The Real Costs of Lubrication

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Presented At The 99th AGM-CIM
Technical Program - M/E Division
April - 1997

Abstract

The paper looks at the basic theory of tribological wear and the role played by the lubricant and its additives. It presents the argument that the "real cost of lubrication" is not the purchase price of the oil or grease, but rather the costs associated with with not using the correct lubricant. A number of examples are presented, covering oils, greases and additives, in which a detailed evaluation of the application has shown that when all of the benefits of using the best lubricant/additive are evaluate, the savings achievable greatly surpass the incremental cost of the superior lubricant. One of the emerging classifications of costs in industry today are those associated with environmental protection. Two of the examples used describe the replacement of more conventional lubricants with environmentally acceptable products, compounded to compete in all aspects of serviceability, but which have the added bonus of being not only biodegradable but also of being of low toxicity.

Introduction

In order to establish the real cost of lubrication, many different factors need to be taken into consideration. The real cost of lubrication of industrial equipment needs to include the initial lubricant requirement, the relubrication amounts and intervals and the cost of maintenance and repairs including downtime in addition to the cost of the lubricant itself. Using a cheaper, but inadequate or even incorrect lubricant in an application can result in much higher costs than had the correct, though more expensive lubricant been installed in the first place.

It was estimated by the British "Jost Report" as early as 1966, that improvements in lubrication and maintenance in industry could save the British economy 500 million pounds sterling per year. A similar report by the Canadian NRC-ACOT in 1986 estimated that, of the five billion dollars lost annually due to friction, 25% could be saved through better application of existing technology and through research and development. Though the problems and costs of friction and wear are industry wide, of the most interest to this forum is the fact that, after agriculture and road transportation, the industry next most affected by the cost of lubrication is mining.

Tribology Theory Costs Real Money

The first recorded use of the term "tribology" was in the "Jost Report" to Her Majesty's Government in Britain in 1966. It was used by the authors, a group of noted academics, who recognized the essential interactions of the three engineering features of friction, lubrication and wear. They coined the term to encompass the interdisciplinary science and technology that are concerned with the behavior of interacting surfaces in relative motion. A simple definition by Nam P. Suh is that "tribology is concerned with the science and the technology of two or more surfaces in relative motion". In simpler words, tribology is concerned with the friction and wear of surfaces of components.

In the Jost Report the authors investigated the ways in which tribological losses occur. Their conclusions have been reiterated by studies more recently conducted in Germany, the USA and in Canada. The Canadian NRC-ACOT report "A Strategy for Tribology in Canada", encompasses a very wide range of industrial products and equipment including ploughs, space equipment, engines, railways, computers and mines. The breakdown by the various main industries are given in Table 1 below.

Table 1 - NRC-ACOT Report of Frictional and Wear Losses

	Friction Losses ($ Million/Year)	Wear Losses ($ Million/Year)	Total Losses ($ Million/Year)
Agriculture:	321	940	1,261
Electric Utilities:	54	189	243
Forestry:	111	158	269
Mining:	**211**	**728**	**940**
Pulp and Paper:	105	382	487
Rail Transportation:	284	467	750
Trucks & Buses:	126	860	986
Wood Industries:	14	189	203
TOTALS:	1,226	3,913	5,139

An American study concluded that the cost of friction and wear in the USA is almost $200 billion each year. This figure was broken down into the following primary categories, frictional losses, wear, lubricant costs and electrical contact wear. The reasons given for the losses of material components are further broken down into the three categories of obsolescence, breakage and surface deterioration. Obsolescence and breakage each account for 15% of the material losses by industry while surface deterioration by wear and corrosion consumes 70% of the lost industrial equipment. A new and very real cost of lubrication that was not taken into consideration in these studies is the cost to the environment. The expense of reclaiming and repairing damage to the environment caused by poor design, maintenance and lubrication practices is becoming more significant. Stricter regulations governing the release of lubricants into the environment and their subsequent clean up are driving costs higher. The release of approximately four hundred litres of oil into a river system in western Canada cost over $80,000 to repair.

Basic Tribology Theory

Around 1500 Leonardo da Vinci deduced the laws of friction for a rectangular block sliding over a flat surface. His works were unpublished and nearly 200 years later Guillaume Amontons, a French physicist and inventor of the hygrometer, was credited with discovering the laws of friction. His works led to the two basic laws of friction. The first law states that the friction force resisting movement is proportional to the normal load on the surfaces. The second law, that the frictional force is independent of the area of contact, has been appended to the apparent macroscopic area. Frictional forces are in fact proportional to the true area of contact which is the sum of all the contact points of the minute hills and valleys on any machine surface.

These hills and valleys are called asperities and their deviation from the mean surface elevation is the most common measure of the roughness of tribological surfaces. Figure #1 shows graphically how the root mean square roughness represented by the Greek letter sigma, σ is determined.

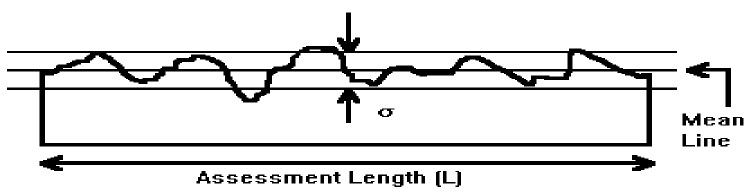


Figure #1 - Root Mean Square Surface Roughness

Since contact between the asperities of two surfaces in relative motion determines the friction and ultimately the wear of those surfaces, tribologists and lubrication engineers attempt to increase the distance between wearing surface to minimize the contact, friction and wear. This is done through the use of fluid lubricants whose purpose is to increase the pressure between the surfaces and force them apart. Obviously, if the surfaces can be kept completely apart at all times, wear from abrasion and adhesion may be nearly eliminated. Friction will, however, remain since it is now governed by fluid mechanics and the internal fluid resistance makes up the bulk of the frictional force. Once the fluid pressure drops in relation to the normal loads on the surfaces, the asperities begin to collide and the most severe wear occurs.

If we determine the thickness of the lubricant film between two surfaces, and we know the surface roughness, we can determine whether asperity contact, increased friction and severe wear is possible. The ratio of the film thickness to the surface roughness is defined as the specific film thickness, λ (lambda).

$$\text{Specific Film Thickness} = \lambda = \frac{\text{Mean Film Thickness}}{\text{Surface Roughness}} \qquad (1)$$

Many different factors affect the specific film thickness, lambda, including the equipment material, design and geometry, the lubricant viscosity, chemistry and additives and the operating conditions of load, speed, temperature, vibration and shock.

As long as the λ ratio is greater than unity, the film thickness exceeds the mean surface roughness and severe wear is minimized. There will still be some wear since the equation uses the mean values and not the absolute maximums. The highest asperities may still collide at λ greater than one. Therefore the value of λ determines the lubrication regime. When λ is less than one, the film thickness is smaller than the asperity height and the most severe wear regime exists. The regime where λ is less than one is the boundary lubrication regime and is unfortunately the most common. When λ becomes greater than four, even the highest asperities will no longer contact and ideal full film or hydrodynamic lubrication exists. In between these two extremes, when λ is greater than one and less than four, is the mixed lubrication regime.

Under the boundary and mixed lubrication regimes, where the majority of friction and wear is generated, many different physical phenomena are manifested. The surfaces become modified by adhesive and abrasive wear and by chemical activity. There can be losses of material from one or both surfaces or even the transfer of material from one surface to the other. Wear debris is created which now contaminates the lubricant accelerating abrasive wear in other areas. All of the above result in the frictional energy which must be conserved. It is therefore converted into other forms including heat and sound energy. The component and the lubricant temperatures are increased by heat from friction. The friction created by contact of wearing surfaces also gives off noise energy, mostly in the region around 40 kHz. This ultrasonic energy has been found to provide excellent early warning of wear in equipment, even before measurable changes in temperature are detected and long before audible noise alerts us to the impending failure. The surface contact also produces significant local forces that contribute to surface fatigue and damaging vibrations in rotating equipment. All of these forces have a detrimental effect on the lubricant and its ability to do its job.

