ZPlus, LLC
Burlington, NC & Corona, CA
www.zddplus.com

ZPlus™ Tech Brief #11

Internal Combustion Engine Lubrication

Lubrication in an Ideally Designed Engine

It is a matter of fact that using PCEO (Passenger Car Engine Oil) to lubricate all of the different parts of an engine is a necessary compromise. In an ideally designed engine, each part requiring lubrication would be given the exact lubricant that best suited the nature of the friction being minimized. The basic principle of lubrication is to keep two bearing surfaces that move relative to each other separated, to minimize wear. Ideally, this would be accomplished in all IC (Internal Combustion) engine bearings with an oil film thick enough to eliminate any contact, but this is not always possible. In an actual engine, the lubricating oil film thickness will range from a continuous film of oil in *Hydrodynamic Lubrication*, (as in crankshaft bearings), to a trapped and thinner film of oil in EHD (*Elasto-HydroDynamic) Lubrication* (found in wrist pins and some high-performance engine crankpin bearings), with actual rubbing contact through the oil film found in *Boundary Lubrication* which happens in some engines with flat lifters. Although in reality these three lubrication modes are part of a continuous range of oil film thickness, they each place different demands on the oil, so we will examine them independently.

Figure 1 gives average loading values for different engine bearing systems and the mode of lubrication they employ. To better understand the different modes of lubrication, we will examine the relationship between film thickness, oil viscosity, load, and speed of movement of the frictional interface.

System	Load in psi	Mode of Lubrication
Crankshaft Bearing	1700	Hydrodynamic
Crankpin Bearing	3300	Hydrodynamic
Wrist Pin	5000	Sliding Hydrodynamic
Cam/Lifter Contact	>150,000	EHD/Boundary

figure 1 - IC Engine Bearing Systems and Loads[1]

The complex relationships between these factors are illustrated in a function called the *Stribeck Curve,* shown in purple on the graph in figure 2. Named after German scientist Richard Stribeck (1861-1950), this curve plots the relationship between the three major factors which determine oil film thickness in bearing systems. Overlaid on this curve is the relative *Coefficient of Friction* curve in blue and relative *Wear* in green.

The horizontal x-axis in figure 2 is a value which increases with rotational speed or oil viscosity, and decreases with an increase in load. As the rotational speed of the shaft increases or the viscosity of the oil is increased, the resulting oil film thickness increases along the purple curve to a higher value. If the load increases, the film thickness decreases along the purple curve to a lower value. It is important to note the system oil pressure is not included in this chart, as the system oil pressure only needs to be adequate enough to replace oil that leaks out of bearing clearances. Pressure above this minimum value does not affect the oil film thickness in a hydrodynamic bearing, as we explain in the hydrodynamic lubrication section of this paper. This leakage is affected by oil viscosity, bearing clearance and pressure.

In figure 2, the oil film thickness curve x-axis is separated into three regions: *Hydrodynamic*, *Mixed*, and *Boundary*. In hydrodynamic lubrication, the two bearing surfaces are completely separated by an oil film. In the mixed region, the oil film thickness drops into the range of the roughness of the bearing surfaces. The peaks of the surface which give it its roughness are called *"asperities."* As you move towards the left in this mixed region, the asperities begin to contact. The left hand region is boundary lubrication, in which the interaction of the asperities between the two bearing surfaces is increasingly the cause of friction. You will notice in figure 2 there are dashed lines superimposed on the left side of

[1] Beardmore, Roy, http://www.roymech.co.uk/Useful_Tables/Tribology/Liquid_Lubrication.htm, 2004

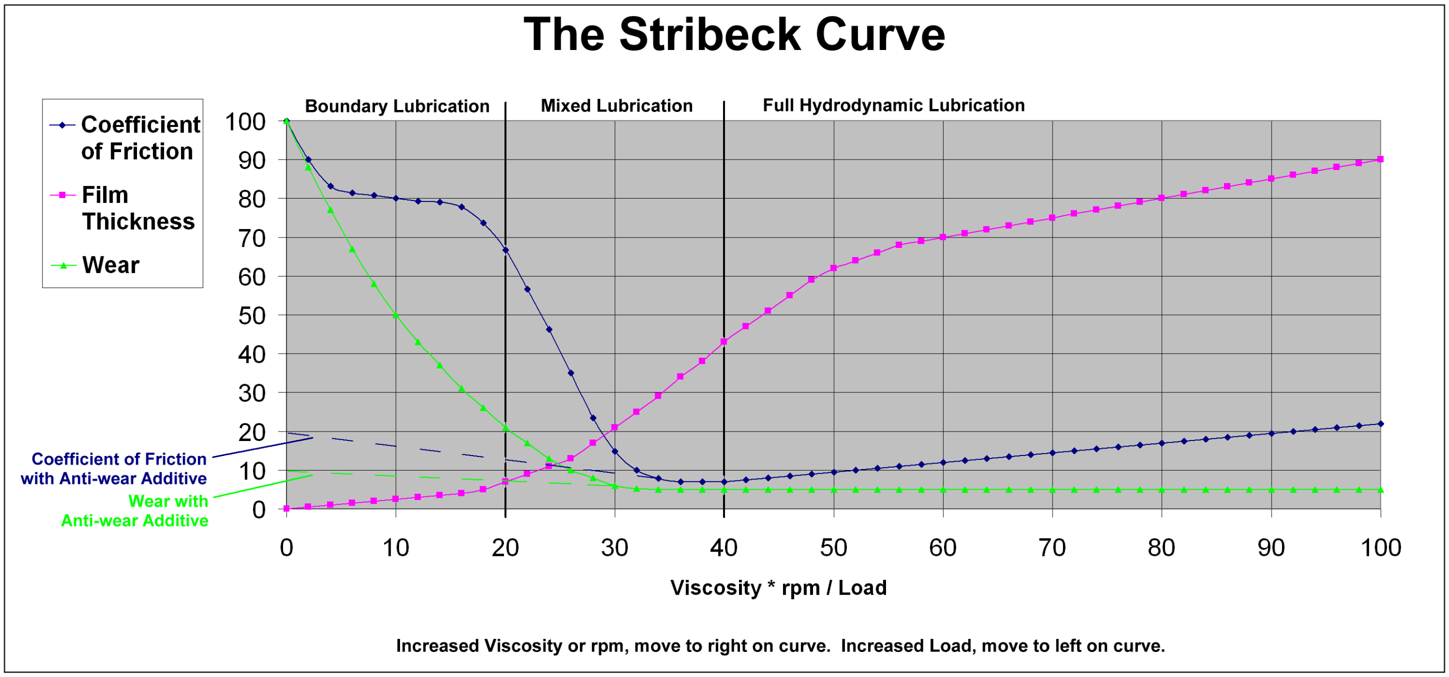

figure 2 - The Stribeck Curve (Plus Coefficient of Friction and Wear Curves)

the graph. They represent the decrease in both coefficient of friction and wear in the presence of the proper anti-wear additive such as ZDDP. Without the anti-wear additive, a decrease in either rotational velocity or viscosity, or an increase in the load ultimately results in destruction of the journal and bearing.

