WHITE PAPER

Lubrication Technology

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Contents

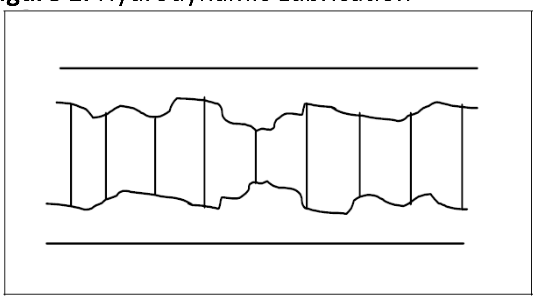
Scope

This paper provides basic lubrication information for industrial gearbox oils and examples of specific recommendations for Sumitomo Drive Technologies (SDT) products. Basic properties, terms from specifications sheets, and maintenance information are also included.

Lubrication Basics

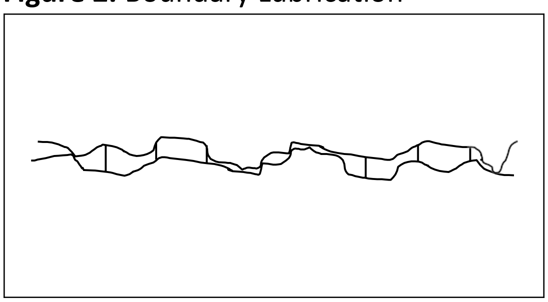
The purpose of lubrication is to reduce friction, transfer heat from bearings and gears, remove dirt and contaminants and prevent wear. There are three basic lubrication regimes: hydrodynamic, boundary, and mixed film lubrication. Film thickness is a function of viscosity, surface roughness, and relative velocity between load-bearing surfaces. Higher viscosity, smoother surfaces, and higher velocity increase film thickness. Figures 1 through 3 illustrate these differences.

Figure 1. Hydrodynamic Lubrication



Hydrodynamic lubrication occurs when lubrication films completely separates two load-bearing surfaces. With no metal-to-metal contact, machine life depends on oil cleanliness. Elasto-Hydrodynamic (EHD) describes instances where load-induced high fluid pressures (~ 500,000 psi) [1] cause the oil to act like solid and elastically deforms the mating surfaces. EHD may occur between gears and between bearing elements and raceways.

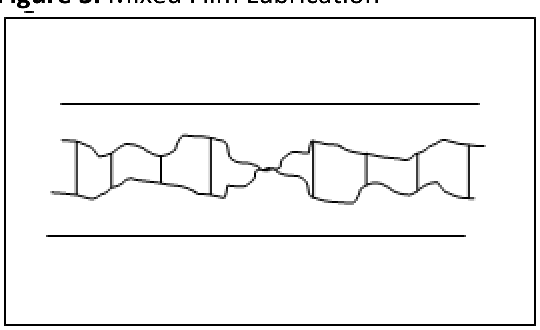
Figure 2. Boundary Lubrication



Boundary lubrication occurs when the load-bearing surfaces come into contact. It can occur when the relative speed between mating surfaces is low, there are high loads, or changes in direction.

Anti-wear (AW) or extreme pressure (EP) additives can reduce friction and wear to adequate levels.

Figure 3. Mixed Film Lubrication



Mixed film lubrication describes the condition where the asperities (peaks) of two surfaces come into contact even though a lubricating film is present. The lubrication film is thicker than in a boundary lubrication—it is a combination of hydrodynamic and boundary lubrication.

Friction can be lower than thick film hydrodynamic lubrication, but mixed film lubrication requires AW additives to reduce wear.

Base Oil

Oil's most important property is viscosity, a quantitative measure of a fluid's flow resistance. Absolute viscosity measures the force required to move a square-centimeter plate parallel to a reference surface at a speed one centimeter per second at a distance of one centimeter. Units are expressed as dyne*sec/cm2 or Poise (P) or centiPoise (cP). Kinematic viscosity measures the time required for a fluid to flow through a tube under the force of gravity and has the units m2/sec. Mathematically, it is also the same as absolute viscosity divided by the fluid density. Kinematic viscosity is expressed as Stokes (one centistoke = 1cSt=10-6 m2/sec). Most catalogs and equipment specification sheets specify kinematic viscosity.

Lubricating oil is a mixture of base oil and additives that modify the base oil properties. The American Petroleum Institute (API) categorizes base oils by type of refining per Table 1.

