Lubricants and Hydraulic Fluids: *Overview*

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This course was adapted from the US Corps of Engineers, Publication No. EM 1110-2-1424, "Lubricants and Hydraulic Fluids", which is in the public domain.

CHAPTER 1

Introduction

1-1. Purpose.

a. This manual provides engineering personnel with design guidance to select, specify, inspect, and approve lubricants and hydraulic fluids used for U.S. Army Corps of Engineers (USACE) machinery and equipment. It provides operation and maintenance staff with guidance for regular and scheduled maintenance of lubricants.

1-2. Applications. This manual applies to all HQUSACE elements, major subordinate commands, districts, laboratories, and field operating activities having responsibilities for the design and construction of civil works projects.

1-3. References. References are in Appendix A. References include technical papers, engineering guidance, engineering manuals, industry standards, and textbooks.

1-4. General.

a. The document is a revision and update of the information presented in the 28 February 1999 version of EM 1110-2-1424. This manual is intended to be a practical guide to lubrication with enough technical detail to allow personnel to recognize and easily discern differences in performance properties specified in manufacturers' product literature so that the proper lubricant for a particular application is selected. The manual is intended for designers and operators of civil works facilities. The manual defines and illustrates friction, wear, and corrosion, and how they damage contact surfaces to cause premature equipment failure.

b. Certain specific chapters are dedicated to the discussion of single topics.

(1) Chapter 2 explores the mechanisms of wear.

(2) Chapter 9 specifically addresses turbine oils for the hydropower industry, including design requirements and selection criteria. Section 9-4 includes guidance on the use of Unified Facilities Guide Specification (UFGS) 48 13 19.00, *Turbine Oil* (May 2014), a guide specification now available for turbine oils.

(3) Chapter 10 provides information on the design and selection of hydraulic fluids, including specialized hydraulic fluids such as fire resistant fluids.

(4) Chapter 12 includes a discussion of self-lubricated bearings.

(5) Chapters 14 through 17 are devoted to lubricant specification and selection, and requirements of lubricants for equipment currently in use at USACE civil works facilities.

(6) Chapter 17 also specifically addresses operation and maintenance considerations.

c. Environmentally Acceptable Lubricants. Environmentally Acceptable Lubricants are becoming more predominant in industry and within USACE. Chapter 13 provides guidance on the design and selection of environmentally acceptable lubricants.

d. Relationship to Other Manuals. This manual supersedes all previous versions of EM 1110-2-1424. It should be used in conjunction with EM 1110-2-2610, *Mechanical and Electrical Design of Navigation Locks and Dams* (30 June 2013), EM 1110-2-4205, *HydroElectric Power Plants: Mechanical Design* (30 June 1995) and all other referenced engineering manuals for the design of hydropower systems, gates, and operating machinery. Appendix A lists other applicable manuals.

1-5. Mandatory Requirements and Deviation from Design Criteria. This manual provides guidance for the protection of U.S. Army Corps of Engineers (USACE) structures and equipment. In certain cases, guidance requirements, because of their criticality to project safety and performance, are considered to be mandatory as discussed in Engineer Regulation (ER) 1110-2-1150. *Engineering and Design for Civil Works Projects* (31 August 1999). Those cases will be identified as "mandatory," or the word "shall" will be used in place of "should."

1-6. Appendices. Appendices to this manual provide useful supplementary material:

a. Appendix A lists references.

b. Appendix B provides grease and oil selection criteria for machinery.

c. Appendix C includes Engineering and Construction Bulletin (ECB) 2006-11, Tainter Gate Trunnion Lubrication.

d. Appendix D provides the Bureau of Reclamation document FIST 2-4, Lubrication of Powerplant Equipment.

e. Appendix E includes the U.S. Environmental Protection Agency (USEPA) document 800-R-11-002, *Environmentally Acceptable Lubricants* (November 2011).

f. Appendix F contains Maintenance Scheduling for Mechanical Equipment Bureau of Reclamation FIST 4-1A January 2009

g. Appendix G contains ERDC/CERL TR-04-28, *Performance Problems with Group II Hydrocracked Turbine Oils in Corps of Engineers Hydropower Facilities* (December 2004)

h. Appendix H contains the Evaluation of Environmentally Acceptable Lubricants (EALs) for Dams Managed by the U.S. Army Corps of Engineers.

i. Appendix I contains the Evaluation of Environmentally Acceptable Lubricants (EALs) Non-Hydropower Uses for Dams Affected by the 2014 Riverkeeper's Settlement

j. Appendix J contains Environmentally Acceptable Lubricant Grease for Hydropower Applications

k. Appendix K provides a glossary of lubrication terms.

CHAPTER 2

Friction and Mechanisms of Wear

2-1. Friction and Objective of Lubrication.

a. Definition of friction. Friction is a force that resists relative motion between two surfaces in contact. In short, it is the resistance to motion. Depending on the application, friction may be desirable or undesirable. Certain applications, such as tire traction on pavement and braking, or when feet are firmly planted to move a heavy object, rely on the beneficial effects of friction for their effectiveness. In other applications, such as operation of engines or equipment with bearings and gears, friction is undesirable because it causes wear and generates heat, which can lead to premature failure of equipment. For purposes of this manual, the energy expended in overcoming friction is dispersed as heat and is considered to be wasted because useful work is not accomplished. This waste heat is a major cause of excessive wear and premature failure of equipment. Two general cases of friction occur: sliding friction and rolling friction. As a general rule, sliding friction will be greater than rolling friction.

b. Asperities. All metal surfaces have some degree of surface roughness. Regardless of how smooth a surface may appear, it has many small irregularities called “asperities.” These are high points and valleys on the metal surface. When two surfaces move past each other, it is the asperities on one surface that are in contact with the asperities on the other surface. In cases where a surface is extremely rough, the contacting points are significant, but when the surface is fairly smooth, the contacting points have a very modest effect. The real or true surface area refers to the area of the points in direct contact. This area is considerably less than the apparent geometric area. One function of a lubricant is to keep the asperities between metal surfaces apart and thus eliminate metal-to-metal contact and reduce friction.

c. Objective of Lubrication. The purpose and objective of lubrication is to reduce friction and wear of metal surfaces and machine parts that move relative to each other. Cooling and debris removal are other important benefits provided by a fluid lubricant. There are a number of ways to reduce friction and these will be discussed in this manual. This includes selecting the proper lubricant, using rolling friction versus sliding friction where possible, and selecting the proper surface finish and material selection.

d. Other Functions of Lubricants. Lubricants also control the temperature of machinery and equipment. Lubricants absorb heat generated at the surface of components and carry it away to be dispersed. Along with heat, lubricants transport particulates, dirt, and other debris away from the friction surface.

2-2. Sliding Friction.

a. Definition. To visualize sliding friction, imagine a steel block lying on a steel table. Initially a force F (action or pushing force) is applied horizontally in an attempt to move the block. If the applied force F is not high enough, the block will not move because the friction between the block and table resists movement. If the applied force is increased, eventually it will be sufficient to overcome the frictional resistance force f and the block will begin to move. At this precise instant, the applied force F is equal to the resisting friction force f and is referred to as the "force of friction" (Figure 2-1)

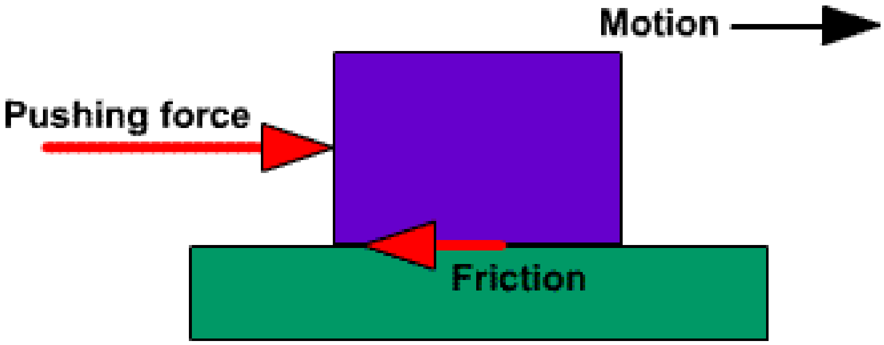

Figure 2-1. Friction Force and Motion.

b. Normal Load and Frictional Force. The discussion of friction and static and dynamic forces are provided in more detail in the *Machinery's Handbook*, 29th ed. (Oberg et al. 2012). In mathematical terms, the relation between the normal load N (weight of the block) and the friction force f is given by the coefficient of friction denoted by the Greek symbol μ. Note that in the present context, "normal" has a different connotation than commonly used. When discussing friction problems, the normal load refers to a load that is perpendicular to the contacting surfaces. For the example used here, the normal load is equal to the weight of the block because the block is resting on a horizontal table. However, if the block were resting on an inclined plane or ramp, the normal load would not equal the weight of the block, but would depend on the angle of the ramp. The relationship between force of friction and the normal force is then:

$$f = \mu \times N \text{ and } \mu = f/N \qquad (2\text{-}1)$$

where:

μ = coefficient of friction

f = frictional force

N = normal load.

c. Inclined Load and Frictional Force. If an object is placed on an inclined plane, the friction between the object and the plane will prevent any movement (provided the angle of inclination is not very large). At a certain angle, the object will barely be stationary. As the angle is increased beyond this, the frictional resistance will be overcome and the force of gravity will allow the block to slide down the inclined plane. The free body diagram shown in Figure 2-2 shows the angle at which this happens, which is termed the "angle of repose" and is typically denoted by the Greek letter θ; therefore $\mu = \tan \theta$.

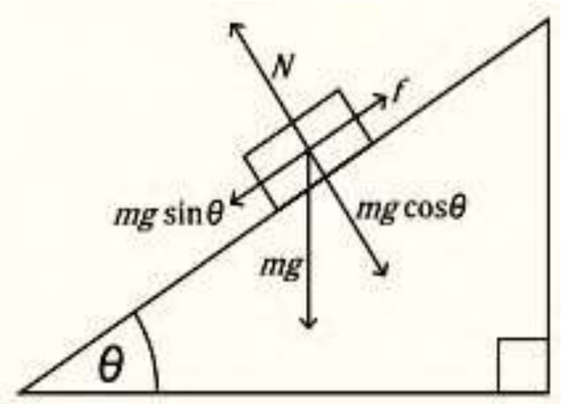

Figure 2-2. Free Body Diagram of a Block on an Inclined Plane,
Where m=mass and g=Gravitational Force.

d. Laws of Sliding Friction. The following friction laws are extracted from the *Machinery's Handbook*, 29th ed. (Oberg et al. 2012).

(1) Unlubricated surfaces. Three laws generally govern the relationship between the frictional force f and the load or weight L of the sliding object for dry surfaces (not lubricated):

(a) "For low pressures (normal force per unit area), the friction force is directly proportional to the normal load between the two surfaces. As the pressure increases, the friction does not rise proportionally; but when the pressure becomes abnormally high, the friction increases at a rapid rate until seizing takes place."

(b) "The friction both in its total amount and its coefficient is independent of the areas in contact, so long as the normal force remains the same. This is true for moderate pressures only. For high pressures, this law is modified in the same way as the first case."

(c) "At very low velocities, the friction force is independent of the velocity of rubbing. As the velocities increase, the friction decreases." This implies that the force required to set a body in motion is the same as the force required to keep it in motion, but this is not true. Once a body is in motion, the force required to maintain motion is less than the force required to initiate motion and there is some dependency on velocity. These facts reveal two categories of friction: static and dynamic. Static friction force is the force required to initiate motion (Fs). Kinetic or dynamic friction force is the force required to maintain motion (F_k).

(2) Lubricated surfaces. The friction laws for well lubricated surfaces are considerably different from those for dry surfaces, as follows:

(a) "The frictional resistance is almost independent of the pressure (normal force per unit area) if the surfaces are flooded with oil."

(b) "The friction varies directly as the speed, at low pressures; but for high pressures the friction is very great at low velocities, approaching a minimum at about 2 feet per second (0.61 meters per second) linear velocity, and afterwards increasing approximately as the square root of the speed."

(c) "For well lubricated surfaces the frictional resistance depends, to a very great extent, on the temperature, partly because of the change in viscosity of the oil and partly because, for journal bearings, the diameter of the bearing increases with the rise in temperature more rapidly than the diameter of the shaft, thus relieving the bearing of side pressure."

(d) "If the bearing surfaces are flooded with oil, the friction is almost independent of the nature of the material of the surfaces in contact. As the lubrication becomes less ample, the coefficient of friction becomes more dependent on the material of the surfaces."

2-3. Friction and Factors Affecting Friction.

a. Definition. The coefficient of friction (μ) depends on the type of material. Tables showing the coefficient of friction of various materials and combinations of materials are available. Common sources for these tables are the *Tribology Handbook* (Neale 1995), Marks Mechanical Engineering Handbook, Tribology Data Handbook, and *Machinery's Handbook*, 29th ed. (Oberg et al. 2012). The tables show the coefficient of friction for clean dry surfaces and lubricated surfaces. It is important to note that the coefficients shown in these tables can vary.

b. Factors Affecting Friction. Multiple factors affect the friction between surfaces and the coefficient of friction. This includes the viscosity of the lubricant, the surface finish, the load, the speed, and the type of motion (sliding versus rolling).

c. Adhesion. Adhesion occurs at the points of contact of surfaces and refers to the welding effect that occurs when two bodies are compressed against each other. This effect is more commonly referred to as "cold welding" and is attributed to pressure rather than heat. A shearing force is required to separate cold-welded surfaces.

d. Shear Strength and Pressure. As previously noted, the primary objective of lubrication is to reduce friction and wear of sliding surfaces. This objective is achieved by introducing a material with a low shear strength or coefficient of friction between the wearing surfaces. Although nature provides such materials in the form of oxides and other contaminants, the reduction in friction due to their presence is insufficient for machinery operation. For these conditions, a second relationship is used to define the coefficient of friction: $\mu = S/P$, where S is the shear strength of the material and P is pressure (or force) contributing to compression. This relationship shows that the coefficient of friction is a function of the force required to shear a material.

e. Stick-Slip. To the unaided eye, the motion of sliding objects appears steady. In reality, this motion is jerky or intermittent because objects slow during shear periods and accelerate following the shear. This process is continuously repeated while the objects are sliding. During shear periods, the static friction force "Fs" controls the speed. Once shearing is completed, the kinetic friction force "F_k" controls the speed and the object accelerates. This effect is known as stick-slip. In well lubricated machinery operated at the proper speed, stick-slip is insignificant, but it is responsible for the squeaking or chatter sometimes heard in machine operation. Machines that operate over long sliding surfaces, such as the ways of a lathe, are subject to stick-slip. To prevent stick-slip, lubricants are provided with additives to make the static friction force "Fs" less than the kinetic friction force "F_k."

