IMPACT OF FRICTION REDUCTION TECHNOLOGIES ON FUEL ECONOMY FOR GROUND VEHICLES

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ABSTRACT

The Department of Defense is a major consumer of petroleum products – over 700 million gallons per day. While the majority of fuel consumed is for aircraft, in terms of logistics and exposure of personnel to hazardous conditions, the amount of fuel consumed in ground vehicles is considerable, with the cost (in-theatre, delivered) ranging from $100 to $600/gallon. This paper addresses the impact that parasitic friction mechanisms (boundary lubrication and lubricant viscosity) have on engine friction and overall vehicle efficiency. A series of mechanistic models of friction losses in key engine components was applied to investigate the impact of low-friction technologies on the fuel consumption of heavy-duty, on-road vehicles. The results indicate that fuel savings in the range of 3 to 5% are feasible by reducing boundary friction and utilizing low-viscosity engine lubricants. The paper will discuss the implications of the studies (as performed for commercial heavy-duty trucks) for military ground vehicles, which have significantly different driving modes. The paper will also discuss the potential of different strategies to implement low-friction/low-viscosity solutions and the impact of these strategies on reliability and durability.

INTRODUCTION

Consumption of petroleum in the US has historically increased with time to the point where it has now reached approximately 37 Quads (quadrillion, 10^{15}, BTUs) annually – equivalent to 17.4 million barrels of oil per day [1,2], or 25% of the world's petroleum consumption. Of this, approximately 28 Quads (13 Mbbl/d) is used in the transportation sector. The federal government is the largest single consumer, accounting for 2% of petroleum consumed [3], with the Department of Defense (DoD) consuming 93% of all federal use – equivalent to 360,000 bbl/d. This severe dependence on petroleum has a significant impact on the US economy and the ability of DoD to fulfill its mission. Consequently, there is considerable interest in developing advanced transportation systems that reduce consumption of petroleum. These technologies include alternative fuel sources (biofuels, hydrogen, etc.) to displace petroleum, lightweight materials to reduce vehicle mass, advanced combustion cycles to improve combustion efficiency, hybrids, and low-friction engine and drivetrain technologies (the subject of this paper).

The concept of reducing parasitic energy losses in engines and drivetrains is not new – numerous studies can be found in the open literature on advanced lubricants, additives, materials, and coatings and their impact on friction. For example, diamond-like carbon (DLC) coatings [4] are finding widespread utilization in vehicles. Initially, they were most commonly used on fuel injection components to improve reliability and durability, but recently they have been applied to components that are oil lubricated with the intention of reducing friction losses and fuel consumption [5]. Advanced lubricants and additives are also under development [6] to improve fuel economy and to accommodate more aggressive engine environments [7]. Additional examples on the fundamentals and technologies involved in lubrication can be found in references 8 and 9.

While great progress is being made in the development of technologies to reduce parasitic energy losses in vehicles, determining the impact of low-friction technologies on fuel economy and durability and reliability can be challenging. A 50% reduction in friction does not imply a doubling of fuel economy. The relationship between friction reduction and fuel economy is complex and dependent on a number of parameters, including engine and drivetrain component design, materials, lubricant/additive packages, and driving schedule. The objective of the research project whose results are presented below was to determine the impact of friction on fuel economy, namely – "If I can reduce boundary film lubrication by X% and/or use a low viscosity engine lubricant, how much fuel would be saved?" With this information (fuel savings as a function of boundary film friction and lubricant viscosity) and data from experimental friction test rigs, we identified several candidate technologies that could potentially provide the level of friction reduction used in the models/codes. The research also examined the impact of advanced additive technologies on the reliability of materials under simulated loss-of-lubricant conditions.

APPROACH

The approach used an integrated suite of computer codes to predict the impact of friction (boundary and viscous) on the fuel economy of heavy-duty (9-12 L) diesel engines, such as those typically used in class 7-8 over-the-road trucks. Integration of four commercial codes (PISDYN, RINGPAK, VALDYN, and ORBIT) was performed by Ricardo Engineering under subcontract to Argonne National Laboratory. These codes were used to model parasitic boundary and viscous (oil shearing) losses in the following engine components:

- Piston skirt to cylinder liner
- Piston rings to cylinder liner
- Cam to cam bearings
- Cam to follower
- Pushrod to rocker arm
- Rocker arm to valve bridge
- Connecting-rod small end bearings
- Connecting-rod large end bearings
- Crankshaft main bearings

PISDYN was used to model the friction forces between the piston skirt and the cylinder liner as the piston travels up and down the liner. It is a time-dependent simulation of piston secondary dynamic motion based on hydrodynamic and boundary lubrication at the skirt/liner interface and wrist pin. The PISDYN calculations used results from a detailed study of a commercial six-cylinder in-line diesel engine (10 L displacement) with articulated pistons.

RINGPAK was used to model the friction between the rings and the liner. Like PISDYN, RINGPAK is a time-dependent simulation of ring motion and incorporates both boundary and hydrodynamic friction mechanisms. The RINGPAK calculations used results from the same engine used in the PISDYN calculations.

ORBIT was used to model time-dependent motions in the main and large end bearings of a six-cylinder in-line (14.6 L) diesel engine. ORBIT uses a time-dependent simulation of boundary and hydrodynamic lubrication in the bearings.

