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THE CAPACITY OF SUPERFINISHED VEHICLE COMPONENTS TO INCREASE FUEL ECONOMY

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ABSTRACT

The lubricant industry is emphasizing the use of low-viscosity lubricants to increase fuel economy. Fuel mileage increases as high as 8% are claimed when conventional engine and driveline lubricants are replaced with new-generation products. Low viscosity lubricants, however, must contain more robust anti-wear and extreme pressure additives to counteract their reduced λ ratio. Consequently, switching to lower viscosity lubricants in order to gain fuel economy entails risk. Should the additive package fail to perform, engine, transmission, and drivetrain components will be seriously damaged.

It seems appropriate then, to attempt to increase the λ ratio for low viscosity lubricants. This, of course, can be done by reducing surface roughness. Superfinishing the surface using chemically accelerated vibratory finishing is a practical and well proven approach for accomplishing this. This paper will present data from both laboratory and field testing demonstrating that superfinished components exhibit lower friction, operating temperature, wear and/or higher horsepower, all of which translate directly into increased fuel economy.

INTRODUCTION

As the cost of gasoline and diesel fuel rises, engineers are exploring many different opportunities for increasing fuel efficiency of gas and diesel-powered vehicles. One area of particular interest is the reduction of frictional losses, often referred to as parasitic friction. Fuel efficiency increases as more energy becomes available to propel the vehicle and less energy is wasted on frictional losses. Currently, less than 15% of fuel energy is converted to useful energy that either propels a car down the road or powers its accessories. The lubrication industry has made great strides in developing low viscosity lubricants to improve fuel efficiency. Replacing the engine oil alone has increased fuel efficiency by 1 to 5% for passenger cars and 4 to 8% for heavy duty trucks. The higher value was for a Class 8 truck in which not only the engine oil but also transmission and drive axle lubrication were replaced. Figure 1 below clearly illustrates that even a 1% increase in fuel savings will have a significant effect not only on the American economy but also on vehicle pollutants.

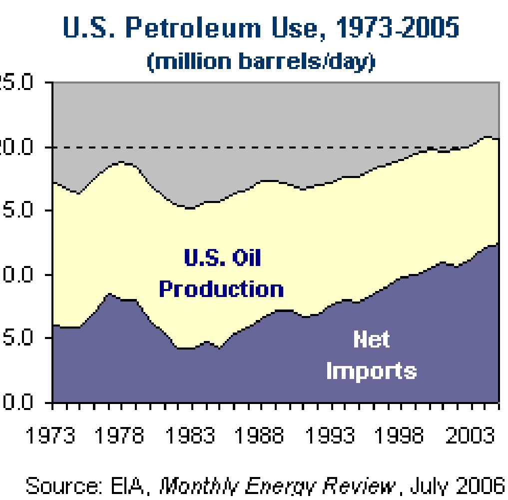


Figure 1: Graph showing U.S. oil use from 1973 to 1975[i].

A typical engine lubricant is composed of 75% to 95% base oil, with the rest being additives. The additives package is a complex blend of chemicals: friction modifiers, anti-wear agents, extreme pressure additives, corrosion inhibitors, antioxidants, detergents, dispersants, anti-foam additives, and viscosity modifiers.[ii] The additives must be chemically compatible with each other as well as with the base oil. Their protective properties must perform under a wide range of operating parameters, and at the same time have little negative impact on the performance of the catalytic converter.[iii] Many traditional additives contain zinc, sulfur, boron and phosphorous, but since some of these elements can lower the performance of the catalytic converter, there is constant regulatory pressure to lower their concentrations. It is important to recognize that additives can also have adverse effects on the system. For example, it has been proposed that anti-wear additives actually penetrate the surface of the metal, causing a reduction in nano-hardness, and that extreme pressure additives can form corrosive chemicals that hasten the onset of corrosion and micropitting.[iv] Although higher viscosity grades of oil provide sufficient wear protection, they also produce higher frictional losses. Lowering viscosity will reduce frictional losses, but unless the additive package is optimized, the product may fail to provide adequate protection because the lubricating film is a good deal thinner. Moreover, most traditional additives are classified as hazardous and may worsen the growing global pollution problem.

Lowering lubrication viscosity must be done with care to avoid inadvertently increasing equipment wear as a result of increased premature micropitting and/or scuffing. A thinner lubricating film, for example, may exacerbate micropitting. Since the phenomenon of micropitting is tied to fatigue, it may not reveal itself until much later in the vehicle's duty cycle. It is conceivable that the vehicle will exhibit increased fuel efficiency in an EPA test cycle, but this could come at the expense of major mechanical overhauls further down the road. For this reason, concocting the ideal package of additives is a balancing act for the lubricant formulator.

It is possible, however, to reduce viscosity while simultaneously maintaining or even increasing the λ ratio. One way to accomplish this is to employ chemically-accelerated vibratory finishing on the working surfaces of gears, shafts and bearings. This superfinishing technique can partially or completely remove peak and valley asperities. Chemically accelerated vibratory finishing, (referred to in this paper as "superfinishing") has been commercially used for more than 20 years to improve the performance of working surfaces. It is important to remember that this process not only creates a planarized surface, but also removes distressed metal left over from machining, grinding and/or heat treatment processes.

Superfinishing is performed in vibratory finishing bowls or tubs in two separate steps: a Refinement Step and a Burnishing Step. In the Refinement Step, proprietary active chemistry is used in the vibratory machine in conjunction with high density, non-abrasive ceramic media. When introduced into the machine, this active chemistry produces a stable, soft conversion coating on the surface of the metal part being processed. The rubbing motion across the part developed by the machine and media effectively wipes the conversion coating off the "peaks" of the part's surfaces, but leaves the "valleys" untouched. (No finishing occurs where media is unable to contact or rub.) The conversion coating is continually re-formed and rubbed off during this stage, producing a surface-smoothing mechanism. This process is continued in the vibratory machine until the surfaces of the part are free of asperities. In the Burnishing Step, the active chemistry is rinsed from the machine with a neutral soap. The conversion coating is rubbed off the part one final time to produce the superfinished surface. In this final step, no metal is removed.

By reviewing the images in Figure 2, it is easy to see how this superfinishing process produces a highly desirable surface with regards to improving the performance of working surfaces like those found on gears and bearings. The starting surface of gears or bearings are machined and/or ground, resulting in a starting surface that has peak and valley asperities as well as a distressed surface layer. In Stage 1 of the superfinishing process, the peak asperities are planarized, resulting in a surface that has significantly improved performance properties since no asperities are present to penetrate the lubricating film. As the superfinishing process is continued, as shown in Stage 2, the valleys begin to disappear. As the superfinishing process advances further, the Final Condition is attained. This is the optimum surface since all peaks and valleys have been removed along with the distressed surface layer, leaving a micro-textured isotropic surface that facilitates lubrication. The rationale for the above was discussed in several other publications. The Vehicle Bloc at the Gear Research Institute at Pennsylvania State University tested rolling sliding contact-fatigue specimens that were superfinished to conditions similar to that shown in Figure 2.[v] The best performance was achieved with a smooth (< 0.1 μm R_a) and textured surface. In another study, a smooth (< 0.1 μm R_a) and non-textured surface was evaluated for scuffing resistance. Although the non-textured surface significantly outperformed the ground baseline specimens, it fell short of the performance achieved by a smooth (< 0.1 μm R_a) and textured surface.[vi] Bell Helicopter Textron recently questioned if the superfinished surface was actually excessively smooth in a way that hinders adequate lubrication. They concluded that the superfinished surface increased the λ ratio such that the gear performed well despite operating under extreme temperature and load conditions[vii].