Hydrodynamic Lubrication

Definition: ***"A system of lubrication in which the shape and relative motion of the sliding surfaces causes the formation of a fluid film having sufficient pressure to separate the surfaces."*** [2]

The crankshaft, crankpin, camshaft bearings, and in some engines rocker shaft bearings, utilize a plain bearing insert and journal which are kept separated by an oil film operating in a hydrodynamic lubrication mode. Contrary to the popular misconception, the journals in these bearings are not kept from rubbing on the bearing inserts by the engine oil system pressure. Using the crankshaft bearings as an example, the oil is introduced into the bearing journal clearance only at a hole in the top bearing insert, although in many engines there is also a groove to distribute the oil half-way around the journal. In this bearing, the oil system pressure is actually trying to push the crankshaft away from the oil hole toward the lower bearing half. If it were not for the presence of hydrodynamic lubrication, the crankshaft would be pushed down into contact with the lower bearing insert. Since crankshaft bearings are not sealed, oil leaks out the side clearance of the bearings, and the engine oiling system merely keeps an adequate supply of oil in the bearing. The hydrodynamic lubrication does the actual work of keeping the bearing and journal separated.

In order to illustrate the principle of hydrodynamic lubrication, consider the bearing and journal system of an IC engine crankshaft journal and bearing, illustrated in figures 3, 4, and 5. The size differential between the bearing shell and journal has been exaggerated to make the principle clear. For the purposes of the example, we are loading the crankshaft vertically opposed to the oil hole. Our initial conditions are that of an engine which has been recently built, but has no oil between the journal and bearing surface.

(Our example starts with dry bearings in order to make the point more clearly but in practice, all correctly assembled engines have the bearings pre-lubricated at assembly, and an engine which has been run will have a thin film of oil in the bearings. In both of these conditions, the bearings will operate in a mixed hydrodynamic/boundary lubrication mode until the oil pressure rises and re-establishes a continuous supply of oil, enabling full hydrodynamic lubrication.)

With the crankshaft stationary, as illustrated in figure 3, the first application of pressure on a cylinder power stroke forces the journal downwards into contact with the bearing. As the crankshaft attempts to spin in the absence of oil, the journal attempts to climb the left side of the bearing due to the mutual friction between the two, as shown in figure 4. Of course

[2] Noria Corporation, "Dictionary of Tribological Terms," 1328 E. 43rd Court, Tulsa, OK 74105

without oil, the journal and bearing will be in rubbing contact, quickly causing damage to both.

To the system in figure 4, we then introduce a supply of oil. The oil characteristics in play in hydrodynamic lubrication are *cohesion* (how well the molecules adhere to each other), *adhesion* (how well molecules stick to other molecules), and *viscosity* (an oil's resistance to flow).

If an oil has poor cohesion, it would have poor film strength and the film could be easily separated. If the oil had absolutely no adhesion to the crankshaft journal, then an oil film would not form on the crankshaft journal. The journal would then be forced under load to the bottom bearing insert and attempt to climb it just as if there was no oil. This pinpoints oil adhesion as a fundamental part of hydrodynamic oil lubrication. Viscosity is the dominant factor in an otherwise suitable oil which determines the resulting film thickness.

As shown in figure 5, the adhesion of the oil to the spinning journal will cause the oil to be continually dragged into the point of contact between the journal and the bearing insert. The oil is entering the contact point from the left in our example, so the oil film thickness will be greater on the left side of the contact than on the right side. This gives the oil film a shape called a "*hydrodynamic wedge.*" With the thickest layer on the left, the pressure in the wedge pushes the journal up and to the right of the load center line.

As the hydrodynamic wedge pushes the journal away from the bearing insert, the pressure in the wedge will drop due to the increased clearance between the journal and bearing insert. At the point where the upward force of the wedge equals the load, the hydrodynamic lubrication is in equilibrium. In this equilibrium, the oil pressure in the hydrodynamic wedge part of the oil film can be many times the engine oiling system pressure.

Since the journal in this system is the pumping component, the pressure in the hydrodynamic wedge is proportional to the speed of rotation. This means the slower a journal rotates, the lower the hydrodynamic pressure becomes, resulting in a thinner oil film, as shown in the Stribeck Curve of figure 2. This is one reason why "*lugging*" an engine at very low rpms and high load can cause bearing damage. In a lugging condition, the oil film being generated by the hydrodynamic action can be too thin to sustain the load, and may allow journal-to-bearing contact.