All three models used a mass-conserving solution to the Reynolds equation to predict oil film pressure and thickness/clearance between mating components. A Greenwood-Tripp model was used to evaluate asperity contact during boundary and mixed-lubrication regimes.

VALDYN was used to calculate friction at key interfaces in the valve train of a six-cylinder in-line (9 L) diesel engine. The valve train in this engine employed a roller-follower with each cam lobe operating two valves. In this case, the models used a simple (boundary) friction model with no hydrodynamic lubrication.

Each code was run at eight engine load/speed conditions to calculate the change in friction forces relative to a baseline condition (baseline condition defined as 40 WT mineral oil with simple, constant boundary friction coefficients – see reference 10 for details on the baseline boundary friction coefficients used for each component). The eight operating conditions are shown in fig. 1.

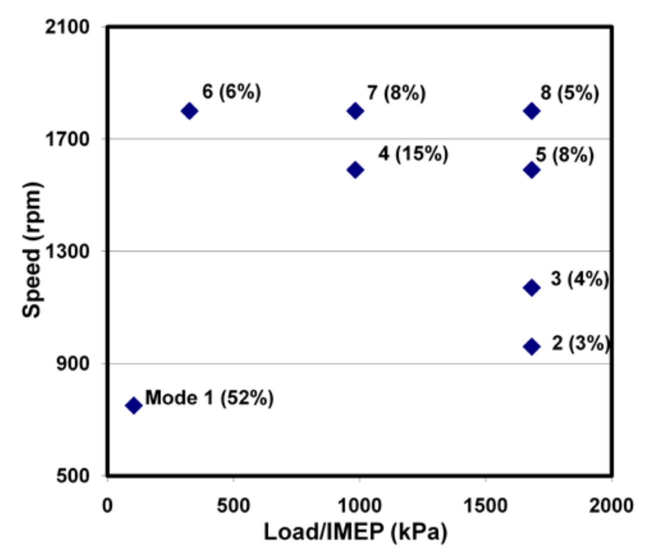

Figure 1: Map Showing Eight Engine Conditions Simulated in Computer Models by Ricardo Engineering

The concept of mean effective pressure (MEP) was used in the modeling effort to normalize the power (indicated, brake, or friction) to the engine displacement. The MEP is the work per cycle divided by the engine displacement. The numbers alongside each of the data points correspond to the mode number (1 through 8) and the weighting factor (in parentheses) – corresponding to the fraction of time each mode contributes to the overall driving schedule. Ricardo has found that the eight-mode approach can be used to simulate different driving schedules. The weighting factors in fig. 1 are those used to simulate a heavy-duty federal test procedure cycle (HD-FTP). Different weighting factors can be used to simulate other driving cycles – for example, the weighting factor for idle in the study by Fox [10] was assumed to account for 52.4% of the time to simulate the HD-FTP cycle; however, the driving cycle for wheeled and tracked vehicles can experience considerably greater idle fractions (as high as 80% [3]) and can account for as much of 60% of the fuel consumed during a mission.

The four models were run under different conditions (boundary friction and lubricant viscosity) at the eight modes shown in fig. 1. These cases correspond to reductions in the boundary friction coefficient of 0, 30, 60, and 90%, and to lubricant viscosities equivalent to single-grade SAE 5, 10, 20, 30, 40, and 50 WT mineral-based engine lubricants. Changes in the friction mean effective pressure (FMEP) relative to the baseline case (40 WT oil with baseline boundary friction coefficients) were calculated over a 720^{o} cycle. A fuel consumption scaling factor (FCSF) was developed for each case, which, in turn, was used to multiply the fuel consumption rate at each of the modes in fig. 1. In these studies, the FCSF was calculated as:

$$FCSF = \frac{IMEP + \Delta FMEP}{IMEP}$$

where IMEP is indicated mean effective pressure. This scaling factor was then applied to the fuel consumption rate (kg/hr) for each of the operating modes, which were then used to reconstruct a cycle-averaged fuel consumption rate based on the weighting factors. Further details on the fuel consumption, weighting factors, speeds, engine loads, and baseline friction coefficients are given in reference 10.

RESULTS

Figure 2 shows an example of the results of the calculations of the friction between the piston skirt and liner. The FMEP is shown as a function of SAE viscosity grade for different assumptions about the boundary film friction. The dashed curve shows the FMEP for viscous (oil shear) losses, whereas the dotted curves show the changes in the boundary friction as a function of oil viscosity. The solid curves show the total FMEP (sum of the viscous + the boundary friction). As seen for the baseline case, the total FMEP has a minimum of around 40 WT. As the viscosity is decreased, the total FMEP increases as the fraction of asperities that come in contact with one another increase, resulting in higher boundary friction. Another aspect to note is that, as the boundary friction coefficient is reduced, the minimum in the total FMEP shifts to lower lubricant viscosities. Thus, decreasing the boundary friction enables the use of low-viscosity lubricants.