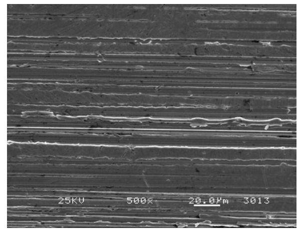
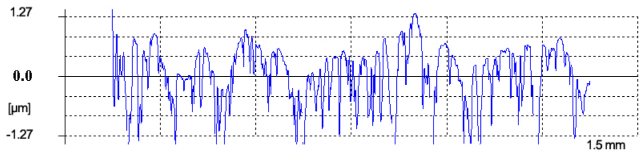
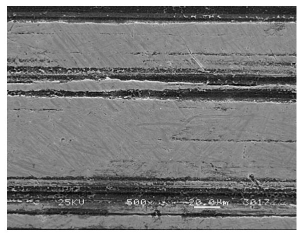
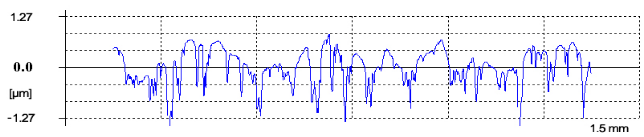
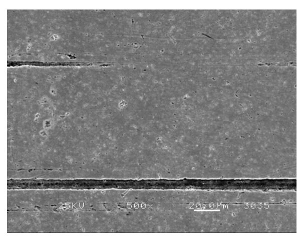
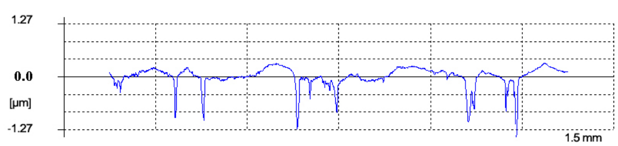
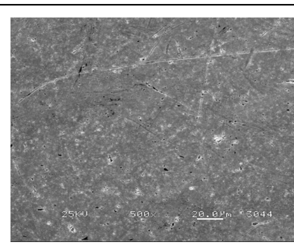
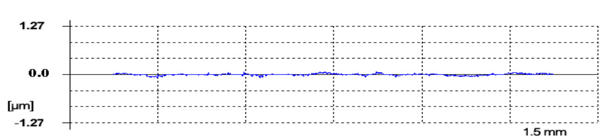
Performance	Surface at 500X Magnification	Roughness Average
Worst **Starting Condition:** Ground surface having peak and valley asperities along with a distressed layer of metal at the surface.		R_a: 0.58 µm R_z: 3.5 µm
		
Good **Stage 1:** Partially superfinished surface where the peak asperities have been planarized.		R_a: 0.30 µm R_z: 2.0 µm
		
Better **Stage 2:** Superfinished where only a few valley asperities remain on the surface.		R_a: 0.13 µm R_z: 1.1 µm
		
Best **Final Condition:** Superfinished surface with all asperities removed while displaying the beneficial and inherent micro-texture.		R_a: 0.025 µm R_z: 0.17 µm
		

Figure 2: SEM images at 500X and profilometer traces of the surface condition of a ground or machined gear or bearing as it progresses through the superfinishing process.

Chemically accelerated vibratory finishing is currently being used by major companies in the automotive racing, aerospace, heavy-axle and wind-turbine industries. In each case, superfinishing was adopted only after a tremendous amount of forethought as well as extensive laboratory and field testing.[viii,ix,x,xi,xii,xiii,xiv] Unfortunately, much of this data is proprietary and confidential.

This paper will present an overview of a number of laboratory and field tests conducted over the years on gears, bearings and test specimens. Some of the data in this paper has been presented in widely scattered publications (full details can be found within the references), while other portions are a result of internal communications with proprietary customers. The results illustrate that superfinishing reduces friction, operating temperature, power loss and wear. It is hoped that this process will be more widely utilized in the future as a way to improve the fuel efficiency of gasoline and diesel-powered vehicles. At the same time, it should make it easier for the lubricant industry to formulate the next generation of lubricant packages since fewer and reduced concentrations of additives will be required for superfinished parts.

TESTING

Friction manifests itself as heat, wear, and power loss. Increased fuel efficiency can be expected when friction is reduced. The following are a series of laboratory and field tests that illustrate superfinishing's contributions to reducing parasitic friction.

Laboratory Testing:

I. Temperature Reduction of Spherical Roller Bearings (SRB)
II. Temperature & Friction Reduction of Roller Bearings (U.S. Patent # 5,503,481)
III. Friction & Wear Reduction - Falex Ring on Block
IV. Friction Reduction - Falex 3 Ball on Flat
V. Friction & Scuffing Reduction - Falex Washer-on-Ring
VI. Power Gain of Spur Gears
VII. Frictional Reduction of NASCAR Transmission
VIII. Horsepower Gain of NASCAR Rear Differential
IX. Efficiency Gain & Temperature Reduction of Rear Axle Ring & Pinion
X. Friction Reduction of Automotive Valvetrain

Field Testing:

I. Temperature Reduction of Truck Rear Axle
II. Temperature Reduction of S-76 Helicopter Transmission

LABORATORY TESTING

I. Temperature Reduction of Spherical Roller Bearings[xv]

The effect of superfinishing on the operating temperature of spherical roller bearings and assemblies was evaluated. Testing was conducted on a Variable Speed Spherical Roller Bearing test rig in a Mobil DTE Extra Heavy Oil bath. Two bearings were superfinished, one having just the rollers superfinished and the other with both the rings and rollers superfinished. The baseline was a standard production honed bearing. Steps were taken to ensure that internal clearance was not a variable in the comparison. The procedure consisted of running each bearing at 2,400 rpm and allowing them to heat stabilize at radial loads from zero to 4,536 kg. After stabilization, the bearing temperature was recorded and load increased by 454 kg. Figure 3 is an example of the test data obtained at 2,400 rpm. Optimum performance benefits were achieved when both the rollers and rings were superfinished. Bearings with only the rollers superfinished showed a notable but lower improvement. At a load of 4,082 kg, the completely superfinished bearing ran 22 °C cooler than the standard production honed bearing.

