PDHonline Course M427 (6 PDH)

Engineering Tribology

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INTRODUCTION:

Tribology is defined as the 'science and technology of interacting surfaces in relative motion and of related subjects and practices'; it deals with every aspect of 1.) Friction, 2.) Wear, 3.) Lubrication and 4.) Adhesion. This term is derived from the Greek word 'tribos' (τρίβοσ) meaning 'rubbing' or to rub. Figure one below will illustrate two types of movement; sliding and rolling, commonplace with a great number of mechanical and electromechanical devices. These sliding and rolling surfaces represent the key to much of our technological society and understanding tribological principles is essential for the successful design of machine elements. When two nominally flat surfaces are placed in contact with each other, surface roughness causes contact to occur at discrete contact spots; thus, interfacial adhesion occurs. Friction is defined as the resistance to motion experienced whenever one solid body moves over another. Wear is defined as surface damage or removal of material from one or both solid surfaces during moving contact. Materials, coatings and surface treatments are used to control friction and wear. One of the most effective means of controlling friction and wear is by proper lubrication, which provides smooth running and satisfactory life for machine elements. Lubricants can be solid or gaseous.

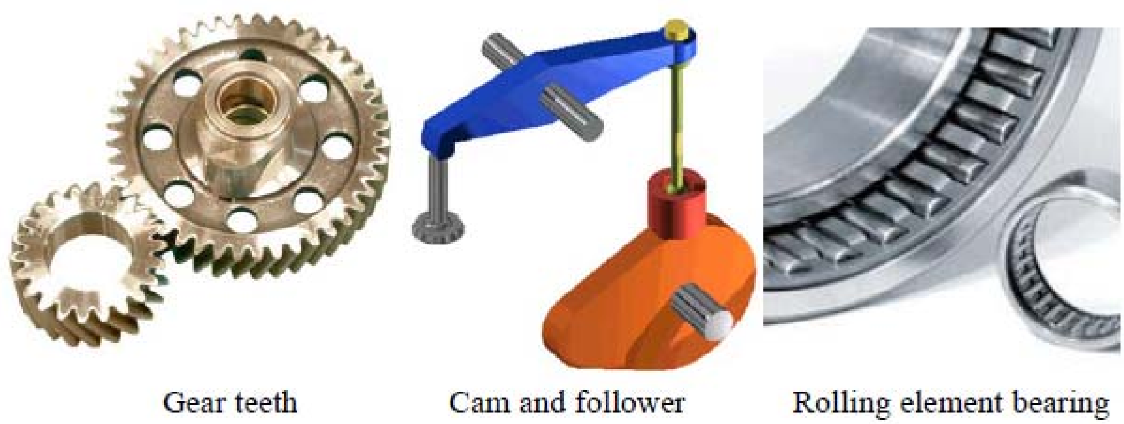

FIGURE 1: SLIDING AND ROLLING MOVEMENT OF PAIRED COMPONENTS

One goal of every designer should be bringing about the transmission of mechanical power with the lowest possible friction losses and with minimal wear of mating surfaces. Even with this being the case, on average, only one hour of instruction over a four year curriculum is taught to mechanical engineering students, relative to the subject. One would think a subject of such importance would be given more recognition during classroom work, but that is definitely not the case in the United States, Canada or Western European countries. The benefits of doing so are appreciable, and we will now take a look at those benefits.

BENEFITS FROM UNDERSTANDING TRIBOLOGY PRINCIPLES:

There is definitely an industrial significance for understanding of tribology. According to some estimates, losses resulting from friction and wear amount to approximately six percent (6%) of the GNP (Gross National Product) in the United States alone. This amounts to approximately $200 million per year. It has been estimated that one-third of the world's energy resources appear as friction in one form or another. According to Dr. Peter Jost, the United Kingdom could save approximately £ 500 million per year by employing better tricological practices. The following table and chart will indicate just where he feels those savings might be expected.

SOURCES	AMOUNT Million £ (POUNDS)
Reduced Maintenance & Replacement	230
Fewer Breakdowns(Increased Reliability)	110
Longer Machine Life	100
Reduced Frictional Dissipation	25
Savings in Investment	15
Savings in Lubricants	5
Savings in Manpower	5

TABLE 1: SAVINGS RELSULTING FROM PROPER UNDERSTANDING AND APPLICATION OF TRIBOLOGY

These values may be seen in graphic form with the representation given below:

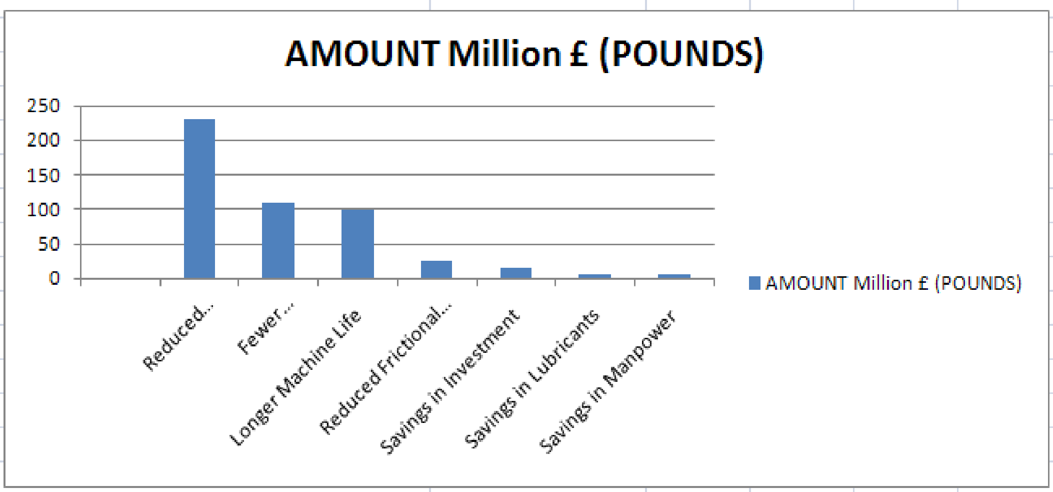

FIGURE 2: SAVINGS RELSULTING FROM PROPER UNDERSTANDING AND APPLICATION OF TRIBOLOGY

A very similar classification of savings will exist for the United States. These two representations give estimated saving, but please note there is a definite benefit due to increased reliability; an improvement in meantime to failure (MTTF) and meantime between failure (MTBF) of moving parts. Figure 3 below will indicate possible savings postulated by manufacturers in the United States. Having been in manufacturing for over 40 years, I can definitely state the 20% savings on maintenance and repair is a viable number. Proper maintenance, on a continuous basis, can extend the life of mechanical parts.

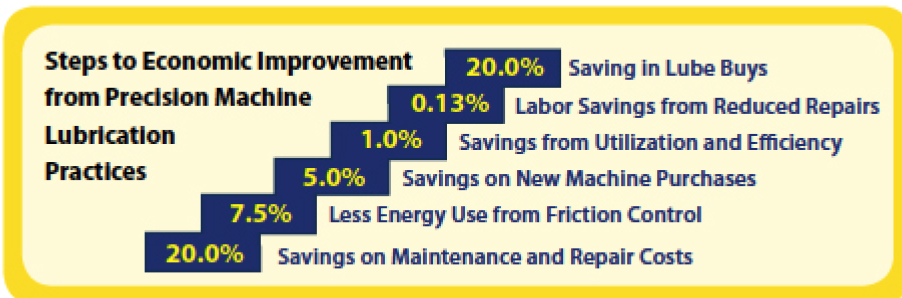

FIGURE 3: H. PETER JOST'S PROJECTED ECONOMIC BENEFITS ASSOCIATED WITH IMPROVED LUBRICATION DESIGN AND PRACTICE.

You can see the potential for cost reduction and improved reliability are significant. Educating the design engineer relative to the study and benefits of tribology can bring good things to the overall design when moving machine elements are needed. It is worth the effort, and more time should be spent at the university level finding solutions to the problem.

HISTORY OF TRIBOLOGY:

There is absolutely no doubt that ever since our ancestors began dragging loads over the ground, they sought methods to lessen friction. There are wall paintings in Mesopotamia and Egypt that depict the transport of huge stone blocks from quarries to the "job site". The very earliest attempt at mitigating friction was cutting trees, laying them down, positioning the load, and then rolling that load to its final destination. This worked well but required back-breaking effort to keep moving the "rotating members" from the rear of the load to the front of the load and in the direction of the desired motion. Later in the evolutionary process, some enterprising individual designed the wheel and eventually got the idea that two wheels were better than one, particularly when stability was desired. Records show the use of wheels dating from 3500 BC. Carts and wagons came next. This fact illustrates our ancestors' concern with reducing friction in translationary motion.

Chinese pictographs, from the second millennium, have been found showing wheeled chariots and carts carrying a variety of loads, i.e. grain, bricks, reeds, etc. We know that each wheel was outfitted with bronze bearings, greased with animal fats and tallow. More useful and less menacing were thrust bearings used in grinding wheels for the production of grain and potter's wheels needed to fashion clay-wear for cooking and holding water. Some of the most illuminating accounts of the implementation of tribological ideas can be found in the writings of Marcus Vitruvius Polio, the Roman architect and engineer, who lived in the first century A.D. His writings were lost during the fall of the Roman Empire but rediscovered in 1920 at Lake Nemi, which is 30 kilometers from Rome.

It is also known that many ideas and inventions seem to have originated in Asia before being introduced into Europe. One such invention was the mechanical clock. It was probably the outstanding engineering achievement during the time period between A.D 400 to 1450. The bearings and escapements must have been "state-of-the-art" for that period of time, and we know that lubricants were used to lessen friction and wear.

The problems of friction and wear were of concern to the greatest mind of the sixteenth century, Leonardo de Vinci (1452—1519). We know from his notebooks, which contain more than 5,000 pages of notes and sketches, that he was definitely aware of the roll friction and wear play when considering rotating and moving mechanisms. Sketches from de Vinci show designs of ingenious rolling-element bearings that form the basis for our modern rotating mechanisms. He deduced the laws governing the motion of a rectangular block sliding over a flat surface. He introduced, for the first time, the concept of the coefficient of friction as the ratio of the friction force to normal load. His work had no historical influence because his notebooks remained unpublished for hundreds of years.