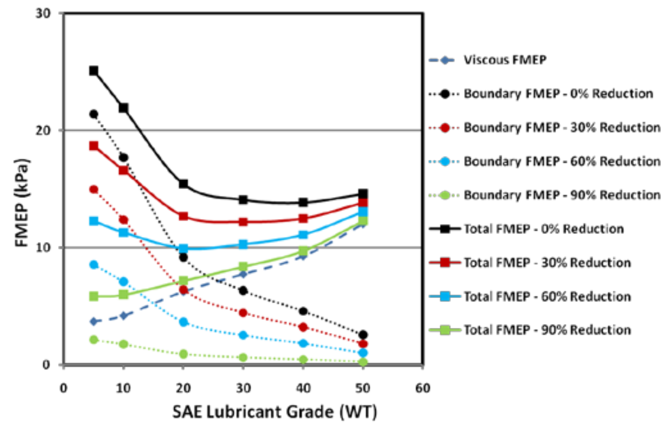

Figure 2: FMEP of Piston Skirt as a Function of Lubricant Viscosity Grade and Boundary Friction Reduction

The results in fig. 2 indicate that the baseline total FMEP between the piston skirt and liner is approximately 14 kPa, which represents approximately 20% of the total FMEP for all components, and 2.3% of the eight-mode FTP average IMEP of 617 kPa.

The relative contributions of the different engine components to the overall engine friction are illustrated in fig. 3 for the baseline case (40 WT oil with baseline boundary friction). As seen in the figure, the predominant source of engine friction is the piston skirt and rings – accounting for close to 70% of mechanical friction in this case. The relative distribution of engine friction shown in fig. 3 is in good agreement with other studies [11,12].

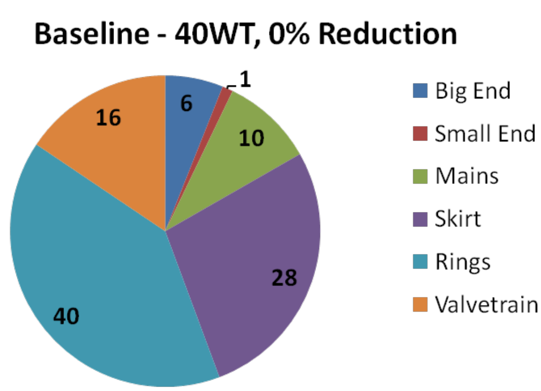

Figure 3: Distribution of Engine Friction by Components

The results of the friction models were combined to examine the impact of boundary friction and lubricant viscosity on fuel consumption/fuel savings. Figure 4 shows the fuel saved per 1000 hours of operation for the weighting factors used to simulate the HD-FTP cycle. All results are relative to the baseline study – 40 WT oil.

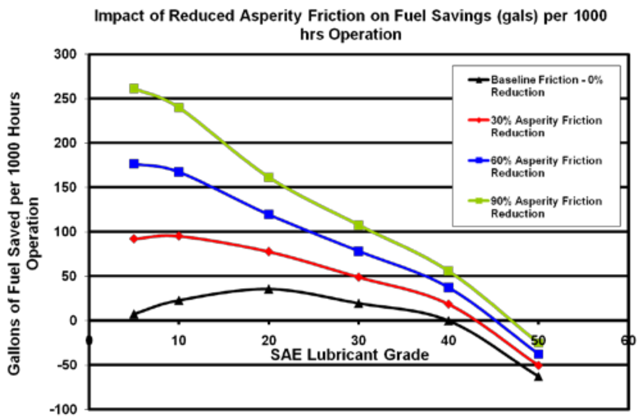

Figure 4: Fuel Saving (per 1000 hours of operation) as a Function of Lubricant Viscosity Grade and Boundary Friction Reduction

Figure 5 shows an alternative approach to illustrating the impact of friction and viscosity on fuel consumption. It shows the change in fuel consumption on a percentage basis (relative to the baseline case – 40 WT oil).

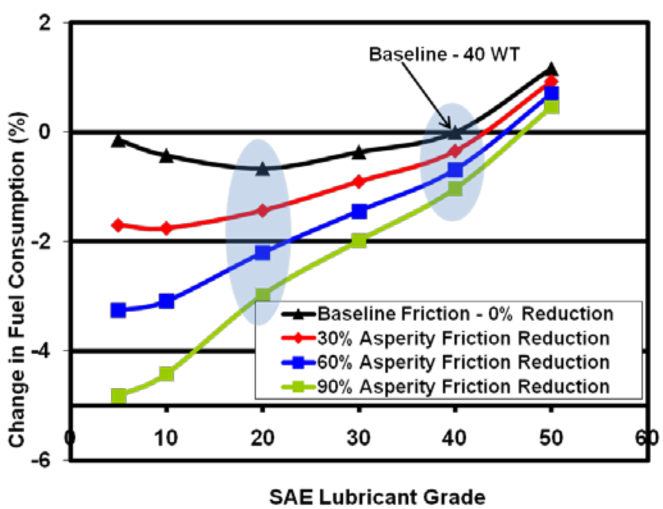

Figure 5: Change in Percent Fuel Consumption as Function of Viscosity and Friction Reduction

As seen for the baseline case, reducing the boundary friction by 30 to 90% alone will reduce fuel consumption up to approximately 1% for the eight-mode HD-FTP cycle. However, reducing both boundary friction and viscosity results in greater savings – up to 3% for a 20 WT oil. Another feature to note is that low-boundary friction enables the use of low-viscosity lubricants. For example, for the baseline case, the fuel consumption reduces slightly as the viscosity is lowered from 40 WT oil to 20 WT oil; however, with further reductions in viscosity (below 20 WT), the fuel consumption starts to increase. However, if the boundary friction can be reduced by 60%, the minimum in fuel consumption does not occur until a 5 WT lubricant is used.