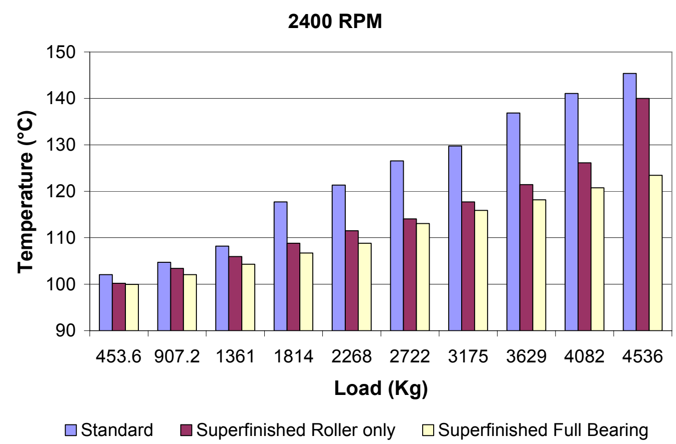


Figure 3: Temperature versus load for superfinished and standard spherical roller bearings.

II. Temperature & Friction Reduction of Roller Bearings (U.S. Patent # 5,503,481)[xvi]

The Timken Co. evaluated superfinished roller bearings for the purpose of reducing friction and thereby enhancing load-bearing capability. The superfinished and the enhanced roller bearings (final grind is carried a bit further on the working surface to reduce the run-in time) were compared. The enhanced-performance superfinished bearings were patented: U.S. Patent # 5,503,481, *Bearing Surface with Isotropic Finish*.

Enhanced roller bearings are typically finished with a final grinding step followed by a honing step to achieve between a 0.075 to 0.2 µm R_a. The superfinished bearings had <0.075 µm R_a. Grinding and honing, however, produce a directional surface texture having peak and valley asperities with these surface irregularities extending in the circumferential direction. The tests proved that a surface with isotropic irregularities (one which has no particular orientation) is superior to those of irregularities extended longitudinally or transversely in the direction of movement. Figure 4 is an image of the superfinished surface taken from the patent. At a magnification of approximately 500X, the SEM image clearly shows the isotropic micro-texturing.

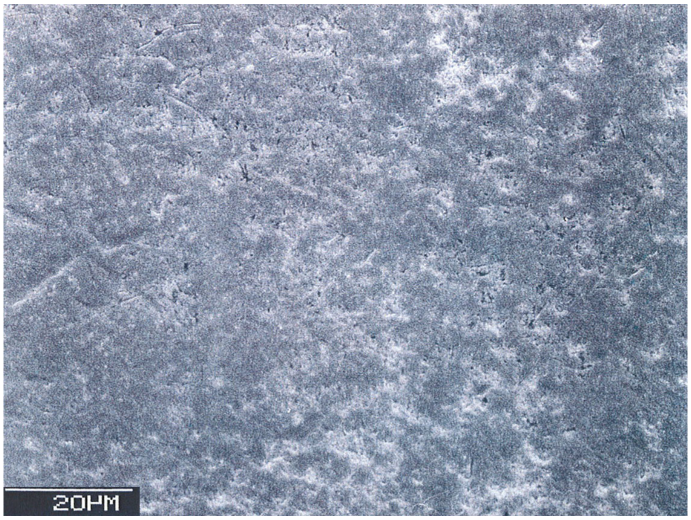


Figure 4: SEM image of the superfinished surface at approximately 500X, showing isotropic micro-texturing.

Figure 5 plots operating temperature versus time; figure 6 plots torque versus time. During run-in, the bearing having the enhanced surface finish shows obvious torque and temperature spikes. Both charts show that the superfinished bearings did not require run-in. No run-in spikes are observed since the superfinished bearing has no peak asperities. Note that the superfinished bearing operates with lower friction and lower temperature than the enhanced surface finish bearing.

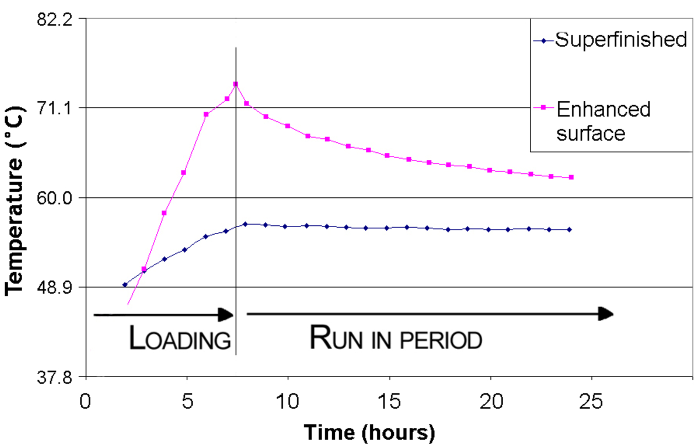


Figure 5. Roller bearing temperature versus time.

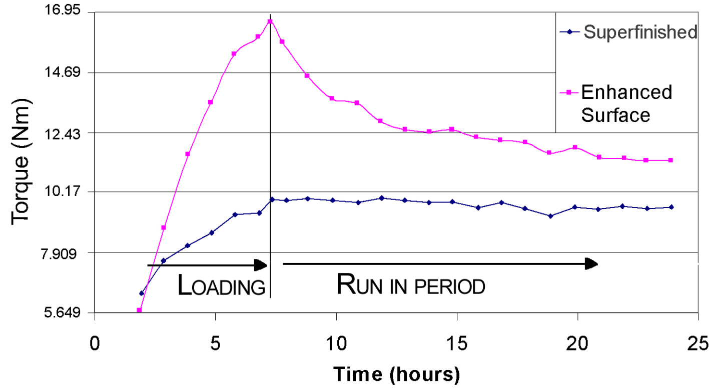


Figure 6. Roller bearing torque versus time.

III. Friction & Wear Reduction - Falex Ring-on-Block[xvii]

This study employed a simple sliding test to examine how superfinishing improves tribological contact performance. Two sets of rings and blocks were used for this evaluation. One set was finished with the conventional grinding method to a 0.6 µm R_a. The other set was superfinished to <0.1 µm R_a. See Figure 7.

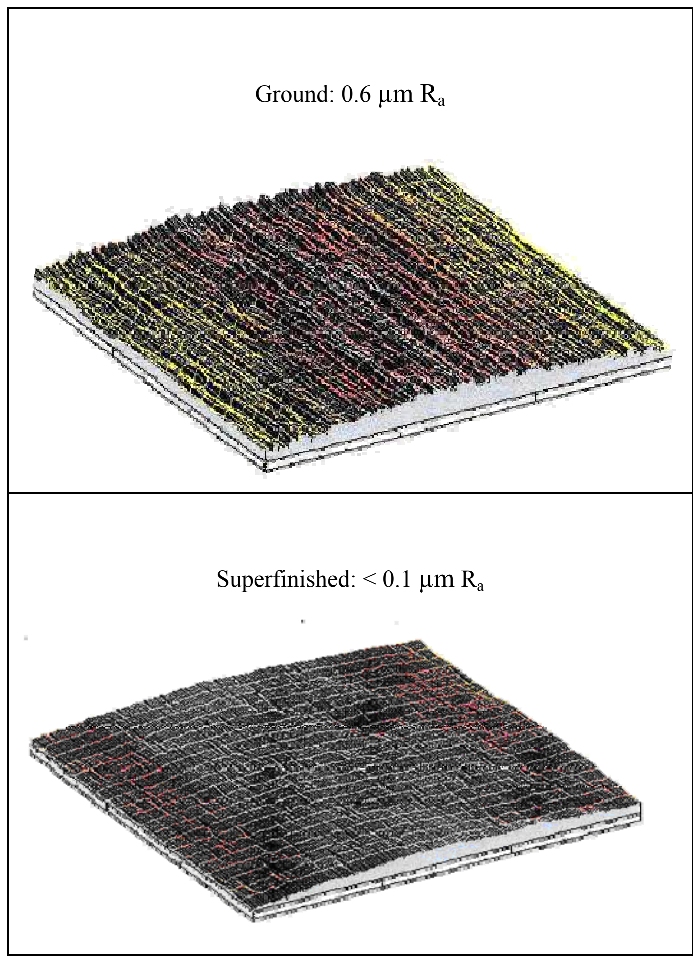


Figure 7. 3D images of a ground and superfinished ring.