In 1699 the French physicist Guillaume Amontons rediscovered the laws of friction after studying dry sliding between two flat surfaces. Two very important concepts were developed by Amontons. First, the friction force that resists sliding at an interface is directly proportional to the normal load. Secondly, the amount of friction force does not depend upon the apparent area of contact. The middle portion of the 17th century was called the "age of reason" and during this period of time the scientific methods of investigation were developed. Attempts to quantify the laws of friction and wear were undertaken. This fact is substantiated by the formation of the Royal Society in England and the Academie Royal des Sciences in France. These societies provided organization and structure necessary to continue research and development in several areas including friction, wear and lubrication. By the close of the century, the laws of friction had been postulated by Sir Isaac Newton. His work became the foundation for a field of study we now know as fluid mechanics.

The concept of viscosity was postulated by Claude Navier and occurred approximately 150 years after Newton did his work on resistance to flow and fluid mechanics. One of the most comprehensive studies of friction during the early part of this period was undertaken by Charles Coulomb in 1785. His work was concerned with practical tribological problems in naval and military matters. Much of his best work was to "fit" various empirical equations to observations. These, for the first time, distinguish between the effects of adhesion and that of deformation. He added a third law that states that friction force is independent of velocity when motion starts. He also made the distinction between static friction and dynamic friction. Other great men of science concerned with friction, wear and lubrication were Robert Hooke, Beauchamp Tower, Osborne Reynolds, Heinrich Hertz, John Theophilius Desanguliers, Leonard Euler, D.B. Hardy and N.P. Petroff. These pioneers brought tribology to a standard, and its laws still apply to many engineering problems today.

As western mechanization continued, there occurred a significant increase in the wear and failure of mechanical devices and substantial loss of money. It became increasingly difficult to keep a manufacturing operation running smoothly and down-time represented a very real problem to plant managers and CEOs. This trend was recognized by specialists involved in the subjects of friction, wear and lubrication. Numerous papers on various facets of these subjects were presented. However, it was not until October 1964 that a Conference on Iron and Steel Works Lubrication, organized by the Lubrication & Wear Group of the Institution of Mechanical Engineers and the Iron and Steel Institute, revealed the magnitude of the problem and its occurrence on an international scale. The situation called for more and better education and for coordinated research on a national scale. We have mentioned before that Dr. H. Peter Jost was the individual most instrumental in initiating and continuing research on the subject of Tribology. A photograph of Dr. Jost is given below:

FIGURE 4: DR. H. PETER JOST

As with any endeavor, Dr. Jost did not work alone. As a result, the founding fathers of Tribology are often considered to be the gentlemen in the following photograph:

FIGURE 5: THE FOUNDING FATHERS OF MODERN TRIBOLOGY

Like most endeavors in any field of science, the progress is evolutionary and not revolutionary. Discovery takes time but the discoveries in today's world occur much faster due to the availability of marvelous research tools such as the scanning tunneling microscope (STM), the Atomic Force Microscope (AFM) and Friction Force Microscope (FFM). These devices have brought about the field of nano-tribology and micro-triboloby with both concentrating on the microscopic properties of friction, wear, lubrication and adhesion. The advent of these devices has led to a remarkable advancement in the field of study with significantly improved understanding of the basic forces involved with applications. We wish now to better define the subject matter. Please note, a survey course such as this can only offer a condensed treatment of this fascinating subject. I do hope to provide enough information to allow further study and give a basic understanding of the principles involved.

FRICTION, WEAR, LUBRICATION AND ADHESION:

I think it is very important to start with fundamental definitions of friction, wear, lubrication and adhesion. In this fashion, we will be grounded and any misunderstanding relative to these important areas of study will be eliminated. It will become evident as to how these factors produce cause-effect relationships.

FRICTION:

Friction is a dissipative process resulting from the relative motion of media in contact with each other. Continued energy input is required to sustain this relative motion. Generally, this motion can be expressed as a combination of sliding (linear displacement tangential to the contact plane) rolling (angular displacement with respect to a tangential axis) and spin (angular displacement with respect to the normal axis). The work expended against friction is often redundant; that is, it makes no useful contribution to the overall operation of the device of which the bodies are a part, and ultimately must be dissipated as waste heat. Consequently, in most tribological designs, our aim is to keep these frictional forces as small as possible. All machine surfaces are rough, relatively speaking, and even the most finely prepared surfaces will demonstrate asperities that induce friction. The real area of contact is made up of a large number of very small regions of contact. These regions are called asperities or junctions of contact. This is where atom-to-atom contact takes place. The “official” definition of asperity is as follows: **asperity, in tribology, a protuberance in the small-scale topographical irregularities of a solid surface.**

The force of static friction between two sliding surfaces is strongly dependent upon the real area of contact. Figure 6 below is a very crude representation of the profile between mating surfaces:

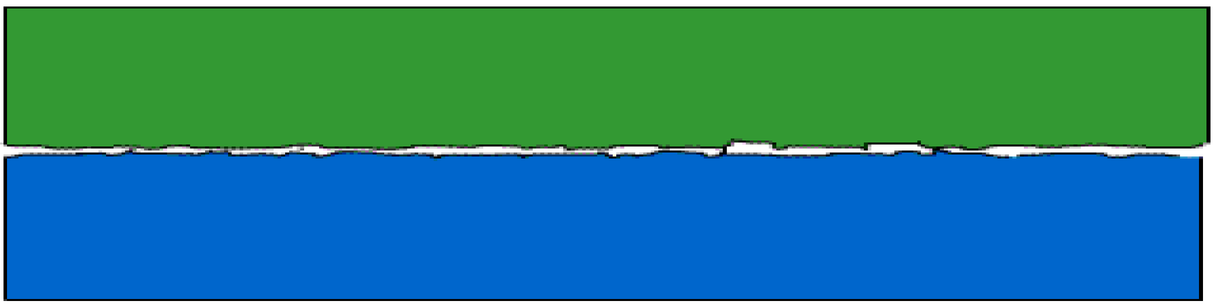

FIGURE 6: ASPERITIES OF MATING MACHINES SURFACES

Another representation is given by Figure 7 below.

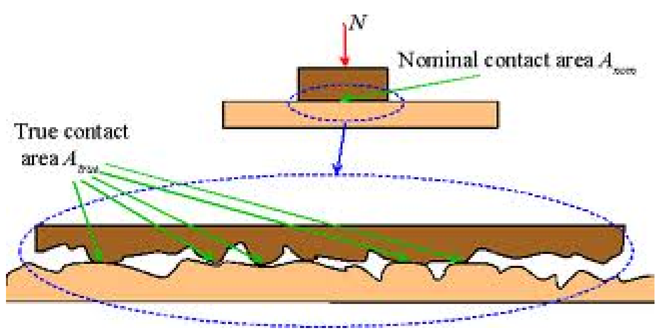

FIGURE 7: TRUE CONTACT AREA AND ASPERITIES OF MATING MACHINED SURFACES

Figure 8 below shows an actual micro-graph of asperities for one machined surface made from tool steel. The unit of measurement is micro-meters, so you can see we are discussing protrusions, ridges and valleys, with very small dimensions.

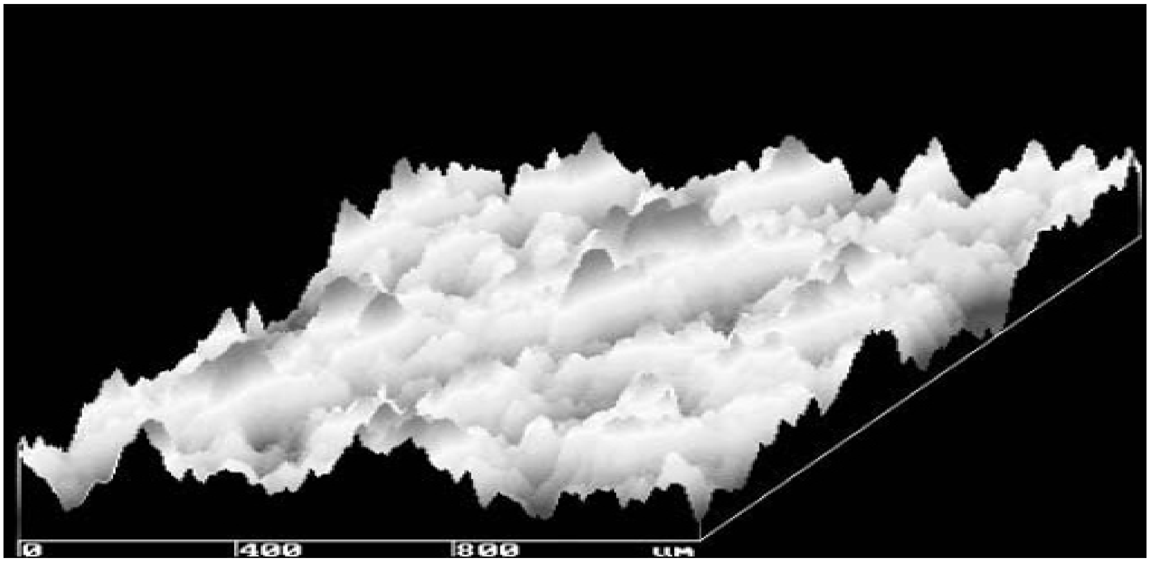

FIGURE 8: ASPERITIES OF MACHINED SURFACE USING ELECTRON MICROSCOPE

Generally, one surface is static (stationary) and one surface is dynamic (moving). This is certainly the case with mechanical members such as indexing slides and bearings. There are several very interesting observations regarding friction; stated as follows:

OBSERVATIONS:

FRICTION

- is essentially an electrostatic force between two surfaces
- never initiates motion; it only responds to motion
- depends on the materials in contact with each other. The coefficient of friction, μ, is a critical property of the materials selected.
- depends on the net force normal pressing the two surfaces in contact (W)
- acts parallel to the surfaces that are (or might have the potential to be) moving with respect to each other
- opposes the direction of motion
- is independent of the area of the surfaces in contact. (First postulated by Coulomb.)
- static friction > kinetic friction > rolling friction for the same combinations of surfaces
- when two surfaces are slipping across each other in the presence of kinetic friction, heat is generated and mechanical energy is not conserved
- when a ball rolls (static friction) without slipping across a surface, mechanical energy is conserved and no heat is generated
- is not dependent upon the surface roughness, or at least surface roughness has a very modest effect on frictional forces.
- is dependent upon the material on both surfaces. Even minute quantities of moisture on the surfaces can reduce friction by 20% to 30%. If there is a layer of grease on the surfaces, friction can be cut by a factor of 10.