2-4. Rolling Friction.

a. Definition. When a body rolls on a surface, the force resisting the motion is termed rolling friction or rolling resistance (Figure 2-3). Experience shows that much less force is required to roll an object than to slide or drag it. Because force is required to initiate and maintain rolling motion, there must be a definite but small amount of friction involved. Unlike the coefficient of sliding friction, the coefficient of rolling friction varies with conditions and has a dimension expressed in units of length.

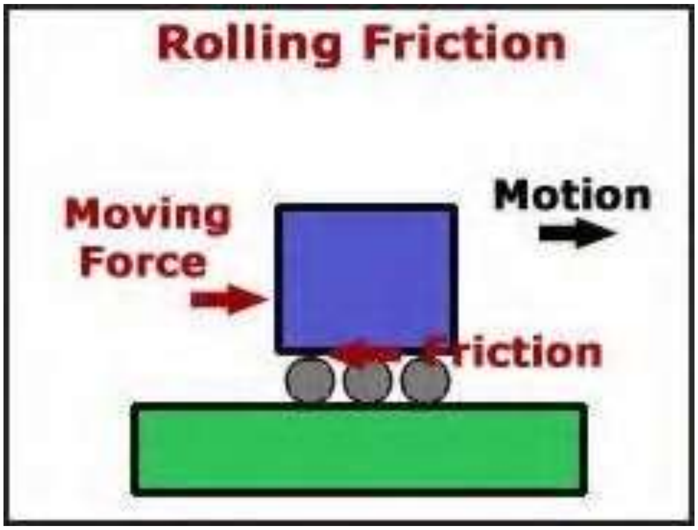

Figure 2-3. Rolling Friction.

(1) Ideally, a rolling sphere or cylinder will make contact with a flat surface at a single point or along a line (in the case of a cylinder). In reality, the area of contact is slightly larger than a point or line due to elastic deformation of either the rolling object or the flat surface, or both. Much of the friction is attributed to elastic hysteresis. A perfectly elastic object will spring back immediately after relaxation of the deformation. In reality, a small but definite amount of time is required to restore the object to original shape. As a result, energy is not entirely returned to the object or surface, but is retained and converted to heat. The source of this energy is, in part, the rolling frictional force.

(2) A certain amount of slippage (which is the equivalent of sliding friction) occurs in rolling friction. If the friction of a rolling object is measured, slippage effects are minimal. However, in practical applications such as a ball or roller bearings, slippage occurs and contributes to rolling friction. Neglecting slippage, rolling friction is very small compared to sliding friction.

b. Laws of Rolling Friction. The *Machinery's Handbook*, 29th ed. (Oberg et al. 2012) provides a more extensive discussion of rolling friction. The laws for sliding friction cannot be applied to rolling bodies in equally quantitative terms, but the following generalities can be given:

(1) The rolling friction force F is proportional to the load L and inversely proportional to the radius of curvature r, or $F = fL/r$, where f equals the coefficient of rolling resistance, in centimeters or inches. As the radius increases, the frictional force decreases.

(2) The rolling friction force F can be expressed as a fractional power of the load L times a constant k or $F = kL^n$ where the constant k and the power n must be determined experimentally.

(3) The friction force F decreases as the smoothness of the rolling element improves. This is why bearings are designed and manufactured to be as smooth as possible.

2-5. Wear.

a. Definition. Wear removes material from working surfaces. Although it is an inescapable process, it can be reduced by appropriate machinery design, precision machining, proper maintenance, and proper lubrication. Wear is defined as the progressive damage resulting in material loss due to relative contact between adjacent working parts. Although some wear is to be expected during normal operation of equipment, excessive friction causes premature wear, and this creates significant economic costs due to equipment failure, cost for replacement parts, and downtime. Friction and wear also generate heat, which represents wasted energy that is not recoverable. In other words, wear is also responsible for overall loss in system efficiency. Wear can be minimized by modifying the surface properties of solids by one or more of “surface engineering” processes (also called surface finishing) or by use of lubricants. Cleanliness of lubricants and viscosity are also critical to minimizing wear.

b. Wear and Surface Damage. The wear rate of a sliding or rolling contact is defined as the volume of material lost from the wearing surface per unit of sliding length. For any specific sliding application, the wear rate depends on the normal load, the relative sliding speed, the initial temperature, and the mechanical, thermal, and chemical properties of the materials in contact. The effects of wear are commonly detected by visual inspection of surfaces. Surface damage can take place without exchange of material or with exchange of material (either loss or gain of material).

c. For the case of surface damage without exchange of material, no material is lost or gained. This includes structural changes such as aging, tempering, phase transformations, and re-crystallization. This also includes plastic deformation such as residual deformation of the surface layer. Finally, it also includes surface cracking, or fractures caused by excessive contact strains or cyclic variations of thermally or mechanically induced strains.

d. Surface Damage with Loss of Material (Wear). This is characterized by wear scars of various shapes and sizes. This can be shear fracture, extrusion, chip formation, tearing, brittle fracture, fatigue fracture, chemical dissolution, and diffusion.

e. Surface damage with gain of material. This can include pickup of loose particles and transfer of material from the opposing surface. It also includes corrosion or material degradation by chemical reactions with ambient elements or elements from the opposing surface.

2-6. Types of Wear. Ordinarily, wear is thought of only in terms of abrasive wear occurring in connection with sliding motion and friction. However, wear also can result from adhesion, fatigue, or corrosion. Normal wear is inevitable whenever there is relative motion between surfaces. However, wear can be reduced by appropriate machinery design, precision machining, material selection, and proper maintenance, including lubrication. The remainder of this manual is devoted to discussions on the fundamental principles of lubrication that are necessary to reduce wear.

a. Abrasive Wear. Abrasive wear occurs when a hard surface slides against and cuts grooves into a softer surface. Abrasive wear can be three-body or two-body type as noted below. Abrasive type wear is also the most common wear problem.

(1) Three-body abrasion occurs when a relatively hard contaminant (particle of dirt or wear debris) of roughly the same size as the dynamic clearances (oil film thickness) becomes embedded in one metal surface and is squeezed between the two surfaces, which are in relative motion (Figure 2-4). When the particle size is greater than the fluid film thickness, scratching, ploughing, or gouging can occur.

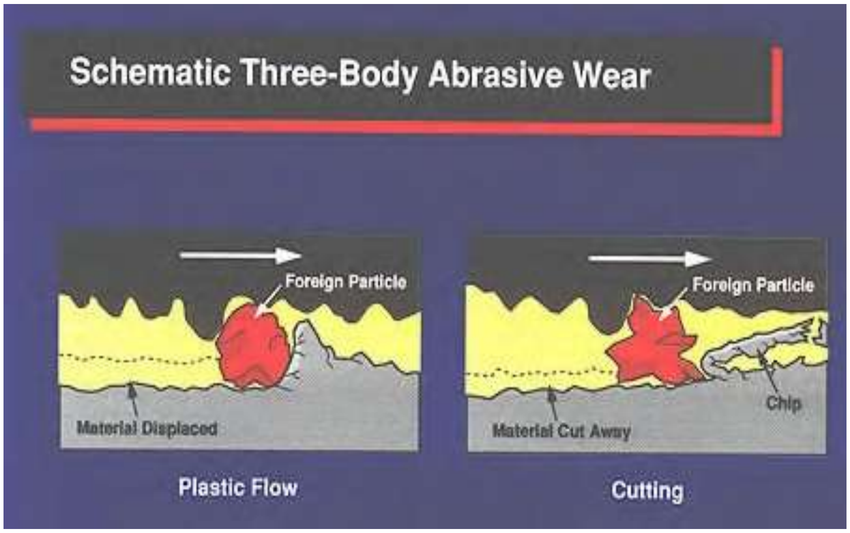

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Figure 2-4. Three-Body Abrasive Wear.

(2) Two-body abrasion occurs when metal asperities (surface roughness, peaks) on one surface cut directly into a second metal surface (Figure 2-5). In this case, a contaminant particle is not directly involved. The contact occurs in the boundary lubrication regime due to inadequate lubrication or excessive surface roughness.

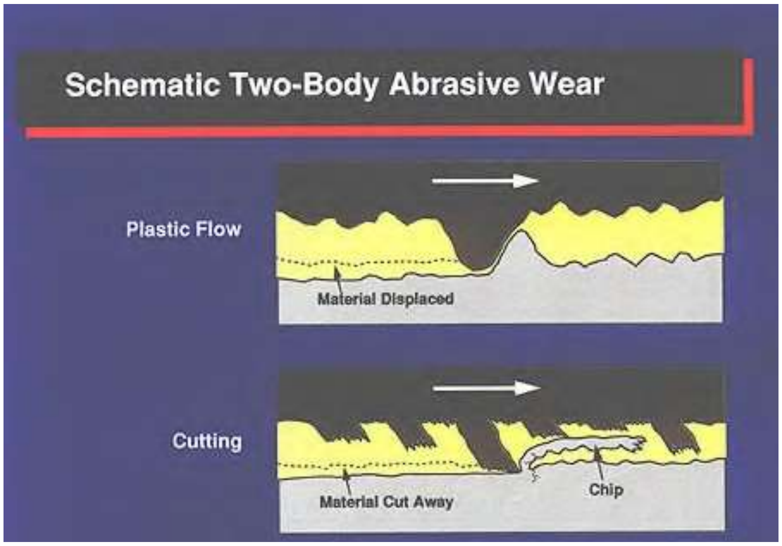

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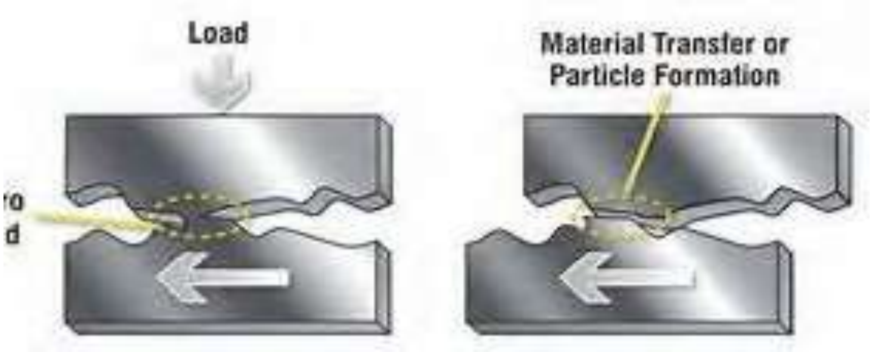
Figure 2-5. Two-Body Abrasive Wear.

(3) Erosion can also be considered a form of abrasive wear. It occurs principally in high velocity, fluid streams where solid particle debris, entrained in the fluid (oil), impinges on a surface and erodes it away. Hydraulic systems are an example where this type of wear may occur.

b. Adhesive Wear. Adhesive wear can be mild or severe. Adhesive wear is the transfer of material from one contacting surface to another (Figure 2-6). It occurs when high loads, temperatures, or pressures cause the asperities on two contacting metal surfaces, in relative motion, to spot-weld together then immediately tear apart, shearing the metal in small, discrete areas. Adhesion occurs in equipment operating in the mixed and boundary lubrication regimes due to insufficient lubricant supply, inadequate viscosity, incorrect internal clearances, incorrect installation, or misalignment. This can occur in rings and cylinders, bearings, and gears.



Courtesy of Noria Corporation and Machine Lubrication Magazine.

Figure 2-6. Diagram of Adhesion Wear.

(1) Mild adhesion generally involves removal of surface film material due to adhesion and subsequent loosening during relative motion. Mild adhesion involves transfer and loosening of surface films only. It is normal and common and generally not preventable. Generally all machine parts are susceptible.

(2) Severe adhesion involves cold welding of metal surfaces due to intimate metal-to-metal contact. This is sometimes referred to as “galling.” Several factors can contribute to severe adhesion including high loads, speeds, and/or temperatures. Other factors include insufficient lubricant, lack of antiscuff additives, no break-in, and surface roughness.

c. Fatigue Wear. Fatigue wear is due to surface failure of a material as a result of stresses that exceed the endurance (fatigue) limit of the material. Two primary types are surface and subsurface. Metal fatigue is demonstrated by bending a piece of metal wire, such as a paper clip, back and forth until it breaks. Whenever a metal shape is deformed repeatedly, it eventually fails. A different type of deformation occurs when a ball bearing under a load rolls along its race. The bearing is flattened somewhat and the edges of contact are extended outward. This repeated flexing eventually results in microscopic flakes being removed from the bearing. Fatigue wear also occurs during sliding motion. While fatigue is generally viewed as a mode of failure, this is not always detrimental. During the break-in period of new machinery, friction wears down working surface irregularities. Parts that are continuously subjected to repeated stress will experience destructive pitting as the material’s endurance limit is reached.

(1) Subsurface fatigue is a form of wear that occurs after many cycles of high-stress flexing of the metal. This causes cracks in the subsurface of the metal, which then propagate to the surface, resulting in a piece of surface metal being removed. Subsurface fatigue failure is the result of a bearing living out its normal life span based on the load, speed, and lubricant film thickness that it is exposed to.