Tests were run at ambient starting temperature and using an applied load of 3.64 kg. The lubricant was SAE 20 grade containing only rust and oxidation inhibitors. It had a viscosity of 55.72 cst at 40° C and 8.15 cst at 100° C. The rotational speed of the ring was 800 rpm and the test duration was 6.0 hours. The test ring diameter was 50 mm. Frictional force and sump temperature were monitored during each test.

Frictional force and sump temperature of the operating system for each test specimen and their respective surface finishes are shown in Table 1 below. The frictional force was approximately nine times higher on the ground set than the superfinished set. The sump temperature showed a reduction of approximately 18° C for the superfinished set.

	Superfinished 0.058 μm R_a	Ground 0.58 μm R_a
Friction Force of Sliding Contacts (Newtons)	6.1	57.6
Temperature (°C)	46.8	65.1

Table 1. Friction and temperature results for Falex Ring-on-Block test: superfinished versus ground.

The study also focused on the wear pattern (scars) present on the blocks after the test was completed. The wear pattern on the ground and superfinished blocks are shown in Figure 8 and Figure 9, respectively. The wear pattern on the ground block was severe and showed significant weight loss. Meanwhile, the superfinished block showed very little wear and almost no weight loss at the end of the test.

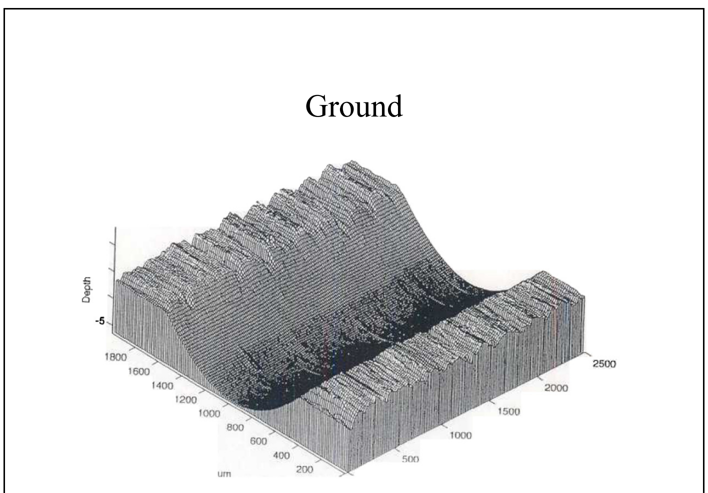


Figure 8. Wear pattern on the ground block at the end of the test.

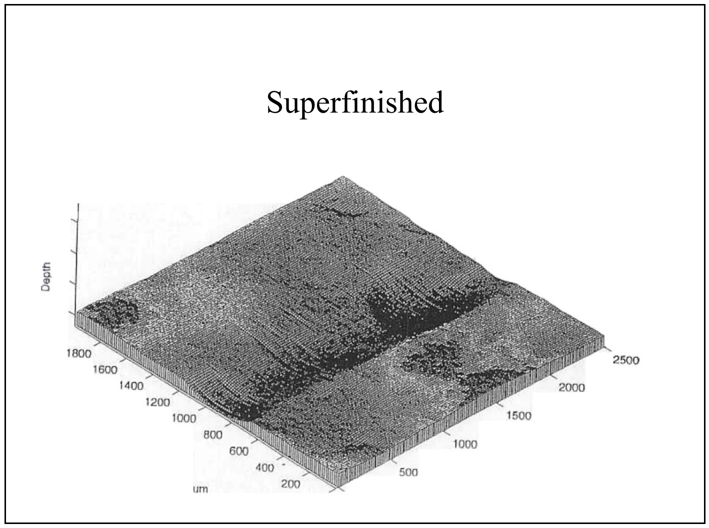


Figure 9. Wear pattern on the superfinished block at the end of the test.

IV. Friction - Falex 3 Ball-on-Flat

An automotive supplier of drivetrain components performed the following Falex 3 Ball-on-Flat test to determine the effects of superfinishing on wear and friction. The specifications for the baseline balls and disks are given in Table 2; test parameters are given in Table 3.

	Balls	**Disk**
Steel	AISI 52100	SAE 8620
Diameter (mm)	12.7	38.0
Surface HRc	64	61
R_a (μm)	0.025	0.525

Table 2. Specifications of the baseline balls and disk.

Rings were superfinished to two different conditions: Full finish (refinement and burnish cycles) and partial finish (refinement cycle only). The full and partial finishes had a 0.04 μm and 0.0725 μm R_a, respectively.

The test results are presented in Figure 10. The baseline samples have an initial coefficient of friction of approximately 0.125. After run in, it falls to approximately 0.11. Both the partial and fully finished superfinished samples had a coefficient of friction of approximately 0.06. Note that the superfinished samples did not show a run-in spike. The fully superfinished samples had a slightly lower coefficient of friction than the partially superfinished samples.

Total load	136 kg
Speed	300 rpm, 0.64 m/s
S/R ratio	Pure Sliding
Hertz stress	438 ksi, 3 GPa initial
Max shear stress	136 ksi, 938 MPa
Lubricant	SAE 75W-90 EP Synthetic
Kinematic viscosity (cst)	119.7 at 40 °C 16.68 at 100 °C
Test time	30 minutes
Temperature	20 °C at start

Table 3. Testing parameters.

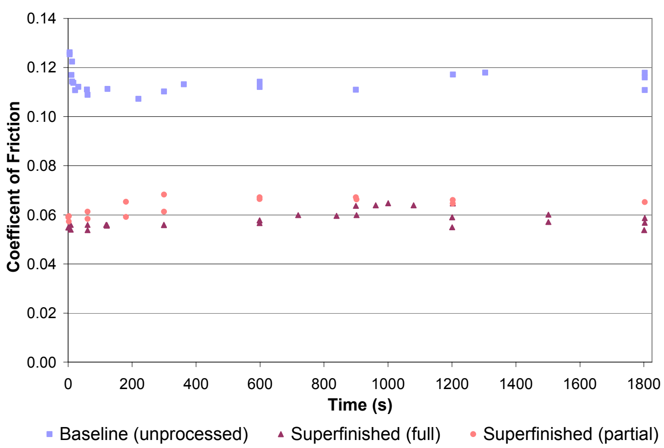


Figure 10. Coefficient of friction versus time for the Falex 3 Ball-on-Flat.