COEFFICIENT OF FRICTION FOR VARIOUS MATERIAL PAIRS:

The first of Amonton's laws states that the friction force "F" between a pair of loaded sliding surfaces is proportional to the normal load "W" or "N" that pair carries. The tangential force required to slide one surface over the other is proportional to the weight of the surface. If the mass of one surface (weight on the planet earth) is doubled, the force required to initiate or maintain the sliding motion is doubled. This law is represented by the following equation:

$$F_f = \mu N$$

EQUATION 1: MATHEMATICAL DEFINITION OF FRICTION.

In this equation, μ is the coefficient of friction. Please note, this value varies depending upon the material for each individual component in the pair. This fact is highlighted with the table given in the Appendix to this document. Please notice from the table that there is a difference between the values for the coefficient of friction for static and sliding or kinetic friction. We are going to discuss static vs. kinetic friction later on in this section.

Let us very quickly look at several other conditions dealing with two and three dimensional friction to make sure we are grounded relative to weight, force and the coefficient of friction. Please take a look at the following:

TWO DIMENSIONAL:

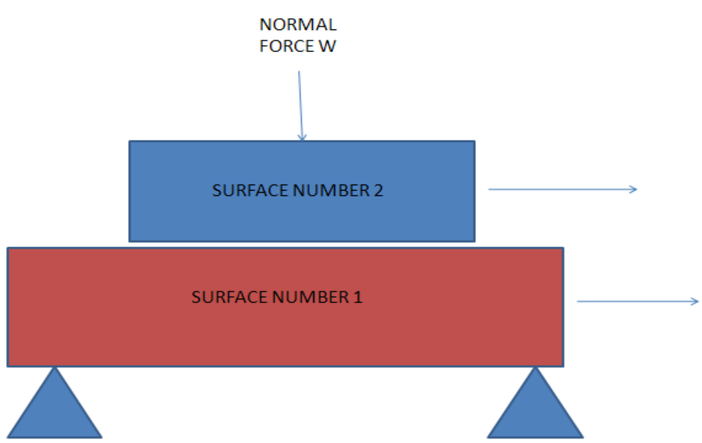

FIGURE 9: SIGNIFICANCE OF FORCE, WEIGHT AND THE COEFFICIENT OF FRICTION

The figure above represents two blocks or two surfaces in which surface number 2 presses down upon surface number 1. A horizontal force is applied to surface number 2 in order to initiate movement. Let us now draw a free-body diagram of the forces applied to the assembly. Again, surface number 1 is static; surface number 2 is dynamic or moving.

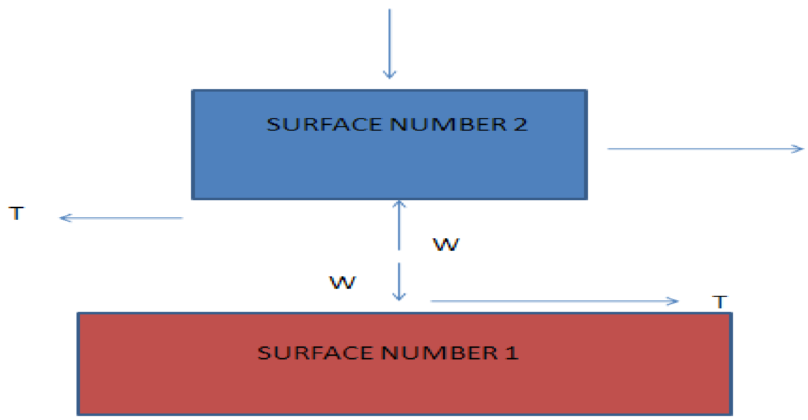

FIGURE 10: TWO-DIMENSIONAL SLIDING COMPONENTS

From observation, we can state the following:

i.) If the two contacting surfaces do not slide, then T(absolute) < μW

ii.) The two surfaces start to slip if T(absolute) = μW

iii.) If the two surfaces are sliding, then T(absolute) =±μW

Please note the sign for these formulas must be selected so that T opposes the direction of motion. Also, as we have discussed previously, μ is called the coefficient of friction for the two materials in contact with each other. For most engineering problems, 0<μ<1. Again, actual values for the coefficient of friction may be seen for various materials from the appendix.

STATIC AND KINETIC FRICTION:

Many textbooks define two different friction coefficients; i.e. static (μs) and dynamic or kinetic (μk). It is definitely true that, for some materials and some material pairs, the static coefficient of friction can be somewhat higher than the coefficient for kinetic friction. This behavior is by no means universal, and in any case, the difference between μs and μk is quite small (on the order of 0.05). The real reason to distinguish between static and kinetic friction is to provide a simple explanation for slip-stick oscillations between two contacting surfaces. When one component of a mating pair slides for a while, sticks, then jumps into motion again—it is called a slip-stick oscillation. If μs and μk were constant, this occurrence would be impossible.

THREE DIMENSIONAL:

We must take a look at a three-dimensional problem because three-dimensional contacts are more complicated. The basic methodology is identical, but the tangential force can have two components. To better describe this mathematically, we introduce {e(1), e(2) and e(3)} with e(1) and e(2) being in the plane of the contact and e(3) being normal to the plane of the contact.

Please look at Figure 12. You will see we have selected as our coordinates e(1)—"X" axis; e(2)—"Y" axis and e(3)—"Z" axis. The tangential force T(1/2) exerted by body (1) on body (2) is then expressed as a component in this basis; i.e.

Again, from Figure 12, we see that the forces opposing motion are the friction forces.

$$T(1/2) = T(1)e(1) + T(2)e(2)$$

EQUATION 2: THREE DIMENSIONAL CONTACT

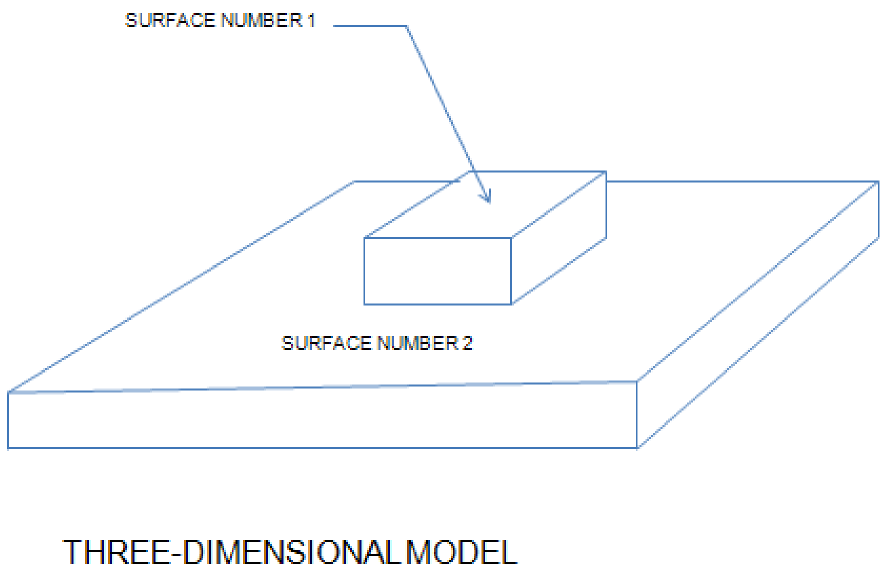

FIGURE 11: THREE DIMENSIONAL MATH MODEL

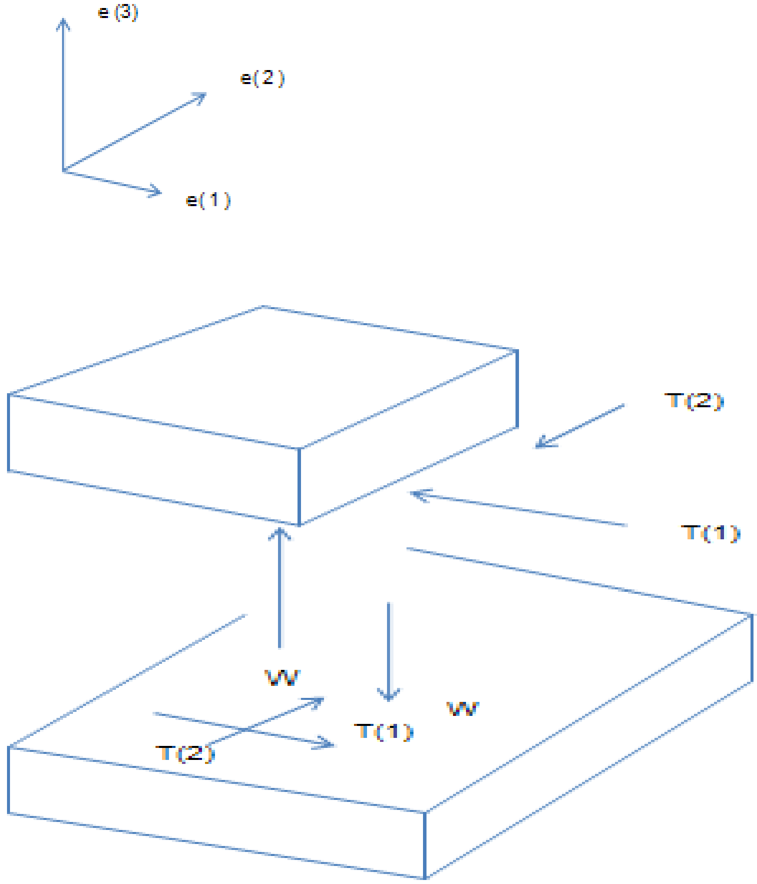

FIGURE 12: THREE DIMENSIONAL MATH MODEL (2)

OBSERVATIONS:

i.) If the two contacting surfaces do not slide, then

[T(1)^2 + T(2)^2]^0.50 <µ W **EQUATION 3: SURFACES DO NOT SLIDE**

ii.) The two surfaces will not start to slip if

[T(1)^2 + T(2)^2]^0.50 = µ W **EQUATION 4: SURFACES WILL NOT SLIP**

iii.) If the two surfaces are sliding, then

[T(1/2) = µ W (V12/|V12| **EQUATION 5: SURFACES SLIDING**

Note: The symbol ^ indicates a quantity raised to a power.