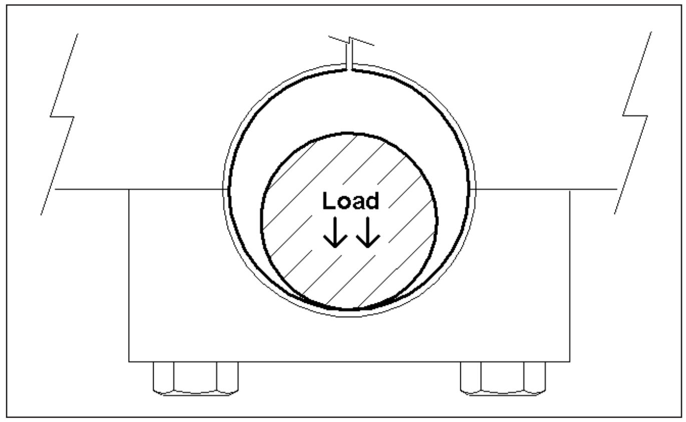

figure 3 - Crankshaft Stationary

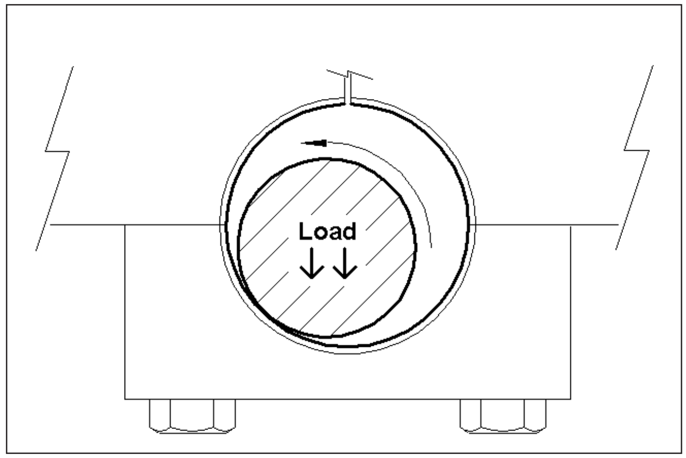

figure 4 - Crankshaft Spinning Without Oil

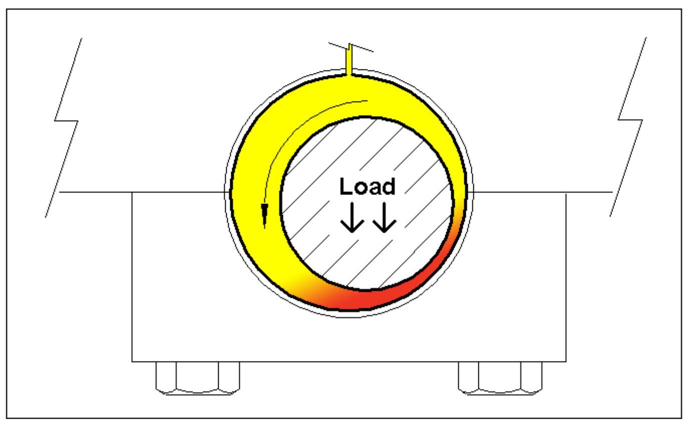

figure 5 - Crankshaft Spinning with Oil

As is the case with all gas or liquid compression systems, energy is used compressing the oil in the wedge, and the oil is heated in the process. This energy is supplied by the engine, and the heat generated in the process adds to the heat in the oil from oil shear. Both must be dissipated in the sump.

The correct viscosity oil is a critical factor to maintaining adequate hydrodynamic lubrication. The factors determining the choice of the correct oil are the maximum load, bearing width, and rpm range in which the bearing will operate. High viscosity oil will allow for high loading at a lower rpm range. Low viscosity oil will allow for higher rpm operation with low shear heating and less power loss.

Factors Which Determine Hydrodynamic Oil Film Thickness

Specified Oil Viscosity

Hydrodynamic oil film thickness is directly proportional to the viscosity of the oil, which in turn is related to the cohesion of oil molecules to each other in combination with the average molecule size. The easiest way to gain load safety margin in the form of additional oil film thickness, is by increasing the viscosity of the oil. The correct choice of oil viscosity is determined by the design load, bearing clearance, and rpm range of the engine.

In general, the higher the load, the higher the appropriate viscosity. The larger the bearing clearance, the higher the appropriate viscosity. The higher the average rpm range, the lower the appropriate viscosity. Low viscosity oil with low shearing losses will increase energy efficiency and prolong oil life. More viscous oil will allow for heavier loading or lower rpm use, but suffer excess oil shearing under high rpm duty.

Engine manufacturers have suggested viscosities which are designed to take these factors into account, as well as factoring in seasonal variations in operating temperatures. Racing oils may seem to violate these constraints if you judge them solely by their viscosity ratings, which can be as high as SAE 70 weight. Racing engines cannot escape from the laws of physics, and these highly viscous oils will suffer from extreme shear, heating, and degradation when the engine is operated at high rpms. This consequence is unimportant in this application, because the oil is often discarded after mere hours of use. Street vehicles operate with very different oil life requirements.

Temperature of the Oil

Oil temperature directly affects the viscosity of the oil, as discussed in ZPlus™ Tech Brief #10. This lowering of viscosity with temperature increase translates directly into a reduction of film thickness. This characteristic is fortunate for a commuter on a cold morning, considering how little oil pressure is available in a cold engine upon start-up. When cold, residual oil in the bearing will have a much greater viscosity than it will at the engine's full operating temperature, which lowers the chance of bearing damage. A hot re-start of an engine poses more of a risk of bearing damage, due to the fact the hot oil film will be thinner than a cold oil film, and may not adequately separate the journal and bearing until proper oil system pressure is re-established.

Rotational Speed of the Journal

As the adhesion of the oil to the spinning journal drags the oil molecules into the hydrodynamic wedge, the cohesion of the molecules will cause a buildup of pressure in the wedge. This increased pressure will directly translate into increased film thickness.

Peak Pressure Due to Loading of the Bearing (Not Oil System Pressure)

A doubling of load will result in a halving of film thickness, if you do not factor in the 'α' or pressure dependent viscosity of the oil. Practically speaking, the viscosity of most PCEO will double from zero pressure to 10,000 psi.[3] Peak pressures in the crank bearings of a normal IC motor are less than 2000 psi, so the 'α' of an average oil will cause a useful but minimal 10% increase in film thickness, relative to that of its uncompressed viscosity.

Polarity of the oil

The polarity of the oil directly affects its adhesion to the parts it is supposed to lubricate. Polar substances can be mutually attractive, which enhances the adhesion of the two.

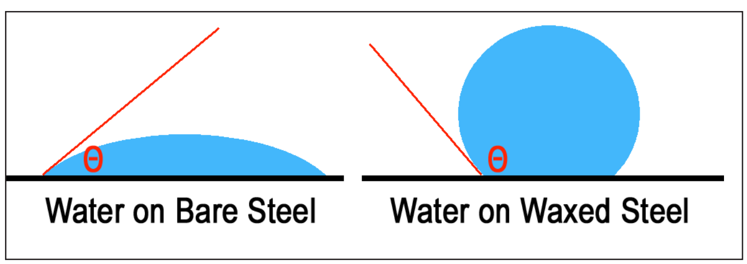

figure 6 - Polarity Affecting Adhesion

As shown in figure 6, when a drop of water is placed in contact with surfaces of different polarities, the resulting interaction of the two determines the degree of wetting.

[3] Totten, G.R., "Fuels and Lubricants Handbook: Technology, Properties, Performance and Testing," *ASTM Manual Series, ASTM International, West Conshocken, PA,* 2003, pg. 189

The greater the adhesion, the more the liquid is attracted to the surface, and the more it will spread out on the surface. The amount of adhesion can be determined by the resulting angle at the junction of the drop as it wets the surface. This wetting can be expressed as a contact angle Θ, with the greatest adhesion giving the lowest angle. This contact angle is also determined by the surface finish, which is discussed in the next section.

Highly polar oil displays greater adhesion to both bearing and journal, resulting in greater boundary layer thickness and higher ultimate film rupture strength, all other characteristics being equal. The cam, lifter, crankshaft, and bearing insert are made from highly polar metals. PCEO polarity is dependent on the polarity of both the base stock and additive package. Mineral-based Group I, II and III base stocks are moderately polar, with polarity decreasing in general as additional refining is added. There are some base stocks, such as Group IV PAO, which are very non-polar. Some Group V oil stocks, like PE (polyol ester) and PAG (polyalkaline glycol), are highly polar *(see ZPlus™ Tech Brief #10 for more oil information)*.