V. Friction & Scuffing Reduction - Falex Washer-on-Ring

An automotive supplier of drivetrain components employed a Falex Washer-on-Ring sliding test to determine how superfinishing affects scuffing resistance. Two baseline sets having 0.425 µm to 0.75 µm Ra were tested and compared to five sets superfinished to 0.0475 µm to 0.0725 µm R_a. The test conditions are presented in Table 4.

Both baseline samples had a coefficient of friction of approximately 0.11. Both baseline samples scuffed at approximately 600 rpm. All five superfinished sets had a coefficient of friction of approximately 0.03 to 0.04. Four superfinished samples did not scuff even at the highest rotational speed. The test was eventually suspended. One superfinished sample scuffed at 580 rpm. No explanation was provided for the single failure, but it was suspected that the starting sample was out of geometric tolerance.

Total Load	256 kg constant
Speed	Ramp up from 100 to 900 rpm at 100-rpm increments.
S/R ratio	2, simple sliding
Contact Stress	3.90 – 5.52 MPa
Lubricant	Mobile 80W-90 Mineral. Oil drained before testing.
Test Time	1.0 minute at each speed
Temperature	20 °C at start

Table 4. Conditions for Falex Washer-on-Ring Test.

VI. Power Gain of Spur Gears[xviii]

General Motors Corporation, Powertrain Division, measured the impact of superfinishing on gear efficiency at the Gear Dynamics and Gear Noise Research Laboratory at The Ohio State University. The superfinished gears were compared to baseline ground gears. The test also compared alternative low-friction lubricants with the standard baseline lubricant. Measurements of power loss under both loaded and unloaded conditions were reported in order to distinguish between load-independent (spin) losses and friction-induced mechanical power losses.

The efficiency test machine was designed to accommodate rotational speeds as high as 10,000 rpm and loads up to 690 Nm, which correspond to a maximum transmitted power of nearly 710 kW. A jet lubrication system was developed to cool and lubricate the gears. A dry sump system minimized churning losses. An external heating and cooling system was used to maintain oil temperature and pressure during testing. The baseline lubricant (Lubricant A) in these tests was synthetic 75W-90 oil supplied to the gearboxes at 110 °C.

This experiment also evaluated how pitch affects efficiency. Fine-pitched gears lower sliding velocity, which has been known to reduce power loss, but at the expense of lower bending fatigue. 23T (coarse-pitch) and 40T (fine-pitch) gears were used for this test. The gears were superfinished to a 0.06 to 0.09 µm R_a while the hard ground surfaces had 0.25 to 0.47 µm R_a. The gear design parameters and test conditions for this study are shown in Table 5 and Table 6, respectively.

Parameter	Fine Pitch 40T	Coarse Pitch 23T
Number of Teeth	40	23
Operating Diametral Pitch	11.104	6.39
Diametral Pitch	10.955	6.43
Module	2.319	3.950
Operating Pressure Angle	26.5	25.9
Pressure Angle	28	25
Face width, mm	26.67	19.4818
Contact Ratio	1.40/1.45	1.53/1.56
Tip Thickness, mm	1.1176	1.2446
Tip Clearance, mm	0.5334	0.6096
Backlash, mm	0.114/0.191	0.119/0.196
Operating Center Distance, mm	91.501	91.501

Table 5. Test gear design parameters.

Test	Speed (rpm)	Torque (N-m)
1	6,000	406
2	6,000	542
3	6,000	677
4	8,000	406
5	8,000	542
6	8,000	677
7	10,000	406
8	10,000	542
9	10,000	677
10	10,000	0
11	8,000	0
12	6,000	0
13	4,000	0
14	2,000	0

Table 6. Test conditions used in this study.

Fine-pitch gears are more efficient than coarse pitch gears by roughly 34% due to the lower sliding velocities generated in these gears. Superfinishing improves the efficiency of both fine and coarse pitch gears by approximately 17%. See Figure 11 and Figure 12. Low-viscosity lubricants can improve spin loss in spur gears through reduction in rolling losses, but the lubricants tested in this study had little effect on loaded power loss.

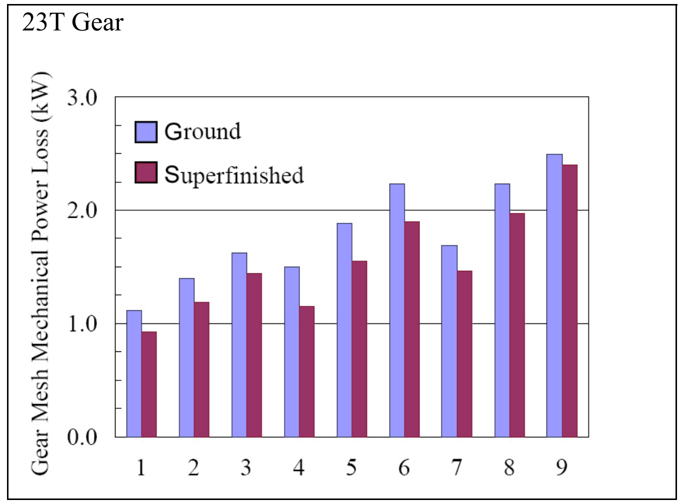


Figure 11. The effect of superfinishing on mechanical gear mesh power losses for 23T Gear; lubricant A at 110 °C.

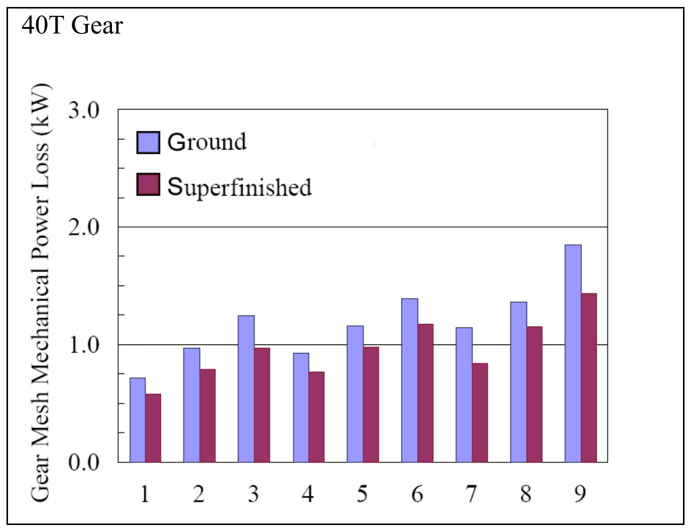


Figure 12. The effect of superfinishing on mechanical gear mesh power losses for 40T Gear; lubricant A at 110 °C.

Two other lubricants (Lubricant B and Lubricant C) were tested with the same gears to quantify their impact on gear efficiency. The viscosity-temperature characteristics shown in Figure 13 display the relative viscosities of lubricants A, B, and C, with lubricant B being the most viscous.