Where T(1/2) denotes the tangential force exerted by body one on body two and V12 is the relative velocity of body one with respect to body two at the point of contact. The relative velocity can be computed from the velocities V(1) and V(2) of the two contacting solids, using the equation:

$$V12 = V(1) - V(2)$$

EQUATION 6: RELATIVE VELOCITY

THERMAL PROCESSES IN FRICTION:

As the asperities in contacting surfaces rub against each other, the dissipated energy transforms into heat. Thermal energy thus produced subsequently diffuses into the mass of both contacting materials. The resulting heating affects the mechanical properties of the materials as well as their micro-structural characteristics. This heating effect may create a very unsafe condition as the temperature exceeds the design limits of the part relative to the material used. Knowledge of these temperatures is certainly desirable and will preclude premature failure of the component and improved performance relative to MTBF and MTTF. If a hard material with a single asperity slides over a smooth or softer surface, a steady state condition may quickly result and heating may be determined through measurements or calculation. A troublesome situation may result when temperature "spikes" occur that drive the components above their design limits. This can anneal the material and actually alter the microstructure. When this happens, reliability and product life are lessened.

WEAR:

Wear is progressive damage, involving material loss, which occurs on the surface of a component as a result of its motion relative to adjacent working parts. Wear is an almost inevitable companion of friction. The economic consequences of wear are widespread and pervasive. They not only involve the costs of downtime but the cost of replacement parts, lost production, and the consequences of lost business opportunities. Wear is customarily noted by "ω" and is defined by the volume of material lost from the wearing surface per unit sliding distance. Its dimensions are consequently, length. For a dry or unlubricated surface, the factors for wear rate are: normal load, the relative sliding speed, the initial temperature of the sliding pair, and mechanical properties of the materials involved. There are many physical mechanisms that can contribute to wear and certainly no simple and universal mathematical model is applicable to all situations. Elements such as entrained dirt, moisture, corrosion of surfaces, de-lamination of component material, etc. can definitely contribute to wear. The Archard wear equation states that ω is directly proportional to the load W on contact but inversely proportional to the surface hardness "H" of the wear material or:

$$\omega = K\,(W/H)$$

FORMULA 7: ARCHARD EQUATION DEFINING WEAR

This mathematical equation indicates that surface hardness is definitely a factor when considering materials for sliding or rotating surfaces.

The dimensionless constant K is known as the wear coefficient. Knowledge of its value is obviously vital in any attempt to apply the above equation in any predictive fashion. Typical values of the dimensionless wear coefficient may be seen as follows:

MATERIALS	COEFFICIENT K
Mild Steel	7×10^{-3}
α/β Brass	6×10^{-4}
PTFE	2.5×10^{-5}
Copper-Beryllium	3.7×10^{-5}
Hard Tool Steel	1.3×10^{-4}
Ferritic Stainless Steel	1.7×10^{-5}
Polyethene	1.3×10^{-7}
PMMA	$7 \times 10\text{-}_{6}$

TABLE 2: COEFFICIENT OF WEAR (ω) FOR VARIOUS MATERIALS

Tables do exist for the approximations of "K" values. As you can see, values do vary, sometimes considerably. Please consult these prior to calculating wear values.

There are two fairly simple classifications of wear; i.e. mild and severe. These are not really based upon mathematical models but observation alone. Distinction between mild and several wear may be seen as follows:

MILD WEAR	SEVERE WEAR
Results in extremely smooth surfaces--often smoother than original surfaces	Results in rough, deeply torn surfaces much rougher than the original surfaces
Debris extremely small, typically only 100nm in diameter	Large metallic wear debris, typically up to 0.01nm in diameter
High electrical contact resistance little true metallic contact	Low contact resistance , true metallic junctions formed

TABLE 3: MILD VS SEVERE WEAR FOR CONTACTING SURFACES

MECHANISMS OF WEAR:

There are several recognizable mechanisms for wear; these are as follows:

SEIZURE:

When metal surfaces are brought into contact with each other, the actual area over which they touch is a comparatively small fraction of the nominal contact area. The high normal pressures generated at these asperity contacts forge metallic junctions which, when sheared by the load tangential to the interface, can grow until the actual area of metallic contact approaches the nominal area. Some degree of seizure can occur. This is where slip-stick oscillations are most prevalent.

MELT WEAR:

Localized melting of the uppermost layer of the wearing solid is always a possibility. At very high velocities, the coefficient of friction can drop, eventually to very low values, as a film of liquid metal forms at the interface. This acts in the same way as a hydrodynamic lubrication film. The heat generated by viscous work in such a melt lubrication situation continues to melt more solid so the wear rate can be very high despite the fact that the coefficient of friction is low.

OXIDATION-DOMINATED WEAR:

Wear may be accelerated by corrosion (oxidation) of the rubbing surfaces.
Increased temperature and removal of the protecting oxide films from the surface during the friction promote the oxidation process. Friction provides continuous removal of the oxide film followed by continuous formation of new oxide film. Hard oxide particles removed from the surface and trapped between the sliding/rolling surfaces additionally increase the wear rate by three body abrasive wear mechanism. This is one reason some lubricants and greases have chemical components that act as rust inhibitors.

MECHANICAL WEAR PROCESSES:

At sliding velocities below 0.10 meters per second, surface heating is negligible and the effect of frictional force is principally to deform the metal surface, shearing it in the sliding direction and ultimately causing the removal of material, usually in the form of small particles of wear debris.

RUNNING-IN:

When mass produced lubricated machine components are run together for the first time, their ultimate load-carrying capacity is often much less than would be the case if they had been preconditioned by running together for an initial period of time at a comparatively light load. This period of time is called running-in or breaking-in. During this regime, the wear rate is often initially high, but as the surfaces become smoother and the more prominent asperities are lost or flattened, the wear rate falls. After a suitable time, the full service conditions can be applied without any sudden increase in wear rate and the steady low-wear rate is maintained for the operational life of the component.

ADHESIVE WEAR:

Adhesive wear is a result of micro-junctions caused by welding between opposing asperities on the rubbing surfaces of the counter-bodies. The load applied to the contacting asperities is so high that they deform and adhere to each other forming micro-joints. The motion of the rubbing counter-bodies result in rupture of the micro-joints. The welded asperity ruptures in the non-deformed (non-cold worked) regions. Thus some of the material is transferred by its counter-body. This effect is called scuffing or galling. When considerable areas of the rubbing surfaces are joined during the friction, a seizure resistance (compatibility) or seizure of one body by the counter-body may occur.

Several factors can lower adhesive wear. These are as follows:

- Lower load.
- Harder rubbing materials.
- Removing contaminates between rubbing surfaces.
- Presence of solid lubricants.
- Presence of lubricating oil.
- Anti-wear additives in oil.

ABRASIVE WEAR:

Abrasive wear is damage to the surface of a component which arises due to the relative motion between those surfaces. Harder asperities or perhaps hard particles trapped at the interface of two surfaces create the wear model. These hard particles may have been introduced between the two softer surfaces as a contaminant from the outside environment or formed due to oxidation or some other chemical process.

Abrasive wear can occur when a harder material is rubbing against a softer material.

- If there are only two rubbing parts involved in the friction process the wear is called **two body wear**. In this case the wear of the softer material is caused by the asperities on the harder surface.
- If the wear is caused by a hard particle (grit) trapped between the rubbing surfaces it is called **three body wear**. The particle may be either free or partially embedded into one of the mating materials.

In the micro-level, abrasive action results in one of the following wear modes:

- **Ploughing**. The material is shifted to the sides of the wear groove. The material is not removed from the surface.
- **Cutting**. A chip forms in front of the cutting asperity/grit. The material is removed (lost) from the surface in the volume equal to the volume of the wear track (groove).
- **Cracking (brittle fracture)**. The material cracks in the subsurface regions surrounding the wear groove. The volume of the lost material is higher than the volume of the wear track.

The following figure will help demonstrate the mechanisms:

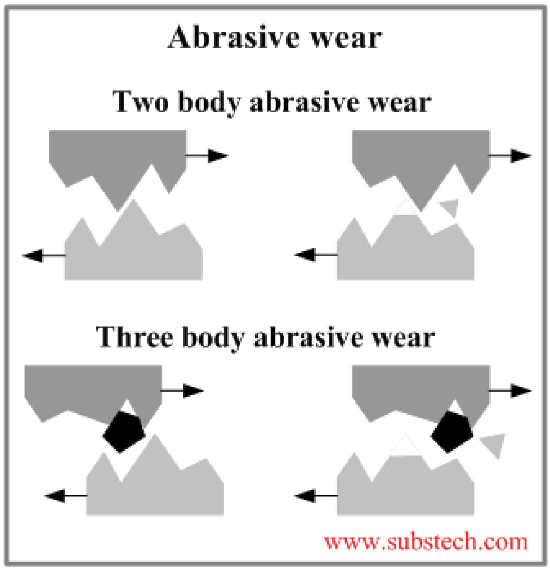

FIGURE 13: TWO- BODY AND THREE BODY WEAR MECHANISMS

FRETTING AND CORROSION WEAR:

Fretting wear is a phenomenon that can occur between two surfaces which have a relative oscillatory motion of small amplitude, usually only a few tens of microns. The main characteristic of a fretting contact in ferrous material pairs is the appearance of reddish-brown debris made up of particles of the hard oxides of iron. These can act as a grinding paste or lap producing highly polished patches on the fretted contact.

EROSIVE WEAR:

Erosive wear is the process involving the removal of material by the impingement of particles, usually at high velocities, on component surfaces. Erosive wear is caused by impingement of particles (solid, liquid or gaseous), which remove fragments of materials from the surface due to momentum effect.

Erosive wear of engine bearings may also be caused by cavitation in the lubrication oil. The cavitation voids (bubbles) may form when the oil exits from convergent gaps between the bearing and journal surfaces. The oil pressure rapidly drops, providing conditions for the formation of voids when the pressure is lower than the oil vapor pressure. The bubbles (voids) then collapse producing a shock wave, which removes particles of the bearing material from the bearing surface.

Erosion can be brought about deliberately by shot-blasting using iron or sand particles but, when encountered unintentionally in service, is often extremely deleterious. Severe erosion can be present during the transport of powders or slurries, the impact of dust particles on the blades of turbo-machinery, and the operation of fluid bed combustors. The particles do not have to be solid. The operation of water droplets on rotating machinery can impart major damage.

LUBRICATION:

A lubricant is any substance that reduces friction by creating a slippery film between two surfaces. Lubricants permit one surface to move easily over the other surface. The dynamic oil film thickness must always be greater than the heights of the combined surfaces in order to avoid frictional energy losses. The ideal condition would be an oil film that is three to five times thicker than the height of the combined surfaces. Component suppliers provide formulas and standardized tools that are useful in establishing minimum viscosity operating requirements. Reputable suppliers provide engineering support to their customers to help refine lubricant selections, and for most applications, the first run selections are not difficult.