Mineral oils are refined from crude oils. Crude oils are a mixture of different types of hydrocarbons (e.g., paraffins, napthenes, aromatics, etc.) with molecules of different weights. Refining removes impurities and sorts hydrocarbons by weight. For a given viscosity grade molecular weight varies considerably, so that the properties are an 'average' of the component fractions.

Figure 4. Mineral and Synthetic Oil.

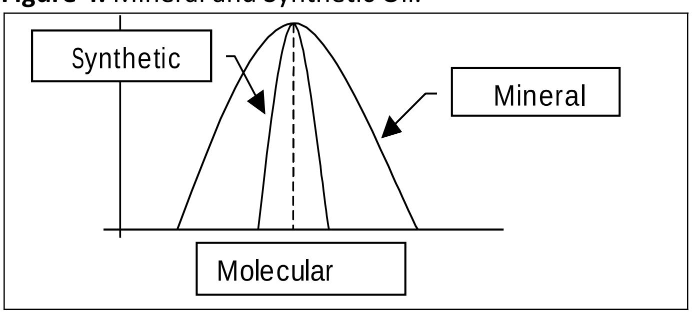


Figure 5. Viscosity Index

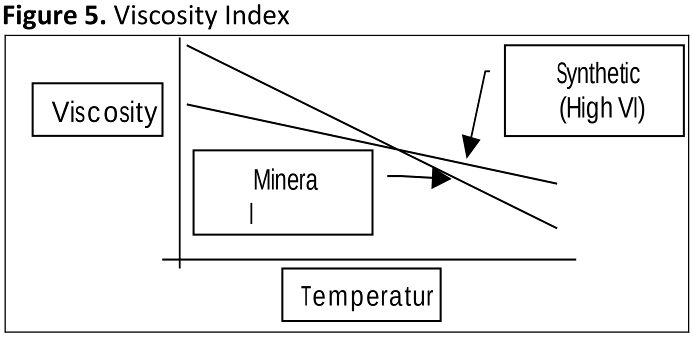


Figure 4 shows a comparison of synthetic and mineral oils. Both have an identical average weight, shown by the dashed line. Compared to synthetics, the lighter fractions in mineral oil cause it to become thinner at high temperature, and the heavier fractions cause mineral oils to be thicker at low temperature.

Figure 5 illustrates a qualitative comparison of viscosity index. Mineral oils have a lower viscosity index (VI) than synthetic oil. The higher the VI, the wider the oil's effective temperature range.

Group III base stocks are highly refined mineral oils that are marketed as synthetic in North America.

Group IV and **V** oils, formed from artificially assembled molecules, have uniform structure and molecular weight, and are much more expensive than mineral oils. Some synthetics are not mixable with other types of oil, may not be compatible with conventional oil seals, and may attack paint and some plastics. Group IV and V synthetics offer higher viscosity indices and good-to-excellent thermal stability and oxidation resistance.

Table 1. Base Oil [1]

Base Oil	Type	Characteristic	Advantage	Disadvantage
Group I	Mineral	Solvent-refined	Inexpensive Excellent solubility	Not as resistant to oxidation and thermal breakdown as others Viscosity Index 80~120
Group II		Hydrocracked	Inexpensive Excellent solubility Better oxidation and thermal stability	Less resistant to oxidation and thermal breakdown than Groups III, IV, and V Viscosity Index 80~120
Group III	Mineral (Synthetic)	Hydrocracked at higher temperature and pressure and ISO Dewaxed	Good oxidation and thermal stability Excellent solubility Higher Viscosity Index	More expensive than Groups I and II
Group IV	Synthetic	Polyalphaolephin (PAO)	Good oxidation and thermal stability Low pour point Higher Viscosity Index	~50% more expensive than Group I May shrink seals Poor additive solubility
Group V		Dibasic Acid Ester	Good oxidation and thermal stability Low pour point Viscosity Index above 140	~150% more expensive than Group Reacts with water May swell seals May remove paint
		Polyalkylene Gylcol	Good oxidation and thermal stability Viscosity Index 160~280	~150% more expensive than Group I Not compatible with other oils May remove some paints
		Polyol Ester	Good oxidation and thermal stability Low pour point Viscosity Index 130~190	~150% more expensive than Group I May swell seals May remove paint Not compatible with other oils

Additives
Oils, by themselves, are unable to provide all the characteristics required by modern machinery. Carefully blended additive packages improve viscosity index and friction, and reduce wear, oxidation, and corrosion. Too high a proportion of one additive may interfere with the benefit of others, or degrade lubrication effectiveness

Rust and Oxidation (R&O)
Oils fortified with additives that protect against rust and oxidations are known as inhibited oils (formerly R&O oils). Rust inhibitors have a polar head that strongly attracts to metal and a hydrophobic tail that repels water. Antioxidants protect oil by either reacting with, or breaking down, peroxides in the oil (initial oxidation products), or by neutralizing oxidation catalysts. Reducing oxidation damage increases oil's useable life.