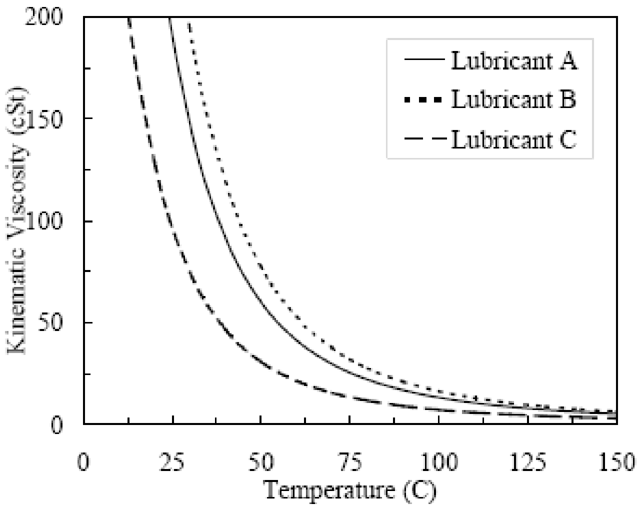


Figure 13. Viscosity-temperature characteristics of lubricants used in this study.

The salient features in Figure 13 are the significantly thinner oil film developed with Lubricant C (due to its lower viscosity) and the effect of superfinishing on the λ ratio. A less viscous lubricant is thought to result in lower rolling losses due to reduced shear resistance, but at the expense of higher sliding losses due to increased asperity contact. Consequently, reducing surface roughness through superfinishing permits the use of a less viscous lubricant while maintaining the same λ ratio. Extending this rationale to the gear-mesh mechanical-power loss measurements of Figures 14 - 17, several conclusions can be drawn:

1. Ground surfaces performed best with Lubricant B because this lubricant resulted in the largest film thickness. Since λ ratio for these gears was consistently less than unity, the reduction in sliding power loss was more significant than the increase in rolling power loss.
2. Superfinished 23 tooth gears performed best with Lubricant C because rolling loss was minimized, but sliding loss was not significantly affected since the λ ratio remained near unity. This observation is further supported by the increasing margin of improvement over the more viscous lubricants as speed increased. Hence, the lower viscosity of Lubricant C results in power-loss improvements if the λ ratio is near unity and pitch-line velocity is sufficiently high.
3. Superfinished 40 tooth gear power-loss trends cannot be explained by the λ ratio alone. Since sliding velocities are significantly lower than 23 tooth gears, the influence of friction coefficient, and therefore sliding power loss, is less significant.

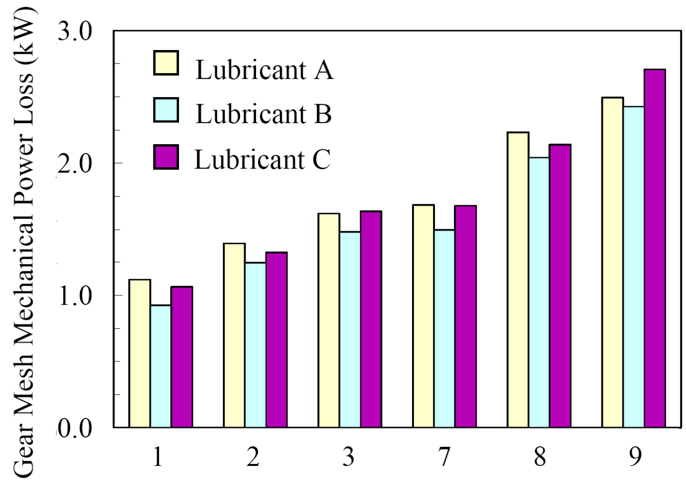


Figure 14. Effect of lubricant type on gear mesh mechanical power losses – 23T ground.

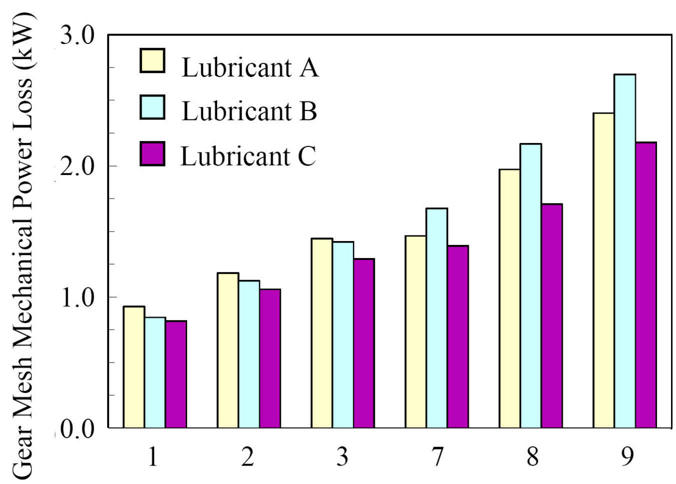


Figure 15. Effect of lubricant type on gear mesh mechanical power losses – 23T superfinished.

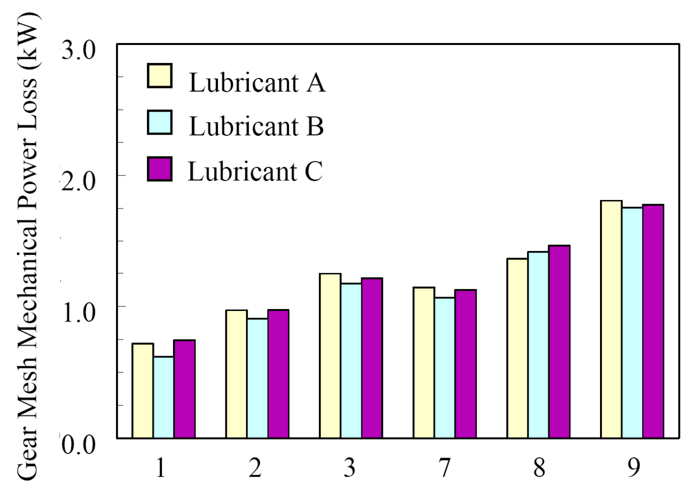


Figure 16. Effect of lubricant type on gear mesh mechanical power losses – 40T ground.

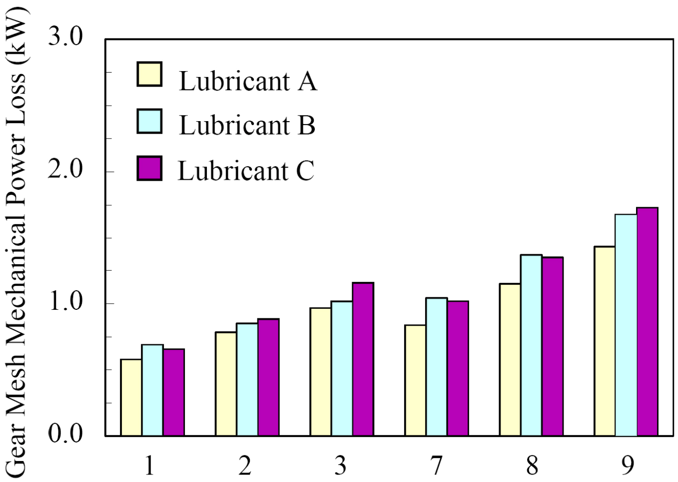


Figure 17. Effect of lubricant type on gear mesh mechanical power losses – 40T superfinished.