Without polar additives, Group IV PAO base stock will exhibit a lower degree of wetting of engine parts compared to other base oils. This is one of the reasons that most current true synthetic Group IV based PCEO is formulated with the addition of another more polar base stock, such as a Group V polyol ester. Of course, the addition of a polar additive such as a detergent or an anti wear additive such as ZDDP will also increase the overall polarity of the oil. Without the addition of these polar substances, a PAO oil film can be more easily displaced due to the lower adhesion and reduced hydrodynamic wedge development. This can result in metal-to-metal contact between the bearing journal and insert. A highly polar liquid will display more thorough wetting of the bearing interface (high adhesion) and displace less under load, other factors being equal.

Surface Finish

A perfectly smooth bearing surface, although impossible to attain, would have no reserve oil capacity, as well as minimal mechanical oil adhesion. The microscopic surface imperfections of asperities and small pits greatly increase the surface area of the journal, and allow for greater oil film adhesion. The limiting factor on the height of these surface imperfections is the thickness of the oil film itself. If the asperities are taller (above the average surface) than the oil film is thick, then contact will be made between the rotating and stationary asperities, causing wear.

Figure 7 details the adhesion of oil to steel of increasing roughness as determined by oil adhesion experiments using a DCA (Dynamic Contact Angle) analyzer.

Surface Finish (RMS)	Smooth (<10nm)	Low (20nm)	High (50nm)
Contact Angle	33°	28°	22°
Recovered Mass (mg)	5	8	10

figure 7 - Surface Finish vs. Oil Adhesion [4]

The DCA used in this University of California Santa Barbara study measures the adhesion between oil and metal strips of varying roughness, by measuring the contact angle or amount of "*wetting*" which occurs between the two. The "*recovered mass*" in the chart is the amount of oil which adhered to the test metal strip due to the combination of polarity and the surface roughness. Since the roughness was the parameter that was varied in the test, the difference between the results is due solely to the surface finish. The rougher the surface finish, the more oil was retained. Although increasing roughness will increase oil adhesion, there is a practical limit. At some amount of roughness, the height of the asperities will begin to approach the thickness of the oil film, reducing the effective oil film thickness. For each bearing system, there is an optimum surface finish which will have maximum oil adhesion without causing contact between the two bearing surface asperities under load.

Engine bearing manufacturers have long known there is an optimum surface finish for the bearing journals which encourages the formation and adhesion of an oil film. The importance of the surface finish is demonstrated by Clevite recommendations[5] regarding the directionality of the finish left by cylindrical grinding and polishing. The process of grinding or polishing at the microscopic level is one of imbedding a hard chunk of abrasive into the relatively soft metal, and then moving it parallel to the surface to remove or push over the metal. Some metal will be removed, and some flows plastically, leaving a directional *"scaled"* appearing surface aligned in the direction of the finishing rotation. This surface feature can help retain oil by mechanically increasing the adhesion of the oil film to the bearing journal and providing microscopic oil reservoirs. These "*scales*" also can appear as microscopic ramps, perhaps helping hydrodynamically pump oil outward on the journal.

---

[4] Victoria Broje and Arturo A. Keller, "Advanced Oil Spill Recovery in Marine Environments," *Bren School of Environmental Science and Management, University of California, Santa Barbara, 2005*

[5] http://mahleclevite.com/it_crankgrind.asp

For these reasons, the direction of grinding and polishing relative to the direction of bearing rotation during operation is critical to the performance of the bearing/oil system.

Width of the Bearing

In a plain bearing of infinite width, there could be no side-leakage of the high pressure in the hydrodynamic wedge. Since all IC engines have bearings of relatively small width compared to the shaft diameter (typ.<0.5 dia.), and fairly large clearance between the journal and bearing insert relative to the size of the oil molecules, there is considerable side-leakage. This leakage allows the high-pressure oil in the hydrodynamic wedge to leak out along the length of the bearing.

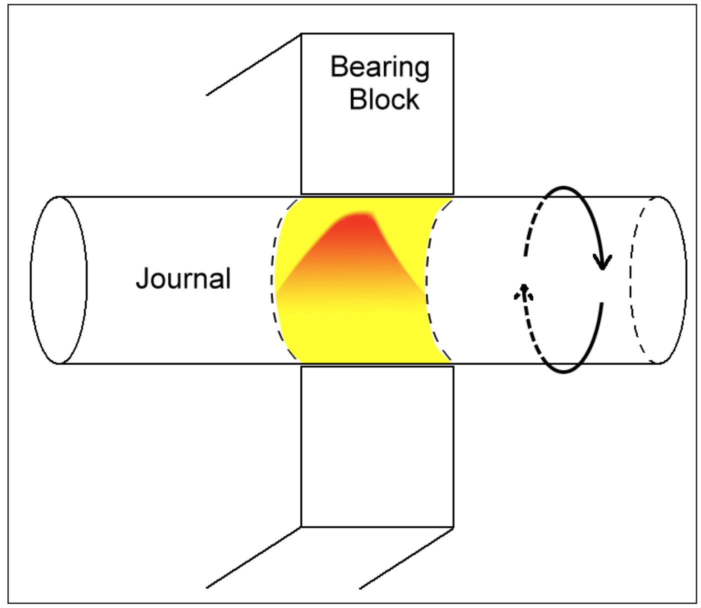

figure 8 - Hydrodynamic Pressure Profile with Side Leakage

As a result of the side leakage, there is a bell-shaped pressure gradient of this oil pressure wedge as viewed perpendicularly to the shaft, shown on figure 8. In this diagram, the red zone of highest pressure is in the center of the bearing, causing a downward pressure gradient as you move to either edge of the bearing. The net result is a reduction in the peak pressure of the bearing surface covered by the wedge. The load-bearing capability of a hydrodynamic wedge is proportional to both bearing width and journal rpm, so a wider bearing will support adequate hydrodynamic lubrication at a lower speed. The correct width for a bearing is one which allows for a hydrodynamic oil film thick enough to support the load with an adequate safety margin, without being so wide as to incur excessive oil shear power loss.

What is the Actual Minimum Film Thickness Needed to Keep the Bearing and Journal Apart?

Figure 9 is a graph plotting measured crankshaft bearing load and minimum oil film thickness as a function of crank angle.[6]
The data in the chart shows graphically how the crankshaft journal does not stay centered in its bearing when loaded.