Anti-wear (AW)
Anti-wear additives protect surfaces operating in boundary lubrication regime by bonding to metal and forming a layer that helps separate the surfaces in boundary lubrication.

Extreme Pressure (EP)
Extreme Pressure additives, also known as anti-scuff additives, form a sacrificial film, under heat and pressure that is worn away by sliding. EP-fortified oil films can support a much greater load and offer greater resistance to scuffing than AW or inhibited oils. This greater resistance to scuffing may cause certain friction-dependent devices – such as backstops – to slip. EP additives are also chemically aggressive. They can be corrosive at high temperatures and some additives can attack 'yellow metals' – such as the bronze wheels in worm gearboxes or brass bearing cages. EP additives trade a small amount of chemical wear for greatly reduced mechanical wear.

Viscosity Index Improvers (VII)
Viscosity Index Improvers are very large polymers used to improve oil's high temperature viscosity. In lower temperatures, VII curl-up and become inert. At higher temperatures, they unwind and entangle oil molecules. VII are used in multi-grade mineral oils. Under high shear, these polymers can align and cause a temporary decrease in viscosity –a phenomenon known as "shear thinning". If the polymers break, a permanent reduction in high-temperature viscosity occurs. Synthetics require very little, if any, VII due to their naturally high viscosity index.

Detergents and Dispersants
Detergents and dispersants suspend sludge and other insoluble compounds in the oil so that it can either be filtered or removed during oil changes. Detergents do not 'clean' dirty gearboxes, but will break up accumulated sludge.

Oil Properties

There are dozens of tests that measure properties such as wear, viscosity, and corrosion. Many are updated and refined by the American Gear Manufacturers Association (AGMA), Deutsche Industrie Norm (DIN), and others. Some tests, such as the Timken, are no longer required by AGMA. No single test accurately indicates how a particular oil will perform in a given gearbox under specific conditions. It is possible that oil with a lower rating for a particular test will, in a real application, out-perform oil with a higher rating.

Table 2 lists some common oil parameters related to viscosity and mechanical wear that may be found on product specification sheets.

Table 2. Common Oil Properties

Property	Description	Purpose
ISO Viscosity Grade	Viscosity classification. Oils are refined so that viscosity is within specific viscosity ranges at a specific temperature. E.g., ISO 460 oil has viscosity 460 cSt at 40oC.	Standardize oil selection.
Viscosity Index	Number that represents an oil’s change in viscosity with respect to temperature. The higher the number, the less viscosity changes.	Provides a method to compare viscosity- temperature performance for different types of oil.
Pour point	3oC above the temperature at which oil will not flow from a horizontal container after five seconds.	Provides the minimum temperature under which an oil may still be considered fluid. At temperatures well above rated pour point, oil may be too thick for use.
Flash point	Temperature at which the lighter molecular fractions may combust.	Provides a reference point for condition testing. If flash point has increased from specification, it indicates that the light oil fractions have boiled off during use and oil may no longer be in grade.
Timken EP Test	Test uses a machine that forces a block against a rotating outer bearing race until a measurable scar forms.	The largest force that does not generate a scar is the Timken OK load; indicates an oil’s anti-scuff property.
4-Ball EP Test	A rotating metal ball is forced against a trio of stationary balls. Two common results are the Weld Load and Load Wear.	
Weld load	Lowest load at which the rotating ball welds to the other balls or experiences extreme scoring.	Provides indication for oil performance in extreme pressure.

Load Wear	Lowest load at which the rotating ball forms a scar.	Provides indication for oil's anti-wear performance.
FZG Scuffing	Test where gear sets are run against one another.	Provides indication of lubricant anti-wear and extreme pressure properties when used with conventional gearing.

Wear [2]

The sections below explain some types of wear and the effect of lubrication.