Lubricants also provide protection from corrosion, dissipation of heat, exclusion of contaminants, and flushing away of wear products. A good lubricant is expected to have high film strength, chemical, thermal and mechanical stability, and corrosion prevention properties. These and other specialized properties are provided in modern lubricants by various refining techniques and additives.

The requirement of a lubricant for rolling element bearings are often more severe than realized. In a rolling element bearing there are conditions of both rolling and sliding with extremely high contact pressures. The lubricant must withstand high rates of shear and mechanical working not generally prevalent in other mechanical components. For these reasons, proper attention to equipment lubrication is vital from design to operation.

The representation of a lubricant doing its job may be seen as follows:

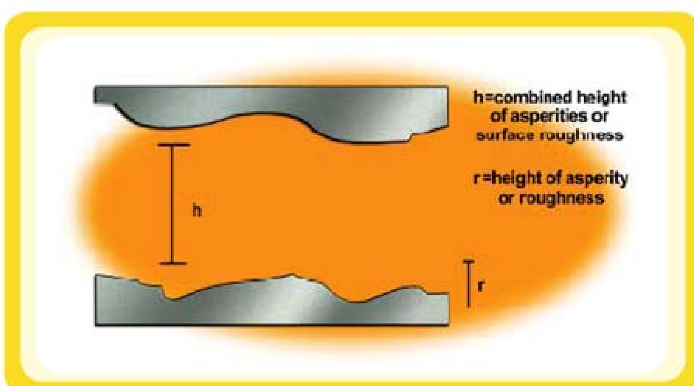

FIGURE 14: LUBRICATION OF TWO INTERACTING SURFACES

Inadequate film conditions occur as a consequence of changes in load, changes in machine operating temperatures, changes in lubricant condition (particularly contamination with gases or fluids), and accidents in lubricant handling and application, which lead to viscosity errors. These condition changes often occur simultaneously, resulting in film collapse, machine component interaction and greatly increased frictional resistance.

METHOD OF LUBRICATION:

OIL-BATH LUBRICATION: The conventional oil-bath system for lubricating ball bearings is satisfactory for low and moderate speed applications. Because this type of system is non-circulating, the static oil level should never be higher than the center of the lowest ball in the bearing being lubricated. A greater amount of oil can cause churning, increase the fluid friction within the bearing and result in excessive operating temperatures. Unless the running level of the oil is known, oil level should be checked only when equipment is dormant as the running level can drop considerably below the static level, depending on the speed of the application.

OIL-SPLASH LUBRICATION: This system of lubrication is used primarily in gear cases where the bearing and gear lubricant are common. The lubrication of bearings in a gearbox, other than one of low speed, is usually not critical because the oil splash from gear teeth is sufficient to lubricate the bearings. Because of constant problems with oil-carrying wear debris, the use of filters and magnetic drain plugs is very helpful in reducing contamination of the bearing.

CIRCULATING-OIL LUBRICATION: This type of system utilizes a circulating pump to assure a positive supply of lubricant to the bearing and is generally used in low to medium speed, heavy-duty bearing applications, as in power transmission equipment. The flow path of the oil in this system is important because a bearing churning a captive volume of oil can generate temperatures capable of causing lubricant breakdown and bearing damage. Due to the inherent possibility of contamination from wear debris in heavy duty applications, suitable oil filters and magnetic drain plugs are necessary to prevent damage to the bearings.

OIL-MIST LUBRICATION: Oil-mist lubrication systems are used in high speed, continuous operation applications. This system permits close control of the amount of lubricant reaching the bearings. The oil may be metered, atomized by compressed air and mixed with air, or it may be picked up from a reservoir using a venture effect. In either case, the air is filtered and supplied under sufficient pressure to assure adequate lubrication of the bearings. Control of this type of lubricating systems is accomplished by monitoring the operating temperatures of the bearings being lubricated.

VISCOSITY:

The classic definition of viscosity is as follows**: 'The measurement of a substance's resistance to flow.** Viscosity is one of the most important factors to consider when selecting a lubricant. To a mechanical engineer, the absolute or dynamic viscosity η is a measure of the resistance a fluid offers to the relative shearing motion where η is defined as the shearing force in the direction of flow between two parallel planes. The importance of this fluid property was suggested by Newton in the seventeenth century: thus, fluids under conditions of uniform temperature and pressure, can be characterized as "Newtonian" when they respond in accordance to the following equation:

$$\eta = \tau / \gamma$$ **EQUATION 8: DYNAMIC VISCOSITY**

where η is the shearing forces n the direction of flow, τ is the shear stress (PSI) and γ is the shear strain. The units for viscosity are generally named poise, after the French doctor Poiseuille who studied the flow of water through very small diameter glass tubing. Water at 20 ° C has a dynamic viscosity of about one hundredth of a poise or 1 centipoise, i.e. cP, which is the unit of measurement for viscosity. 1 cP = 1 x 10^-12 poise = 1 x 10^-3Pa s = 1 mPa s. Lubricating oils have typical viscosities one hundred times that of water or 100 cP.

The kinematic viscosity of a fluid is defined as follows:

Kinematic viscosity = absolute viscosity / density

The viscosity of a fluid depends upon the "local" values for temperature and the "local" pressure. Both of these variables are very important in determining the viscosity of the lubricant separating sliding or rotating mechanical components. A very important classification of lubricant viscosities may be made by using the Viscosity Index or VI. That index is defined as follows: A scale used to classify the viscosity of industrial oils. The Viscosity Index (VI) measures the rate of change of a substance's viscosity in relation to a change in temperature. The higher the number, the smaller the viscosity change which means the better the oil protects the surfaces. The number does not indicate the actual viscosity in high and low temperature extremes of the oil but represents the rate of viscosity change with temperature change. A depiction of the VI is given in the appendix to this course. Please take a look at this time to familiarize yourself with the general layout of the table.

Viscosity Improvers are viscous chemical compounds called polymers or polymeric compounds that decrease the rate at which oils change viscosity with temperature. These viscosity modifiers extend motor oil's operating temperature range and make multi-grade or all season-oils possible.
The VI is measured by comparing the viscosity of the oil at 40°C (104°F) with its viscosity at 100°C (212°F). VI can provide insight into oil's ability to perform at high and low temperatures.
Without this additive treatment, oils low enough in viscosity to meet the low-temperature requirements of SAE 5W or 10W motor oil will be unable to meet the high temperature viscosity requirements of SAE 30 or heavier oil. That's because the normal rate of viscosity over the required range of ambient starting to engine operating temperatures is simply too large.

LUBRICANT CLASSIFICATION:

We will not get into the very technical and individual classification of lubricants but suffice it to say these are as follows:

MINERAL LUBRICANTS--Mineral fluid lubricants are based on mineral oils. Mineral oils (petroleum oils) are products of refining crude oil. There are three types of mineral oil: paraffinic, naphthenic and aromatic.

- **Paraffinic Oils**—Produced either by hydro-cracking or solvent extraction process. Most hydrocarbon molecules of paraffinic oils have non-ring long-chained structure. Paraffinic oils are relatively viscous and resistant to oxidation. They possess high flash point and high pour point. Paraffinic oils are used for manufacturing engine oils, industrial lubricants and as processing oils in rubber, textile, and paper industries.

- **Naphthenic Oils--** Produced from crude oil distillates.
  Most hydrocarbon molecules of naphtenicnic oils have saturated ring structure. Naphthenic oils possess low viscosity, low flash point, low pour point and low resistance to oxidation.

Naphthenic oils are used in moderate temperature applications, mainly for manufacturing transformer oils and metal working fluids.

- **Aromatic Oils--** Products of refining process in manufacture of paraffinic oils.
  Most hydrocarbon molecules of aromatic oils have non-saturated ring structure.
  Aromatic oils are dark and have high flash point.
  Aromatic oils are used for manufacturing seal compounds, adhesives and as plasticizers in rubber and asphalt production.

- **Semi-Fluid Lubricants (Greases)--** Semi-fluid lubricants (greases) are produced by emulsifying oils or fats with metallic soap and water at 400-600°F (204-316°C). Typical mineral oil base grease is Vaseline. Grease properties are determined by a type of oil (mineral, synthetic, vegetable, animal fat), type of soap (lithium, sodium, calcium, etc. salts of long-chained fatty acids) and additives (extra pressure, corrosion protection, anti-oxidation, etc.). Semi-fluid lubricants (greases) are used in variety applications where fluid oil is not applicable and where thick lubrication film is required: lubrication of roller bearings in railway car wheels, rolling mill bearings, steam turbines, spindles, jet engine bearings and other various machinery bearings.

- **Solid Lubricants**--Solid lubricants possess laminar structure preventing direct contact between the sliding surfaces even at high loads. Graphite and molybdenum disulfide particles are common Solid lubricants. Boron nitride, tungsten disulfide and polytetrafluorethylene (PTFE) are other solid lubricants. Solid lubricants are mainly used as additives to oils and greases. Solid lubricants are also used in form of dry powder or as constituents of coatings.

SYNTHETIC LUBRICANTS

- **Polyalphaolefins (PAO)--** Polyalphaoleins are the most popular synthetic lubricant. PAO's chemical structure and properties are identical to those of mineral oils. Polyalphaoleins (synthetic hydrocarbons) are manufactured by polymerization of hydrocarbon molecules (alphaoleins). The process occurs in reaction of ethylene gas in presence of a metallic catalyst.

- **Polyglycols (PAG) --** Polyglycols are produced by oxidation of ethylene and propylene. The oxides are then polymerized resulting in formation of polyglycol. Polyglycols are water soluble. Polyglycols are characterized by very low coefficient of friction. They are also able to withstand high pressures without EP (extreme pressure) additives.

- **Ester Oils**--Ester oils are produced by reaction of acids and alcohols with water.
  Ester oils are characterized by very good high temperature and low temperature resistance.

- **Silicones--**Silicones are a group of inorganic polymers, molecules of which represent a backbone structure built from repeated chemical units (monomers) containing Si=O moieties. Two organic groups are attached to each Si=O moiety: methyl+methyl ($(CH_3)_2$), methyl+phenyl ($CH_3 + C_6H_5$), phenyl+phenyl ($(C_6H_5)_2$). The most popular silicone is polydimethylsiloxane (PDMS). Its monomer is $(CH_3)_2SiO$. PDMS is produced from silicon and methylchloride. Other examples of silicones are polymethylphenylsiloxane and polydiphenylsiloxane. Viscosity of silicones depends on the length of the polymer molecules and on the degree of their cross-linking. Short non-cross-linked molecules make fluid silicone. Long cross-linked molecules result in elastomeric silicone. Silicone lubricants (oils and greases) are characterized by broad temperature range: -100ºF to +570ºF (-73ºC to 300ºC).

