimized lubricant formulations in the engine possible.

This program explores pragmatic application of available lab technologies for use in lubricant development. This program presents opportunities for optimizing engine and lubrication system efficiency without significant cost. Potential savings in energy used and operating costs could be substantial.

This program prepares mechanical engineering students and researchers to integrate lubricant technology (traditionally a chemical engineering focus) into future design. Our engineers will be better prepared and more competitive in the global economy and industry.

The advisory group model of involving industry participants facilitates the transfer of technologies. It also improves industry responsiveness to these efforts and the likelihood that they may be adopted in the future.

13.2. Papers and presentations, dissemination of knowledge

Dissemination of knowledge to parties of interest was an important project goal. The team delivered the following papers and presentations:

- Plumley, M. J., Wong, V., Molewyk, M., and Park, S.-Y., "Optimizing Base Oil Viscosity Temperature Dependence For Power Cylinder Friction Reduction," SAE Technical Paper 2014-01-1658, 2014, doi: 10.4271/2014-01-1658.
- Molewyk, Wong. "In Situ Control of Lubricant Properties for Reduction of Power Cylinder Losses through Thermal Barrier Coating" SAE Paper 2014-01-1659
- Plumley, Wong. "Analysis of Shear-thinning on Engine Friction Using Mineral and PAO Base Oils". Presentation. STLE 69th Annual Meeting and Exhibition, May 18-22, 2014, Orlando, FL.
- Plumley, Wong. "Optimizing Base Oil Viscosity for Power Cylinder Friction Response Temperature Dependency" Presentation. STLE 69th Annual Meeting and Exhibition, May 18-22, 2014, Orlando, FL.
- Martins, Plumley, Wong. "Engine Lubricant Viscosity Optimization for Valvetrain and Power Cylinder Systems". Presentation. STLE 69th Annual Meeting and Exhibition, May 18-22, 2014, Orlando, FL.
- Molewyk, Wong. "In Situ Control of Lubricant Properties for Reduction of Power Cylinder Losses through Thermal Barrier Coating". Presentation. STLE 69th Annual Meeting and Exhibition, May 18-22, 2014, Orlando, FL.
- Gu, Wong. "Development and application of a lubricant composition model to study effects of oil transport, vaporization, fuel dilution, and soot contamination on lubricant rheology and engine friction". Presentation. STLE 69th Annual Meeting and Exhibition, May 18-22, 2014, Orlando, FL.
- Cheng, Wong, Plumley, Martins, Molewyk, Gu, Park. "Lubricant Formulations to Enhance Engine Efficiency in Modern Internal Combustion Engines Project ID FT019", Presentation. US Department of Energy 2014 Annual Merit Review, June 19, 2014, Washington, DC
- Martins, T., "Enhanced Engine Efficiency Through Subsystem Lubricant Viscosity Investigations," Massachusetts Institute of Technology, Cambridge, MA, 2014.
- Gu, G. X., "Development and application of a lubricant composition model to study effects of oil transport, vaporization, fuel dilution, and soot contamination on lubricant rheology and engine friction," Thesis, Massachusetts Institute of Technology, 2014.
- Molewyk, M. A., "In situ control of lubricant properties for reduction of power cylinder friction through thermal barrier coating," Thesis, Massachusetts Institute of Technology, 2014.
- Plumley, M., "Design and Prototype of Dual Loop Lubricant System To Improve Engine Fuel Economy, Emissions, and Oil Drain Interval," Thesis. Massachusetts Institute of Technology, Cambridge, MA, 2015.
- Plumley, M. J., Wong, V., Martins, T, "Design and prototype of modern diesel engine lubricating system to improve fuel economy, wear, and emissions performance," Accepted, 70th STLE Annual Meeting and Exhibition, May 17-21, 2015, Dallas, TX.