Stribeck plotted the frictional force against the specific film thickness, λ (see Figure #2). The curve demonstrates the different lubrication regimes and the relative frictional forces acting within these regimes. The highest friction exists where λ is less than one in the boundary regime. As λ increases, the friction drops throughout the mixed regime until it reaches 4 where elastohydrodynamic lubrication reigns. This is a portion of hydrodynamic lubrication where some elastic surface deformation is possible. EHD is somewhat geometry dependent as it requires very high specific loads. As λ increases the minimum friction is achieved at the point where the fluid film is just thicker than the highest asperities. As the fluid film increases beyond the optimal value, the internal resistance of the fluid (the viscosity) causes increased friction.

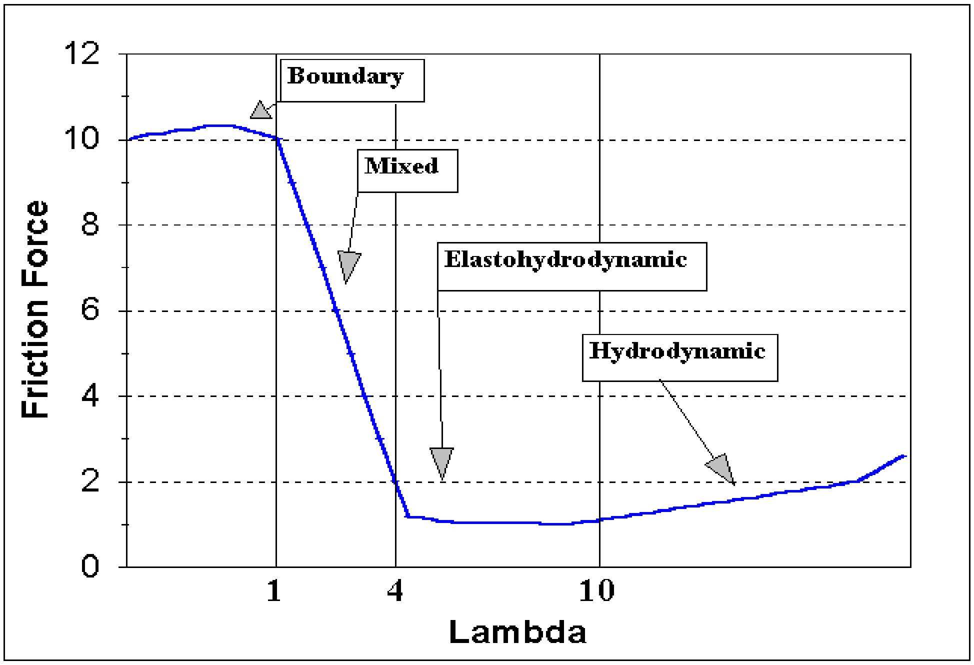


Figure #2 - Stribeck Curve

To look at the above curve from a practical application viewpoint, the region below λ=1 corresponds to the start up of equipment. No lubricant film is present between the surfaces and the most severe wear occurs at cold start. As lubricant flow increases and the temperature rises after start up, the lubricant film thickness increases slowing the frequency of asperity contact. When the component is at normal temperature and operating with little load (at idle) the film thickness is at its optimum and very little wear occurs. Then as the speed and load on the component increase, the friction and wear increases again.

From a lubricant standpoint the same curve provides an explanation of the fluid mechanics. Fluid lubricants separate wear surfaces by being drawn into the constricting "wedge" between the moving surfaces. As the restriction increases so too does the pressure on the fluid. This fluid pressure, exerted in all directions, forces the surfaces apart. When λ is greater than four and hydrodynamic lubrication reigns, low loads and , adequate viscosity maintain a full fluid film and minimize wear and friction. Beyond the optimum, increases in either fluid viscosity or surface speed results in greater frictional drag forces. In the mixed regime the viscosity keeps the surfaces apart except when vibration and shock loads exceeds the fluid pressure and momentary contact occurs. When λ is less than one, the fluid pressure is insufficient to counter the normal pressure or load on the component and boundary contact occurs. Here the viscosity is no longer the dominant parameter and the activity of the lubricant additives becomes the only protection from severe wear.

To summarize, the factors that affect friction and wear of equipment include the surface condition, the operating and maintenance practices and the lubricant properties. Through better design of equipment we can shift the Stribeck curve to the right and operate largely in the hydrodynamic range. Proper installation and maintenance will ensure that contaminants, corrosion, misalignment or improper lubrication do not contribute to increased wear. Finally improving the quality of the lubricants in the system can significantly improve the efficiency and life of nearly all equipment. While any one of the above can cause premature equipment failures, usually combinations of two or more are responsible. The remaining sections of this paper will describe several applications where improving lubricant quality can reduce friction, wear and most importantly the cost of lubrication.

Gen-Sets Operating on Low Sulfur Diesel Fuel

The first lubricant application discussed here is one that was particularly localized in early 1993. The state of Alaska was the first in the USA to mandate the use of low sulfur diesel fuel in order to improve the quality of engine exhaust emissions. The new diesel fuel was required to have a sulfur content lower than 0.05%. This was significantly lower than the previous maximum of 0.5%. Since that time, the use of low sulfur diesel fuel has become mandated across North America and the significant number of wear problems related to the new fuels has spread.

Before presenting the background of this specific application, it should be stated that not all low-sulfur diesel fuels have poor or inadequate lubricity. Some low sulfur fuels have excellent lubricity while some high sulfur fuels may have poor lubricity. Most of the data and information from a variety of sources does suggest that the general statement can be made that low sulfur fuels have poorer lubricity than their higher sulfur predecessors. The poorer lubricity stems partly from the loss of sulfur and aromatic compounds in the fuel hydrotreating process.

Shortly after the use of low sulfur diesel fuel was required in Alaska, there was a significant increase in the amount of diesel fuel system problems and repairs. The operators of equipment also complained of poorer fuel economy, harder starting, low power and other related problems.

Laboratory Formulation and Testing - On investigating a few injection pump failures it was obvious that the pumps were simply wearing out prematurely. Samples of the fuel were obtained and they were tested for their lubricity using the ASTM D5001 Ball on Cylinder Lubricity Evaluator (BOCLE) test. In this method, designed to evaluate the lubricity of kerosine jet fuels, a 0.5 inch steel ball bearing is worn, under light load, against a rotating steel cylinder. It was determined in these tests that the new low sulfur diesel fuel consistently produced larger wear scars than the regular sulfur fuel.

The research project to formulate an additive for the fuel that would restore the previous wear results was initiated. The restrictions were that the product needed to be sulfur free, soluble in diesel fuel, and not affect the other characteristics of the fuel. A prototype product was developed that improved the wear results in the lab and had no adverse effect on the physical properties of the fuel. BOCLE test data showed that the wear scar was reduced from an average 0.615 mm to 0.55 mm in diesel fuel containing 0.021% sulfur. Correlative data from fuel pump wear tests indicates that an acceptable wear scar should be smaller than 0.6 mm to prevent fuel pump damage. The addition of the additive made a potentially unacceptable fuel acceptable in terms of lubricity measured by this method.