VEGETABLE--Vegetable lubricants are based on soybean, corn, castor, canola, cotton seed and rape seed oils. Vegetable oils are environmentally friendly alternatives to mineral oils since they are biodegradable. Lubrication properties of vegetable based oils are identical to those of mineral oils. The main disadvantages of vegetable lubricants are their low oxidation and temperature stabilities.

ANIMAL LUBRICANTS--Animal lubricants are produced from the animal's fat. There are two main animal fats: hard fats (stearin) and soft fats (lard). Animal fats are mainly used for manufacturing greases.

TYPES OF LUBRICATION:

Boundary

Boundary Lubrication (sometimes referred to as thin film lubrication) is a condition in which the lubricant film becomes too thin to provide total separation. This may be due to excessive loading, speeds, or a change in the fluid's characteristics. In such a situation, contact between surface asperities (peaks and valleys) occurs. Friction reduction and wear protection is then provided through chemical compounds rather than the properties of the lubricating fluid. Boundary lubrication often occurs during the startup and shutdown of equipment, or when loading becomes excessive. This condition can commonly be observed in certain types of gear sets that need to withstand sliding pressures and shock loading, such as hypoid gears found in automotive differentials.

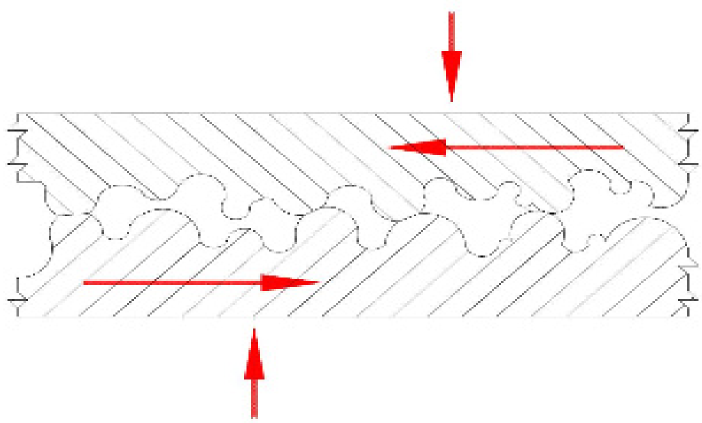

FIGURE 15: SLIDING PAIR WITHOUT LUBRICATION

Mixed Film

Mixed Film Lubrication is a combination of both hydrodynamic and boundary lubrication. In such a situation, only occasional asperity contact occurs. This condition can be the result of lubricant breakdown or increased load placed upon the lubricant.

Elastohydrodynamic

Elasatohydrodynamic Lubrication (EHD or EHL) occurs as pressure or load increases to a level where the viscosity of the lubricant provides 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 increases the effectiveness of the lubricant.

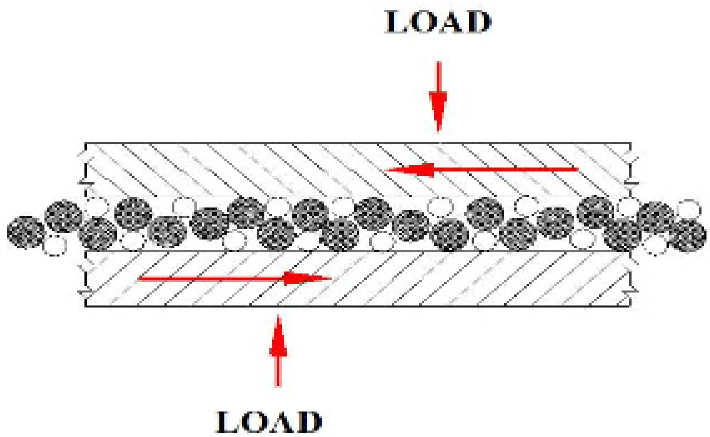

FIGURE 16: LUBRICANT COMPONENT INTERACTION

Although hydrodynamic lubrication is the ideal situation, in many instances it cannot be maintained. Factors which affect hydrodynamic lubrication include Lubricant Viscosity, Rotation Speed or 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.

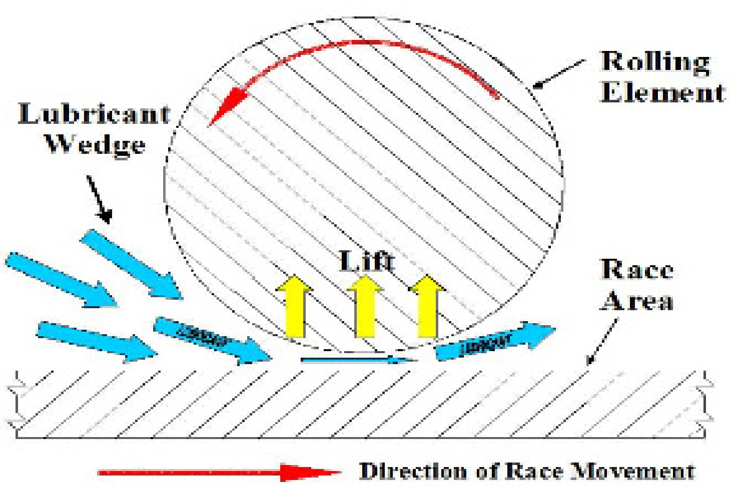

FIGURE: 17 ROLLING ELEMENT LUBRICATION

A good understanding of these different types of lubrication can assist greatly in the selection of the proper lubrication for a specific application and the prevention of equipment failure.

STRIBECK CURVE:

When lubrication is applied to reduce the wear/friction of moving surfaces, an increasing load can shift the lubrication from several regimes such as Boundary, Mixed and Hydrodynamic Lubrication. 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. It is this process that determines the lubrication regime. How the regimes react to friction is shown in what is called a Stribeck curve. To evaluate lubricants and their reaction with applications, the Stribeck Curve can be identified using a Pin On Disk Tribometer. A pin-on-disk apparatus looks as follows:

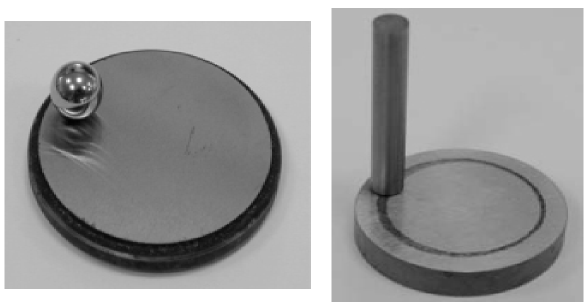

FIGURE: 18 PIN ON DISK DETERMINATION OF VISCOSITY

The unit consists of a gimbaled arm to which the pin is attached, a fixture which accommodates disks up to 165 mm in diameter & 8 mm thick, an electronic force sensor for measuring the friction force, and a computer software (on Labview platform) for displaying the parameters, printing, or storing data for analysis. The motor-driven turntable produces up to 3000 rpm. Wear is quantified by measuring the wear groove with a profilometer (to be ordered seperately) and measuring the amount of material removed. Users simply specify the turntable speed, the load, and any other desired test variables such as friction limit and number of rotations.

Designed for unattended use, a user need only place the test material into turntable fixture and specify the test variables. A pre-determined Hertzian pressure is automatically applied to the pin using a system of weights. Rotating the turntable while applying this force to the pin includes sliding wear as well as a friction force. Since pins can be fabricated from a wide range of materials, virtually any combination of metal, glass, plastic, composite, or ceramic substrates can be tested.

Software included with this model provides for quick calculation of the Hertzian pressure between the pin and disk. The cup-like (housing) enclosed fixture permits the use of liquid lubricants during a wear test (optionally). The graphical output for this exercise may be seen as Figure 19—Stribeck Curve.

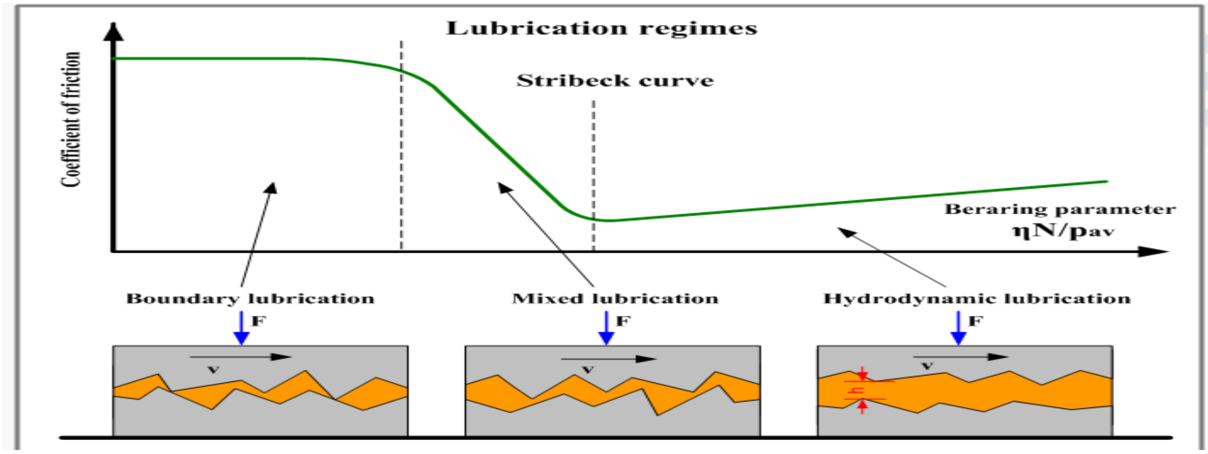

FIGURE: 19 STRIBECK NUMBER

SAE GRADES:

The Society of Automotive Engineers (SAE) Viscosity Grade is a system based on viscosity measures taken from a variety of tests. It developed 11 distinct motor oil viscosity classifications or grades: SAE 0W, SAE 5W, SAE 10W, SAE 15W, SAE 20W, SAE 25W, SAE 20, SAE 30, SAE 40, SAE 50 and SAE 60. These are single grade or single viscosity oils. These grades designate the specific ranges that the particular oil falls into. The "W" indicates the grade is suitable for use in cold temperatures. (Think of the "W" as meaning "Winter".) The classifications increase numerically, readily indicating the difference between them and what the difference means. Simply put, the lower the number, the lower the temperature at which the oil can be used for safe and effective protection. The higher numbers reflect better protection for high heat and high load situations. SAE 20 and SAE 20W are two separate classifications.
Single grade oils have a limited range of protection and, therefore, a limited number of uses.
With today's well-refined, high viscosity index oils, however, an SAE 20 oil usually will meet the viscosity requirements of SAE 20W and vice versa. Those that do are classified SAE 20W-20.
This multi-grade or multi-viscosity ability increases oil's usefulness, because it meets the requirements of two or more classifications. Examples of multi-viscosity oils are SAE 5W-30, SAE 10W-30, SAE 15W-40 and SAE 20W-50. The number with the "W" designates the oil's properties at low temperatures. The other number characterizes properties at high temperatures. For instance, a multi-viscosity or multi grade oil such as 10W-30 meets the 10W criteria when cold and the 30 criteria once hot. SAE 10W-30 and SAE 5W-30 are widely used because under all but extremely hot or cold conditions, they are light enough for easy engine cranking at low temperatures and heavy enough to protect at high temperatures.