Figure 1. Two-Body Abrasion

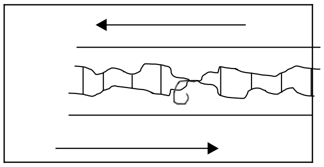
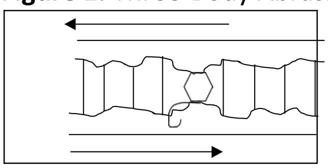


Figure 2. Three-Body Abrasion



Abrasive wear can be characterized as two-body or three body. Two-body wear (Figure 1) is abrasion between a hard surface and a softer one in boundary or mixed film lubrication. Three-body wear (Figure 2) describes wear that results when hard contaminants, such as dirt or wear particles, become embedded in one surface, and abrade the mating surface. As three- body wear progresses, more wear particles are introduced into the contact zone, and these new particles accelerate the wear rate.

Figure 3. Flaking

Figure 4. Micropitting

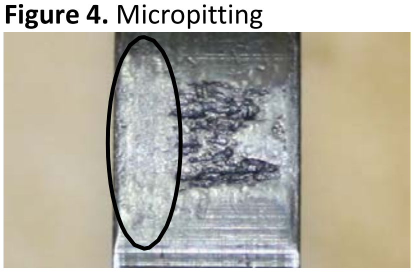


Figure 5. Pressure marks.

Surface fatigue describes a process where repeatedly applied stress initiates a subsurface crack. The crack propagates until it reaches the surface, where a section of the surface breaks free, leaving a pit. When several pits combine, they form a spall. Figure 3 shows flaking damage on a ball bearing element.

Stress may come from contaminates pressed between load-bearing surfaces and from contact of surface asperities. Figure 4 shows an SDT Cyclo® slow speed shaft roller with pressure marks contaminants.

With very hard surfaces, the pits may be small and shallow – on the order of 10□m deep. Areas of micropitting may appear ‘frosted’. The circled area in Figure 4 shows micropitting on the lobe of an SDT Cyclo® reducer disc.

There are several theories [4] to explain surface fatigue initiated by rolling elements.

1. Fine crack theory – oil forced into a crack creates force against mating surfaces generates a large hydraulic force that causes crack propagation.
2. Oil film peeling theory – oil between roller and surface imparts a tangential force along the surface that initiates a crack.
3. Thermal stress theory – friction-induced temperature creates thermal stress that initiates and propagates cracks.
4. Surface buckling theory – high contact pressures cause the surfaces to buckle and deform plastically. The deformed surfaces fatigue and crack.

"Corrosive wear" [2] is chemically induced and controlled by oil and its additives. Some wear-control additives chemically attack metal surfaces. By damaging the surface finish, corrosion makes it easier for cracks to form and propagate.

"Adhesive wear" [2] occurs when, under boundary lubrication conditions, the asperities in the opposing load-bearing surfaces weld together, then break apart. Small pits are left on one surface, small projections on the other, and wear particles are formed that can lead to abrasive wear. Adhesion wear is confined to the oxide layers on metallic surfaces.

"Scuffing" [2] is a severe type of adhesive wear where the disruption in machine surfaces extends through the oxide layers to bare metal. Running a gearbox under ½ load for approximately ten hours will smooth load-bearing surfaces and help prevent scuffing. For severe loading conditions, EP-type gear oils may be required.

"Polishing" [2] is a type of wear where the load-bearing surfaces are polished to a mirror-like finish. It can be caused by either very fine abrasive particles or chemical attack by EP additives. The surfaces may look good, but if the polishing is excessive, machine geometry may be affected. Countermeasures include cleaning the oil via filtration or oil changes and using less chemically aggressive EP additives.

Table 3 shows a summary of common wear modes and countermeasures. Increasing oil viscosity can mitigate most failure modes at the expense of higher oil temperatures due to the increased churning losses. The higher oil temperatures increase the breakdown and oxidation rate of oil and additives.

Table 3. Wear and Countermeasures [2]

Wear	Description	Higher viscosity	No Sulfur-Phosphorus additives	Reduce EP additives	Increase EP additives
Abrasion	Debris abrades surface	X			
Micro pitting	Very fine, asperity removal	X			
Macro pitting	Subsurface fatigue	X			
Corrosive	Chemical attack			X	
Adhesion	Transfer of material from one surface to another	X	Low speed apps		
Scuffing	Severe adhesion wear	X			X
Polishing	Very fine wear			X	

Selecting Oil

The most important factor in lubrication selection is oil film thickness. Gearbox manufacturers do not specify film thickness – they specify viscosity. As can be seen from Table 3, insufficient film thickness can lead to an increase in abrasion, pitting, and adhesion wear. If too high a viscosity is selected, gearbox-churning losses will lead to higher oil temperatures, and consequently shorter oil and additive life.