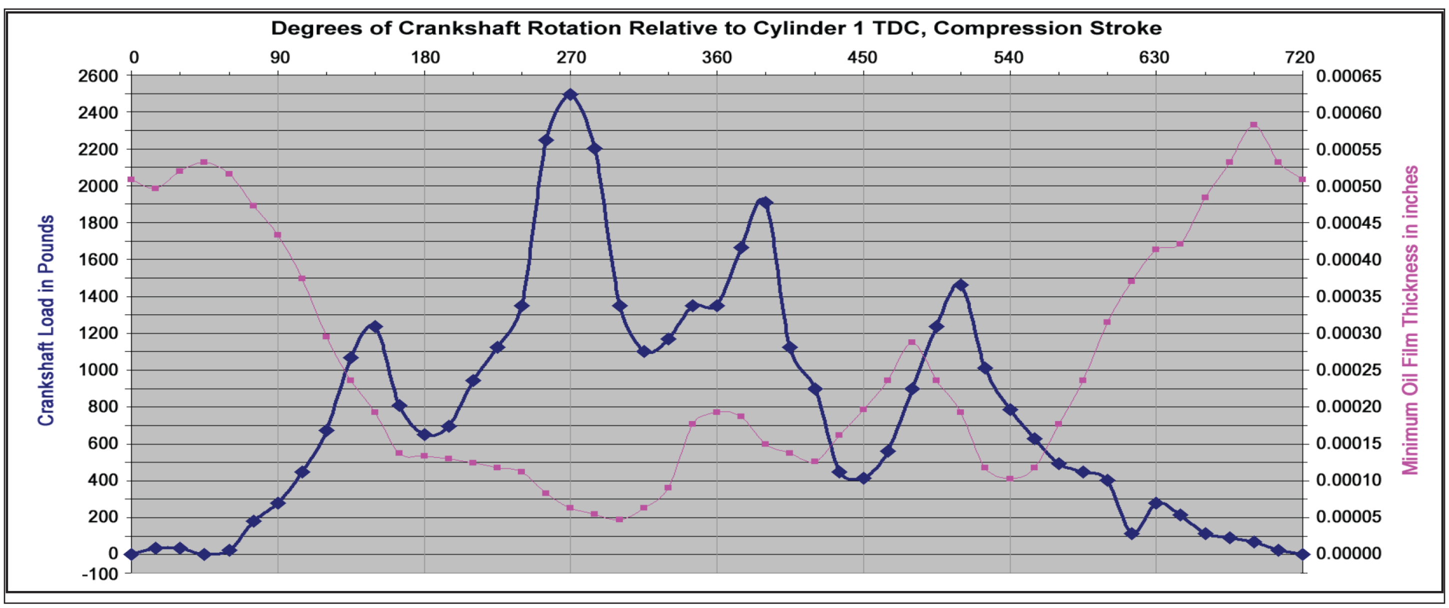

figure 9 - Crankshaft Load and Oil Film Minimum Thickness vs. Crankshaft Angle

[6] Paranjpe, Rohit S., Tseregounis, Spyros I. and Viola, Michael B. (2000), "Comparison between Theoretical Calculations and Oil Film Thickness Measurements Using the Total Capacitance Method for Crankshaft Bearings in a Firing Engine," Tribology Transactions, 43:3, pp. 345-356

Note the chart does not specify at what location that minimum occurs, just the angle of the crankshaft in its rotation relative to TDC (Top Dead Center) when the minimum oil film thickness occurs.

By this we mean the minimum film thickness will, of course, be located near the middle of the bearing cap during the power stroke, and at the top near the oil hole during the intake stroke. But both events cause a relative minima on the graph. The data in the chart was obtained by testing with 5W-20 oil, and results show an average minimum film thickness of 0.000264″. The test engine was a GM 3800 V6, with radial main bearing clearance of 0.0012″. This means in the initial test, due to radial forces in the crankshaft, the actual journal spends most of its time rotating off-center in the bearing housing an average of 0.000936″, which is most of the 0.0012″ clearance! The critical piece of information in this chart is the absolute minimum oil film of 0.000047″, which occurs in this particular bearing position at around 300°. Retesting using 10W-30 oil gave a minimum of 0.000054″, and 20W-50 oil resulted in a minimum film thickness of 0.000070″. This means in this particular engine using 5W-30 oil, if the asperities on either the crankshaft or bearing insert were higher than 0.000047″ total, there could be contact between them at this time. Once the asperities of the journal contact those on the bearing, some will break off and be carried in the oil as debris which may cause additional damage. This is one of the reasons the current high-performance engine design emphasizes the roundness, surface finish, and accuracy of machining of journals and bearings which are supported by hydrodynamic lubrication.

Since the sum of the machining, bearing manufacturing tolerances, and installation tolerances far exceed this minimum oil film thickness, why does a crankshaft bearing not fail immediately upon startup? The answer is the bearing insert surface material is relatively soft and compliant, and when an engine is first fired up, the crankshaft DOES make contact with enough force to literally pound the high spots and misalignments into a close fit with the journal. Once they have been reformed to a height less than the oil film thickness, the crankshaft is fully supported by the hydrodynamic film.
It is obvious that any and all precision employed during assembly benefits bearing life.

Oil Shear

Figure 10 shows the relative velocity at different layers in the oil film. Referring to the illustration, the oil film directly in contact with the bearing insert is adhered to the metal of the bearing and does not move. In a liquid, molecules can be moved relative to each other, so the layer of oil molecules adhering to the stationary layer does have some freedom of movement. The layer of oil at the crankshaft journal is adhered to the journal, and will move at the full rotational speed of the shaft. The intermediate layers of oil, starting with the stationary bearing insert layer, have gradually increasing velocity crossing the thickness of the film towards the crankshaft. These oil molecules, in the act of moving past each other, will adhere to each other then break free. This action of pulling the oil molecules apart is called “*shearing,*” and requires energy proportional to the viscosity of the oil. The constant shearing of the oil around a spinning crankshaft absorbs energy and is called “*viscous drag.*” The energy of the shearing (or the “*shear energy*”) is dissipated as heat in the oil.