(2) Surface fatigue begins with reduced lubrication regime and a loss of the normal lubricant film. The oil film is reduced to boundary or a mixed regime. Some metal-to-metal contact and sliding motion occurs. Surface damage occurs. The high points of the metal surface asperities are removed, which initially appear as a matted or frosted surface. This is not smearing, as in adhesion. This type of surface damage is usually visible with a magnification of three to five times.

d. Corrosive Wear. Corrosive wear occurs as a result of a chemical reaction on a wearing surface. The most common form of corrosion is due to a reaction between the metal and oxygen (oxidation); however, other chemicals may also contribute. Corrosion products, usually oxides, have shear strengths different from those of the wearing surface metals from which they were formed. The oxides tend to flake away, resulting in the pitting of wearing surfaces. Ball and roller bearings depend on extremely smooth surfaces to reduce frictional effects. Corrosive pitting is especially detrimental to these bearings. Corrosion of metals can be defined as the destructive attack of a metal through interaction with its environment. Corrosion is a process that occurs when a material deteriorates due to its interaction with the surrounding environment in which an electrochemical reaction consumes the material through oxidation. The corrosion of metals is an electrochemical process. That is, it is an electrical circuit where the exchange of electrons (electricity) is conducted by chemical reactions in part of the circuit. These chemical reactions occur at the surface of the metal exposed to the electrolyte. Oxidation reactions (corrosion) occur at the surface of the anode and reduction reactions occur at the surface of the cathode. Corrosion is often caused by the contamination or degradation of lubricants in service. Most lubricants contain corrosion inhibitors that protect against this type of attack. When the lubricant additives become depleted due to extended service or excessive contamination by moisture, combustion, or other gases or process fluids, the corrosion inhibitors are no longer capable of protecting against the acidic (or caustic) corrosive fluid and corrosion-induced pitting can occur. The pits will appear on the metal surface that was exposed to the corrosive environment. Additives can also be depleted by filtering at incorrect temperatures or in conditions where additives have stopped being suspended. Factors that can contribute to corrosion include a corrosive environment, corrodible metals, and high temperatures.

e. Galvanic Corrosion. Galvanic corrosion is an electrochemical process in which one metal corrodes preferentially to another when both metals are in electrical contact and immersed in an electrolyte. The same galvanic reaction is exploited in primary batteries to generate a voltage. When a galvanic couple forms, one of the metals in the couple becomes the anode and corrodes more quickly than it would alone, while the other becomes the cathode and corrodes more slowly than it would alone. For galvanic corrosion to occur, three conditions must be present:

- Electrochemically dissimilar metals must be present.
- These metals must be in electrical contact.
- The metals must be exposed to an electrolyte.

f. Cavitation. This is a special form of erosion in which vapor bubbles in the fluid form in low pressure regions and are then collapsed (imploded) in the higher pressure regions of the oil system. The implosion can be powerful enough to create holes or pits, even in hardened metal if the implosion occurs at the metal surface. This type of wear is most common in hydraulic pumps, especially those that have restricted suction inlets or that are operating at high elevations. Cavitation occurs in liquid when bubbles form and implode in pump systems or around propellers. Pumps put liquid under pressure, but if the pressure of the substance drops or its temperature increases, it begins to vaporize, just like boiling water. Bubbles in the liquid cannot escape so they implode, causing physical damage to parts of the pump or propeller.

CHAPTER 3

Principles of Lubrication

3-1. Lubrication and Lubricants.

a. Purpose of Lubrication. The study of lubrication and mechanism wear is called tribology. The primary purpose of lubrication is to reduce wear and heat buildup between contacting surfaces in relative motion. While wear and heat cannot be completely eliminated, they can be reduced to negligible or acceptable levels. Because heat and wear are associated with friction, both effects can be minimized by reducing the coefficient of friction between the contacting surfaces. Lubrication is also used to reduce oxidation and prevent rust, to provide insulation in transformer applications, to transmit mechanical power in hydraulic fluid power applications, and to seal against dust, dirt, and water. This chapter discusses four types of lubrication regimes. Two of the preferred and most common types of fluid related lubricant films are hydrodynamic (sometimes called full film lubrication) and elastohydrodynamic. Hydrodynamic films are present between sliding contacts. The most common examples would be a journal bearing and thrust bearings at hydropower plants. Elastohydrodynamic lubrication relates to rolling contact. Other lubrication regimes include boundary lubrication and mixed lubrication.

b. Lubricants. Reduced wear and heat are achieved by inserting a lower viscosity (shear strength) material between wearing surfaces that have a relatively high coefficient of friction. In effect, the wearing surfaces are replaced by a material with a more desirable coefficient of friction. Any material used to reduce friction in this way is a lubricant. Lubricants are available in liquid, solid, and gaseous forms. Industrial machinery ordinarily uses oil or grease lubricants. Solid lubricants such as molybdenum disulfide or graphite are used when the loading at contact points is heavy. In some applications, the wearing surfaces of a material are plated with a different metal to reduce friction.

c. Lubrication Regimes. An understanding of lubrication regimes is fundamental to tribology. It is critical and important to understand what lubrication regime equipment and machinery are operating in. The fluid viscosity, the load that is carried by the two surfaces, and the speed that the two surfaces move relative to each other combine to determine the thickness of the fluid film. This, in turn, determines the lubrication regime. How these factors all affect the friction losses and how they correspond to the different regimes is shown on a Hersey-Stribeck curve discussed in the next section. The Society of Tribologists and Lubrication Engineers (STLE) web site also provides extensive discussion of lubrication regimes. In addition to fluid film lubrication, there is solid film lubrication, in which a thin solid film separates two surfaces. The basic regimes of fluid film lubrication are:

- Hydrodynamic lubrication, in which two surfaces are separated by a fluid film.
- Elastohydrodynamic lubrication, in which two surfaces are separated by a very thin fluid film; this regime is related to rolling contact such as roller bearings.
- Mixed lubrication, in which two surfaces are partly separated and partly in contact.

- Boundary lubrication, in which two surfaces mostly are in contact with each other even though a fluid is present.

3-2. Hersey-Stribeck Curve.

a. Definition. Engineers often use the Hersey-Stribeck curve (Figure 3-1) to evaluate lubricants, to design bearings, and to understand lubrication regimes. Depending on the type of intervening film and its thickness, a number of lubrication regimes can be identified. Stribeck performed comprehensive experiments on journal bearings around 1900. He measured the coefficient of friction as a function of load, speed, and temperature. Some years later, Hersey performed similar experiments and devised a plotting format based on a dimensionless parameter.

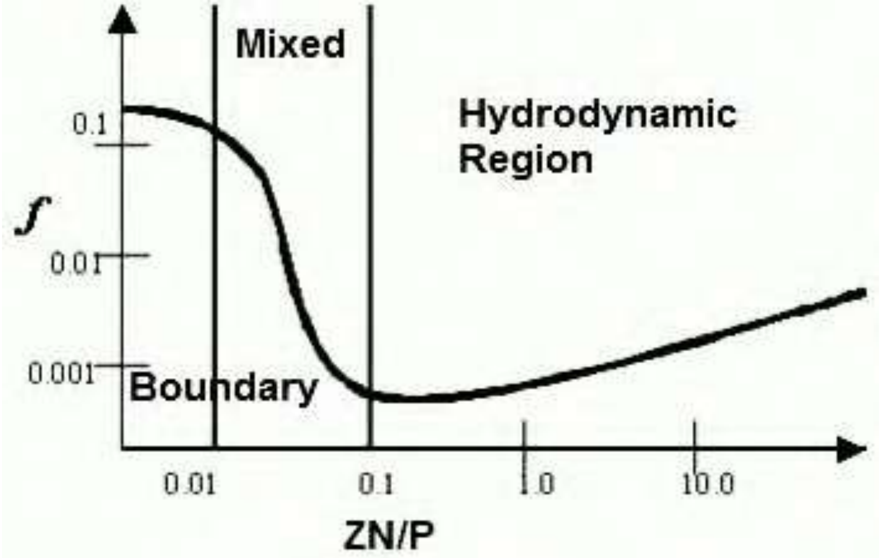


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Figure 3-1. Hersey-Stribeck Curve.

b. The Hersey-Stribeck Curve is a plot of the friction as it relates to viscosity, speed, and load. On the vertical axis is the friction coefficient. The horizontal axis shows a dimensionless parameter that combines the other variables: ZN/P. In this formula, Z is the fluid viscosity (Z is often used when viscosity is expressed in centipoise), N is the relative speed of the surfaces, and P is the load on the interface per unit bearing width. Boundary lubrication is shown on the left side of the curve. Hydrodynamic lubrication is shown on the right side of curve. As you move to the right on the horizontal axis, the effects of increased speed, increased viscosity, or reduced load are seen. The zero point on the horizontal axis corresponds to static friction.

c. The combination of low speed, low viscosity, and high load will produce boundary lubrication. Boundary lubrication is characterized by little fluid in the interface and large surface contact. Boundary lubrication results in very high friction.

d. As the speed and viscosity increase, or the load decreases, the surfaces will begin to separate, and a fluid film begins to form. The film is still very thin, but acts to support more and more of the load. Mixed lubrication is the result, and is seen as a sharp drop in friction coefficient. The drop in friction is a result of decreasing surface contact and more fluid lubrication. The surfaces will continue to separate as the speed or viscosity increase until there is a full fluid film and no surface contact. The friction coefficient will reach its minimum and there is a transition to hydrodynamic lubrication. At this point, the load on the interface is entirely supported by the fluid film. There is low friction and no wear in hydrodynamic lubrication since there is a full fluid film and no solid-solid contact.

e. The curve shows the friction increasing in the hydrodynamic region. This is due to fluid drag (friction produced by the fluid). A higher speed may result in thicker fluid film, but it also increases the fluid drag on the moving surfaces. Likewise, a higher viscosity will increase the fluid film thickness, but it will also increase the drag.

f. Machinery will see boundary lubrication at startup and shutdown (low speeds and thin film), before transition to hydrodynamic lubrication at normal operating conditions (high speeds and thick film). The Stribeck curve shows that a machine will see the most friction and wear during startup and shutdown.

3-3. Hydrodynamic Lubrication.

a. General. In hydrodynamic lubrication, sometimes referred to as fluid film lubrication, the wearing surfaces are completely separated by a film of oil. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a sufficiently high velocity. This type of lubricating action is similar to a speedboat operating on water. When the boat is not moving, it rests on the supporting water surface. As the boat begins to move, it meets a certain amount of resistance or opposing force due to viscosity of the water. This causes the leading edge of the boat to lift slightly and allows a small amount of water to come between it and supporting water surface. As the boat's velocity increases, the wedge-shaped water film increases in thickness until a constant velocity is attained. When the velocity is constant, water entering under the leading edge equals the amount passing outward from the trailing edge. For the boat to remain above the supporting surface, there must be an upward pressure that equals the load.

b. Definition. When a fluid lubricant is present between two rolling and/or sliding surfaces, a thicker pressurized film can be generated by the movement of the surfaces (velocities). The non-compressible nature of this film separates the surfaces resulting in no metal-to-metal contact. The condition in which surfaces are completely separated by a continuous film of lubricating fluid is commonly referred to as hydrodynamic lubrication. In this regime, the lubricant immediately adjacent to each surface travels at the same speed and direction of each surface. As you travel through the width of the film, differential speed graduates and direction is reversed. The thickness of the oil film increases as relative velocity increases between surfaces. Hydrodynamic lubrication does not depend on the introduction of the lubricant under pressure, but it does require an adequate supply of lubricant at all times. The lubricant film thickness must be larger than roughness of the two moving surfaces and any dirt, debris, or wear particles at operating temperatures, speeds, and loads.

c. In Figure 3-2, each surface has tiny asperities or peaks that will contact each other if they are placed together. If one of the surfaces were to slide over the other, then friction would increase, the asperities would break, and the surfaces would wear. In hydrodynamic lubrication, a fluid film separates the surfaces, prevents wear, and reduces friction. The hydrodynamic film is formed when the geometry, surface motion, and fluid viscosity combine to increase the fluid pressure enough to support the load. The increased pressure forces the surfaces apart and prevents surface contact. Therefore, in hydrodynamic lubrication, one surface floats over the other surface. The increase in fluid pressure that forces the surfaces apart is hydrodynamic lift.

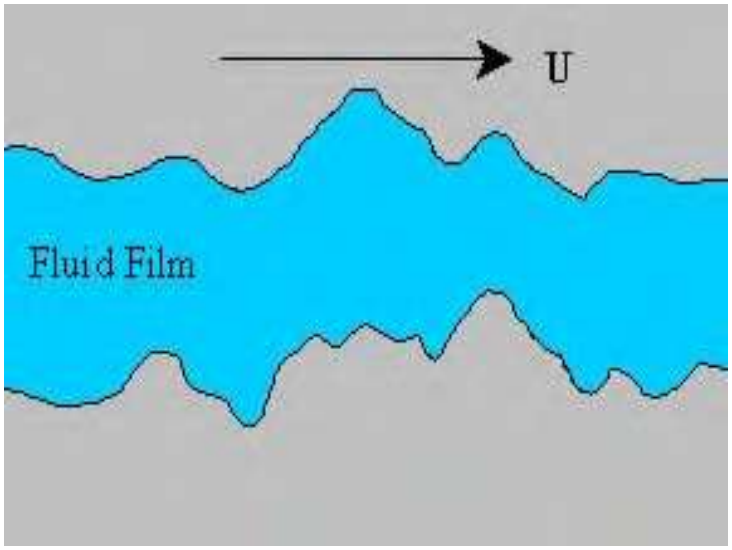

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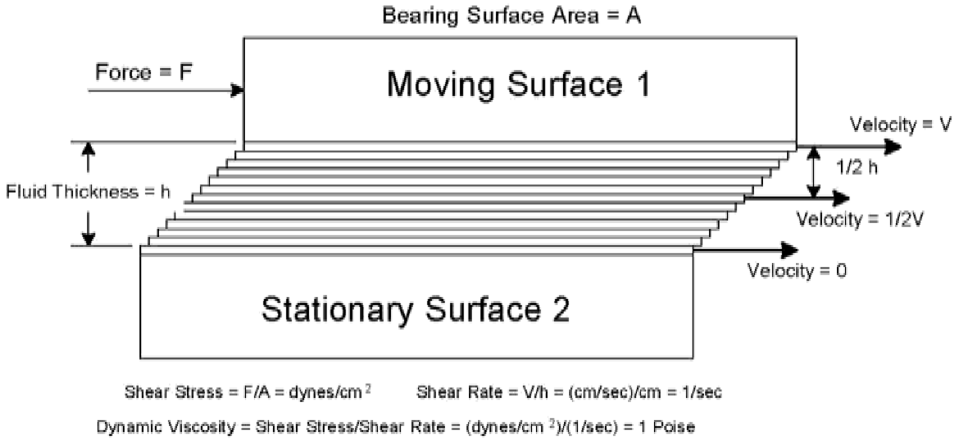
Figure 3-2. Hydrodynamic Lubrication and Fluid Film Thickness.

d. Hydrodynamic lubrication reduces friction between moving surfaces by substituting fluid friction for mechanical friction. To visualize the shearing effect taking place in the fluid film, imagine the film is composed of many layers similar to a deck of cards and assume h = fluid film thickness (Figure 3-3). Then assume that the fluid layer in contact with the moving surface clings to that surface and that they both move at the same velocity. Similarly, assume that the fluid layer in contact with the other surface is stationary. The layers in between then move at velocities directly proportional to their distance from the moving surface. For example, at a distance of ½ h from Surface 1, the velocity would be ½ V. The force F required to move Surface 1 across Surface 2 is simply the force required to overcome the friction between the layers of fluid. This internal friction, or resistance to flow, is defined as the viscosity of the fluid and is one of the most important properties. Viscosity will be discussed in more detail in Chapter 4.

e. Hydrostatic versus Hydrodynamic Lubrication. In heavily loaded bearings such as thrust bearings and horizontal journal bearings, the fluid’s viscosity alone is sometimes not sufficient to maintain a film between the moving surfaces. In these bearings, higher fluid pressures are required to support the load until the fluid film is established. If this pressure is supplied by an outside source, it is called hydrostatic lubrication. Hydrostatic lubrication is obtained by introducing the lubricant into the load-bearing area under a high enough pressure to separate the two surfaces. Alternatively, as discussed above, if the pressure is generated internally, that is, within the bearing by dynamic action, it is referred to as hydrodynamic lubrication. In hydrodynamic lubrication, a fluid wedge is formed by the relative surface motion of the journals or the thrust runners over their respective bearing surfaces. The guide bearings of a vertical hydroelectric generator, if properly aligned, have little or no loading and will tend to operate in the center of the bearing because of the viscosity of the oil.