DISCUSSION

The model calculations presented above indicate the impact of lubricant viscosity and boundary friction for a HD-FTP transient (on-road) driving cycle. Military ground vehicles have significantly different driving cycles, which can include lengthy periods at or near idle, where up to 60% of the fuel used during a mission is consumed [3]. In contrast, commercial trucks, while they may operate for considerable times at idle (up to 50% idle), rarely consume more than 5% of the fuel at idle. Furthermore, the non-idle portion of a military driving cycle will be significantly different from the non-idle portion of the on-road driving cycle. A significant portion of the driving cycle for a military vehicle will be spent on secondary roads, trails, and cross-country driving – conditions that favor higher engine loads at lower engine speeds. From a tribological viewpoint, both of these conditions (higher loads and lower speeds) can promote greater levels of mixed and boundary lubrication, where friction losses are higher.

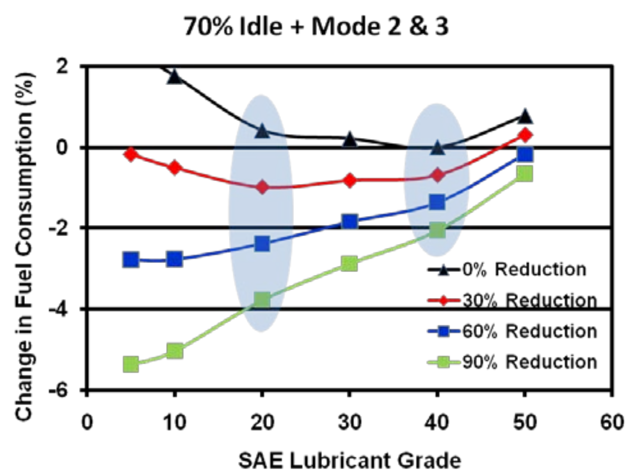

Figure 6: Impact of Lubricant Viscosity and Friction Reduction on Fuel Economy for a High-Idle, High-Load Medium-Speed Cycle

To examine the impact of friction on off-road driving cycles, the weighting factors for the eight-mode FTP simulation were modified. Instead of the weighting factors shown in fig. 1, the idle weight factor was increased to 70%, and the remaining eight-mode cycle was replaced with a simple condition of high-load, two medium speeds (e.g., 70% at mode 1, 15% at mode 2, and 15% at mode 3 – fig. 1). The results in fig. 6 indicate that friction has a greater impact on the fuel economy. For the 40 WT condition, reducing boundary friction up to 90% will reduce fuel consumption up to 2%, compared with 1% for the HD-FTP condition in fig. 5. At 20 WT, fuel savings are 4% (compared to 3% in fig. 5).

Engine idle, by definition, is a high friction condition. It represents the condition where all the indicated power is consumed by friction (and pumping losses) – the brake horsepower is zero. At idle (mode 1), approximately 50% of the indicated horsepower is consumed by engine friction. Figure 7 shows the FMEP as a fraction/percentage of the IMEP for the baseline case (40 WT oil).

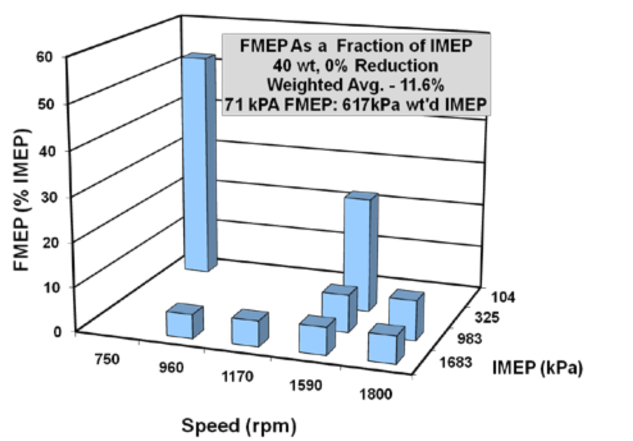

Figure 7: FMEP as a Function of Engine Mode (speed and load)

For the majority of the load/speed conditions shown in fig. 7, the FMEP is a small fraction of the IMEP – typically less than 5%. However, several conditions (idle and low-load, moderate-speed case) exhibit relatively high friction losses, where the FMEP is 25 to 50% of the IMEP. For the eight-mode HD-FTP cycle considered in these studies, the FMEP is approximately 71 kPa, which represents approximately 11.6% of the weighted IMEP produced during the FTP cycle. Studies have shown that reducing the boundary friction by 90% relative to current technologies, coupled with a change to a 20 WT oil, will reduce the FMEP for the same driving cycle by approximately 21 kPa or 8.1% of the IMEP. This equates to a fuel savings of 3.5%.

The impact on military vehicles has not been studied in detail – our studies focused on commercial vehicles. Nevertheless, these studies show a strong dependence on the driving cycle, in particular, the amount of idle. Figure 8 illustrates this impact in greater detail, where the change in fuel consumption is shown as a function of time spent idling.