III. SUMMARY AND CONCLUSIONS

The study affirmed many findings available in the literature. The majority of power cylinder friction was determined to be hydrodynamic in nature. This was shown extensively with modeling results, as well as experimental results on a 16 hp twin cylinder diesel engine. Measured friction reductions corresponded to reductions in hydrodynamic friction anticipated as a result of reduction in the HTHS 150 viscosity of the lubricant. Experimental results also showed that HTHS 150 was a better indicator of friction reduction than viscosity values measured at lower shear rates or lower temperatures. For instance, HTHS 150, the viscosity measured at 150ºC and 10^6 s^{-1}, better predicted friction response than the viscosity measured at 100ºC and 10^6 s^{-1}.

The use of conventional friction models, with detailed consideration of parameters in modern heavy duty diesel engines, indicated different functional requirements may exist for different portions of the cylinder liner. Friction along the liner due to ring and skirt travel was shown to be predominantly hydrodynamic, particularly in the presence of low cylinder pressures loads against the top ring. The exception is that boundary loads, and therefore wear concerns, are greatest near TDC. Opportunities exist to reduce friction and wear through in situ control options such as the use of Thermal Barrier Coating (TBC) as well as tailored lubricant compositions aimed at taking advantage of the different requirements along the liner. In a related study by one of the participants, additional fuel economy gains through in situ control of liner temperatures were identified. Thermal barrier coating (TBC) insulation on the liner was shown to 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.

There may be considerable benefit to varying the ratio of light to heavy hydrocarbons, and even viscosity modifying polymers, to take advantage of the potential wear reduction benefits near TDC of the liner for a specific vaporization environment. First of its kind lubricant composition friction modeling indicated several opportunities for formulation improvement. More advanced coupling of detailed composition and rheological models to those used for friction could provide for key insights into optimizing formulations for particular applications. A general approach is suggested which requires further development. Contaminants, including fuel dilution and soot, may significantly change the viscosity.

Base oil viscosity reductions significantly reduced engine friction and improved mechanical efficiency. For the 1800 rpm, 45% load test condition on the subject engine reducing HTHS 150 by 19% in the power cylinder, through use of a 5W20 lubricant, while maintaining the prescribed 15W40 lubricant in the valve train, resulted in a 7.1% reduction in total engine friction and a corresponding 3.7% improvement in mechanical efficiency. Brake specific fuel consumption benefits were also measured directly, based on gravimetric measurements of fuel consumption, when switching from a 15W40 to a 10W30 lubricant.

Investigations into the effects of temporary shear thinning indicated significant benefits may also be achieved from optimizing high shear properties of lubricants. Two lubricants of different base oils, a group II/III and a group IV, with the same additive packages and kinematic viscosities, were used to investigate this behavior. For the low speed, low load test condition on the subject engine increasing HTHS 150 by 24% in the power cylinder, using a pure Newtonian SAE 40 instead of a multigrade 15W40, resulted in a 9.0% increase in total engine friction and a corresponding reduction in mechanical efficiency of 3.6%.

The study experimental results indicated a potential fuel economy benefit from increasing friction modifier concentrations, or reducing those of anti-wear additives. The benefits were relatively modest, reducing friction on the order of 1 percent, which was within the uncertainty of the test apparatus. Small friction reductions have also been reported in the literature for each case. Gains will be greater when lower viscosity lubricants are used in the power cylinder, so additional investigation, particularly when very low viscosity lubricants are used, is warranted.

The experimental results indicated the single overhead camshaft configuration for the test engine exhibited predominantly mixed lubrication behavior. Modeling, as well as tear down tests, indicated that journal bearings exhibited hydrodynamic behavior, as expected, yet other components such as rocker arms and valve guides exhibited greater degrees of boundary behavior. Results indicated that viscosity reduction had a limited impact on valve train friction, findings in literature suggesting that friction modifiers may be of greater value in this subsystem. Experimental results in this study also indicated that use of higher viscosity lubricants in the valve train reduced friction, and therefore may reduce wear, particularly at lower speeds. The use of a PAO base SAE 40 lubricant in the valve train reduced valve train bench test friction by up to 25%. Benefits in the fired engine were more modest, at 10%. The effect may be a result of higher viscosity at high shear rate, or more sophisticated response of the base oil under high pressures.