Source: Bureau of Reclamation FIST 2-4.

Figure 3-3. Fluid Film Lubrication.

f. Factors Influencing Film Formation. Although hydrodynamic lubrication is the ideal situation, in many instances it cannot be maintained. Factors that affect hydrodynamic lubrication include lubricant viscosity, rotation speed (Revolutions per Minute [RPM]), oil supply pressure, and component loading. An increase in speed or viscosity increases oil film thickness. An increase in load decreases oil film thickness. The following factors are essential to achieve and maintain the fluid film required for hydrodynamic lubrication:

(1) The contact surfaces must meet at a slight angle to allow formation of the lubricant wedge.

(2) The fluid viscosity must be high enough to support the load and maintain adequate film thickness to separate the contacting surfaces at operating speeds.

(3) The fluid must adhere to the contact surfaces for conveyance into the pressure area to support the load.

(4) The fluid must distribute itself completely within the bearing clearance area.

(5) The operating speed must be sufficient to allow formation and maintenance of the fluid film.

(6) The contact surfaces of bearings and journals must be smooth and free of sharp surfaces that will disrupt the fluid film.

g. Theoretically, hydrodynamic lubrication reduces wear to zero. In reality, the journal tends to move vertically and horizontally due to load changes or other disturbances and some wear does occur. However, hydrodynamic lubrication reduces sliding friction and wear to acceptable levels.

3-4. <u>Thrust Bearings and Journal Bearings</u>.

a. Thrust Bearings. The principle of hydrodynamic lubrication can also be applied to a more practical example related to thrust bearings used in the hydropower industry (Figure 3-4). These types of bearings are used throughout the hydropower industry. A thrust bearing assembly is also known as tilting pad bearings (Figure 3-5). These bearings support the axial load of the generator and are designed to allow the pads to lift and tilt properly and provide sufficient area to lift the load of the generator. As the thrust runner moves over the thrust shoe, fluid adhering to the runner is drawn between the runner and the shoe causing the shoe to pivot, and forming a wedge of oil. As the speed of the runner increases, the pressure of the oil wedge increases and the runner is lifted as full fluid film lubrication takes place. In applications where the loads are very high, some thrust bearings have high pressure pumps to provide the initial oil film. Once the unit reaches 100% speed, the pump is switched off. Some applications will have the pumps turn back on during shutdown.

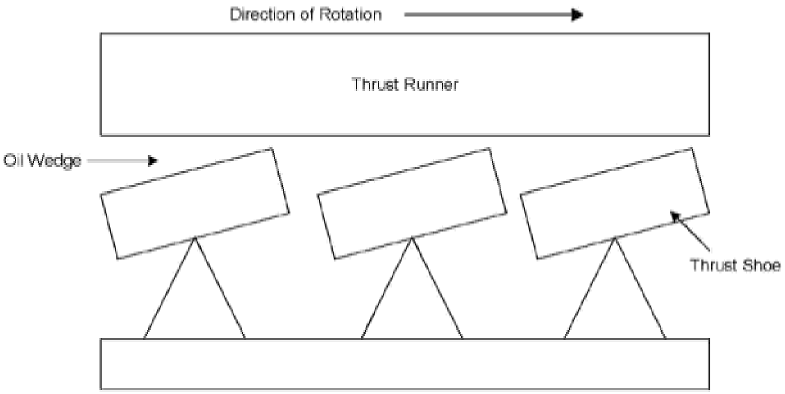


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Figure 3-4. Pivoting Shoe Thrust Bearing.

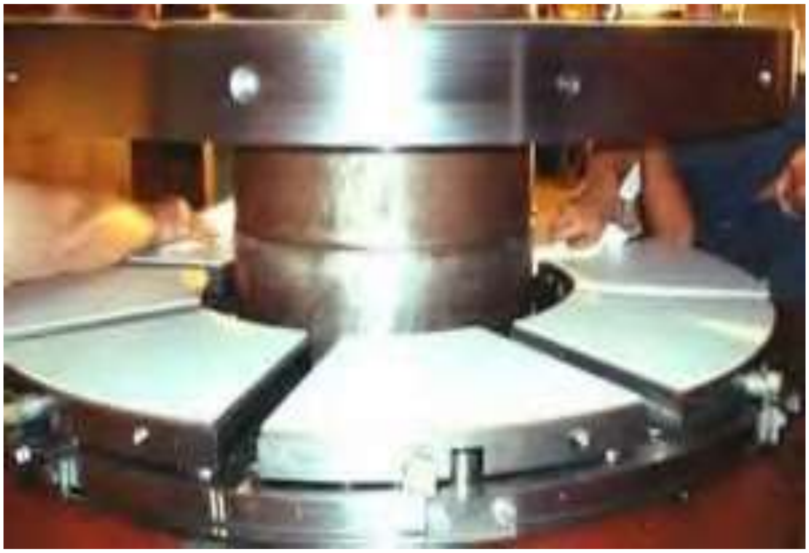

Figure 3-5. Hydropower Tilting Pad Bearings.

b. Journal Bearings. The operation of journal or sleeve bearings is also an example of hydrodynamic lubrication. Before motion begins, the journal rests on the bearing centered on the vertical diameter (Figure 3-6). When the journal rotates, oil adhering to the journal causes a buildup of pressure. As the velocity of rotation increases, this pressure lifts the journal to provide a curved wedge-shaped film that prevents contact between the journal and its bearing. Eventually, velocity becomes constant and the journal rides on a film of oil sufficient to prevent contact with the bearing surface. The journal is not only lifted vertically, but is also pushed to the side by the pressure of the oil wedge. The minimum fluid film thickness at full speed will occur at a point just to the left of center and not at the bottom of the bearing. In both the pivoting shoe thrust bearing and the horizontal journal bearing, the minimum thickness of the fluid film increases with an increase in fluid viscosity and surface speed and decreases with an increase in load. Simplified equations have been developed to provide approximations of film thickness with a considerable degree of precision. Regardless of how film thickness is calculated, it is a function of viscosity, velocity, and load. As viscosity or velocity increases, the film thickness increases. When these two variables decrease, the film thickness also decreases. Film thickness varies inversely with the load. If the load is increased, film thickness decreases. Viscosity, velocity, and operating temperature are also interrelated. If the oil viscosity is increased the operating temperature will increase, and this in turn has a tendency to reduce viscosity. Thus, an increase in viscosity tends to neutralize itself somewhat. Velocity increases also cause temperature increases that subsequently result in viscosity reduction.

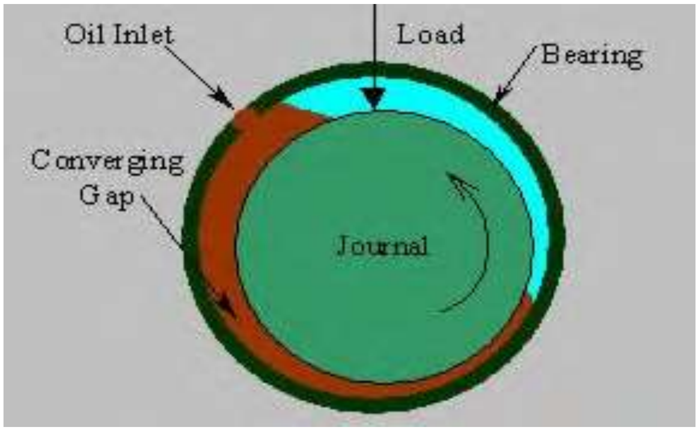

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Figure 3-6. Journal Bearing.

3-5. Boundary Lubrication.

a. Definition of Boundary Lubrication. When a complete fluid film does not develop between two moving surfaces, the film thickness may be reduced to permit contact between wear surface high points or asperities (Figure 3-7). The lambda ratio is discussed below. Boundary conditions occur when λ is less than 1. Boundary lubrication is generally associated with metal-to-metal contact between two sliding surfaces. This condition is characteristic of boundary lubrication and generally friction will be at its greatest in this regime. Boundary lubrication occurs whenever any of the essential factors that influence formation of a full fluid film are missing. One example would be using a lubricant with too low of viscosity. The most common example of boundary lubrication includes bearings, which normally operate with fluid film lubrication, but which experience boundary lubricating conditions during routine starting and stopping of equipment. Other examples include gear tooth contacts and reciprocating equipment. Boundary lubrication occurs when the lubricating film is about same thickness as the surface roughness such that the high points (asperities) on the solid surfaces contact. This is generally an undesirable operating regime for a hydrostatic or hydrodynamic bearing, since it leads to increased friction, energy loss, wear, and material damage. But, most machines will see boundary lubrication during their operating lives, especially during startup, shutdown, and low speed operation. Special lubricants and additives have been developed to decrease the negative effects of boundary lubrication. These are discussed further in Chapter 5. This includes using antiwear and extreme pressure additives.

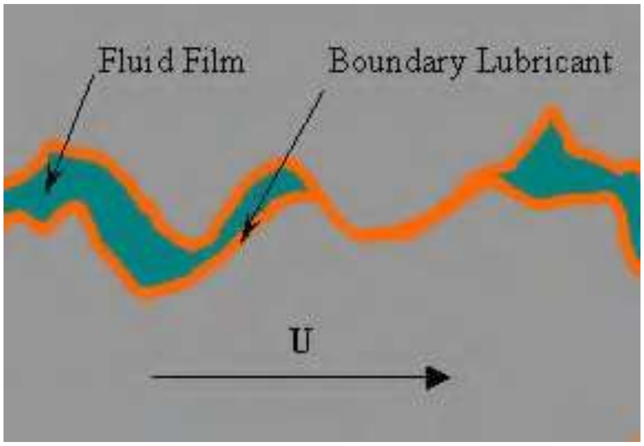

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Figure 3-7. Example of Boundary Lubrication between Two Moving Surfaces — Asperities in Contact between Two Surfaces.

b. Lubricity. Lubricants required to operate under boundary lubrication conditions must possess an added quality referred to as “lubricity” to lower the coefficient of friction of the oil between the rubbing surfaces. Lubricity is the measure of the reduction in friction of a lubricant. It is an oil enhancement property provided through the use of chemical additives known as antiwear (AW) agents. AW agents have a polarizing property that enables them to behave in a manner similar to a magnet. Like a magnet, the opposite sides of the oil film have different polarities. When an AW oil adheres to the metal wear surfaces, the sides of the oil film not in contact with the metal surface have identical polarities and tend to repel each other and form a plane of slippage. Boundary lubricants generally have long, straight, polar molecules, which will readily attach themselves to the metal surfaces. Most oils intended for use in heavier machine applications contain AW agents. Examples of equipment that rely exclusively on boundary lubrication include reciprocating equipment such as engine and compressor pistons, and slow-moving equipment such as turbine wicket gates. Gear teeth also rely on boundary lubrication to a great extent.

3-6. Elastohydrodynamic (EHD) Lubrication.

a. Definition of EHD Lubrication. The lubrication principles that are applied to rolling bodies, such as ball or roller bearings, are known as elastohydrodynamic (EHD) lubrication. Elastohydrodynamic lubrication occurs as pressure or load increases to a level where the viscosity of the lubricant provides a higher shear strength than the metal surface it supports. This regime can occur in roller bearings or gears as the lubricant is carried into the convergent zone approaching a contact area or the intersection of two asperities. As a result, the metal surfaces deform elastically in preference to the highly pressurized lubricant, which increases the contact area and thus increasing the effectiveness of the lubricant. EHD films are present in rolling contacts, such as ball bearings or roller bearings. In this situation, the softer material makes up the rolling element, which actually deforms for a split second to enlarge the contact area between mating surfaces. The oil film thickness under these conditions is very small on the order of 1 micron. Because of the small oil film thickness, bearings are very susceptible to particle and water contamination.

b. Rolling Body Lubrication. Although lubrication of rolling objects operates on a principle considerably different from that for sliding objects, the principles of hydrodynamic lubrication can be applied, within limits, to explain lubrication of rolling elements. Rolling elements typically require much less lubrication than sliding contacts. An oil wedge, similar to that which occurs in hydrodynamic lubrication, exists at the lower leading edge of the bearing (Figure 3-8). Adhesion of oil to the sliding element and the supporting surface increases pressure and creates a film between the two bodies. Because the area of contact is extremely small in a roller and ball bearing, the force per unit area, or load pressure, is extremely high. This is referred to as the Herztian contact or Hertzian stress. Roller bearing load pressures may reach 5000 psi (344.75 bar) and ball bearing load pressures may reach 1,000,000 psi (68,950 bar). Under these pressures, it would appear that the oil would be entirely squeezed from between the wearing surfaces. These extreme pressures greatly affect the viscosity of the lubricant. Viscosity increases that occur under extremely high pressure prevent the oil from being entirely squeezed out. Consequently, a thin film of oil is maintained.