The new High Frequency Reciprocating Rig (HFRR) Test data also proved that the treated fuel had significantly better lubricity than the neat samples. The HFRR test is one that uses a steel ball wearing against a flat plate in reciprocating motion. It is most likely to be accepted by the ISO and the ASTM as the new standard method for evaluating diesel fuel lubricity. The addition of the additive to the low sulfur fuel improved the wear scar from 0.37 mm to only 0.21 mm. The photographs in Figure #3 and #4 show the improvement in scar size in each of these two lubricity tests.

Figure #3 - BOCLE Wear Scar Results

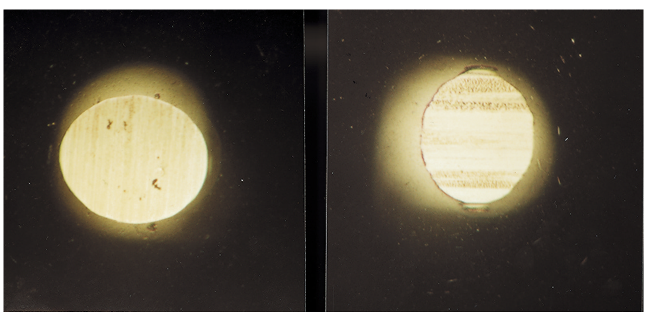
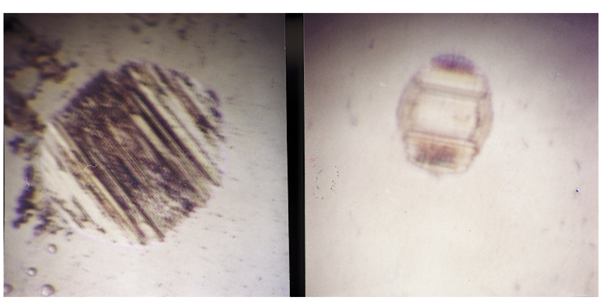


Figure #4 - HFRR Wear Scar Results



Field Test - The initial field test was arranged and was carried out on a stationary Caterpillar 3408, four cycle, V8 diesel engine that powers an electrical generator set. Phased vibration readings were taken at each plunger in the fuel pump and at the top of each power cylinder. The first three sets of data (averaged into one reading) were taken in approximately one hour intervals. The prototype diesel fuel lubricant was added at a 1:500

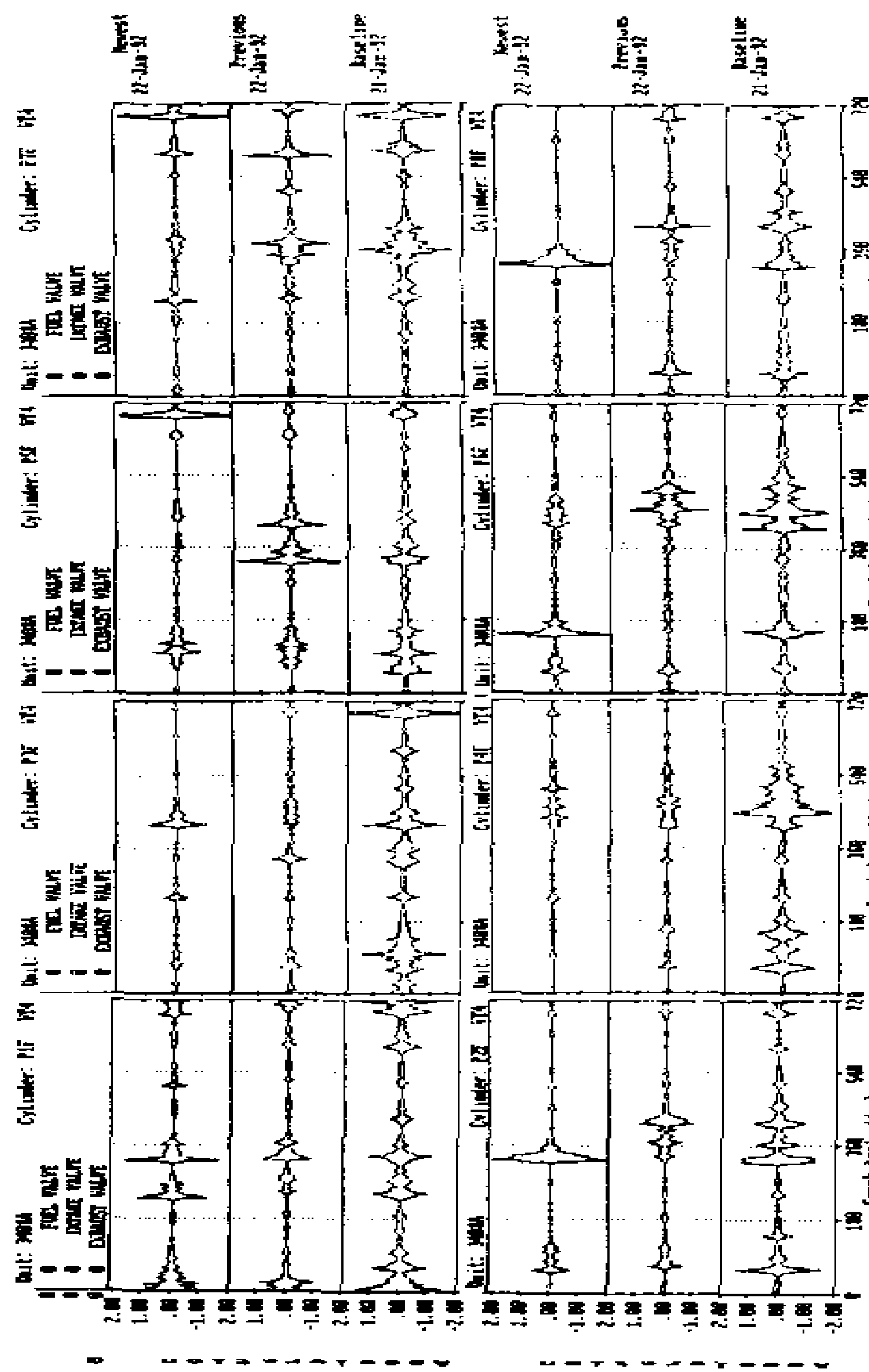
ratio and then three more sets of data were gathered. After twelve hours had elapsed, a third set of readings was taken, averaged and recorded.

The Caterpillar 3408's fuel pump is located on top of the engine, away from the heat of combustion. A fuel transfer pump pulls fuel from the fuel tank, through a fuel filter and then into the inlet of the injection pump. A camshaft controls the timing of the individual plungers. The fuel pump piston plungers are capable of generating between 4500 and 9000 psi fluid pressure as the diesel fuel is charged to each power cylinder's fuel injector. This high pressure causes significant loading on the plunger and creates a boundary lubrication condition where metal to metal contact may occur, prematurely wearing the pump plungers. Only the diesel fuel itself provides the lubrication in this wear interface.

The vibration readings were taken by an independent vibration analysis company from whose report the following comments are quoted. Most importantly, "after (the lubricity additive) was introduced to the 3408's fuel supply, there occurred a gradual reduction in the crosstalk recorded. Crosstalk is the vibration noted on the vibration signatures that emanated from another source other than the one being monitored". In other words, the overall amount of vibration from the fuel pump decreased once the lubricity additive was introduced to the fuel supply. These reductions in overall vibration at the fuel pump can be seen in the eight charts of Figure #5. The reduction in vibration is evident as the test progresses from the "baseline" data (without the additive) on the bottom, to the "previous" data (shortly after additive introduction) and to the "newest" data taken 12 hours later. In fact, the vibration consultant's report concluded that "The reduced crosstalk may indicate that the fuel pump plungers are functioning smoother, that is, experiencing less metal to metal friction".