At low-to-moderate idle times, a 90% reduction in boundary friction will reduce fuel consumption up to 4-5%, similar to the data shown in fig. 5. However, above 75% idle times, the impact of a low-friction (low boundary friction) technology is significantly greater, with improvements above 10 to 20% predicted.

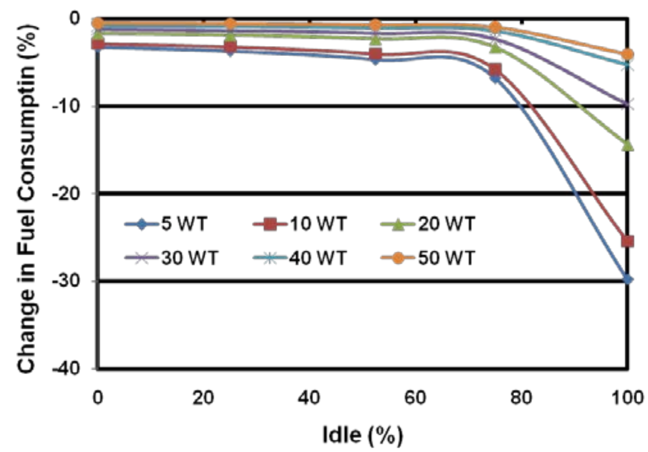

Figure 8: Impact of a 90% Reduction in Boundary friction on Fuel Consumption as a Function of Idle

The ability to predict/model the impact of viscous and boundary friction on fuel economy can be used to estimate the potential impact of advanced technologies on fuel consumption. For example, the US consumes approximately 13 Mbbl of petroleum per day for transportation split among passenger cars (4.7 Mbbl/d), light trucks (4.3 Mbbl/d), and heavy trucks (2.5 Mbbl/d), with the balance used by rail, marine, and air. The data shown above indicate friction consumes approximately 11-12% of the IMEP for heavy trucks. Similar studies [12] indicate approximately 10% of the fuel is lost to friction in passenger cars. These estimates vary strongly depending on the driving cycle, but a good rule-of-thumb is that friction consumes approximately 10% of the petroleum consumed – or roughly 1.3 Mbbl/day is lost to friction at our current level of use. At a price of $100/bbl, this implies that engine friction alone costs $50B/yr. Drivetrain friction is not included in this estimate, and studies have shown the drivetrain friction is about half that of engine friction, approximately 5%. Overall, friction thus consumes up to 15% of fuel consumed in transportation at a cost of $75B/year.

While the magnitude of fuel consumption by the US military is small in comparison to the total US consumption, it nevertheless is significant – 360,000 bbl/day [3]. Ground vehicles consume approximately 10% or 36,000 bbl/day. Assuming comparable engine friction losses (e.g., 10% rule of thumb), 3,600 bbl/day is lost to engine friction in military ground vehicles for an annual cost of $0.13B/year (at $100/bbl). The cost of engine friction is even higher for high-idle (low-load) conditions, where friction can consume up to 50% of the IMEP.

These costs, however, only include the cost of the petroleum, not the costs associated with operations and maintenance of delivery systems. When these are factored in, the cost of delivering fuel to its point-of-use ranges from $20-$25/gal for aircraft to $100-$600/gal for ground vehicles [3] – up to 100 times greater than the cost of commercial fuel in the US. Thus, the annual cost due to engine friction in military ground vehicles is significantly greater than the cost of the petroleum itself.

Additional costs that cannot be quantified include the cost in lives to deliver fuel to front-line operations and costs associated with not accomplishing military missions.

DURABILITY

The discussion presented above focused on modeling the impact of low-friction technologies on fuel consumption and fuel economy. This section will provide comments on the implications of low-friction technologies (in particular, the use of low-viscosity lubricants); the approaches being used to identify technologies that can achieve the levels of friction reductions assumed in the models (30, 60, 90%); and the impact of lubricant additives during off-normal, extreme tribological conditions.

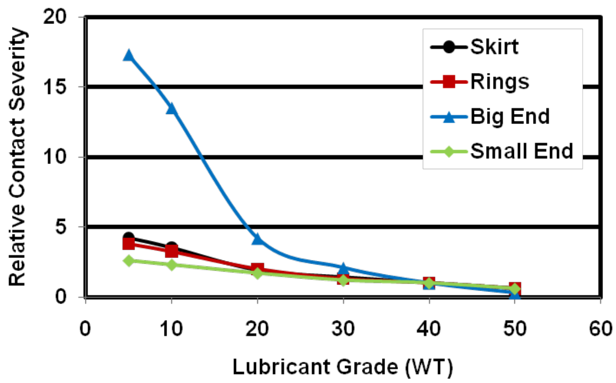

Figure 9: Contact Severity as a Function of Viscosity for FTP Cycle

The data presented above (figs. 4-6) indicate improving fuel consumption requires not only a low-friction boundary film, but also a low-viscosity lubricant. Low-friction boundary films by themselves reduce fuel consumption by 1-2% (see baseline case for 40 WT oil) depending on the driving cycle. Greater savings are achieved using a low-viscosity lubricant in conjunction with a low-friction boundary film – up to 4-5%. While a low-viscosity lubricant is beneficial in terms of reducing frictional losses, a drawback is that its use results in thinner oil film thicknesses and greater occurrence of contact between surface asperities on mating surfaces; hence, higher wear of components is possible, assuming the low-friction boundary films do not impart additional wear resistance.