VII. Frictional Reduction of NASCAR Transmission[xix]

An independent testing facility conducted this test using a dynamometer to measure the effect of superfinishing on horsepower. A pair of T101 transmissions (Nextel Cup Auto Racing) was used as the test articles. Third gear was selected for the comparison tests. One transmission had standard ground gears while the other had superfinished components. The tests were carried out in sequence on the same dynamometer test stand with the same engine. During each test, the engine and transmission were preheated by idling until the transmission oil (Unocal 90W) reached 82 °C. The engine was then cooled back down to normal operating temperature before running with the applied load to measure horsepower losses through the transmission. Driveshaft angles were monitored to ensure a consistent angle for all runs. Figure 18 shows that approximately 1.0% horsepower was recovered with superfinished transmission components.

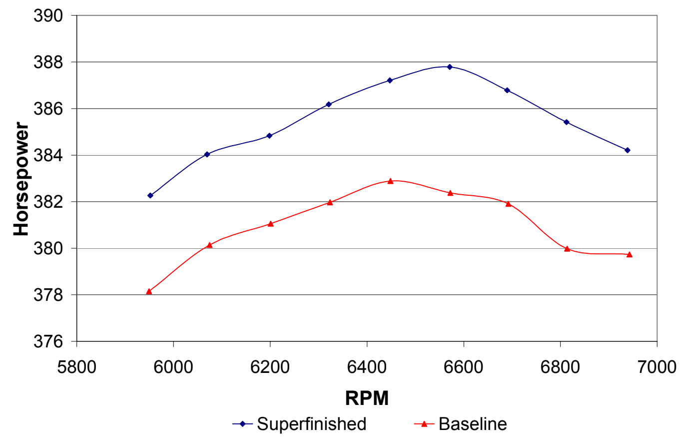


Figure 18. T-101 horsepower output of a superfinished versus standard ground transmission in third gear measured on a chassis dynamometer.

VIII. Increased Horsepower at NASCAR Rear Differential[xx]

A paper presented at the 2003 SAE Performance Racing Industry (PRI) Conference and later published by Circle Track Magazine documented the increased efficiency of a high performance differential containing a superfinished ring and pinion gearset. The test specimens used were a standard quick-change differential with standard bearings and seals, and a

	Standard Quick-Change Bearings & Seals					**Superfinished Gears with Low-Friction Bearings and Seals**				
	Run #1	Run #2	Run #3	Run #4	Run #5	Run #1	Run #2	Run #3	Run #4	Run #5
Max. HP	286.5	284.4	282.3	285.3	280.9	296.8	297.16	296.80	299.80	300.10
Coast @ 137 km/h	N/A	N/A	17.5	16.5	16.1	9.60	8.56	8.13	8.00	7.70
Coast @ 160 km/h	N/A	N/A	24.0	25.5	22.3	13.40	11.66	10.97	10.67	10.40
Average:										
Maximum HP			**283.88**					**298.13**		
137 km/h Coast			**16.70**					**8.40**		
160 km/h Coast			**23.93**					**11.42**		
Gains:										
Maximum HP		**+14.25**	= 5% gain in HP							
137 km/h Coast-down		**-8.30**	= 50% reduction							
160 km/h Coast-down		**-12.51**	= 52% reduction							

Table 7. Results of chassis dynamometer testing of standard and superfinished quick-change differentials in a NASCAR late model car.

quick-change differential with low friction bearings, seals, and superfinished ring and pinion gears known as a "Tiger" in the racing industry. This study used a Dynojet Model 248 chassis dynamometer (as used to monitor NASCAR Nextel Cup cars) output of the engine on run-up in fourth gear and then the amount of resistance at two speed intervals, 137 and 160 km/h while the car was coasting. The run was stopped after coasting down to 137 km/h. The results showed an average gain of 14.25 horsepower and a reduction in parasitic friction losses through the rear end by 50 percent at 137 km/h and 52 percent at 160 km/h. These test results are shown in Table 7.

IX. Efficiency Gain & Temperature Reduction of Rear Axle Ring & Pinion[xxi]

Ford tested the effect of superfinishing rear axle ring and pinion gears on overall fuel efficiency. Results are published in the *Handbook of Lubrication and Tribology volume 1*. A rear axle ring and pinion gearset was superfinished to a 0.07 µm R_a. Chassis roll dynamometer tests under metro/highway cycles were conducted. Figure 19 compares axle efficiency between the superfinished processed gears and production gears at 1000 rpm.

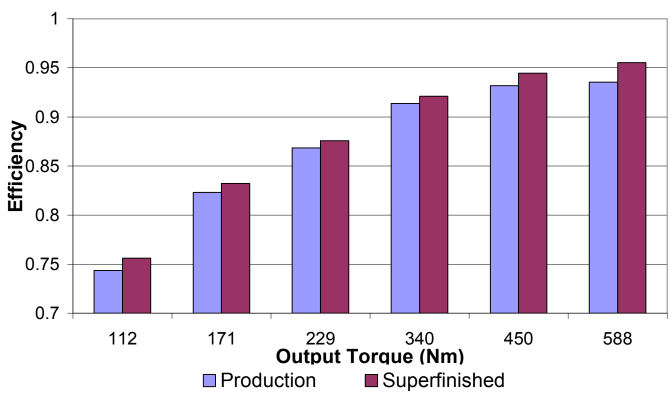


Figure 19. Axle efficiency of the superfinished ring and pinion.

The superfinished ring and pinion showed improved efficiency that improved fuel economy by about 0.5%. The operating temperature of the rear axle was significantly reduced with superfinished gears. See U.S. patent application 2005/0202921 for more details.[xxii]

X. Friction Reduction of Automotive Valvetrains[xxiii]

Only 6 to 10% of an engine's total frictional loss occurs in the valvetrain. Even though this is a small number, it can be significantly and easily reduced by superfinishing the components. Virtually all major racing teams, including those in NASCAR and Formula 1, use superfinishing on their entire valvetrain (camshaft and lifters). Unfortunately, due to the competitive nature of the sport, no published data was found and a NASCAR Late Model with a 5.7-liter engine running a two barrel carburetor. For this test, the dynamometer was used to measure horsepower as engine speed climbed and during the coast-down. The goal was to measure maximum horsepower documenting fuel efficiency gains. It can be said with confidence, however, that many racing teams are superfinishing their valvetrain components to <0.025 µm R_a. The composite roughness of their mating components is therefore <0.035 µm R_a.