(See Page 10, Figure #5)

Figure #5 - Phased Vibration Signatures at the Fuel Pump



Phased vibrations at the head of each power cylinder were also monitored. The individual cylinder vibrations taken during the test are shown in Figure #6. Typical combustion cycle event vibration signatures are given in Figure #7. Overall, there is little difference in the vibration patterns from the Baseline to Previous to Newest results. There are two notable exceptions. Though not dramatic, there are increases in the vibration amplitudes at each cylinder near the 650 degree crank angle. This corresponds approximately to the point at which fuel is injected into the cylinder. The increase in vibration at this point in the combustion cycle indicates that there is better fuel injection. This suggests that the fuel injection pump is generating higher fuel pressures due to fewer frictional losses in the pump. The vibration analyst's report states that "As the level of crosstalk decreased, the vibration due to fuel being ejected from the fuel pump to the injectors increased. This phenomenon may indicate that the fuel pump was generating more head as a result of less energy being dissipated to overcome frictional forces namely those experienced by the fuel pump plungers".

Figure #6 - Phased Vibration Signatures at the Cylinder

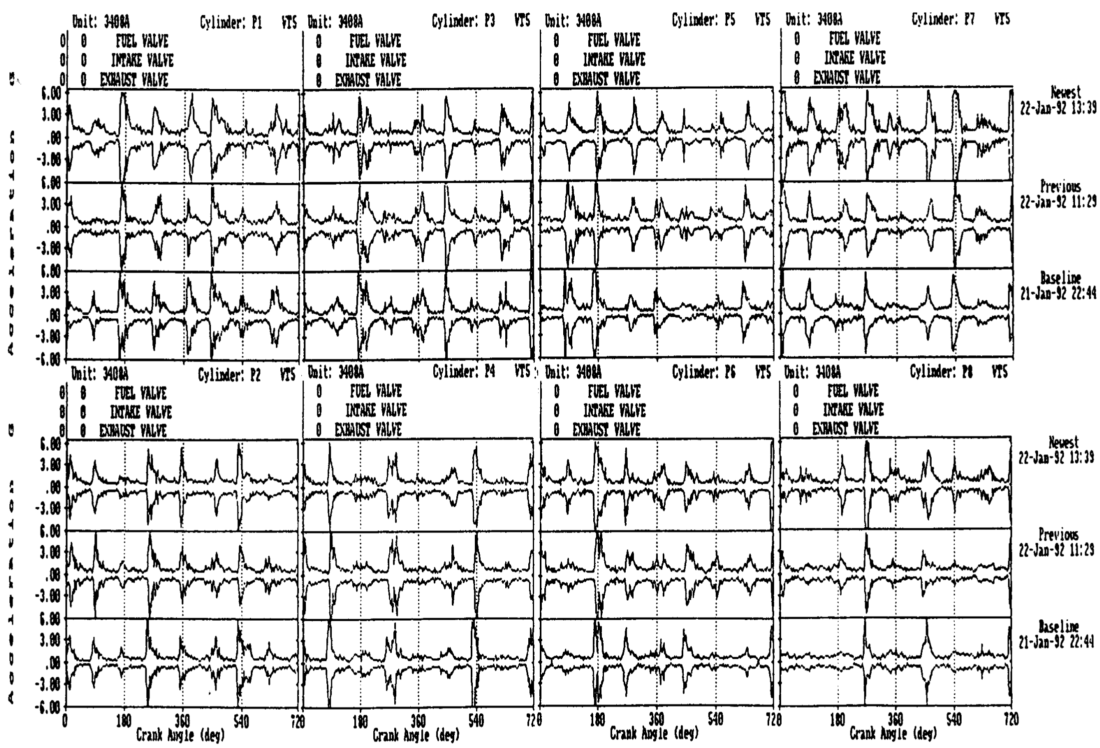
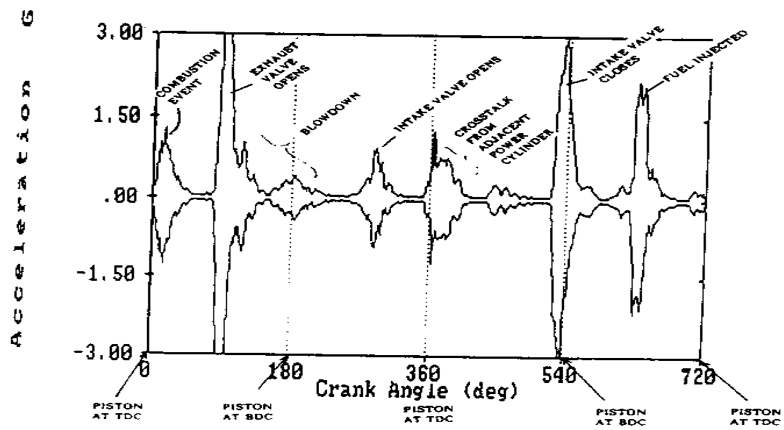


Figure #7 - Combustion Event Vibration Signatures



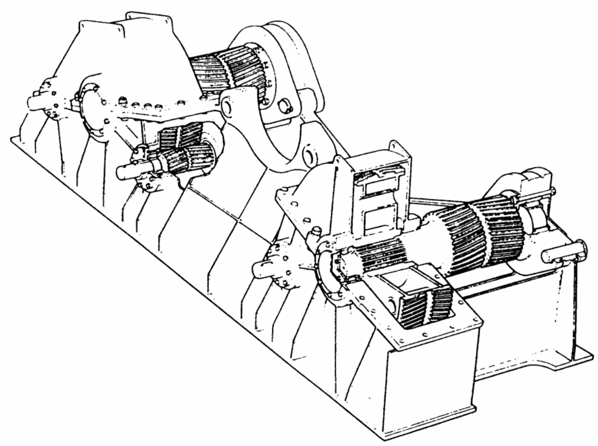
The vibration patterns from the cylinder heads also shows that there is an overall increase in the amount of vibration at the TDC position of 720 degrees. This is the position where the combustion event itself takes place. The increase in combustion vibration correlates with the improved fuel injection patterns. The better injection and swirling of the fuel should provide a better burn resulting in improved power and efficiency. The overall efficiency of the engine is expected to improve and not just the life of the injection pump.

The laboratory fuel lubricity tests and the field vibration test results both provide very positive evidence that the use of a fuel lubricity additive can improve the performance, efficiency and service life of diesel engines using new generation, low-sulfur fuel.

Dragline and Conveyor Gearbox Applications

Mining equipment is being asked to work harder and longer than ever before in order to improve process efficiency. Capital restraints require that equipment designed for certain operating and loading conditions be used under more severe conditions and higher loads. Often, the results are higher than normal operating temperatures and increased wear, shortening equipment life. A large oil sands project was experiencing high temperatures and increased power consumption due to increased loads on conveyor systems and on the draglines used in mining.

Figure #8 - Drag Drum Gearbox Configuration



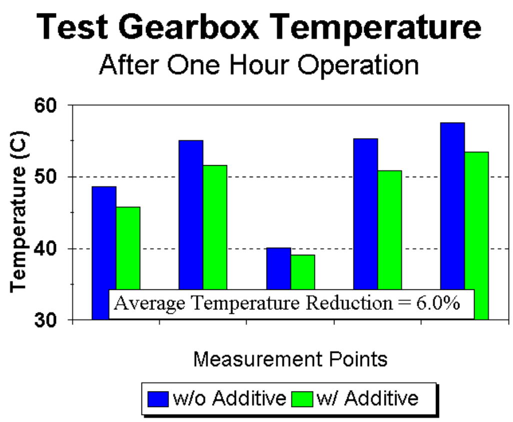
Drag Drum Gearbox Test - The drag drum gearboxes on a Marion Dragline 3 (Figure #8 above) were tested to evaluate the friction reducing properties of a conventional Extreme Pressure (EP) gear oil and the same oil treated with an additive formulated to improve the boundary lubrication (or EP) characteristics of the oil. Gearbox temperature and ultrasonic wear noise readings were taken at several different points on the dragline gearbox.