Film thickness is a function of viscosity, pressure, temperature, relative velocity, surface roughness, and other factors. All else being equal – geometry, velocity, load, and viscosity- some base oils will provide a thinner oil film. Mineral oils will provide the thickest film, followed by PAO, PAG, and esters [3, 5]. For a given viscosity grade (e.g., ISO 320) the synthetic base stock will offer higher viscosity at operating temperature, but film thickness may be thinner than if using a mineral base oil. If changing from mineral to synthetic, one cannot assume that the same viscosity grade will be acceptable.

This in no way implies that synthetic oils always have thinner films or inferior performance. In practice, for a given temperature, mineral and synthetic oil will not have the same viscosity. The synthetics' higher viscosity at elevated temperatures will offset some or all of the inherent film reduction. Conversely, mineral oil's poor VI implies higher viscosity – and thicker oil films – at lower temperatures. This topic will be the subject of future papers.

If choosing between a mineral and synthetic, one must weigh synthetics' cost disadvantage against their superior thermal stability, high viscosity index, and oxidation resistance. Though synthetic oils may offer extended life, they may not offer extended drain intervals on unfiltered lubrication systems. Independent of a lubricant's remaining useful life, contaminants and debris accumulation may require the same oil change intervals as mineral oil. If the user requirements dictate operation over a wide temperature range, or oil changes are impractical, then synthetic offers clear advantage over mineral.

The next thing to consider is additive package. Inhibited oils are often used in hydraulic cylinders or other applications where there will always be hydrodynamic lubrication. Hydrodynamic lubrication is not always present in gearboxes during starting, reversing, or when experiencing shock load. AW oils have chemically mild additives that will form weak bonds with metal and help separate load bearing surfaces under boundary conditions. AW oils offer minimal corrosion and the additives activate at lower temperatures than EP additives. EP additives can provide protection under very heavy and shock loading conditions. Below their relatively high activation temperature, EP additives are inert. When local temperatures are high enough for the additives to bond to metal, they offer superior protection against scuffing, but at the expense of corrosive wear. Under light loading conditions and lower temperatures, EP oils, due to corrosion, may offer increased wear compared to AW oils. [1]

Gearboxes in applications where there may some incidental contact with food will require National Science Foundation (NSF®) H1 class lubricants. The base oils may be mineral or synthetic, but there are significant restrictions to the additives. Many common EP additives are not allowed. Higher viscosity oil may compensate.

Selecting Oil for SDT Gearboxes

All manuals make recommendations based on ambient temperature and sometimes other factors. Specific, factory-tested oils are listed. If it becomes necessary to choose non-catalog, it is best to consult the appropriate SDT Product or Application Engineer.

The Cyclo® Reducer, Cyclo®HBB, Cyclo® BBB and the Paramax require an EP-type oil with a viscosity at start-up no higher than 4300 cSt for most applications, or 2200 cSt when oil is supplied via a shaft driven oil pump. At operating temperature, viscosity should be no lower than 15 cSt. Synthetic and NSF® H1 approved oil may require an increase of one or two viscosity grades.
HSM reducers require AW mineral or synthetic oil. EP type oils may be used in units without an integral backstop. Integral backstop units can only use approved EP type oils.

Recommended Maintenance Schedules

Oil in all gearboxes should be changed after the first 500 hours in order to remove debris from initial wear. The tables below summarize subsequent oil change recommendations.

Table 4. Cyclo® and HSM

Less than 10 hrs/day	Every six months
10 to 24 hrs/day	Every 2500 hours
High ambient temperatures, high humidity, or atmospheres with corrosive gases	Every one to three months

Table 5. Paramax.

Second change	2500 hours
Subsequent changes, oil temperature less than 70oC	5000 hours
Subsequent changes, oil temperature greater than 70oC	2500 hours

Conclusion

Properly sized for the application, gearbox life is often limited by maintenance. Proper maintenance includes using the correct oil and, by either oil changes or filtration, keeping contaminants and wear debris concentration low. SDT reducers have lasted many years. Figures 6 shows a Cyclo® disc a gearbox that completed 2000 hours of continuous full load testing. Please note the absence of pitting and visible wear.

Figure 6. Cyclo® disc

References

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Lee is a graduate of Virginia Tech and has over 14 years experience in the power transmission industry ranging from motors to cycloidal and hypoid type gear reducers.