Ford Research Laboratory conducted a study to determine the effect of superfinishing valvetrain components on friction reduction. The valvetrain components (cams and tappets) are made of nitrided steel. One objective of the test was to evaluate the effect of superfinishing on frictional torque compared with a production tappet insert. The production tappet insert had an 8-µm manganese phosphate coating to facilitate break-in. The superfinished tappet insert had a surface finish of approximately 0.06 µm R_a. The researchers compared production and superfinished valvetrain components with regular SAE 5W-30 and a special version of SAE 5W-30 formulated by adding a friction modifier, MoDTC. A new broken-in (i.e., not superfinished) cam lobe having a nominal 0.55 µm R_a was used for each tappet insert material. It should be noted that between the two sets of experiments using the different lubricants, some modifications were made to the apparatus to prevent oil leakage and a new higher-sensitivity torque meter was used. Therefore the frictional torque measurements from the two experiments may not be directly comparable.

The test rig ran at speeds ranging from 400 to 1,600 rpm for 50 hours. Frictional torque was recorded for both superfinished and production tappet inserts using both a modified low-friction lubricant and a non-modified lubricant. See Figure 20.

Using SAE 5W-30 without friction modification, the superfinished valvetrain showed a significant decrease in frictional torque compared to the standard production insert. Interestingly, when using the special friction-modified SAE 5W-30, the standard production insert performed as well as the superfinished. As stated, however, the frictional torques of the two experiments cannot be compared. Nevertheless, it is important to note that optimum performance benefits are expected when both mating surfaces (cam and insert) are superfinished to a <0.025 µm R_a. Whereas racing teams are superfinishing their valvetrain components to <0.035 µm composite R_a, the composite R_a in this study was approximately 0.55 µm. It would be interesting to repeat this testing with both the cam and the inserts superfinished to a <0.025 µm R_a.

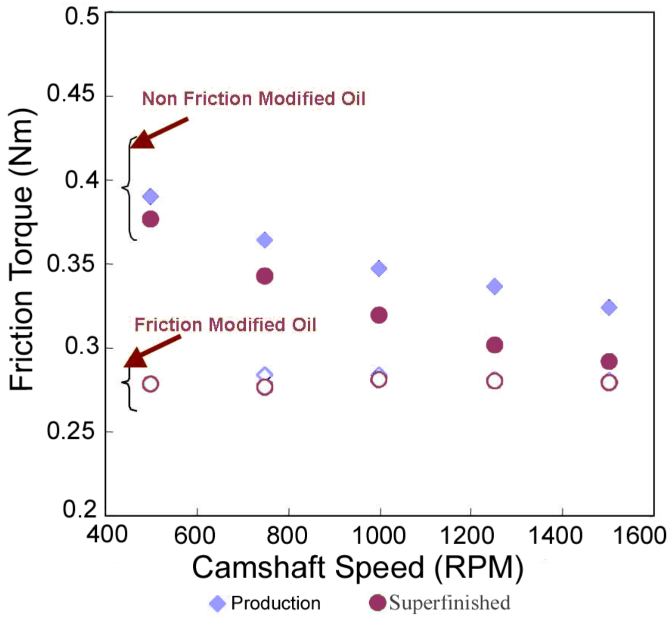


Figure 20. – Friction reduction achieved with superfinished tappets.

FIELD TESTING

I. Temperature Reduction of a Truck Rear Axle

A leading truck manufacturer conducted this experiment to determine the effect of superfinishing on rear-axle operating temperature in their full-size pickup truck while towing a 4,091-kg trailer in hot weather and hilly terrain. The testing was conducted with the vehicles tuned to the 2008 power output. Their current production vehicles are tuned for 514 Nm, the 2008 model will produce 549 Nm. The vehicles used for this testing were tuned to produce 549 Nm torque.

A high-speed tow test was used to determine the effect of high speeds and minor elevation changes. The loop was 151 km in length with a 228 m change in elevation. Test speed was dependent on wind conditions and ranged from 96 to 120 km/h. A 4086-kg trailer with a large frontal area was used. The test truck was equipped with sensors for axle sump temperature, ambient air temperature, axle torque of each rear wheel, transmission temperature, and vehicle speed.

The superfinished axle ran 32° C cooler than the standard axle.

Hill test: The test route consisted of steep grades and high winds. The test route rose 853 meters in 19 km and quickly produced extremely high axle temperatures. Once over the peak, the truck was driven 8 km to a turn-around area. The return run also generated high temperatures due to the axle temperature generated on the climb. Test speed was limited to 64 km/h by the vehicle "failsafe" once engine temperature reached three-quarters hot. A 4086-kg trailer with a large frontal area was used. The test truck was equipped with sensors for axle sump temperature, ambient air temperature, axle torque of each rear wheel, transmission temperature, and vehicle speed. Test conditions for hill testing were not consistent due to changes in wind conditions and when the vehicle would enter failsafe mode. The variable test conditions made it difficult to produce repeatable results.

The superfinished axle ran 53° C cooler than the standard axle.

II. Temperature Reduction of S-76 Helicopter Transmission[xxiv]

Sikorsky performed this test to determine how superfinishing affects the temperature of the S-76 helicopter drive train. Acceptance Test Procedure (ATP) was employed to simulate typical torque loading experienced by the main gearbox, measure oil-out temperature at various torque loadings, and measure vibration levels (noise levels). The surface roughness data of the ground gears and superfinished gears are given in Table 8 and Table 9, respectively. The gearbox passed the 200-hour endurance test and has become flight operational.

	Bevel Pinions & Gears	Spur Pinions	Bull Gears
	µm	µm	µm
R_a	0.330 to 0.457	0.0406 to 0.432	0.033 to 0.432
R_z	2.032 to 2.921	2.463 to 2.565	2.108 to 2.870
R_{max}	2.769 to 4.242	3.023 to 3.277	3.378 to 3.581

Table 8. Roughness parameters for ground aerospace gears.

	Bevel Pinions and Gears	Spur Pinions	Bull Gears
	µm	µm	µm
R_a	≈ 0.089	≈ 0.076	≈ 0.076
R_z	≈ 0.533	≈ 0.584	≈ 0.559
R_{max}	≈ 0.759	≈ 0.739	≈ 0.658

Table 9. Roughness Parameters for superfinished aerospace gears.

The superfinished transmission components showed the following performance improvements: a 2.8° C reduction of the oil-out temperature, a 3.7 dB reduction at the second stage bevel gears, and a 7.0 dB reduction of the first harmonic of the bull gear.

CONCLUSIONS

1. Extensive laboratory and field testing have documented that superfinished gears and bearings have significantly reduced friction, temperature and wear.
2. The superfinished surface is the ideal surface for low-viscosity lubricants since peak asperities are absent, resulting in a higher λ ratio.
3. The aerospace, high-performance-racing, wind-turbine and heavy-axle-vehicle industries are making use of this unique surface to reduce friction, temperature, power loss and wear with the accompanying gains in fuel efficiency.
4. Lubricant formulations can be simplified for gears and/or bearings that have their working surfaces superfinished. Superfinished surfaces operate at lower temperatures and are much less prone to micropit, scuff or wear. Therefore, fewer and/or lower concentrations of detergents, friction modifiers, antioxidants, anti-wear, extreme-pressure additives, etc. are required. These lubricants will be more environmentally friendly.

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