Temperature is obviously a measure of the relative friction in the gearbox, while the ultrasonic noise measurement is also a means of monitoring the frictional wear within a mechanical system. A brief introduction to ultrasonics and the relevance thereof is provided in Appendix A.

The drag drum gearbox test was carefully controlled to ensure that the measured effects on temperature and ultrasonic noise emissions are from friction and not external factors. Identical 95 ton loads were applied to the drag bucket and the operator endeavoured to extend and retrieve the bucket in the most consistent manner possible. The same number of bucket extensions and retrievals were performed over the same one hour period in both the before and after tests. As a secondary control, since the ambient temperature was expected to change over the course of the test, only one of the two drag drum gearboxes was treated while the other was left as a control.

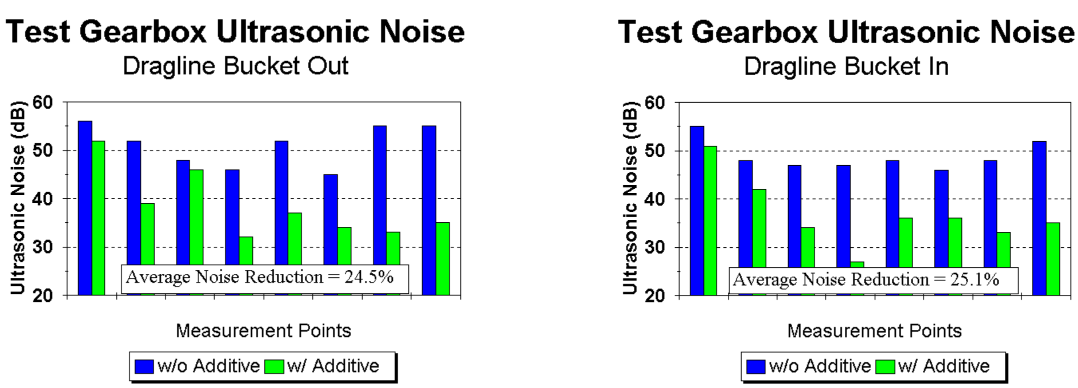
After the fixed number of extensions and retrievals over a period of one hour, the first set of temperature and ultrasonic noise readings were taken on the two drag drum gearboxes. The oil in the drag drum gearbox 1B was then treated with 5% of the boundary lubrication additive and the dragline was loaded under the same set of conditions as with the neat oil. The same set of readings was taken at the same points with the boundary lubricant additive applied. The temperatures at different points on the gearbox treated with the lubricant additive are given in Figure #9 below.

Figure #9 - Drag Drum Test Gearbox Temperatures



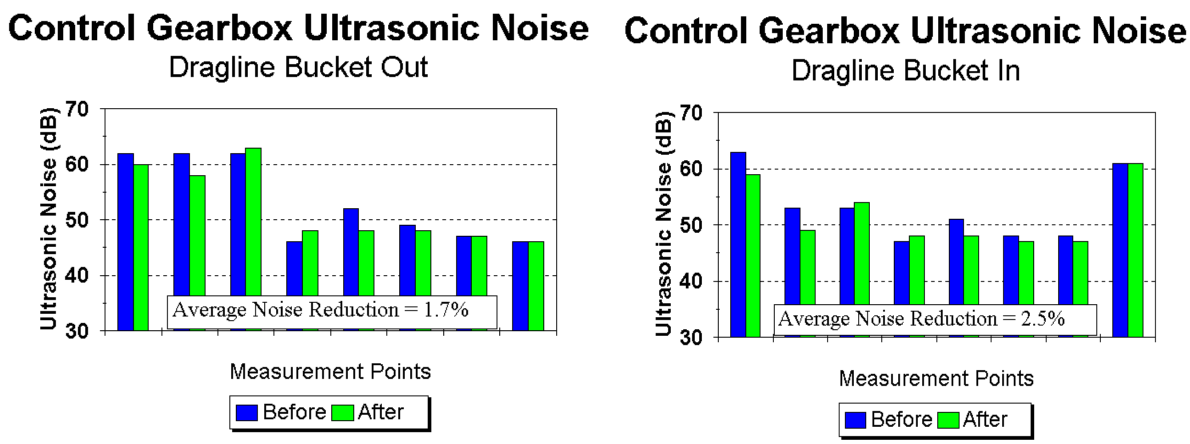
It is obvious that the temperatures in the test gearbox were reduced with the application of the boundary additive. The average reduction in temperature for the five measurement points is 3.2^{0}C or 6.0% with the additive, while the control gearbox temperature dropped by only 0.5^{0}C or 0.9%. The significantly cooler operation in the treated gearbox indicates that there has been a reduction in the amount of friction and wear in the unit.

Figure #10 - Drag Drum Test Gearbox Ultrasonic Noise Readings



The ultrasonic noise readings shown in the above figure were taken as the bucket was both extended and retrieved and complete sets of data are presented in Tables #5 and #6 in Appendix B. Five individual ultrasonic noise readings were taken at each point and the middle quantitative value recorded and reported. The ultrasonic noise readings taken corroborate the significant reduction in friction and wear occurring in the test gearbox compared with that in the control gearbox. The average reduction in ultrasonic noise emissions is 25.1% with the dragline bucket being retrieved, with a high of 42.6% and a low of 7.3%. With the dragline bucket extending out, the average reduction in wear noise is 24.5%. The greatest reduction was 40.0% and the lowest reduction was 7.1%. The control gearbox readings show virtually no change in the amount of wear noise from the various measurement points before and after. The overall reductions in ultrasonic wear noise are given in Figure #11 below and are less than 2.5%.

Figure #11 - Drag Drum Control Gearbox Ultrasonic Noise Readings



While there is no known correlation between ultrasonic noise readings and the amount of wear or friction in a piece of equipment, it is known that reductions in ultrasonic wear indicate reduced friction and lower rates of wear. In this test, the reduction of both gearbox temperature and ultrasonic noise emissions when the boundary additive was applied, provide good evidence of reduced friction, wear and therefore extended equipment and lubricant service life. The mining operation that initiated this test is now using the boundary lubrication additive in three of four draglines and in their conveyor system's gearboxes with good success.

Saw Guide Lubricant Application

The environmental and public relations advantages in using biodegradable, non-toxic process fluids in the lumber industry are obvious. It is strange that the switch to these fluids by several western Canadian sawmills has been justified by improved performance and lower cost operation. Oil is mixed with water and sprayed on the saw guides during cutting to keep them lubricated, cool and clean. The conventional petroleum based lubricant used was very thick and left gummy residue that was difficult, time consuming and expensive to clean up. The heavier oil mixed with the saw dust from cutting, plugged the saw guides causing them to overheat, requiring increased guide maintenance. A typical saw box contains two stacks of six large blades separated and stabilized by babbitt saw guides. Friction in this interface generates a significant amount of heat causing varnishing, rusting and vibration of the blades. The current mineral oil based saw guide oil works, but has caused some health problems with workers. A natural and synthetic ester based lubricant specifically designed for saw guide lubrication was substituted in several mills and has solved, not only the saw guide maintenance problems, but other process problems as well.

The new biodegradable lubricant is much more soluble in water and can be cut with considerably more water while still providing adequate lubrication. The higher water content provides better cooling and clearing of the saw dust. In fact the length between saw blade guide changes has been increased by a small margin. Some users have been able to use thinner blades with no loss of blade stability creating less lumber loss to sawdust.