API CLASS:

The American Petroleum Institute (API) developed a classification system to identify oils formulated to meet the operating requirements of various engines. The API system has two general categories: S-series and C-series.

The **S-series service classification** emphasizes oil properties critical to gasoline or propane fueled

engines. If oil passes a series of tests in specific engines (API Sequence tests), the oil can be sold bearing the applicable API service classification. The classifications progress alphabetically as the level of lubricant performance increases. Each classification replaces those before it. SL oil may be used in any engine, unless the engine manufacturer specifies a “non detergent” oil. SA and SB are non detergent oils and are not recommended for use unless specified.
New car warranties from 1980 to 1989 require SF oils, while new car warranties from 1990 to 1993 require SG oils. New car warranties beginning with the 1994 model year require oils with an API SH performance rating. Beginning with 1997, new car warranties require and API SJ oil. The year 2001 brought the introduction of SL oils. SL oils are designed to increase fuel economy, reduce emissions and protect hot, hard-working engines over the course of a very long warranty period.

C-series classifications pertain to diesel engines. They are: CA, CB, CC, CD, CD-II, CE, CF, CF-II, CF-4, CG-4, CH-4 and CI-4. All are obsolete except CF, CF-II, CH-4, and the new CI-4 performance rated oils. However, oils used in turbo charged gasoline engines retain CF as part of their performance designation: SH, CF. Unlike S-series classifications, C-series classifications do not supersede one another. The current classifications, CF, CF-2, CH-4 and CI-4 are specified for various applications.

CF for Indirect Injected Diesel Engine Service. Service Category CF denotes service typical of indirect injected diesel engines and other diesel engines that use a broad range of diesel fuels in off-road applications, including diesel fuel with greater than 0.5 percent sulfur by weight. CF oils may be used in place of CD oils.

CF-2 for Two-Stroke Diesel Engine Service. This service category is typical of two-stroke engines requiring highly effective control over cylinder and ring-face scuffing and deposits. CF-2 oils may be used in engines for which CD-II oils are recommended.

CI-4 for Severe Duty Diesel Engine Service. CG-4 typically is required in high speed four-stroke diesel engines used in heavy-duty on- and off-highway applications. CI-4 oils are especially effective in engines designed to meet 2000 exhaust emission standards. CI-4 oils may be used in place of CD, CE, CF-4, CG-4 and CH-4 oils.

These classification systems aim to help motorists choose the right oil for their needs. The choice depends on the engine, the outdoor temperature and the type of driving the engine must withstand. SJ and SL are the current API class.
SJ and SL oils are widely available and ensure the best engine protection available.

ADHESION:

Adhesion is a term relating to the force required to separate two bodies in contact with each other. Desanguliers proposed adhesion as an element in the friction process, a hypothesis which appeared to contradict experiments because of the independence of friction on the contact area. The real area of contact is made up of a very large number of small regions of contact, called asperities or junctions of contact. This is where atom to atom contact takes place. It has been shown that the force of static friction between two sliding surfaces is strongly dependent on the real area of contact. This is called the asperity theory of friction. When two rough surfaces, in contact with each other, are made to move tangentially to the contact plane, a friction force is produced as a result of the interaction between the asperities of the contacting surfaces. If the contacting surfaces are maintained fully separated by a layer of lubricating material, the friction can be reduced significantly as friction is due to the internal resistance to shear/sliding of the lubricating material under those conditions. The stress concentration, at those points of contact, lead rapidly to the formation of asperity junctions. At an asperity junction, inter-atomic interactions lead to adhesive bonding of the mating materials. To make the materials move relative to each other, either the adhesive bonds must be ruptured by the applied shear force or weaker material underneath one of the surfaces has to yield or break. If the shear strength at the weak spot of the junction is τ, the coefficient of friction associated with the adhesive bonding μ (ad) is then

$\mu(ad) = A(r)\tau/W(n) = \tau/P(r)$ **<u>EQUATION: 9 COEFFICIENT WITH ADHESIVE BONDING</u>**

where $P(r) = W(n)/A(r)$ is the mean real pressure. Plastic deformation (or wear) plays a major role in most cases. If plastic deformation is involved, P(r) is equal to the indention hardness "H" of the softer material. The indention hardness in turn is approximated by the following formula:

$H = 3\sigma(y)$ **<u>EQUATION: 10 INDENTION HARDNESS</u>**

Where $\sigma(y)$ is the uniaxial yield stress. If the real contact area is increased due to asperities being pushed together, the order of surface area is increased by orders of magnitude. If one material is much harder than the other, ploughing may take place. The amount of force required for ploughing often exceeds the one required to break the adhesive bonds and can sometimes explain the experimentally observed value of μ.

ENGINEERING SURFACES:

No matter how smooth you think a surface is, there are always imperfections: i.e. peaks and valleys, etc. that must eventually be accounted for. When two mating surfaces are placed in contact with each other, these peaks make contact and wear begins. There is an associated roughness also with each surface as a result. High-powered microscopes can show these peaks and valleys. An example of this is given in Figure 6 of this study. Large numbers of engineering components either deteriorate progressively or fail catastrophically through surface-related phenomena; i.e. wear, fatigue and corrosion, etc. This fact has led to the establishment of the interdisciplinary subject called surface engineering. Surface engineering is defined as the application of both traditional and innovative surface technologies to produce a composite material with properties unattainable in either the base or surface materials individually. Broadly speaking, these techniques can be divided into those involving the modification (whether by mechanical, thermal, or chemical means) of the existing surface of the component, and those that involve the deposition of additional or overlay material, often in the form of a very thin layer or coating over the bulk substrate. A great number of treatments and overlay coatings have important application relative to tribology. These are as follows:

SURFACE TREATMENTS:

Thermal—Induction hardening, flame hardening and laser hardening.

Thermo-Chemical Diffusion—Carburizing, nitriding and Carbo-nitriding.

Mechanical—Shot-peening and Cold working.

Ion Implantation

Laser Glazing

OVERLAY COATINGS:

Plating—Electroplating and Mechanical plating

Weld Cladding—Oxyacetylene, Tungsten Inert Gas (TIG), Metal Inert Gas (MIG)

Thermal Spraying—Flame Spraying and Plasma Spraying

Chemical Vapor Deposition(C VD)

Physical Vapor Deposition (PVD)—Sputtering and Evaporation

HARDNESS OF MATERIALS:

Surface hardness, as we have seen, is one extremely important factor in determining component wear.

The American Society of Metals Handbook, Volume 1, ***Properties and Selection***, defines hardness as follows*: "resistance of metal to plastic deformation usually by indentation. However the term may be applied to stiffness or temper, or to resistance to scratching, abrasion or cutting. Indention hardness may be measured by various hardness tests, such as Brinell, Rockwell and Vickers."* Basically every substance has a property we call hardness. Several measurements exist indicating the hardness of a substance. These are as follows: Rockwell, Knoop, Brinell, Vickers, Shore and Mohs. The table below will give a very basic indication as to comparisons between the most-used types and the ASTM standards governing determination of material hardness.

TEST	TEST METHOD	TEST FORCE RANGE	INDENTER TYPES	ASTM TEST METHOD	MEASURE METHOD
Rockwell	Regular	60, 100, 150 kgs	Conical Diamond & Small Ball	E 18	Depth
	Superficial	15, 30, 45 kgs	Conical Diamond & Small Ball	E 18	Depth
	Light Load	3, 5, 7 kgs	Truncated Cone Diamond	N/A	Depth
	Micro	500, 100 grams	Small Truncated Cone Diamond	N/A	Depth
	Macro	500 to 3000 kgs	5, 10 mm Ball	E 103	Depth
Micro-Hardness	Vickers	5 to 2000 grams	136° Pyramid Diamond	E 384	Area
	Knoop	5 to 2000 grams	1300 x 1720° Diamond	E 384	Area
	Rockwell Type	500, 3000 grams	Truncated Cone Diamond	N/A	Depth
	Dynamic	.01 to 200 grams	Triangular Diamond	N/A	Depth
Brinell	Optical	500 to 3000 kgs	5mm, 10 mm Ball	E 10	Area
	Depth	500 to 3000 kgs	5mm, 10 mm Ball	E 103	Depth
Shore	Regular	822 (A), 4550 (D) grams	35° Cone (A) 30° Cone (D)	D 2240	Depth
	Micro	257 (A), 1135 (D) grams	35° Cone (A) 30° Cone (D)	N/A	Depth
IRHD	Regular	597 grams	2.5 mm Ball	D 1415	Depth
	Micro	15.7 grams	.395 mm Ball	D 1415	Depth

TABLE 4: HARDNESS MEASUREMENT COMPARISONS

It is important to note that there is definitely a correlation between hardness and tensile strength. The truncated chart below is representative of what is available relative to the literature and will indicate the type of correlations available.

TENSILE STRENGTH TO HARDNESS CONVERSION CHART						
Brinell		Vickers or Firth Hardness No.	Rockwell		Scleroscope No.	Approximate Tensile Strength 1000 psi
Dia. (mm): 3000-kg Load 10-mm Ball	Hardness No.		C 150-kgLoad 120° DiamondCone	B 100-kg Load 1/16" dia. Ball		
2.05	898					440
2.10	857					420
2.15	817					401
2.20	780	1150	70		106	384
2.25	745	1050	68		100	368
2.30	712	960	66		95	352
2.35	682	885	64		91	337
2.40	653	820	62		87	324

TABLE 5: TENSILE STRENGTH VS HARDNESS

The hardness for metals is generally measured using Rockwell or Brinell hardness scales. The test equipment needed for determining material hardness is quite sophisticated but “user friendly”

Accurate testing can be made with excellent repeatability. Several devices for determining hardness are given with the following photographs.