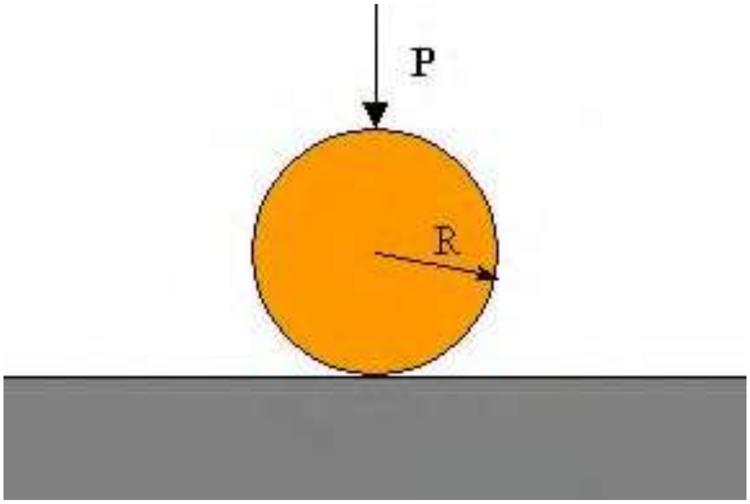

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Figure 3-8. Rolling Body Diagram.

c. Film Thickness Ratio and Effect of Film Thickness and Roughness. The roughness of the wearing surfaces is an important consideration in EHD lubrication. Rough surfaces usually wear more quickly and have higher coefficients of friction than do smooth surfaces. The *Machinery's Handbook*, 29th ed. (Oberg et al. 2012) is also a good resource for this topic.

Roughness is defined as the arithmetic average of the distance between the high and low points of a surface, and is referred to as Ra or roughness profile. This value is also sometimes called the centerline average (CLA). The root mean squared value Rq is also used to denote surface finish. As film thickness increases in relation to roughness, fewer asperities make contact. Engineers use the ratio of film thickness to surface roughness to estimate the life expectancy of a bearing system. This is also referred to as lambda (λ). The lambda ratio can also be used for sliding contacts in addition to rolling contact. The relation of bearing life to this ratio is very complex and not always predictable. In general, life expectancy is extended as the ratio increases. Full film thickness is considered to exist when the value of this ratio is between 2 and 4. Bearings should always be designed for λ greater than 2. Bearing life can be double or tripled as the lambda ratio reaches a value of 4 or more. It is also another reason that bearing surfaces should be designed to be as smooth as possible and kept as clean as possible. When lambda values are greater than 2, fatigue failure is generally due entirely to subsurface stress. However, in most industrial applications, a ratio between 1 and 2 is achieved. The values of λ between 1 and 2 are generally attributed to mixed lubrication discussed below. At these values surface stresses occur and asperities undergo stress and contribute to fatigue as a major source of failure in antifriction bearings. Boundary lubrication occurs when λ is less than 1.

3-7. Mixed Lubrication.

a. Definition. Hydrodynamic lubrication and EHD lubrication can degenerate into a condition called "mixed lubrication." Here the oil film is so thin that some asperities come into contact permitting some metal transfer and wear. Mixed regime can result in excessive wear of contacting surfaces. As noted above, mixed lubrication occurs at values of λ between one and two. Mixed film lubrication occurs as a machine is started up and bearings and shafting gradually come up to speed. Selecting too low of a viscosity of a lubricant can also result in mixed lubrication.

b. Film Thickness. Mixed lubrication occurs between boundary and hydrodynamic lubrication, as the name would suggest. The fluid film thickness is slightly greater than the surface roughness so that there is very little asperity (high point) contact, but the surfaces are still close enough together to affect each other. In a mixed lubrication system, the surface asperities themselves can form miniature non-conformal contacts. Mixed lubrication often occurs in gear sets as the lubrication regime transfers from boundary conditions to hydrodynamic conditions. The gear meshes roll across each other at the pitchline, which falls into the EHD lubrication regime. The gear sets slide across each other above and below the pitchline, which results in both boundary and hydrodynamic lubrication. For bearings, mixed lubrication conditions can result from the following conditions:

- Viscosity that is too low.
- Bearing speed that is too low.
- Bearing that is overloaded.
- Bearing clearances that are too tight.
- Journal and bearing that are not properly aligned.

CHAPTER 4

Lubricating Oil Basics and Production

4-1. Types of Oil.

a. A lubricant is a balanced mix of a number of components. The composition of this mix that a production plant or manufacturer must follow is called the “formulation.” The formulation of a lubricant is made up of base oils and additives that combine to determine behavior when in use, both in terms of performance and duration.

b. Petroleum is a mixture of chemical compounds formed from the remains of ancient animals and plants. It is trapped in porous rock formations and is usually associated with concentrated saltwater. Petroleum generally belongs to the chemical class called hydrocarbons, which are chains of carbon atoms compounded with hydrogen. The number of carbons may range from 1 (in methane gas) to well above 50. Hydrocarbons are further broken into three classes: paraffins, naphthenes, and aromatics.

c. Lubricating oils are produced by a blending process between a base oil and additives. All lubricants start with a base oil. The characteristics of the finished (blended) lubricant is determined by the type of base oil used and additives. Oils are generally classified as refined, synthetic, or biobased. Paraffinic and naphthenic oils are refined from crude oil while synthetic oils are manufactured. Biobased lubricants are also referred to as vegetable or crop-based lubricants and are discussed further in Chapter 13. Mineral oil comes from crude oil and its quality depends on the refining process. There is a grading scale for oil and different applications require different oil quality. Mineral oil is mainly made up of four different types of molecules including paraffinic, branched paraffinic, naphthenic, and aromatic. Paraffinic oils have a long, straight chained structure, while branched paraffinic oils are the same with a branch off the side. These are used mainly in engine oils, industrial lubricants, and processing oils. Naphthenic oils have a saturated ring structure and are most common in moderate temperature applications. Aromatic oils have a non-saturated ring structure and are used for manufacturing seal compounds and adhesives. Synthetic oils are man-made fluids that have identical straight chained structures, much like the branched paraffinic oils. One of the benefits of a synthetic is that the molecular size and weight are constant while mineral oils vary greatly; therefore the properties are very predictable. The distinguishing characteristics between these oils are noted below.

d. Paraffinic molecules tend to form waxy, crystal-like particles at low temperatures and elevate the pour point, the temperature at which an oil ceases to flow. However, paraffinics have a better viscosity index. Naphthenics behave in an opposite manner; they do not form waxy particles as easily and have a lower pour point, but their viscosity index is poor. Viscosity index is defined in this chapter in Paragraph 4-2.b.

e. Paraffinic Oils. Paraffinic oils are distinguished by a molecular structure composed of long chains of hydrocarbons, i.e., the hydrogen and carbon atoms are linked in a long linear series similar to a chain. Paraffinic oils contain paraffin wax and are the most widely used base stock for lubricating oils. They will have a higher resistance to oxidation. In comparison with naphthenic oils, paraffinic oils have a higher pour point, higher viscosity index, low volatility, high flash points, and low specific gravities.

f. Naphthenic Oils. In contrast to paraffinic oils, naphthenic oils are distinguished by a molecular structure composed of "rings" of hydrocarbons, i.e., the hydrogen and carbon atoms are linked in a circular pattern. These oils do not contain wax and behave differently than paraffinic oils. Naphthenic oils are generally reserved for applications with narrow temperature ranges and where a low pour point is required. Naphthenic oils have:

- Lower pour point due to absence of wax.
- Lower viscosity indexes.
- Higher volatility (lower flash point).
- Higher specific gravities.

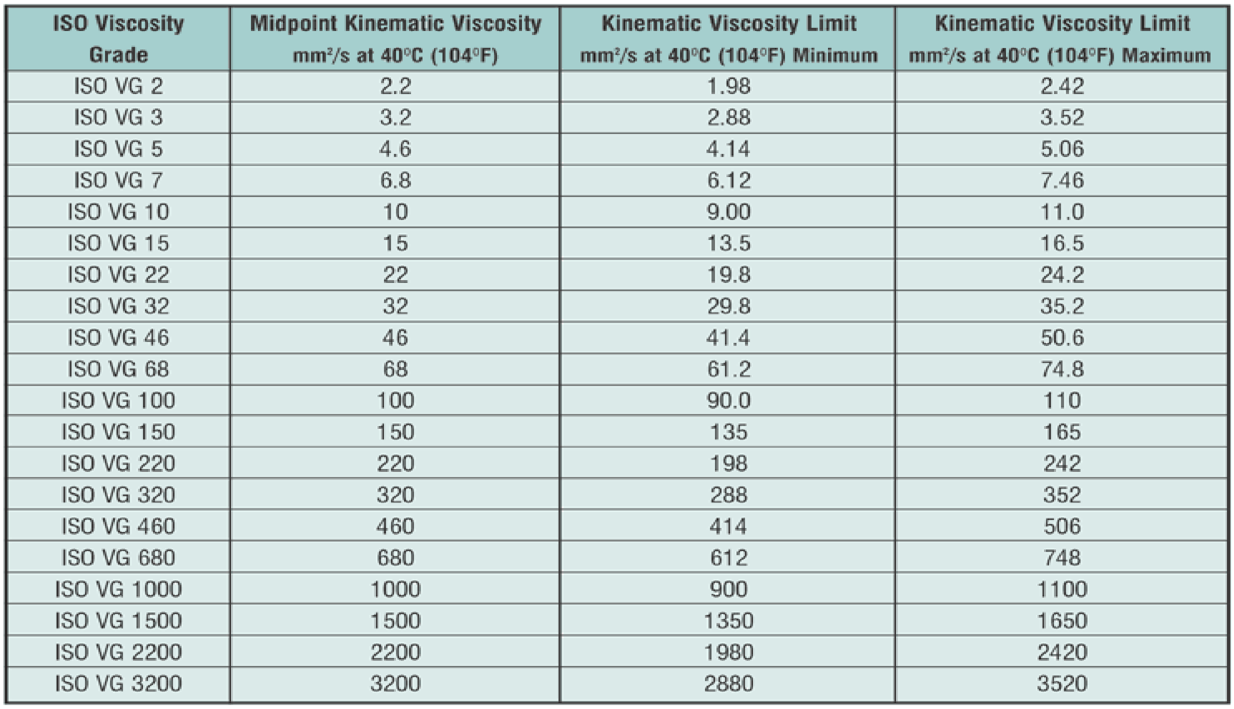
4-2. Properties and Characteristics of Lubricating Oils.

a. Viscosity. Viscosity is the most important property of a lubricant. Technically, the viscosity of an oil is a measure of the oil's resistance to shear. Viscosity is more commonly known as resistance to flow. If a lubricating oil is considered as a series of fluid layers superimposed on each other, the viscosity of the oil is a measure of the resistance to flow between the individual layers. A high viscosity implies a high resistance to flow while a low viscosity indicates a low resistance to flow. Viscosity varies inversely with temperature. Viscosity is also affected by pressure; higher pressure causes the viscosity to increase, and subsequently the load-carrying capacity of the oil also increases. This property enables use of thin oils to lubricate heavy machinery. The load-carrying capacity also increases as operating speed of the lubricated machinery is increased. Two common methods for measuring viscosity are shear and time.

(1) Shear. When viscosity is determined by directly measuring shear stress and shear rate, it is expressed in centipoise (cP) and is referred to as the absolute or dynamic viscosity. Dynamic viscosity can also be considered an oil's internal resistance to flow and shear. It is the measure of how the fluid behaves under pressure, such as a pressurized hydraulic line.

(2) In the oil industry, it is more common to use kinematic viscosity, which is the absolute viscosity divided by the density of the oil being tested. Kinematic viscosity is expressed in centistokes (cSt) and can also be thought of as an oil's resistance to flow and shear by the forces of gravity. Water has a low viscosity of 1 cSt (1 mm^2/s). The kinematic viscosity is related to the time required for a fixed volume of lubricant to flow through a capillary tube at a given test temperature, usually 40 °C (100 °C), under the influence of gravity. Viscosity in centistokes is conventionally given at two standard temperatures: 104 and 212 °F (40 and 100 °C). The viscosity at 104 °F (40 °C) is also an indicator of the International Standards Organization (ISO) grade. For example, an oil with a viscosity of approximately 32 cSt (32 mm^2/s) at 104 °F (40 °C) is an ISO Grade 32 oil. Various organizations have developed grading systems for lubricant viscosity. The ISO Standard 3448 defines various viscosity grades (see Table 4-1). This classification defines 20 viscosity grades in the range of 2 to 320,000 cSt (2 to 320,000 mm^2/s) at 104 °F (40 °C). The Society of Automotive Engineers (SAE) defines viscosity grades for automobile oils. The American Gear Manufacturers' Association (AGMA) also defines viscosity grades. Table 4-1 provides a cross reference.

Table 4-1. ISO Viscosity Grades.



ISO Viscosity Grade	Midpoint Kinematic Viscosity mm^2/s at 40°C (104°F)	Kinematic Viscosity Limit mm^2/s at 40°C (104°F) Minimum	Kinematic Viscosity Limit mm^2/s at 40°C (104°F) Maximum
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.46
ISO VG 10	10	9.00	11.0
ISO VG 15	15	13.5	16.5
ISO VG 22	22	19.8	24.2
ISO VG 32	32	29.8	35.2
ISO VG 46	46	41.4	50.6
ISO VG 68	68	61.2	74.8
ISO VG 100	100	90.0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650
ISO VG 2200	2200	1980	2420
ISO VG 3200	3200	2880	3520

Courtesy of Noria Corporation and Machine Lubrication Magazine.