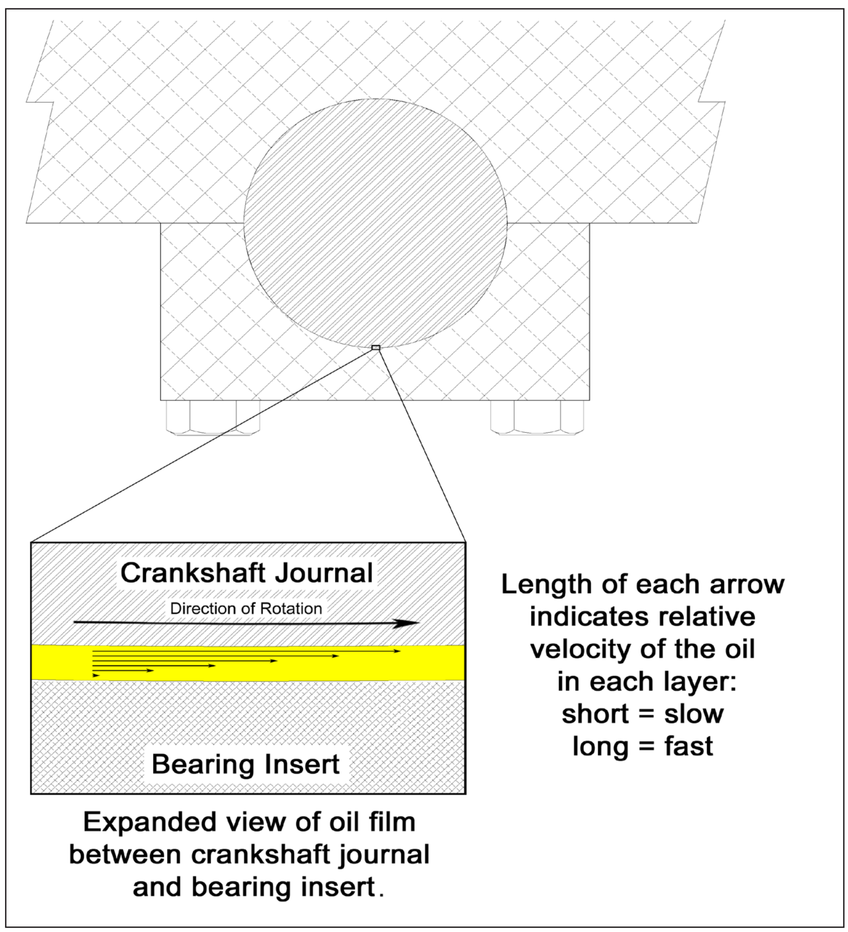

figure 10 - Oil Shear Between Moving Journal and Bearing

The heat caused by shear is a primary factor in the choice of oil viscosity. If too high viscosity oil is chosen for a specific bearing clearance and shaft speed, the oil will be heated by the shearing to a temperature which may cause decomposition of either the base oil or an additive. The longest, highest weight molecules in the oil are subject to the most shear, and it is often the VI (Viscosity Improver) additive molecules which suffer the greatest degradation as a result of shearing. If the VI molecules degrade, the oil loses the effect of the VI additive and will drop in viscosity as well as show a greater change in viscosity with temperature change.

In order to gain an appreciation of how important shear is to efficient engine design, consider that in one paper [7] it was calculated that a standard European 2 liter 4 cylinder engine running 7500 rpm using 10W-30 oil is stated to waste more than 4000 watts (5.4 horsepower) shearing the oil in the bearings alone. The same engine in the same operating conditions using 0W-20 oil would waste 2700 watts (3.9 horsepower).

Summarizing Factors Affecting Hydrodynamic Oil Film Thickness

A hydrodynamic oil film will support a journal until a critical design point of: increasing load, too low rotational speed, or insufficient oil viscosity. At that point, the oil film will be thinner than the asperities are tall. Once the asperities of the journal contact those on the bearing, there is the potential for damage to both surfaces.

Sliding Hydrodynamic Lubrication

The pushrod ends, rocker ball, rocker arm ends and oil pump gears operate in a sliding, relatively lightly-loaded hydrodynamic/hydrostatic lubrication regime. In a manner similar to rotating hydrodynamic lubrication, the film of oil between the parts is pressurized by the actual contact pressure. In this case, the pressurized oil moves from side to side or back and forth in an arc, creating a hydrodynamic wedge of oil which keeps the surfaces separate. As the oil film attempts to squeeze out ahead of the applied pressure, the movement reverses which reverses the direction of the oil wedge, keeping the oil (except for side leakage) in the bearing gap. In these systems, there is an ample supply of oil to replenish the film on the edge of the moving parts. Due to the fact there is relatively little shear in the small movements, these systems would benefit from an oil with high viscosity. In normal gear operation, the lubrication is mixed boundary/elasto-hydrodynamic. In a gear oil pump, one gear is driven and the other is actually driven by the oil film itself and is not coupled to a load other than the compressed oil, resulting in relatively low loading.

The lifter bore, piston rings, skirt and valve stem also operate in a lightly-loaded sliding contact hydrodynamic regime. Unlike the pushrod ends and rocker balls, the movement of these parts relative to the stationary engine block is large. To keep power losses low, these parts would benefit from a low-to-moderate viscosity oil chosen for its temperature stability and low shear loss. Due to the relatively large amount of contamination to which these parts are subject, the oil must have a good amount of chemical stability as well.

EHD (Elasto-HydroDynamic) Lubrication

Definition: ***"The opposing surfaces are separated but there occurs some interaction between the raised solid features called asperities, and there is an elastic deformation on the contacting surface enlarging the load bearing area whereby the viscous resistance of the lubricant becomes capable of supporting the load."***

For roller and ball bearings: ***"In rolling element bearings, the elastic deformation of the bearing (flattening) as it rolls, under load, in the bearing race. This momentary flattening improves the hydrodynamic lubrication properties by converting point or line contact to surface-to-surface contact."*** [8]

EHD lubrication occurs when a volume of oil in a film is trapped by surface characteristics or the velocity of the bearing surfaces. Since it cannot escape, the load pressure of the bearing is transmitted through the trapped film of oil, and can momentarily become great enough to exceed the strength of the base metal, causing deformation of the metal. In other EHD lubrication scenarios, under sufficient pressure, oil with a large 'α' factor will have a viscosity increase and in doing so transfer the load between bearing members without metal-to-metal contact. An important aspect of EHD, as the definition points out, is the pressure which can be transferred through the trapped or thickened lubricant film can be great enough to deform the two metal bearing surfaces. In doing this, a small contact patch will broaden into a larger one, spreading the pressure over a larger area. This lowers the effective pressure per unit area, which can allow the film to carry the load without rupture. This characteristic is valuable in rolling contact bearings, such as ball or roller bearings, where the point or line contact can cause extreme localized pressures and temperatures at the contact point.