From an occupational health and safety perspective the new fluid is a great improvement. The previous petrochemical fluid caused some respiratory health problems with some of the operators and skin rashes with a number of saw filers working on the blades. Since switching to the new, environmentally friendly fluid, both of these problems have disappeared. The low toxicity of the product is evidenced by one filing superintendent's report that earthworms were found eating the oil soaked wood pulp. In addition, the mill's permit allows the biodegradable oil soaked sawdust to be incinerated. The previous oil was not disposable in this manner.

Table #2 - Cost of Biodegradable and Mineral Oil Lubricants

	Mineral Oil	**Biodegradable Oil**
Initial Cost	$ 30 / 20 litre pail	$ 100 / 20 litre pail
Oil Consumption	3.33 litres per shift	0.83 litres per shift
Cost per Shift	$5.00 per shift	$4.16 per shift

All of these advantages in operation and safety have not come at any increased cost in many of the mills that have switched to the environmentally friendly fluid. The figures in Table #2 above were provided by an Alberta mill. They indicate that the higher initial cost of the biodegradable lubricant have been more than offset by the reduced consumption. Not accounted for is the lower disposal cost for the oil soaked saw dust generated daily in the saw boxes, the advantages of fewer health problems and associated downtime or the increased production from extended saw blade change intervals and the ability to use narrower blades creating less timber wastage.

Eco-System Compatible Railroad Wheel-Flange Grease

The fourth application discussion covers a very specialized grease application requiring not only excellent anti-wear, extreme pressure and water resistance, but very low temperature fluidity combined with low toxicity and ready biodegradability. In the early years of railroad engineering there was concern that the steel-wheel/steel-track interface would not provide enough friction to allow for sufficient tractive force. Experience soon dispelled these concerns and it was determined that, in fact, the amount of friction was potentially problematic, especially in the sliding contact of the rail gauge-face/wheel-flange interface. It is estimated by E. Rabinowicz that a loaded train of one hundred cars travelling from Chicago to Los Angeles and back would result in the loss of three quarters of a ton of metal![1] The cost of this type of wheel flange and rail wear is not limited to wheel and track maintenance, but must also include the even more significant losses in train momentum causing increased fuel consumption. Both of these losses are of special interest to those mining operations that use rail cars to transport ore and overburden to and from mine sites and processing facilities.

On major rail systems trackside lubricators were developed and installed to dispense grease onto the rail gauge face, especially near curved sections in the track where the wheel-flange/rail friction is most detrimental. The grease applied to the rails needs to have several important properties to provide optimum wear and friction protection. Not only are anti-wear and extreme pressure protection important, but the grease needs to be resistant to water, have sufficient tackiness that it is picked up by the wheel flanges and is carried some distance down the rail, and it needs to be dispensable at the cold temperatures of the Canadian climate. The current standard rail grease is one formulated with a mineral base oil thickened by a calcium or lithium complex soap and fortified with solid molybdenum and/or graphite additives.

These rail greases are used predominantly in sections of track that are highly curved due to rapid elevation changes or proximity to rivers or lakes. These areas are, by their very nature, environmentally sensitive eco-systems that can be permanently affected by contamination by toxic or bio-accumulating products. It is estimated that over half a million kilograms of petroleum based greases are distributed in these wayside lubricators each year in Canada alone. The bulk of this lubricant ends up being washed from the ballast and often ends up in enviromentally sensitive areas since the railroads often follow river and lake systems. The development of a non-toxic, rapidly biodegradable grease was initiated with the cooperation of a national railway.

Laboratory Formulation and Testing - Due to the proprietary nature of the formulation of this product very little detail of the formulation process is provided. The initial design objectives were defined around the currently available mineral oil based grease. The new grease needed to provide comparable anti-wear and extreme pressure protection, be pumpable from conventional wayside lubricators at temperatures down to -30^{0}C or lower and be rapidly biodegradable and non-toxic to the flora and fauna. In addition, the grease needed to have a dropping point higher than that currently available, be carried down the rail a comparable distance and reduce the rail-wheel flange friction factor.

Small pilot batches of a comparable grease were developed using biodegradable synthetic ester base fluid, additives to achieve the desired performance and a thickener system. In order to advance the testing program, a pilot grease manufacturing facility was constructed capable of producing 12 kilogram batches of grease from scratch. The plant includes a

[1] 1 Proceedings Rail & Wheel Lubrication Symposium, December 1-2,1981, Memphis, TN; published by Railway Systems & Management Association, Northfield, NJ, p. 98. Values confirmed by private communication with Dr. E. Rabinowicz, July, 1996.

heated, cooled and stirred kettle and secondary mixing and colloid mill units. After considerable optimization, the process produces consistent pilot batches of finished grease.

The first two criteria that needed to be met by the grease were the dropping point and the low temperature pumpability of the grease. Since these two properties are most influenced by the grease complex thickening reaction, they were the first evaluation step for the prototype formulation. The grease was determined to have a dropping point of greater than 110^0C, higher than the 94^0C dropping point of the current grease. The relative cold temperature pumpability of the greases was evaluated using a simple test apparatus.

Two square plastic containers with large plastic syringes attached at the bottom were charged with equal amounts of the two greases and placed in a laboratory freezer at -29^0C for extended periods of time. After only a few days at this temperature the current grease mirrored the known field performance and failed to flow into the vacuum cavity left when the syringe plunger was retracted. Even after several months of continuous storage, the prototype grease flowed easily into the cavity. When the new grease was later tested in the wayside lubricators at winter temperatures, it was found to be much more easily pumpable than the current standard. In fact, it was mentioned that the current grease often needed to be diluted with kerosine in order to pump adequately in very cold temperatures.

Three separate laboratory wear tests were used to compare the anti-wear and extreme pressure properties of the prototype grease with that of the conventional grease. ASTM D2266 Four Ball Wear test results showed the new prototype to meet or exceed the performance of the current grease in wear prevention properties under moderate loading conditions. The ASTM D2596 Four Ball EP test is a more severe loading condition used to evaluate the extreme pressure protection offered by lubricating greases. The result was again that the new prototype equalled the load carrying capability of the standard mineral based grease.

The final ASTM test procedure used was the ASTM D2509 Timken OK load test which was conducted. As had been anticipated, the prototype grease failed this test. The failure is rooted in the mechanism by which the EP additives function in the prototype grease. Rather than using the conventional solid graphite and molybdenum compounds, the grease utilizes an EP package that will, when activated by the heat generated by the friction of initial asperity contact, form a sacrificial and protective boundary lubrication film on the wear surface. This boundary film then reduces subsequent wear. Due to the required activation energy from initial wear, there is an initial scuffing and light scoring of the metal interface. The defined pass criterion for the Timken OK load test is the absence of any evidence of scoring on the test block after a ten minute run period. The grease behaved as predicted and allowed a small amount of scoring prior to providing EP and anti-wear protection. The test laboratory did report that the scoring would be best described as moderate and not severe.

Shorter term, modified Timken tests were carried out that suggest that the EP characteristics may well be equivalent to the current grease under slightly different wear conditions. As the product is still in the development stage, this data cannot yet be released in this forum. Overall, the acceptable result in two of the three ASTM standard wear tests and in the fourth modified procedure prompted the pursuit of field testing to establish the true performance.