(3) Time. Another method used to determine oil kinematic viscosity measures the time required for an oil sample to flow through a standard orifice at a standard temperature. Viscosity is then expressed in SUS (Saybolt Universal Seconds). SUS viscosities are also conventionally given at two standard temperatures: 100 and 210 °F (37 and 98 °C). As previously noted, the units of viscosity can be expressed as centipoise (cP), centistokes (cST), or Saybolt Universal Seconds (SUS), depending on the actual test method used to measure the viscosity. Figure 4-1 shows a chart of the viscosities of various types of oils, for comparison.

b. Viscosity Index. The viscosity index, commonly designated VI, is an arbitrary numbering scale that indicates the changes in oil viscosity with changes in temperature. The higher the VI, the less the viscosity will change as a result of a temperature change. VI is an arbitrary number and is calculated from results of the kinematic viscosity at 104 °F (40 °C) and 212 °F (100 °C). VI properties can be enhanced with addition of additives. Viscosity index can be classified as follows: low VI usually < 35; medium VI usually 35 to 80; high VI usually 80 to 110; very high VI usually > 110. A high viscosity index indicates small oil viscosity changes with temperature. A low viscosity index indicates high viscosity changes with temperature. Therefore, a fluid that has a high viscosity index can be expected to undergo very little change in viscosity with temperature extremes and is considered to have a stable viscosity.

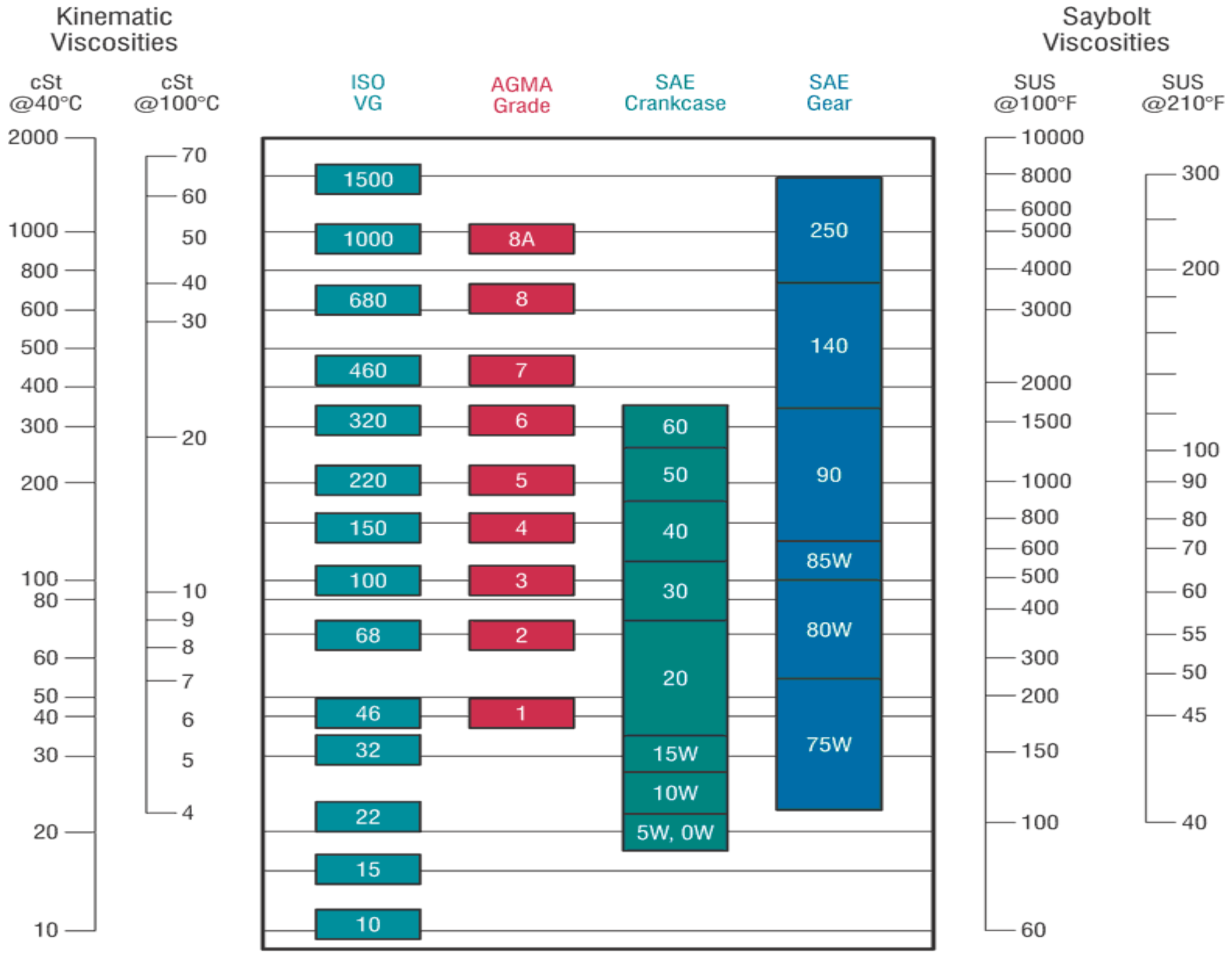


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Figure 4-1. Viscosity Comparison Chart.

A fluid with a low viscosity index can be expected to undergo a significant change in viscosity as the temperature fluctuates. For a given temperature range, say 0 to 100 °F (–18 to 37 °C), the viscosity of one oil may change considerably more than another. An oil with a VI of 95 to 100 would change less than one with a VI of 80. Knowing the viscosity index of an oil is crucial when selecting a lubricant for an application, and is especially critical in extremely hot or cold climates. Failure to use an oil with the proper viscosity index under great temperature extremes may result in poor lubrication and equipment failure. Typically, paraffinic oils are rated at 100 °F (38 °C) and naphthenic oils are rated at 0 °F (–18 °C). Proper selection of petroleum stocks and additives can produce oils with a very good VI.

c. Pour Point. The pour point is the lowest temperature at which an oil will flow. This property is crucial for oils that must flow at low temperatures and in cold weather applications. In paraffinic oils, the pour point is the result of the crystallization of waxy particles. In naphthenic oils, the pour point is the result of the increase in viscosity caused by a decrease in temperature. A commonly used rule of thumb when selecting oils is to ensure that the pour point is at least 50 °F (10 °C) lower than the lowest anticipated ambient temperature.

d. Cloud Point. The cloud point is the temperature at which dissolved solids in the oil, such as paraffin wax, begin to form and separate from the oil. As the temperature drops, wax crystallizes and becomes visible. Certain oils must be maintained at temperatures above the cloud point to prevent clogging of filters.

e. Flash Point and Fire Point. The flash point is the lowest temperature to which a lubricant must be heated before its vapor, when mixed with air, will ignite but not continue to burn. The fire point is the temperature at which lubricant combustion will be sustained. The flash and fire points are useful in determining a lubricant's volatility and fire resistance. The flash point can be used to determine the transportation and storage temperature requirements for lubricants. The flash point is not necessarily the safe upper temperature limit. Instead, it is a relative indication of the fire and explosion hazard of a particular oil. The flash point can also be used as an indication of the evaporation losses that can be expected under high temperature applications. Lubricant producers can also use the flash point to detect potential product contamination. A lubricant exhibiting a flash point significantly lower than normal will be suspected of contamination with a volatile product. Products with a flash point less than 100 °F (37.7 °C) will usually require special precautions for safe handling. The fire point for a lubricant is usually 8 to 10% above the flash point. The flash point and fire point should not be confused with the auto-ignition temperature of a lubricant, which is the temperature at which a lubricant will ignite spontaneously without an external ignition source.

f. Acid Number or Neutralization Number. The acid or neutralization number is a measure of the amount of potassium hydroxide required to neutralize the acid contained in a lubricant. It can be considered the measure of the acidity of an oil and is the amount, in milligrams, of potassium hydroxide (KOH) required to neutralize 1 gram of oil. A relative increase in the neutralization number indicates oxidation of the oil. Acids are formed as oils oxidize with age and service. The acid number for an oil sample is indicative of the age of the oil and can be used to determine when the oil must be changed.

g. Oxidation stability is a chemical reaction that occurs with a combination of the lubricating oil and oxygen. The rate of oxidation is accelerated by high temperatures, water, acids, and catalysts such as copper. The rate of oxidation increases with time. The service life of a lubricant is also reduced with increases in temperature. Oxidation will lead to an increase in the oil's viscosity and deposits of varnish and sludge. The rate of oxidation depends on the quality and type of base oil as well as the additive package used. Some synthetics, such as polyalphaolefins (PAOs), have inherently better oxidation stability than do mineral oils. This improved oxidation stability accounts for the slightly higher operating temperatures that these synthetic oils can accommodate. Generally, oxidation will reduce the service life of a lubricant by half, for every 18 °F (10 °C) increase in fluid temperature above 140 °F (60 °C). Severely hydrotreated base oils also have poor solubility characteristics. Without proper formulation, additives may not remain suspended, and some additive drop out could occur. Oxidation products including varnish and sludge remain in suspension or deposit on surfaces rather than being dissolved in the fluid. Several methods may be used to determine or evaluate the oxidation stability of an oil, which is usually regarded as the number of hours until a given increase in viscosity is noted or until there is a given increase in the acid number (AN).

4-3. Synthetic Oils. Synthetic oils are making up a larger portion of the lubrication market. The use of synthetic oils is increasing due to their popularity with many lubricant end users and marketing by manufacturers. Synthetic lubricants will typically be double or more the cost of petroleum-based lubricants. Synthetic base oils include many different types of compounds; some are limited to a single specific application. PAOs and Polyalkylene Glycols (PAGs) are two products produced in the synthetic base oil market and the most common. Tables 4-2 and 4-3 summarize the most common synthetic lubricants. Detailed performance data and specification data are not provided in this manual for synthetic lubricants. The detailed performance data can be obtained from lubricant manufacturers and from handbooks such as the *Tribology Handbook* (Neale 1995). Synthetic oils are a man-made fluid with scientifically designed molecules with a specific identified structure. As such, fluid properties are very predictable. Synthetic lubricants have greater hydrolytic stability, i.e., the ability to resist chemical decomposition (hydrolysis) in the presence of water.

a. Synthetic lubricants are produced from chemical synthesis rather than from the refinement of existing petroleum or biobased (vegetable) oils. These oils are generally superior to petroleum (mineral) lubricants in most circumstances and usually can extend the life of the lubricant. The primary drawback is the higher cost. Synthetic oils perform better than mineral oils in that they have:

- Better oxidation stability or resistance.
- Better viscosity index.
- Much lower pour point, as low as –50 °F (–46 °C).
- Lower coefficient of friction.
- Better high temperature stability and protection against breakdown.
- Better low temperature viscosity.
- Lower operating temperatures.

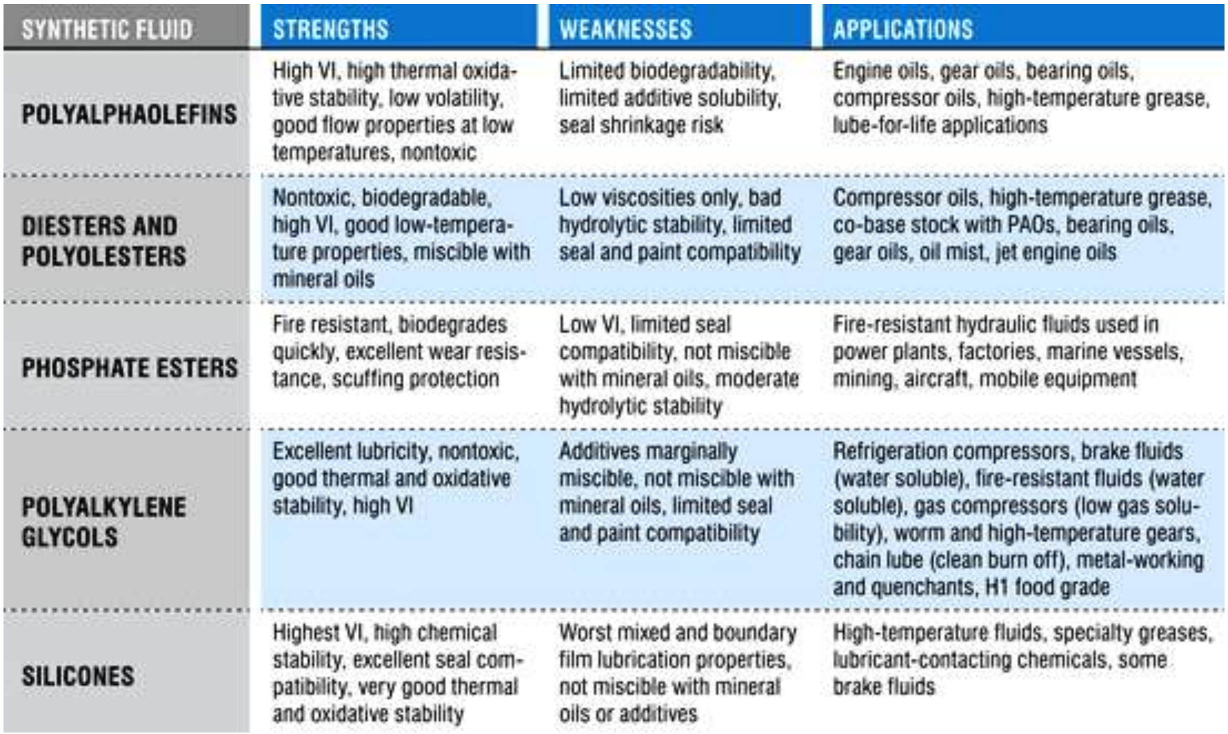
b. The advantages offered by synthetic oils are most notable at either very low or very high temperatures. Good oxidation stability and a lower coefficient of friction permits operation at higher temperatures. The better viscosity index and lower pour points permit operation at lower temperatures. There are two API (American Petroleum Institute) base oil categories that include synthetics. API Group Types are discussed further later in this chapter. The first is API Group IV. The only synthetic base oil included in this group is PAO. The second category is API Group V. These are non-PAO synthetic bases and include diesters, polyolesters, alkylated benzenes, phosphate esters, etc. Some API Group III oils are also marketed as synthetics especially in the automotive industry. These are highly processed crude oils that are very close in property to Type IV oils.

c. The major disadvantage to synthetic oils is their initial cost, which is approximately two times (or more) higher than the cost of mineral-based oils. However, the initial higher cost is usually recovered over the life of the product, which can be about three times longer than conventional lubricants. The higher cost makes it inadvisable to use synthetics in oil systems experiencing leakage.