[7] Taylor , R.I., "Lubrication, Tribology & Motorsport," *Shell Global Solutions (UK), Cheshire Innovation Park, PO Box 1, Chester, CH1 3SH, UK,* 2002-01-3355, pg. 10

[8] Noria Corporation, "Dictionary of Tribological Terms" 1328 E. 43rd Court Tulsa, OK 74105

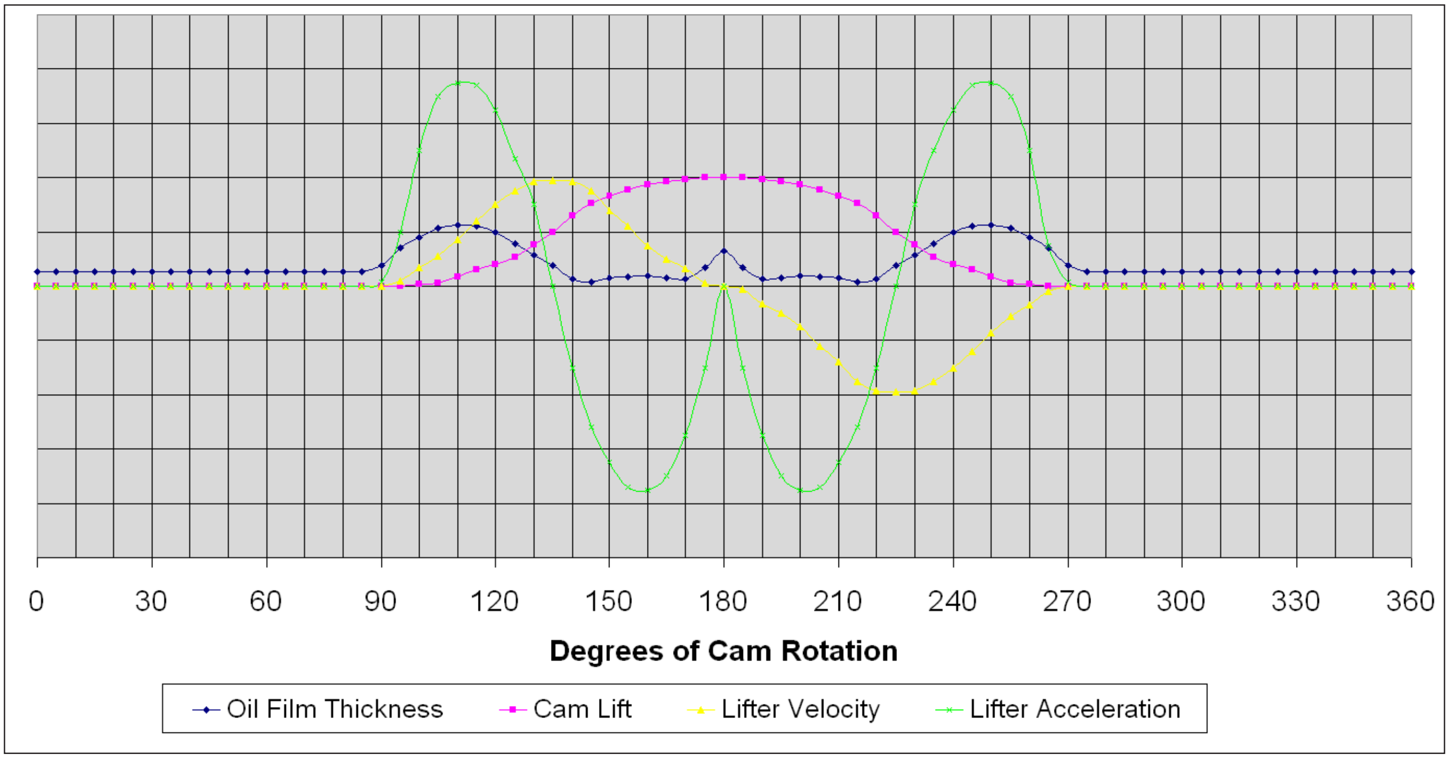

figure 11 - Cam and Lifter Characteristics and Timing

Crank and crankpin bearings usually operate in hydrodynamic lubrication regime where the peak pressures are lower, and cause little viscosity increase.

In some high-performance engines, especially at high rpms, the crankpin bearings can effectively employ EHD lubrication, since the connecting rod is more flexible than the engine casting in the main bearing area. This allows the crankpin to effectively distort the rod bearing material, thereby enlarging the contact area.

The cam/lifter interface, and in certain diesel engines, heavily-loaded wrist pins are the only engine systems where the oil is subjected to sufficient load for the pressure/viscosity factor 'α' to be a major factor in film thickness. Some loads typical of those found in an IC engine are shown in figure 1. Even 5000 psi at a heavily-loaded wrist pin bearing surface is insufficient to radically change the oil viscosity. The cam/lifter interface, which is a sliding point or line contact interface, operates in a mixed elasto-hydrodynamic/boundary lubrication regime. This system would operate best with a very high viscosity oil with a high 'α', or grease, either with the appropriate anti-wear agent. Many engines with low spring pressures or mild cam profiles have low enough lifter foot pressures to employ EHD lubrication alone at all cam angles (*refer to figure 11*). Modern engines with roller lifters employ EHD lubrication at the rolling point of contact between the lifter and camshaft.

Boundary Lubrication

Definition: ***"Form of lubrication between two rubbing surfaces without development of a full-fluid lubricating film. Boundary lubrication can be made more effective by including additives in the lubricating oil that provide a stronger oil film, thus preventing excessive friction and possible scoring."*** [9]

In order to better understand the lubrication modes employed in the cam lobe/lifter contact, examine figure 11, which plots the valve lift, velocity, acceleration and oil film thickness of the cam lobe/lifter interface. The oil film thickness is consistent in the base circle area of rotation, as the lifter slides on a wedge of oil preceding the contact point. This is a fully-flooded hydrodynamic lubrication mode.

As the lifter begins to climb the entrance ramp of the lobe, the wedge shaped gap formed by the spherical lifter foot and cam lobe trap oil. As the follower climbs the lobe, and the spring compresses, the lifter foot pressure increases, and at some point puts the trapped oil film into an EHD lubrication mode. As the lifter foot continues to climb the cam lobe entrance ramp, the increasing spring pressure reduces the oil film thickness, reaching minima at 145° and 215°. The

[9] Noria Corporation, "Dictionary of Tribological Terms" 1328 E. 43rd Court Tulsa, OK 74105

Loughborough University study data concluded the oil film thickness at these minima was less than 2×10^{-7} meters (0.2 microns) or about 0.000008″, which is 8 micro inches.

The study concluded of the cam lobe/lifter contact: "The peak transient contact pressures remain in the gigapascal range."[10] One gigapascal is equal to 145,037.738 pounds per square inch. This contact pressure was calculated for what we would consider to be typical of the valve train in a low-performance engine, with only 135 pounds of valve open lifter foot pressure.