Field Testing and Evaluation - In conjunction with the railroad track maintenance personnel, the prototype grease was installed at two carefully selected locations. The locations were chosen to provide good comparisons with nearby locations using the current grease. Though the curves were not identical, it was believed, from the track curvatures, grades, distances from lubricators and wear histories, that the comparisons in wear rates are comparable. A track wear gauge of our design was used in conjunction with a rail profilometer from the railroad. The rail wear over a period of 10 weeks was monitored and both methods showed reduced rail wear with the new prototypical grease. The results of the wear measurements are given in Table #3 below.

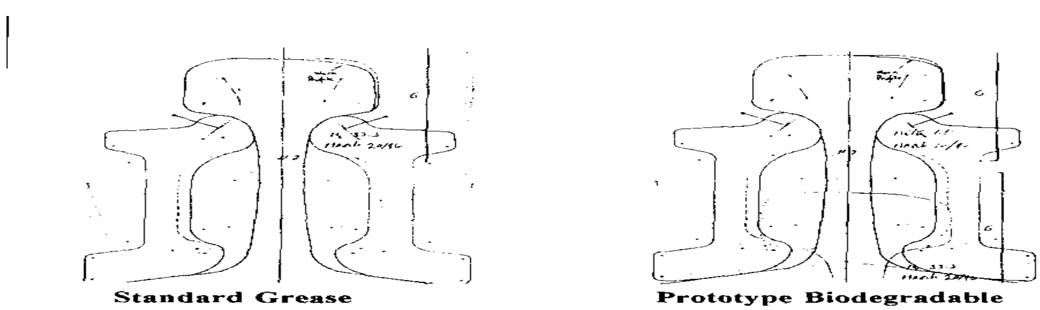
Table #3 - Field Test Rail Wear Data

	Curve A - Prototype Grease		Curve B - Standard Grease	
Date	Rail Width	Wear	Rail Width	Wear
Dec 13	73.90 mm		73.90 mm	
Jan 31	73.05 mm	0.85 mm	72.90 mm	1.00 mm
Feb 29	73.15 mm	(0.10 mm)	72.85 mm	0.05 mm

The wear data shows that over the first six week period there is a lower rate of rail wear with the Prototype grease as compared with the standard. Over the second period of time it appears that there has been metal added to the rail. In fact, what had happened was that a rail grinding unit had passed over both sections of rail drastically altering the profiles. The final rail width still clearly shows less wear with the new grease.

The profilometer traces in Figure #12 on the next page also corroborate the fact that, after three months, the rail lubricated with the biodegradable grease has less gauge face wear and a lower wear profile. These profilometer traces are taken directly from the contour of the rail by following it with a metal stylus. Note the minute difference in gauge width from the new rail profile for the test grease compared with the mineral based standard.

Figure #12 - Overlay of Rail Profiles at Mile 33.3 and Mile 1.8 (prototype)



The actual friction coefficient (•) of the rail gauge face can be measured using a special instrument called a tribometer, obtained from the American Association of Railroads. The unit is attached to the rail and by means of a spring loaded roller, it measures the friction factor. This unit was used to establish how far the grease was carried down the rails providing lubrication to the gauge face. It was established in these tests that the new, biodegradable grease is carried as far, if not further, than the current grease. The friction coefficient with the biodegradable grease was improved (lowered) from an average of 0.329 to an average of 0.259. Field testing will continue and it is expected that the new biodegradable rail grease may actually be an economically viable alternative and not just an ecologically preferable lubricant. Though the product cost is higher, the potential savings in rail wear alone appear to justify its use. A conservative estimate of the savings affordable by the reduced wear with the new lubricant was calculated. Based on the wear results obtained in the field test reported earlier, and information on the cost of replacing rail, the savings on rail replacement alone could exceed $25,000 per mile of curved track per year. This value does not include other primary and secondary cost benefits including site clean up, labor to free frozen lubricators or traffic rerouting or delays caused during rail replacement.

Comprehensive Engine Bench Tests

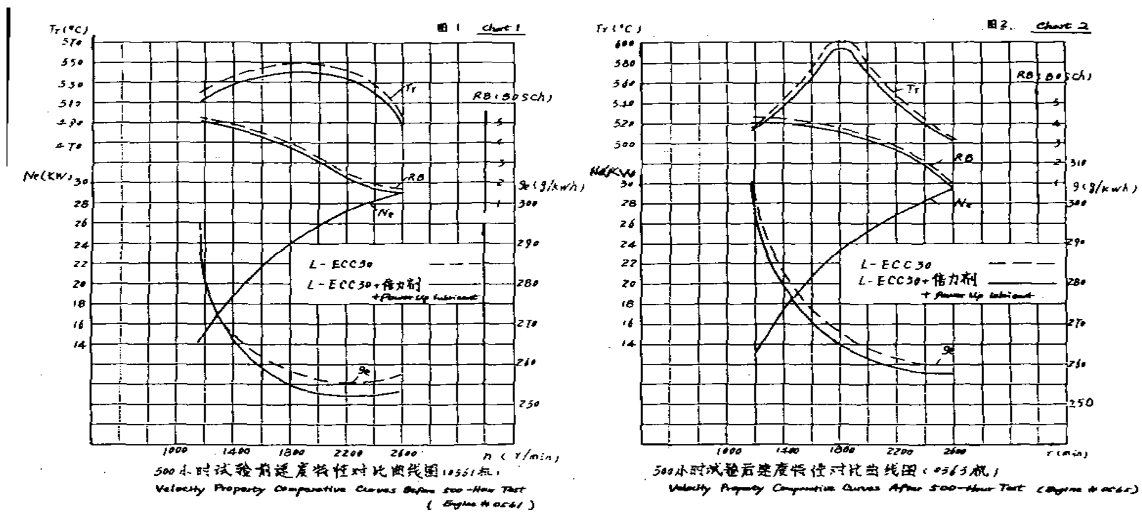
The final example presented herein is a comprehensive engine bench test program that is being performed by the Shanghai Internal Combustion Engine Research Institute. The program involves the long term testing of two identical N485 high speed diesel engines manufactured by the Wujin Diesel Plant. The initial report on the testing documents the first 500 hours of engine testing. The test is still ongoing and has now reached a total of over 1500 hours. The SICERI plans on continuing the bench tests until at least 5000 hours.

Test Procedure and Equipment - Two identical N485 diesel test engines were mounted on laboratory bench stands and attached to dynamometers to enable consistent engine loading over their full rpm range. Test engine #0561 was used as the main test engine, while engine #0565 was largely a control engine for reference comparison. Test engine #0561 was tested over the full range of engine speeds at full load, first with the reference lubricant L-ECC30 only and then again with the reference lubricant and the boundary lubrication additive. Once the before and after data was gathered this engine was run for the full 500 hour period. The engine oil was changed at running times of 150, 300, 400 and 500 hours and analyzed for wear metal content and size distribution and lubricant quality. The lubricant quality was determined using infrared spectrographic analysis and other physical property tests.

The control engine #0565 was run in under the same conditions as engine #0561 and then run for the same 500 hour period, with the same oil change intervals. For this engine the full load tests were done over the range of engine speeds engine at the end of the oil change interval. The used oil was then treated with the boundary lubricant additive and the full load tests were repeated. Throughout the 500 hour test period, the fuel consumption rates, exhaust smoke opacity and exhaust temperatures of each engine were monitored. Samples of both lubricants were taken at each oil change interval and analyzed for wear metal content and lubricant degradation. After the 500 hour interval both engines were disassembled and the wear of certain parts was measured by micrometer. The cleanliness of the pistons was graded and reported at the end of 500 hours.

Engine Load Test Results - The engine performance curves for test engine #0561 are given on the left hand side of Figure # 13.