Table 4-2. Summary of Synthetic Lubricants for Industrial and Automotive Applications.

Type	**Properties**
Synthesized Hydrocarbons	API Group IV - PAOs Polybutenes
Organic Esters	API Goup V Dibasic acid esters (Diesters) Polyol esters
Severely Hydroprocessed	API Group III (some manufacturers classify these as synthetic — API Group Types are discussed later in this chapter)
Others	API Group V PAGs Phosphate esters Silicones

Table 4-3. Synthetic Lubricant Summary.



SYNTHETIC FLUID	STRENGTHS	WEAKNESSES	APPLICATIONS
POLYALPHAOLEFINS	High VI, high thermal oxidative stability, low volatility, good flow properties at low temperatures, nontoxic	Limited biodegradability, limited additive solubility, seal shrinkage risk	Engine oils, gear oils, bearing oils, compressor oils, high-temperature grease, lube-for-life applications
DIESTERS AND POLYOLESTERS	Nontoxic, biodegradable, high VI, good low-temperature properties, miscible with mineral oils	Low viscosities only, bad hydrolytic stability, limited seal and paint compatibility	Compressor oils, high-temperature grease, co-base stock with PAOs, bearing oils, gear oils, oil mist, jet engine oils
PHOSPHATE ESTERS	Fire resistant, biodegrades quickly, excellent wear resistance, scuffing protection	Low VI, limited seal compatibility, not miscible with mineral oils, moderate hydrolytic stability	Fire-resistant hydraulic fluids used in power plants, factories, marine vessels, mining, aircraft, mobile equipment
POLYALKYLENE GLYCOLS	Excellent lubricity, nontoxic, good thermal and oxidative stability, high VI	Additives marginally miscible, not miscible with mineral oils, limited seal and paint compatibility	Refrigeration compressors, brake fluids (water soluble), fire-resistant fluids (water soluble), gas compressors (low gas solubility), worm and high-temperature gears, chain lube (clean burn off), metal-working and quenchants, H1 food grade
SILICONES	Highest VI, high chemical stability, excellent seal compatibility, very good thermal and oxidative stability	Worst mixed and boundary film lubrication properties, not miscible with mineral oils or additives	High-temperature fluids, specialty greases, lubricant-contacting chemicals, some brake fluids

Courtesy of Noria Corporation and Machine Lubrication Magazine.

d. Factors to be considered when selecting synthetic oils include pour and flash points; demulsibility; lubricity; rust and corrosion protection; thermal and oxidation stability; antiwear properties; compatibility with seals, paints, and other oils; and compliance with testing and standard requirements. It is important to note that synthetic oils are as different from each other as they are from mineral oils. Their performance and applicability to any specific situation depends on the quality of the synthetic base oil and additive package. Synthetic oils are not necessarily interchangeable with each other.

e. Synthetic Lubricant Categories. Table 4-3 identifies several synthetic oils and their properties. This can be broken down into five general categories. These categories are listed in Table 4-3 and discussed in the remainder of this paragraph. As noted, PAO synthetics are the most common type.

(1) Synthesized hydrocarbons. PAOs and dialkylated benzenes are the most common types. Of these, PAOs are probably the most common synthetic lubricant and one of the oldest type of synthetic lubricants. As noted, PAO lubricants are classified as API Type IV lubricants. These lubricants provide performance characteristics closest to mineral oils and are compatible with them. PAO lubricants are a synthetic hydrocarbon (made from ethylene gas), but do not contain any wax, sulfur, or nitrogen compounds. This results in a lubricant with a very high viscosity index and a non-polar base oil. Applications include engine and turbine oils, hydraulic fluids, gear, and bearing oils, and compressor oils. Disadvantages include their:

- Limited ability to dissolve some additives.
- Tendency to shrink rubber-based seals and hoses. (Special seal and hose material is needed with these fluids.)
- Requirement for addition of antioxidant additives to resist oxidation.
- Limited properties on boundary lubrication.
- Poor fire resistance.
- Poor biodegradability.

(2) Organic esters. Diabasic acid and polyol esters are the most common types. There are saturated esters and unsaturated esters. Saturated esters are the most applicable to hydraulic systems on USACE navigation structures. The properties of these oils are easily enhanced through additives. Applications include crankcase oils and compressor lubricants.

(3) Diesters: Produced by reaction of an oxygen containing acid with an alcohol. Applications includeType-1 Jet engine oils, compressor oils, and high temperature greases.

(a) Properties include their:

- Composion, which contains no sulfur, phosphorus, waxes, or metals.
- Excellent low temp fluidity (Pour Point in range from -58 to -85 °F (-50 to -65 °C).
- High Viscosity Index (above 140).
- Good lubricating properties (high shear resistance).

- Good thermal & oxidation resistance.
- Lower volatility than mineral oils.
- High solvency (tend to dissolve varnish and sludge deposits).

(b) Some disadvantages include their:

- Poor hydrolytic stability.
- Tendency to shrink rubber-based seals and hoses. Special seal and hose material is needed with these fluids.
- Requirement for addition of antioxidant additives to resist oxidation.
- Limited properties on boundary lubrication.

(4) Polyol esters. Reaction of an acid with an alcohol containing two or more alcohol groups (OH) in its structure. Applications include second generation jet engine oils (Type 11), compressor lubricants, blends for automotive engine oils, and base oil for high temperature range greases.

(a) Some properties include their:

- Composition, which contain no sulfur, phosphorus, waxes, or metals.
- Excellent low temp fluidity. (Pour Point in range from -58 to -94 °F (-50 to -70 °C).
- High Viscosity Index (above 120-160).
- Good lubricating properties (high shear resistance).
- Excellent thermal and oxidation resistance (better than Diesters).
- Lower volatility than mineral oils.
- Better hydrolytic stability than Diesters.
- (Some varieties) biodegradability (i.e., Trimethylolpropane [TMP]).
- Perfume odor.

(b) Some disadvantages include their:

- Tendency to swell seals. (Seal material selection needs to take this into account.)
- Requirement for special additives. (Does not work with additives developed for mineral oils.)
- Tendency to affect paints and finishes.
- Compatibility only with Polyol Esters.

(5) Phosphate esters. Produced by reaction between alcohols and phosphoric acid. These lubricants are suited for fire resistance applications. Applications include: fire resistant industrial hydraulic fluids and hydraulic fluids for aviation systems. They can also be used as wear and friction reducing additive in grease.

(a) Some properties include their:

- Excellent fire resistance.
- Pour Point, which is in the range from -13 to -58 °F (-25 to -50 °C).
- High Viscosity Index (above 120-160).
- Excellent boundary lubrication properties.
- Good thermal stability.
- Fair hydrolytic stability.
- Specific gravity >1. (Water contamination floats on top.)

(b) Some disadvantages of phosphate esters include their:

- Very low VI (ranging from 60 to -30).
- Limited capabilities at higher temperatures.
- Decomposition byproducts are corrosive.
- Poor compatibility with mineral oils.
- Degradation products, which include phosphate soaps that have a black, sludge-like consistency.

(6) Polyalkylene Glycols (PAGs). The terms "polyalkylene glycol" and "polyglycol" are often use interchangeably. These can be water soluble and water insoluble. They have very different properties from each other. PAG can also be used as an environmentally acceptable lubricant. See Chapter 13. Applications include gears, bearings, and compressors for hydrocarbon gases.

(a) Other applications include:

- Fire resistant hydraulic fluids (solubles).
- Hydraulic brake fluids (solubles).
- Metal working fluids (solubles).
- Heat transfer fluids (insolubles).
- Base fluid for high temp. hydraulic fluids and bearing oils (insolubles).
- Screw-type refrigeration compressors and hydrocarbon gases.
- Screw-type compressors handling hydrocarbon gases.

(b) Some properties of PAG lubricants include their:

- Tendency to volatilize rather than form deposits or sludge at high temperatures.
- Wide thermal range and high thermal conductivity.
- High Viscosity Index (150).

- Low solubility with hydrocarbon gases and some refrigerants.
- Good biodegradability and aquatic toxicity.
- Ability to hold large amounts of water (hydraulic fluid) in solution.

(c) Some disadvantages of PAG lubricants include their:

- Poor performance at low temperatures when compared with other synthetics.
- Incompatibility with mineral oils or their additives.
- Incompatibility with PAOs or esters.
- Considerable compatibility risk with paints and finishes.
- Considerable compatibility risk with many types of seals.
- Expense, which can be 6-8 times more costly than mineral oils.

(7) Silicones. Silicone lubricants are chemically inert, nontoxic, fire resistant, and water repellant. They also have low pour points and volatility, good low temperature fluidity, and good oxidation and thermal stability at high temperatures. They can operate over a wide temperature range. Silicones have a polymeric composition with the carbons in the backbone replaced by silicon. Silicones are one of the older types of synthetic fluids. Some applications include high temperature oils and greases, lubricants exposed to radiation and oxygen, specialty hydraulic applications requiring compressible fluids, and brake fluids (limited compressibility).

(a) Some properties include their:

- Very high Viscosity Index (300 and more).
- Chemically inert.
- Nontoxicity.
- Good low temperature fluidity.
- Fire resistant.
- Low volatility.
- Water repellent properties.
- Good thermal and oxidation stability up to high temperatures.

(b) Some disadvantages of silicone lubricants include their:

- High compressibility (low lube properties).
- Oxidation products (silicon oxides), which are abrasive.
- Low surface tension. (They do not form adherent lubricating films.)
- Poor antiwear protection.
- Poor additives response.

4-4. Other Properties of Oils. This section will provide a general discussion of lubricating oils. Turbine oils are specifically discussed in Chapter 9. Hydraulic oils are discussed in Chapter 10. Other properties of lubricating oil that are important, but for which direct measurement of their quantitative values is less significant, are described below.

a. Color. New lubricating oils, including turbine oils, are normally light in color. Oil will gradually darken in service. This is due to oxidation. This is accepted as normal, but oil oxidation should be monitored. However, a significant color change occurring in a short time indicates that something has changed. For example, if oil suddenly becomes hazy, it is probably being contaminated with water. A rapid darkening or clouding may indicate that oil is contaminated or excessively degraded.

b. Foaming Characteristics. Foaming characteristics are measured by American Society for Testing and Materials (ASTM) Test Method D892. This test will show the tendency of oil to foam and the stability of the foam after it is generated. Foaming can result in poor system performance and can cause serious mechanical damage and along with air entrainment can cause oxidation of the oil to occur at a faster rate. Most lubricants contain antifoam additive to break up the foam.

c. Water Content. Lubricating oil should be clear and bright. Most turbine oil will remain clear up to 75 ppm water at room temperature. A quick and easy qualitative analysis of insoluble water in oil is the hot plate test or crackle test. A small amount of oil is placed on a hot plate. If oil smokes, there is no insoluble water. If it spatters, the oil contains free or suspended water. This test is useful to find oils with very high water content generally with a threshold of 500 ppm and above. Moreover, a test that results in an audible spatter can indicate 2000 ppm free or suspended water.

d. Inhibitor Content. The stability of lubricating oil is based on the combination of high quality base stock with highly effective additives. Oxidation inhibitors include phenols and amines. Therefore, it is very important to monitor the oxidation of the oil. ASTM Test Method D2272, “Rotating Pressure Vessel Oxidation Test,” is very useful for approximating the oxidation inhibitor content of the turbine oil. The remaining useful life of the oil can be estimated from this test. Acid number tests are also used to monitor this. Other tests include membrane patch colorimetry. Chapter 9 discusses additional tests for turbine oils.

e. Wear and Contaminant Metals. Quantitative spectrographic analysis of used oil samples may be used to detect trace metals (and silica) and to identify metal-containing contaminants. System metals such as iron and copper can be accurately identified if the sample is representative and if the metals are solubilized or are very finely divided. A high-silica level generally indicates dirt contamination.

f. Percent of Saturates. These are very stable oil molecules. Higher saturates % means oil will resist oxidation and remain in service longer. On the other hand, oils high in saturates are correspondingly low in aromatics.

g. Sulfur Content. Sulfur contaminants can be corrosive to machinery under certain conditions since the presence of sulfur combined with water can lead to the formation of acid.

4-5. Lubricating Oil Degradation. A lubricating oil may become unsuitable for its intended purpose as a result of one or more processes. This discussion provides a general overview. Chapters 9 and 10 discuss turbine oil and hydraulic oil degradation respectively.

a. Oxidation. Oxidation occurs by chemical reaction of the oil with oxygen. The first step in the oxidation reaction is the formation of hydroperoxides. Subsequently, a chain reaction is started and other compounds such as acid, resins, varnishes, sludge, and carbonaceous deposits are formed. Oxidation is the primary limiting factor on remaining usable life for oil. The rate at which oxidation occurs is not steady. Oxidation inhibitors are consumed during the process of oxidation. Oil oxidation rates increase over time, than drastically increase as the last of the remaining oxidation inhibitors are consumed. When nearing this point, the oil is at the end of its useful life.

b. Water Contamination. Water may be dissolved or emulsified in oil. Dissolved water indicates it is below the saturation level. Water contamination can be dissolved and the fluid may look clear at lower water content, below the saturation point. At higher water content levels, above the saturation point, the water is “free” or “emulsified” and the fluid looks milky. Water affects viscosity, promotes oil degradation and equipment corrosion, and interferes with lubrication. The higher the oil temperature, the more water that can be dissolved in the oil. Water in hydraulic oil can compromise oil viscosity, can cause acid levels to increase, and can cause corrosion in system components (such as cylinders, valves, filter housings, pumps). In reservoirs, water can spoil antifoam additives, compress into vapor, can damage oil and internal components, and can compromise system performance. Results of water contamination in fluid systems include:

- Fluid breakdown, such as additive precipitation and oil oxidation.
- Reduced lubricating film thickness.
- Accelerated metal surface fatigue.
- Corrosion.
- Jamming of components due to ice crystals formed at low temperatures.
- Loss of dielectric strength in insulating oils.