Figure 9 illustrates the impact of viscosity on the relative contact severity. The contact severity is based on calculations of the contact loads for the different components (skirt, rings, large end bearing, and small end bearing). If one assumes the durability is inversely proportional to the load, then the results in fig. 9 can be used to estimate the relative improvement in the durability required for the components to survive relative to the baseline (40 WT) case. For example, operation with a 10 WT engine lubricant would require the use of a ring tribological system (combination of materials, coatings, lubricant additives, surface texture/finish, and/or geometry) that is 3.25 times more durable (wear resistant), while the large end bearings would need to have a tribological system that is 13.5 times more wear resistance than current systems. The results in fig. 9 indicate the critical component that would be impacted by the use of low-viscosity lubricants is the connecting-rod large end bearings. The rings, skirt, and liner will also be impacted, but the degree of improvement in wear resistance required to function is not as severe.

TECHNOLOGY ASSESSMENT

The above modeling efforts examined the impact of low-friction technologies on fuel economy, but did not specifically identify technologies that can provide the assumed friction reductions. Numerous approaches to reduce boundary friction have been and are under development. These approaches include the use of (i) chemical compounds that are added to lubricants to promote the formation of low-friction compounds on surfaces, (ii) low-friction materials and/or coatings used to fabricate engine components, and (iii) advanced surface finishes that promote operation in low-friction hydrodynamic regimes.

The development and validation of advanced tribological systems involve numerous stages that range from fundamental discovery of new materials and compounds to detailed durability and performance tests (engine and fleet studies). The cost associated with the engine and fleet studies can be quite high, and thus lab tests are often used to screen candidate technologies before performing engine and fleet tests.

The results presented in this section describe one of the techniques used to screen candidate materials and additives that show potential to reduce boundary layer friction. The reciprocating test used is illustrated in fig. 10 and has been used to measure the friction between rings and liners, and piston skirts and liners. The technique uses segments of rings, skirts, and liners obtained from commercial components. The segments are reciprocated back-and-forth over the liner at loads, speeds, and temperatures prototypical of engine combustion chambers. A load cell attached to the ring or skirt segment monitors the friction forces continuously. The electrical contact resistance between the ring/skirt segments and the liner is monitored and provides information on the formation of tribochemical films on the surface.

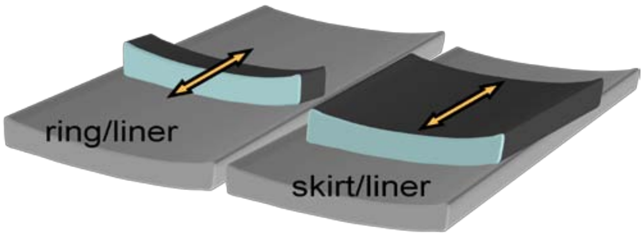

Figure 10: Schematic Illustration of a High-Frequency Reciprocating Test Rig

Figure 11 [13,14] shows an example of the data collected during a series of tests designed to evaluate the impact of two lubricant additives on the friction coefficient. The friction coefficient of a fully formulated 15W/40 oil is shown in black, along with the friction behavior of the same oil treated with two different additives (an emulsion based boric-acid additive – in red, and a liquid molybdate ester – in blue). The tests were run under fully submerged conditions, at 100°C, with a stroke of 2 cm, a load of 250 N, and a reciprocating speed of 120 rpm – conditions that resulted in boundary lubrication over the entire stroke.

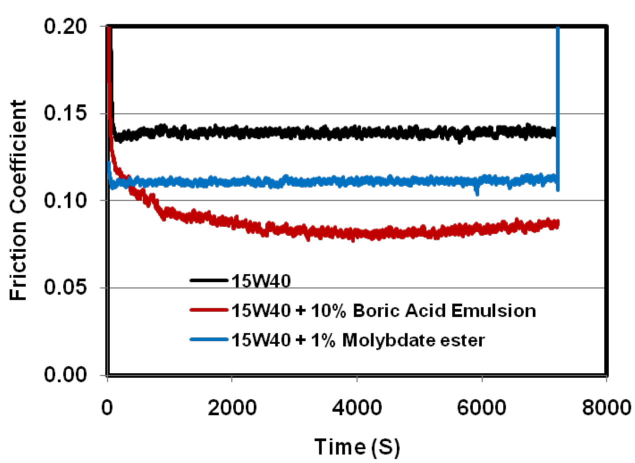

Figure 11: Friction Coefficient as a Function of Time for 15W/40 Diesel Engine Oil

The friction coefficient for the formulated 15W/40 (black curve) was approximately 0.14, compared to the baseline coefficient of 0.08 assumed in the PISDYN simulations. This discrepancy is not of great concern since this specific engine lubricant is intended for use as a transmission fluid, and thus a higher friction coefficient is not unexpected. Tests on commercial non-mil-spec lubricants, that are blended specifically for engine use only, show low friction coefficients (in the 0.1 to 0.12 range). As seen in fig. 11, the molybdate ester additive exhibited friction around 0.11, while the emulsion based boric-acid exhibited friction near 0.08 – a 40% reduction compared to the baseline 15W/40 lubricant.