A common classic and high-performance engine found in the U.S. is the Chevrolet 350 engine. We obtained a flat-lifter camshaft with lifters in order to get an idea of the lifter foot pressure which could be found in one of these engines. Using an optical comparator, we examined the foot of the GM lifter and cam lobe. Figure 12 shows a simplified view of the relationship between the two, with both the sphericity of the lifter foot and the taper of the cam lobe exaggerated for clarity. The lobe taper and spherical lifter foot profile, together with the lifter offset from the cam lobe center line, induce a spinning/skidding motion to the lifter. This motion reduces wear by several methods. First, it spreads the contact point over a large circular area on the lifter foot instead of concentrating it on a single patch. It also helps to bring lifter metal with a fresh oil coating into the contact point. This ensures that an effective EHD film is maintained in lightly-loaded conditions, and in heavily-loaded conditions, that fresh anti-wear agent is brought to the contact point.

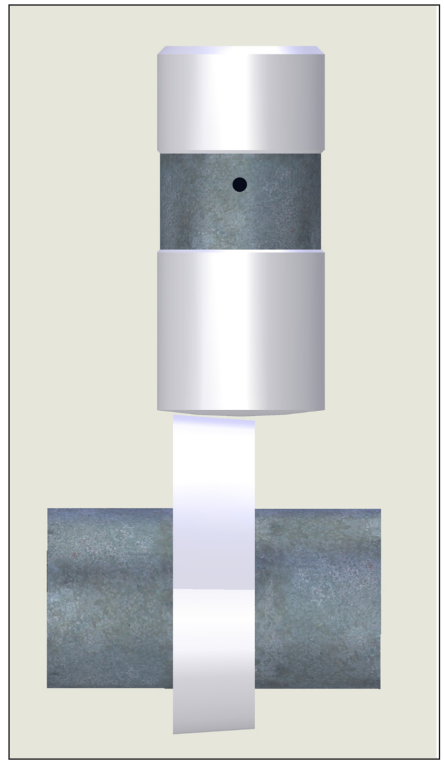
figure 12 - Ideal Cam / Lifter Contact

The geometry of the lifter/cam lobe contact causes a tight mating of the two, which encourages the contact patch to be as broad as possible, as well as encouraging the formation of an EHD oil wedge. The GM lifter foot was ground with a 36″ radius spherical profile, and the cam lobe had a face taper of 0.003″ across the 0.450″ cam lobe width. This means the angle of the cam's lobe surface is 0.387° relative to the axis of the cam, and the spherical foot of the lifter will touch the lobe at a point with a tangent of 0.371°. With little or no pressure, the two meet at a point. But, under the high loading of the valve spring, the resulting plastic deformation of the cam's surface and lifter foot cause the point contact to lengthen into a line or rectangular patch contact. The cam and lifter we obtained are often used with springs which can have as much as 335 pounds of pressure at full lift. With 1.5:1 ratio rocker arms, that results in 500 pounds of lifter foot pressure, or 3.7 times that used in the Loughborough study. We measured a static contact patch of 0.3 mm by 1.3 mm under the 500 pound load. This would indicate the metal of the cam lobe and lifter were each being deformed by several microns, and the resulting equilibrium pressure was in the 200,000-400,000 psi range. The actual pressure will be limited by the amount of deformation of the cam lobe and lifter. This is because in response to pressure the contact patch gets larger, spreading the pressure over a larger area, which in turn lowers the peak pressure per unit area.

Due to the difficulty of actually measuring the area of the contact point as it moves dynamically, as well as the complex deformation of the cam and lifter metal in response to pressure this great, we cannot know the exact pressure at the contact point using our GM parts. We can state with confidence that a 500 pound lifter foot pressure will cause a contact pressure not less than that stated in the Loughborough study using a 135 pound lifter foot load, and possibly up to 3.7 times as much. This would mean the contact pressure is in the range of 145,000-536,000 psi. At these pressures, the deformation of the metal will limit the actual pressure per unit area to a value less than the yield limit of the metals.

We have seen a number of failed lifters and cams, both in OEM engines and in engines employing aftermarket flat cam/lifter systems which have aggressive ramps and high spring pressures. This indicates to us, unlike the low-performance engine used as an example in the Loughborough study, effective EHD films are not being sustained in many high-performance engine applications. In these cases of cam lobe destruction, it is clear that regular API SM oils are not able to provide EHD film lubrication at the points in the cam rotation where the oil film is minimum thickness. If there is no EHD oil film, the only mode of lubrication which can inhibit wear under these conditions is boundary. In boundary lubrication mode, the protection for the two bearing surfaces will be provided by the anti-wear agent in the oil, which has historically been ZDDP. Current API SM oils do indeed still have ZDDP, but the evidence says there is not a sufficient amount to protect high-performance engines from wear. The characteristics of ZDDP cause it to decompose into a glassy high-strength film in the presence of the high pressure and temperatures at the cam lobe and lifter contact point. This film, while not as low-friction a lubricant as an oil film, is the last line of defense against wear, and effectively forms a

---

[10] M Kushwaha and H Rahnejat, "Transient elastohydrodynamic lubrication of finite line conjunction of cam to follower concentrated contact," *Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, Loughborough, UK*, pg. 2882

sacrificial layer to protect the base metal of the cam lobe and lifter foot. Since it is sacrificial, the rate of film build will be partially dependent on the amount of ZDDP available in the oil. High spring pressures or rpms will wear away the film at a greater than normal rate, increasing the required amount of ZDDP needed to ensure low wear on the cam lobes and lifter foot.

Lubricating an Engine in Reality

Back here in reality, all of our engines are designed with a single lubrication system which is required to handle ALL of the lubrication chores in all engine systems. In addition, the oil is asked to move many kilowatts of heat from critical engine parts to the sump, where it is expected to transfer the heat to the sump walls and return cooled for more abuse. As if this wasn't enough of a job, the oil is also supposed to capture all blow-by and atmospheric moisture condensates, absorb, neutralize and store them until the next oil change, and protect all internal components from corrosion. The oil is asked to perform these chores for 3000 to 7500 miles or more, depending on how often you empty the sump. With the most recent innovations in precision engine design and fuel injection, there is a large push in the industry, especially in Europe, for sealed-for-life oil charges in the sump. We can expect PCEO to evolve with engine design towards this goal.

The quality and performance of today's base oil stocks far exceeds anything available in the past. Unfortunately, the change in engine design and the resulting change in oil composition means the newest oils do not necessarily contain the best additive package for older engines. Specifically in the area of anti-wear agents for the boundary lubrication regime, the newest oils exhibit less anti-wear capabilities than do the heaviest duty oils containing high amounts of ZDDP.

Obviously, a single oil cannot optimally lubricate all engine systems, while at the same time performing all of the other jobs it is asked to do. This means that like all other engineered substances, it is the product of compromises in design.

In many cases, the very best oil to use is the one that provides optimum protection for the weakest part of your specific engine. In the case of flat-tappet engines, especially those with high valve-spring pressures, you must be careful to ensure an adequate level of the anti-wear agent ZDDP.