(1) Effects of water on bearing life. Studies have shown that the fatigue life of a bearing is greatly reduced by even small amounts of water contained in a petroleum-based lubricant. The *Noria Practical Handbook of Machinery Lubrication* (Noria 2011) documents that an oil with 500 ppm of water reduces the life of a rolling element bearing by 60%. Other studies have shown similar results. This is discussed further in Chapter 15.

(2) Effect of water and metal particles. Oil oxidation is increased in a hydraulic or lubricating oil by the presence of water and particulate contamination. Small metal particles act as catalysts to rapidly increase the neutralization number of acid level. Water has many negative effects including increased oxidation and chemical reactions causing additive loss.

(3) Sources of water contamination. There are a number of sources of water contamination in a lubrication system. This includes heat exchanger leaks, seal leaks, condensation of humid air,

inadequate reservoir covers, or desiccant breathers, and temperature drops changing dissolved water to free water.

(4) Typical oil saturation levels

- Hydraulic: 200 to 400 ppm (0.02 to 0.04%).
- Lubricating: 200 to 750 ppm (0.02 to 0.075%).
- Transformer: 30 to 50 ppm (0.003 to 0.005%).

c. Air Contamination. Air in oil systems may cause foaming, slow and erratic system response, microdieseling, and pump cavitation. In an oil system, air can be present in four forms. This includes dissolved, free, foam, and entrained. Forms of air in oil include:

- Free air (pockets of air in the system).
- Dissolved air (up to 12% in petroleum oils).
- Entrained air (suspended in oil, bubbles typically less than 1 mm diameter).
- Foam (bubbles rise to surface, typically greater than 1 mm diameter).

d. All oils have some dissolved air in them. The two forms of air of most concern are foam and entrained air. Foam consists of air bubbles on the surface of the oil. Entrained air can rise to form foam. Some causes of foam and air entrainment include oil levels that are too high and oil viscosity that is too high. The introduction of air can accelerate oil oxidation and formation of varnishes. Formation of varnish provides seed particles further increasing the formation of bubbles. This causes a self propagating, accelerating degradation process. Foam and air entrainment can cause:

- Foaming.
- Slow system response with erratic operation.
- A reduction in system stiffness.
- Higher fluid temperatures.
- Pump damage due to cavitation.
- Inability to develop full system pressure.

e. Loss of Additives. Two of the most important additives in turbine lubricating oil are the rust and oxidation (R&O) inhibiting agents. Without these additives, oxidation of oil and the rate of rusting will increase. Water will cause antifoam additives to agglomerate and lose effectiveness and be filtered out.

f. Accumulation of Contaminants. Lubricating oil can become unsuitable for further service by accumulation of foreign materials in the oil. The source of contaminants may be from within the system or from outside. Internal sources of contamination are rust, wear, and sealing products. Outside contaminants are dirt, weld spatter, metal fragments, etc., which can enter the system through ineffective seals, dirty oil fill pipes, or dirty makeup oil.

g. Biological Deterioration. Lubricating and hydraulic oils are susceptible to biological deterioration if the proper growing conditions are present. Procedures for preventing and coping with biological contamination include cleaning and sterilizing, addition of biocides, frequent draining of moisture from the system, avoidance of dead-legs in pipes. The *Tribology Handbook* (Neale 1995) is also a good reference document for more details on biological deterioration and contamination of lubricating oils.

4-6. Oil Classifications and Grading Systems. Professional societies classify oils by viscosity ranges or grades. The most common systems are those of the SAE, AGMA, ISO, and ASTM. Other grading systems include ILSAC (International Lubricant Standardization and Approval Committee) and API. For example, the API classifies engine oils based on service grades. The API Service Symbol, also known as the "Donut," is clearly marked on oil packaging when the product meets the service grade performance requirements. Reference: http://www.api.org/certification-programs/engine-oil-diesel-exhaust-fluid/service-categories.aspx

a. The variety of grading systems used in the lubrication industry can be confusing. A specification giving the type of oil to be used might identify an oil in terms of its AGMA grade, for example, but an oil producer may give the viscosity in terms of cSt or SUS. Conversion charts between the various grading systems are readily available from lubricant suppliers. Conversion between cSt and SUS viscosities at standard temperatures can also be obtained from ASTM D2161.

b. API Base Types Oils.

(1) API base type oils with higher numbers are more refined than those with lower number (Table 4-4). The API has established guidelines to define different types of mineral and synthetic base oils. These guidelines or "Groups" define base oils by their physical and chemical properties:

- Group 1: Solvent-Refined mineral oils - Oldest and least refined base oils.
- Group 2: Undergo further refining to increase % of saturates.
- Group 3: Severely hydroprocessed. Technology improves performance characteristics to the point where they can be marketed as synthetics.
- Group 4: PAOs are chemical synthetics called "synthesized" hydrocarbons.
- Group 5: Man-made synthetics (diesters, polyol estes, phosphate esters, silicones, polyglycols).

(2) Table 4-4 lists saturates by percent. These are very stable oil molecules. Higher saturates percentage means oil will resist oxidation and remain in service longer. Sulfur content is also noted. Sulfur is a contaminant that can be corrosive to machinery under certain conditions. Hydroprocessed and hydrocracking are discussed in the paragraph below under "Oil Refining."

Table 4-4. API Group Types.

Group	**Description**	**Saturate wt %**	**Sulfur wt %**	**Viscosity Index**
I	Solvent-Refined Mineral Oil	< 90	> 0.03	80 to < 120
II	Hydroprocessed	≥ 90	≤ 0.03	≥ 80 to ≤ 120
III	Hydrocracked	≥ 90	≤ 0.03	≥ 120
IV	PAOs	100	0	≥ 130
V	All others — other synthetics			≥ 130

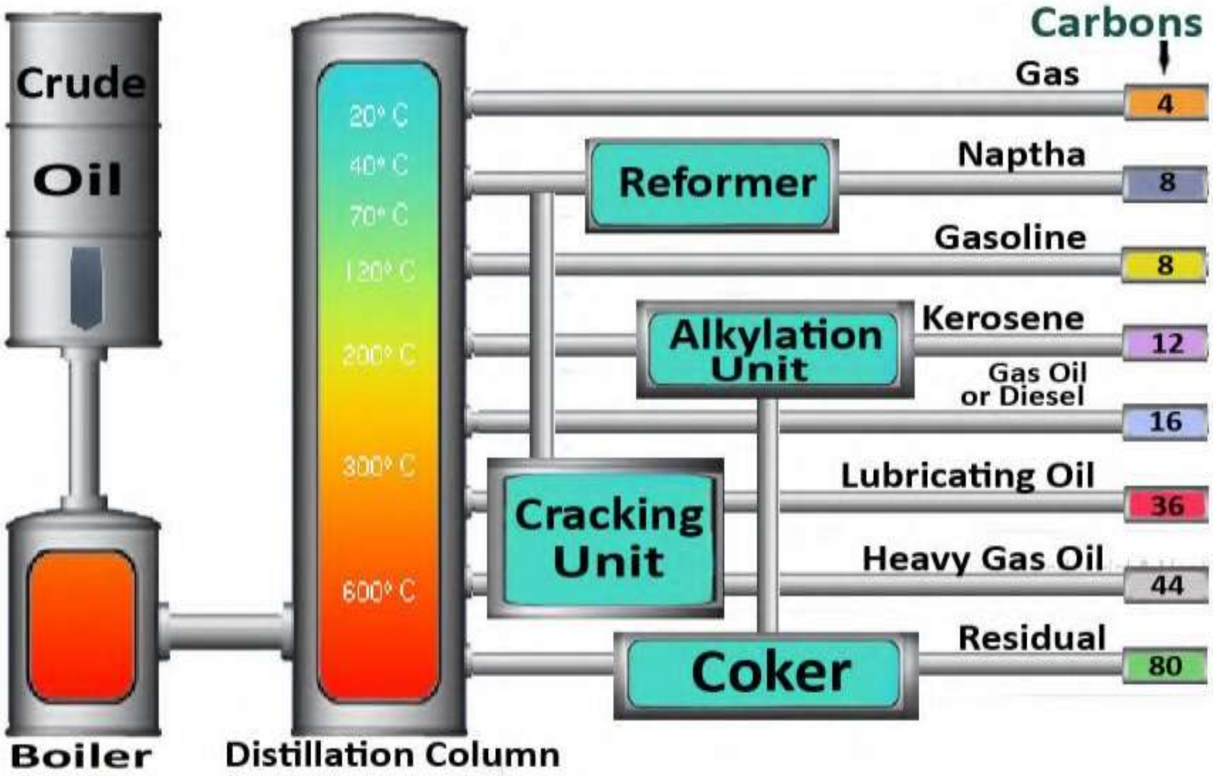
4-7. Oil Refining.

a. Mineral oils are obtained from distillation of crude petroleum. Due to the wide variety of petroleum constituents, it is necessary to separate petroleum into portions (fractions) with roughly the same qualities. The refining process (Figure 4-2) separates crude oil into the following basic components:

- Natural gas liquids.
- Gasoline.
- Naphtha.
- Kerosene.
- Gas oils.
- Lubricating base oils.
- Paraffin.
- Asphalt.
- Coke.

(1) Crudes are segregated and selected depending on the types of hydrocarbons in them. Starting from petroleum crude oil, the typical process for making a lubricant base oil is:

- Separation of lighter boiling materials, such as gasoline, diesel, etc.
- Distillation to give desired base oil viscosity grades.
- Selective removal of impurities, such as aromatics and polar compounds.
- Dewaxing to improve low temperature fluidity.
- Finishing to improve oxidation resistance and heat stability.



Courtesy of Noria Corporation and Machine Lubrication Magazine.

Figure 4-2. Typical Refining Process.

(2) The selected crudes are distilled to produce fractions. A fraction is a portion of the crude that falls into a specified boiling point range. Due to the wide variety of petroleum constituents, it is necessary to separate petroleum into fractions with roughly the same qualities. This is accomplished by distillation which is discussed further below.

(3) Each fraction is processed to remove undesirable components. The processing may include:

- Solvent refining to remove undesirable compounds.
- Solvent dewaxing to remove compounds that form crystalline materials at low temperature.
- Catalytic hydrogenation to eliminate compounds that would easily oxidize.
- Clay percolation to remove polar substances.

(4) The various fractions are blended to obtain a finished product with the specified viscosity. Additives may be introduced to improve desired characteristics. Chapter 5 of this manual discusses the various types of and uses for additives.

b. Separation into Fractions. Separation is accomplished by a two-stage process: crude distillation and residuum distillation. There are two distinct stages in distillation. The first stage produces raw gasoline, kerosene, and diesel fuel. The second stage involves distilling the portion of the first stage that did not volatilize (the residuum). Lubricating oils are obtained from the residuum.

(1) Crude distillation. In the first stage, the crude petroleum is mixed with water to dissolve any salt. The resulting brine is separated by settling. The remaining oil is pumped through a tubular furnace where it is partially vaporized. The components that have a low number of carbon atoms vaporize and pass into a fractionating column or tower. As the vapors rise in the column, cooling causes condensation. By controlling the temperature, the volatile components may be separated into fractions that fall within particular boiling point ranges. In general, compounds with the lowest boiling points have the fewest carbon atoms and compounds with the highest boiling points have the greatest number of carbon atoms. This process reduces the number of compounds within each fraction and provides different qualities. The final products derived from this first stage distillation process are raw gasoline, kerosene, and diesel fuel.

(2) Residuum distillation. The second stage process involves distilling the portion of the first stage that did not volatilize. Lubricating oils are obtained from this portion, which is referred to as the residuum. To prevent formation of undesired products, the residuum is distilled under vacuum so it will boil at a lower temperature. Distillation of the residuum produces oils of several boiling point ranges. The higher the boiling point, the higher the carbon content of the oil molecules in a given range. More importantly, viscosity also varies with the boiling point and the number of carbon atoms in the oil molecules.

c. Impurity Removal. Once the oil is separated into fractions, it must be further treated to remove impurities, waxy resins, and asphalt. Oils that have been highly refined are usually referred to as “premium grades” to distinguish them from grades of lesser quality in the producer’s line of products. However, there are no criteria to establish what constitutes a premium grade.

d. Solvent Refining. Solvent-refined oils will typically have very poor biodegradability and a low viscosity index. Developed over 70 yrs ago, this process attempts to remove the undesirable components by solvent extraction. Initially, light oils such as gasoline, diesel, etc. are separated from crude petroleum by atmospheric distillation. The resulting material is charged to a vacuum distillation tower, where lubricant fractions of specific viscosity ranges are taken off. These fractions are then treated individually in a solvent extraction tower. A solvent furfural is mixed with them and extracts about 80% of the aromatic material present. After reducing the aromatic content, the solvent extracted lube fraction is dewaxed by chilling to a low temperature, which removes much of the wax and so improves the low temperature fluidity of the product. Finally, the dewaxed lube fractions are sometimes finished to improve their color and stability, depending on the application requirements. One common method of finishing is mild hydrofinishing. Hydrofinishing process is used to remove remaining impurities (sulfur, nitrogen) and aromatics, and to make molecule size more uniform. Solvent-refined oils will usually have high aromatic content, often 20-30% and an aniline point less than 108 °F (42 °C).

e. Hydrocracking. Advances in refining technology and an industry push for extended oil changes and long life lubricants has resulted in hydrocracked oils that have similar performance characteristics to those of PAO synthetics, but at a cost that is closer to traditional solvent-refined base stocks. Hydrocracked base oils biodegrade more quickly than solvent-refined oils and will have a higher viscosity index. Hydrocracking is a reaction of crude oil and hydrogen at high temp (approx. 1000 °F [537.7 °C]), and pressure (approximately 1000 psi [68.95 bar]) in presence of a catalyst. Contaminants are eliminated and majority of aromatics are converted to paraffins. Some characteristics of severely hydrocracked oil are that it:

- Is 99.99 % saturated.
- Has an aromatic content much lower than Group 1. Its aniline point is greater than or equal to 108 °F (42 °C).
- Has no heavy metals content.
- Has virtually no aromatics.
- Has a low pour point.
- Has better oxidation resistance.
- Exhibits performance similar to synthetics.