Variation of lubricant temperature and flow rate was investigated. Doubling lubricant flow, through greater feed pressures, reduced valve train friction by approximately 10%. Reducing lubricant temperature in the valve train, and therefore effectively increasing viscosity, provided a friction reduction benefit at low speeds. At higher speeds, when camshaft journal friction became more significant, overall valve train friction increased with lower temperatures.

The dual loop lubricating system configuration facilitated an improvement in mechanical efficiency by allowing the use of PAO base oil in the valve train, with total engine friction reductions of up to 2% possible based on cylinder head bench test studies. The configuration facilitated PAO use by eliminating the adverse effects associated with giving up shear thinning in the power cylinder subsystem, allowing for greater cost benefit due to longer oil change intervals in the valve train, and allowing for more refined temperature control in local regions.

The dual loop lubricating system also allowed for improved mechanical efficiency by allowing the use of higher viscosity multigrade oils in the valve train and lower viscosity multigrades in the power cylinder, allowing for improved wear protection in the head. The hesitation to shift to lower viscosity lubricants in the engine is driven by wear concerns. Elastohydrodynamic pressures in the power cylinder subsystem should be lower than in the valve train.

Additive optimization resulting from decoupling of functional requirements in a dual loop lubricating system provides for improved mechanical efficiency. Some friction modifiers may be incompatible with certain dispersants according to literature, hence segregated oil systems may allow introduction of previously incompatible modifiers to the valve train. Reduction of detergents in the valve train may allow for less competition with ZDDP as well. Conversely, in the power cylinder, overbased detergents may be allowed which comparable wear protection under the lower elastohydrodynamic pressures encountered in this subsystem.

Dual loop lubricating systems offer opportunities for improved oil performance including improved oil drain intervals. The study demonstrated low water levels in each subsystem despite operation at lower temperature in the head. The study also demonstrated protection of each subsystem from contaminants of the other. In the subject engine case, soot was effectively removed as a wear concern from the valve train subsystem. Lower total acid numbers were also realized in the valve train, indicating detergent reduction may be possible in this subsystem. The study showed oxidation may be a concern in the valve train. This would need to be studied through additional long term tests, as fuel dilution in the subject engine may have led to increased oxidation.

Novel engine configurations may provide immediate emissions benefit by allowing use of low and zero SAPS lubricants. The use of different coatings and component designs was not specifically studied in this program, but they may warrant future research with regard to segregated oil systems. Incorporation of zero ZDDP lubricants in the power cylinder, while retaining formulations benefiting valve train performance, would allow an immediate improvement in emissions performance with no loss, or even an increase in, valve train wear protection.

IV. RECOMMENDATONS FOR FUTURE RESEARCH

Results indicate significant fuel economy improvements can be gained through optimization of lubricant properties in particular engine subsystems. Additional investigation into the practical implementation of dual loop lubricating systems may provide immediate opportunities to take advantage of these findings using conventional lubricants already available. Development of in situ control techniques, such as local temperature control in specific engine regions, could also allow the opportunities to be leverages.

The current program's development of novel composition friction modeling demonstrated potential for significant fuel economy and engine durability gains through lubricant formulation tailored to specific engine subsystems. These findings highlight the utility of such models in optimizing lubricant systems for improved performance.

The current program has possible spinoffs for other industries, including automotive, railroad, marine, or diesel power generation applications. Consortia or working groups involving various government agencies and University-Industry should be continued to ensure that the full potential for friction reduction in these areas will be fully explored and further areas for study identified.

Potential deployment of dual loop engines, and their related lubricants, in the near term is not only possible, but quite probable, if future continued funding is made available in this area.

V. ACKNOWLEDGEMENT

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