Figure #13 - Engine Performance Curves



The top curve is the exhaust gas temperature in °C with the dashed line the reference oil alone and the solid line representing the results with the reference oil/lubricant additive mixture. The exhaust temperature is an average of 5.2°C lower with the boundary additive. The second curve is a measure of the exhaust smoke opacity measured using the Robert Bosch scale. Once again, the opacity of the exhaust is improved by the additive by an average of 7.9%. The third curve labelled Ne is the power output curve for the engine throughout the rpm range. The two curves are identical and overlap, proving consistent loading in the before and after tests. The final lower curve provides the brake specific fuel consumption. The bsfc is the normalized fuel consumption and provides the best measure of engine efficiency by removing variations in load. The curves are similar from 1200 to 1400 rpm above which the fuel consumption drops considerably with the additive present. The greater reduction in consumption under load is important since the overall average of 1.0% includes the lack of improvement at idle speeds.

The second chart on the right of Figure #13 is the result of the full load testing done at the end of the 500 hour interval with engine #0565. The oil was changed and the tests performed on the worn engine with the reference oil alone and then the oil treated with the subject additive. Once again, the addition of the boundary lubricant to the reference oil improved the performance curves in each measured parameter. The exhaust temperature was lowered an average of 9.8°C, the exhaust opacity was improved by 14% and the fuel consumption was decreased 1.4%.

500 Hour Reliability Results - Except for the first 80 hours, the fuel consumption curve for the test engine was lower than that for the control engine virtually throughout the 500 hours. The initial increase (though less than 0.8%) is attributed to different wearing in patterns in two newly rebuilt engines. An average improvement of about 1.2% corresponds well with the results in the load tests (and a previous SAE J1376 Fuel Economy test which showed and improvement of 1.34% in bsfc for a highway tractor).

Bearing, journal, piston and cylinder sleeve dimensions were measured before and after the 500 hour test period for each engine. The reductions in wear totalled 7.3% in the cylinder sleeves, 47.7% average for the main and connecting rod journals, 31.4% for the piston pins and 16.5% for the connecting rod bearings. The pistons in each engine were removed and rated for their cleanliness. As expected, the engine treated with the improved additive package was much cleaner. The piston ring lands were 5.9% cleaner, the skirts 6.7% cleaner and the piston crown had 70.5% fewer deposits than the reference engine's. The additive contains detergent and dispersant components to control deposits. These significant reductions in wear rates of engine components and improved piston cleanliness with the additive/reference oil mixture suggest that the engine life will be considerably improved and that maintenance and repair costs will be lowered over the extended engine life.

Spectrographic oil analysis of the final oil sample at the end of the 500 hour test period also corroborates the measured wear rates. The iron, copper, chromium and aluminum wear metal concentrations were determined by atomic absorption spectroscopy and are listed in Table #4 below.

Table #4 - Used Oil Wear Metal Concentrations by AA

Engine	Iron (ppm)	Copper (ppm)	Chromium (ppm)	Aluminum (ppm)
0565 (Ref. Oil)	31.6	4.0	2.2	9.2
0561 (Ref Oil + Add.)	18.7	2.8	0.7	7.8

Not only did the concentrations of the four wear metals decline with the additive, the particle size distribution also improved. Using a direct reading ferrograph to separate small iron particles (less than 5 microns) from large particles greater than 5 μm, the number of particles per millilitre of oil was determined. The control engine using only the reference oil had a small particle count of 21.1/ml and a large particle concentration of 46.4/ml. The test engine lubricated with the reference oil and boundary additive contained only 8.1 and 21.5 particles per ml respectively.

Finally, infrared spectrographic analysis was used to determine the extent to which the two lubricants were degraded in service. The soot, oxidation and nitration levels were measured for both used lubricant samples. The soot level in the test engine was 4.93 abs/cm compared with 5.01 abs/cm in the control engine. The oxidation level was also lower at 0.92 compared with 2.75 abs/cm. Finally, the nitration rate was improved from 3.27 abs/cm to only 2.21 abs/cm. These lower degradation levels indicate lower peak lubricant temperatures, reduced blow-by gas contamination and lower soot loading from incomplete combustion.

In order to quantify these results economically a simple fuel consumption calculation based on reference data from Detroit Diesel was completed. For a 300 bhp, Series 60 engine operating at an average load of 150 hp, the fuel consumption in one 250 hour oil change interval is 6205 litres (at 0.310 lb fuel/bhp hr). Using the 1.2% average fuel consumption improvement from the engine tests, the fuel savings are 74.5 litres or $36.50 (at $0.49/L). The cost of the additive to treat this engine for one interval is only $33.50. The net savings based on fuel consumption alone are $3.00 per interval. Not a huge payback, until one factors in the lower maintenance, repair and downtime costs associated with the wear reductions shown.

Conclusion

Though conventional lubricants provide excellent performance over a wide range of operating conditions, there are applications where, for reasons of extreme operation or impracticality, specialized lubricants can provide solutions to maintenance and equipment wear problems. There are also many specialized applications, where benefits from the application of specialized lubricants are possible. The use of niche products or additives in place of the acceptable but not exceptional conventional lubricant products are often and economically beneficial alternative. Applied in the right places for the right reasons, additives and specialty products have a growing place in the maintenance and lubrication fields, a fact that is evidenced by the appearance of several specialty additive products from original equipment manufacturers and lubricant suppliers.

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APPENDIX A

Ultrasonic Noise Measurement

Most humans can hear sounds in the range from 20 to 16,000 Hz (Hertz). The world is full of sounds from 20,000 Hz (20 kHz) and up, that humans cannot hear, this noise is called Ultrasound.

Most frictional sounds produce maximum amplitude (loudest noise) right around the 40 kHz range. The newest and very reliable method of testing for optimum lubrication, wear patterns and predictive maintenance was developed by NASA and is called Ultrasonic Detection. Hundreds of companies worldwide are using the ultrasonic detector as a method of predictive maintenance.

The ultrasonic detector used in these tests measures ultrasound in the range from 36.7 to 41.7 kHz. This is done for three reasons:

- it covers the range of high amplitude metal to metal frictional noises,
- it eliminates a lot of noise that is not relevant to frictional measurement and
- ultrasound at 40 kHz is very directional and only travels short distances.

The detector converts the ultrasound amplitude to corresponding audible sounds so that the operator can hear it using headphones. If the intensity and pitch of the ultrasound changes, then the audible sound will change in direct proportion. The converted audible sound wave intensity is displayed on an LED screen in dB between 0 and 100. The sound wave intensity is in direct proportion to the incipient wear occurring in the component. There is, however, no known equation or correlation between ultrasonic noise reduction and wear or friction reduction.

It is important to note that the decibel scale is logarithmic and as such covers a wide range of actual sound wave intensities. An increase in ultrasound of 10 dB is actually a ten-fold increase in sound wave intensity.

Example: An increase from 40 to 50 dB is an increase in noise of 10 dB but is actually a ten times increase in intensity.

Ultrasonics is a relatively new predictive maintenance technique and new information and methods are being continually developed.

APPENDIX B

Table #5 - Test Gearbox Readings

Reference Points	Without Additive (dB)		With Additive (dB)	
	Bucket In	Bucket Out	Bucket In	Bucket Out
2	55	56	51	52
3A 3B	48 47	52 48	42 34	39 46
4	47	46	27	32
5	48	52	36	37
6	46	45	36	34
7A 7B	48 52	55 55	33 35	33 35

Table #6 - Control Gearbox Readings

Reference Points	Control Gearbox - Same Test as Above without Additive (dB)			
	Bucket In	Bucket Out	Bucket In	Bucket Out
2	63	62	59	60
3A 3B	53 53	62 62	49 54	58 63
4	47	46	48	48
5	51	52	48	48
6	48	49	47	48
7A 7B	48 61	47 46	47 61	47 46