Research continues on the use of this approach to evaluate other additive technologies, coatings, and surface textures. Studies [14] on uncoated and coated piston skirts show that graphite-resin coated pistons actually exhibit slightly higher friction than uncoated pistons, while hydrogenated *a*-carbon coated pistons exhibited lower friction (30% decrease at 120°C in a commercial 10W/40 lubricant).

RELIABILITY

Fuel economy and durability are important to commercial and military applications. Another property of critical importance that can have significant impact on the ability to accomplish military missions is reliability – especially under harsh, off-normal conditions. In contrast to durability, which involves gradual degradation of components over time and is predictable, reliability issues typically involve sudden, unpredictable degradation of a system. Scuffing is a prime example. Scuffing [15, 16] is characterized as a sudden catastrophic failure of the lubricated sliding surface characterized by a sudden rise in friction, contact temperature, vibration, and noise, resulting in surface roughening through severe plastic flow and loss of surface integrity. Once scuffing occurs, the friction remains high even though the operating conditions are returned to pre-scuffing values. Scuffing is a common failure mechanism that is addressed in the design of vehicle systems, in particular, combustion chamber components [17], as well as gears and valve-train components.

Military ground vehicles must often operate under extreme tribological environments, including environments where the vehicle lubrication can be compromised or disabled to the point where the engine and/or drivetrain operate under starved lubrication. In such cases, scuffing can occur and lead to sudden failure/seizure of critical components and thus impact the ability to complete a mission safely.

A potential solution to mitigate the impact of sudden catastrophic failure on mission critical systems is the application of advanced additives to increase scuffing resistance. Results are presented below on a series of bench-top experiments designed to investigate the impact of advanced additives on the critical scuffing load of a qualified mil-spec 15W/40 diesel engine lubricant.

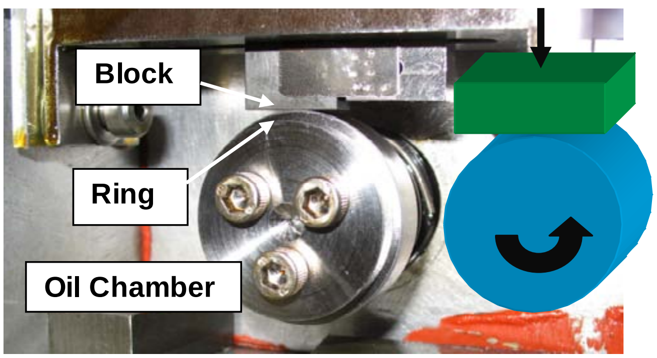

Figure 12 illustrates the block-on-ring [16, 18] technique that was used to evaluate the scuffing load performance. The technique involves pressing a flat block against a rotating ring, producing a highly stressed line contact. The load is increased in 25 N increments every 60 sec until scuffing occurs (as denoted by a sudden increase in friction). The technique is used to compare the effectiveness of additives relative to a baseline condition.

During operation, the rotating speed was held constant (at speeds of 750, 1000, and 1500 rpm), while the applied load was increased from 0 up to 2000 N. The block-and-ring specimens were contained in an enclosure that was filled with the lubricant (and additives) to a level approximately 1/3 of the way up from the bottom of the ring. The oil wetted the ring and was transported to the contact region between the block and ring.

This technique was used to quantify the scuffing load of a engine/driveline lubricant based on a mil-spec mineral. Tests were performed on the as-received oil and oil mixed with five additives (emulsion based boric-acid, tricresyl phosphate, graphite, boron nitride, and molybdenum disulphide). The oil-to-additive ratios were 5:1, 10:1, and 25:1 by weight.

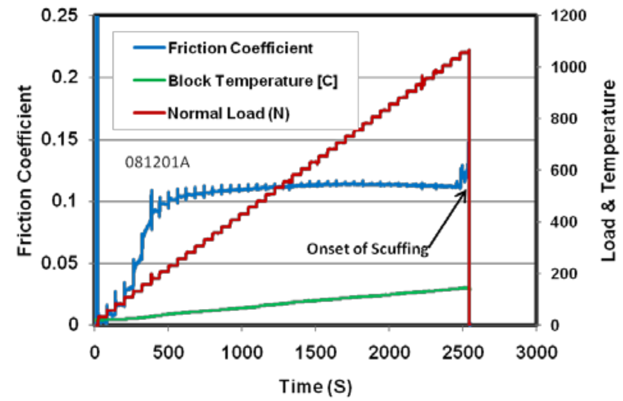

Figure 13: Friction, Applied Load, and Block Temperature as a Function of Time during a Block-on-Ring Test at 750 rpm

Figure 13 shows an example of the friction, applied load, and block temperature as a function of time during a test on the as-received formulated 15W/40 lubricant. Initially, the friction is low, but then increases to steady-state values around 0.12, until scuffing occurs at a critical load around 1050 N. At that point, the friction increases rapidly, and the test is stopped.

Figure 14 summarizes the results for the series of tests designed to evaluate the impact of the additives on the scuffing load of the formulated 15W/40 mil-spec diesel engine lubricant. The data shown in fig. 14 represent the

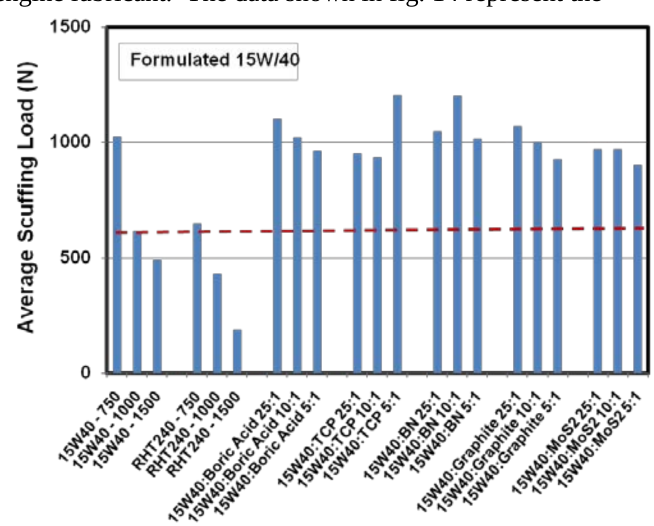

Figure 14: Average Scuffing Load of Formulated 15W/40 and 15W/40 with Additives

average scuffing load obtained from a minimum of three repeat tests at each run. The first three tests show the impact of speed on scuffing, while the next three show the impact of speed on the scuffing load of the unformulated base fluid used in the as-received formulated oil. The remaining tests show the average scuffing load for the five additives at the three oil-to-additive levels. The dashed red line in fig. 14 represents the average scuffing load for the formulated 15W40 oil at a speed of 1000 rpm – the speed used for the tests with the different additives. As seen in fig. 14, all of the additives increased the scuffing load of the as-received formulated lubricant. The magnitude of the improvement ranged from 50% to 95% depending on the additive and treat level.

Figure 15 shows comparable information on the impact of the five additives used in the formulation of the 15W/40 formulated lubricant, e.g., the base fluid without the additive package added at the factory. The dashed green line in fig.15 represents the average scuffing load of the base fluid at 1000 rpm – the speed used for the tests on the base fluid with the five additives. In contrast to the formulated 15W/40 oil, the graphite and MoS_2 additives were ineffective in improving the scuffing load.

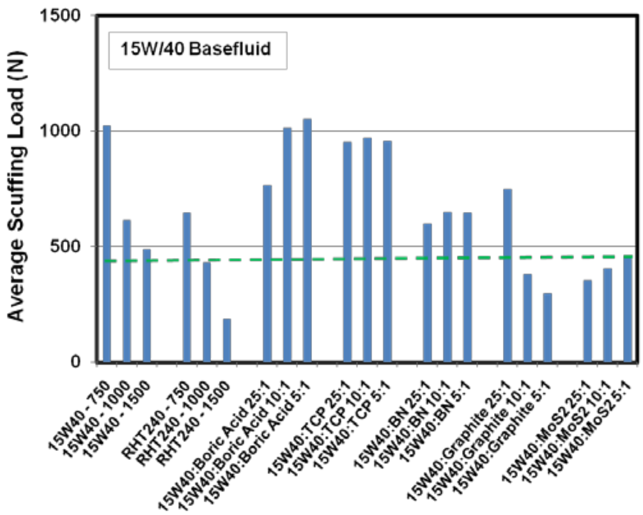

Figure 12: Average Scuffing Load of 15W/40 Base Fluid and 15W/40 Base Fluid with Additives

Research is continuing to better understand the behavior seen in figs. 14 and 15, as well as to investigate the behavior under starved lubrication conditions, e.g., during "oil-off" tests. The oil-off tests provide a method to quantify the impact of additives on extending the time-to-fail/scuff after the oil is drained from the lubrication cup – simulating a starved lubrication condition.

CONCLUSION

A suite of four mechanistic friction models has been integrated and used to examine the impact of boundary film friction and lubricant viscosity on fuel consumption and economy of heavy-duty diesel engines. Analysis of the results indicates the following trends:

- Parasitic friction mechanisms (oil shearing and metal-to-metal asperity friction) consume approximately 10% of fuel used in transportation. Another 5% is consumed by drivetrain friction.
- The losses can be significantly greater for vehicle operating cycles that involve long periods of idle, where power is required for hotel power.
- Application of low-friction boundary-film technologies will lower fuel consumption by 1% for an on-highway commercial truck. Greater fuel savings (up to 2%) can be realized for high-idle driving cycles that involve off-road conditions.
- The application of low-friction technologies that lower friction in the boundary-lubrication regime (Stribeck curve) enables the use of low-viscosity fluids resulting in potential fuel savings up to 3-4% for commercial driving cycles – provided suitable low-friction technologies are available.
- While low-viscosity lubricants are beneficial in reducing parasitic friction losses, caution must be exercised to offset the increased contact severity and potential durability/reliability issues associated with increased contact loads that occur with low-viscosity fluids.
- Potential solutions to improve fuel economy, such as lubricant additives and low-friction materials/coatings, have been identified in lab studies, and need further effort to implement them industrially.
- The use of advanced additives in formulated mil-spec lubricants has been observed to increase the scuffing resistance in lab-based tests and may represent a potential solution to enhancing the survivability of ground vehicles under extreme tribological environments.

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