BRINELL HARDNESS TEST:

Dr. J. A. Brinell invented the Brinell test in Sweden in 1900. The oldest of the hardness test methods in common use today, the Brinell test is frequently used to determine the hardness of forgings and castings that have a grain structure too coarse for Rockwell or Vickers testing. Therefore, Brinell tests are frequently done on large parts. By varying the test force and ball size, nearly all metals can be tested using a Brinell test. Brinell values are considered test force independent as long as the ball size/test force relationship is the same.

In the USA, Brinell testing is typically done on iron and steel castings using a 3000Kg test force and a 10mm diameter carbide ball. Aluminum and other softer alloys are frequently tested using a 500Kg test force and a 10 or 5mm carbide ball. Therefore the typical range of Brinell testing in this country is 500 to 3000kg with 5 or 10mm carbide balls. In Europe, Brinell testing is done using a much wider range of forces and ball sizes. It's common in Europe to perform Brinell tests on small parts using a 1mm carbide ball and a test force as low as 1kg. These low load tests are commonly referred to as baby Brinell tests. Figure 20 shows the device used for measuring hardness with the Brinell system.

FIGURE 20: BRINELL HARDNESS TESTER

ROCKWELL HARDNESS TEST:

Stanley P. Rockwell invented the Rockwell hardness test. He was a metallurgist for a large ball bearing company, and wanted a fast non-destructive way to determine if the heat treatment process they were doing on the bearing races was successful. The only hardness tests he had available were Vickers, Brinell and Scleroscope. The Vickers test was too time consuming, Brinell indents were too big for his parts and the Scleroscope was difficult to use, especially on his small parts.

To satisfy his needs he invented the Rockwell test method. This simple sequence of test force application proved to be a major advance in the world of hardness testing. It enabled the user to perform an accurate hardness test on a variety of sized parts in just a few seconds. A Rockwell test apparatus is shown by Figure 21.

FIGURE 21: ROCKWELL HARDNESS TESTER

KNOOP AND VICKERS HARDNESS TEST:

The **Knoop hardness test method**, also referred to as a **microhardness test method**, is mostly used for small parts, thin sections, or case depth work. The Vickers method is based on an optical measurement system. The Microhardness test procedure, ASTM E-384, specifies a range of light loads using a diamond indenter to make an indentation which is measured and converted to a hardness value. It is very useful for testing on a wide type of materials as long as test samples are carefully prepared. A pyramid shaped diamond is used for testing in the Knoop scale. This indenter differs from the pyramid indenter used on a Vickers test. The Knoop indenter is more elongated or rectangular in shape. The Knoop method is commonly used when indentations are closely spaced or very near the edge of the sample. The width of the Knoop indentation can provide more resolution for measurement and the indentation is also less deep. Consequently, it can be used on very thin materials.

FIGURE 22: Knoop, Vickers, Brinell Hardness Tester

Comparison tables like the one below will indicate what is available relative to the various classifications.

ROCKWELL				SUPERFICIAL ROCKWELL			BRINELL		VICKERS OR FIRTH DIAMOND HARDNESS NUMBER	SCLERO-SCOPE	TENSILE STRENGTH
							10 m/m Ball				
Diamond Brale			1/16" Bull	"N" Brale Penetrater			3000 kgm Load				
150 kgm C Scale	60 kgm A Scale	100 kgm D Scale	100 kgm B Scale	15 kg Load 15N	30 kg Load 30N	45 kg Load 45N	Diam. Of Ball Impression in m/m	Hardness Number			Equivalent 1000 lb. Sq. In.
80	92	87		97	92	87			1865		
79	92	86			92	87			1787		
78	91	85		96	91	86			1710		
77	91	84			91	85			1633		
76	90	83		96	90	84			1556		
75	90	83			89	83			1478		
74	89	82		95	89	82			1400		
73	89	81			88	81			1323		
72	88	80		95	87	80			1245		
71	87	80			87	79			1160	99	
70	87	79		94	86	78			1076	98	
69	86	78		94	85	77			1004	97	
68	86	77			85	79			942	96	
67	85	76		93	84	75			894	95	
66	85	76		93	83	73			854	93	
65	84	75		92	82	72	2.25	745	820	91	
64	84	74			81	74	2.3	710	789	88	
63	83	73		92	80	70	2.3	710	763	87	
62	83	73		91	79	69	2.35	682	746	85	
61	82	72		91	79	68	2.35	682	720	83	
60	81	71		90	78	67	2.4	653	697	82	
59	81	70		90	77	66	2.45	627	674	80	326
58	80	69		89	76	65	2.55	578	653	78	315
57	80	69		89	75	63	2.55	578	633	77	304
56	79	68		88	74	62	2.6	555	613	75	294
55	79	67		88	73	61	2.6	555	595	74	287
54	78	66		87	72	60	2.65	534	577	72	279
53	77	65		87	71	59	2.7	514	560	71	269
52	77	65		86	70	57	2.75	495	544	69	261
51	76	64		86	69	56	2.75	495	528	68	254

TABLE 6: HARDNESS SYSTEM COMPARISON

CONTACT BETWEEN SURFACES:

When two surfaces come in contact with each other, there will always be some distortion of each. These surfaces may be elastic or may involve some additional plastic, and so permanent, changes in shape. Rolling element bearings, with inner and outer races, represent an excellent example of this type of contact in which distortion will occur. In attempting to predict the likely damage to components and their life-expectancy, knowledge of the true stresses experienced by the material is crucial. If the material is loaded beyond its elastic limit, permanent deformation may occur thus possibly increasing the friction and wear. Again, for the best MTBF and MTTF, a suitable lubricant must be used.

BEARINGS:

Since rolling contact is so prevalent in the design of machinery, we will now take a very brief look at four bearing configurations found in most complex dynamic systems.

HYDROSTATIC BEARINGS:

A hydrostatic bearing is one in which the leaded surfaces are separated by a fluid film, which is forced between them by an externally generated pressure. Formation of the film requires the fluid supply to be pumped continuously, but does not depend on the relative motion of the surfaces. Such bearings are used consistently by engineers due to the smoothness of operation relative to both dynamic surfaces. Hydrostatic bearings are used where normal leads are high and the advantage of low friction at zero speeds is a premium. Stiffness and vibration can definitely be controlled with this type of bearing.

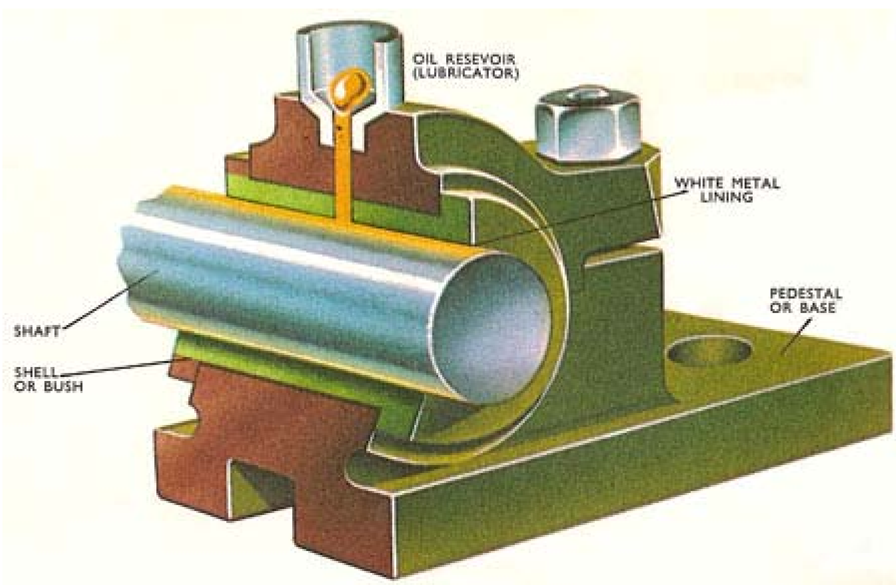

FIGURE 23: HYDROSTATIC BEARING

HYDRODYNAMIC BEARINGS:

Successful operation of a hydrodynamic bearing depends upon the presence of a converging, wedge-shaped gap into which the viscous fluid is dragged by the relative motion of the two solids. A pressure is generated, which tends to push the faces of the wedge apart with the integrated effect being to balance the normal load on both bearing surfaces. Lubrication and the type of lubricants for hydrostatic and hydrodynamic bearing is critical and will spell the difference between success and failure.

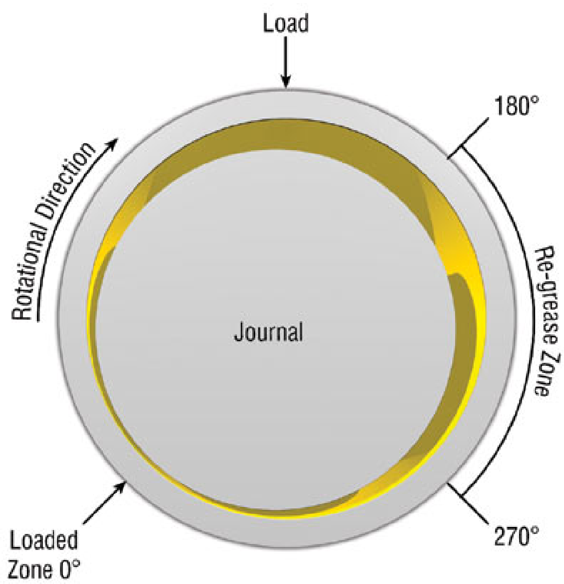

Figure 5

FIGURE 24: HYDRODYNAMIC BEARING

GAS BEARINGS:

Gas-lubricated bearings operate on the very same principles as those using liquid lubricants. Since the viscosity of a gas is generally so much less than that of a liquid, it follows that pressures and specific loads in a gas bearing are very low. Frictional or traction stresses will be similarly reduced. Air as an operating fluid has definite advantages, one being it is plentiful and cheap and will not tarnish or corrode solid surfaces. It can also be exhausted safely to the atmosphere without the need for special arrangements for its collection and return. Air will not freeze at low temperatures or boil at high temperatures. Now the downside—since air pressures are typically 10^5 Pa, an air bearing must occupy about 100 times the area of an oil-lubricated design in order to support a comparable load. It is also the case that the gap between the two opposing solid surfaces is much smaller with air than an oil-lubricated bearing.

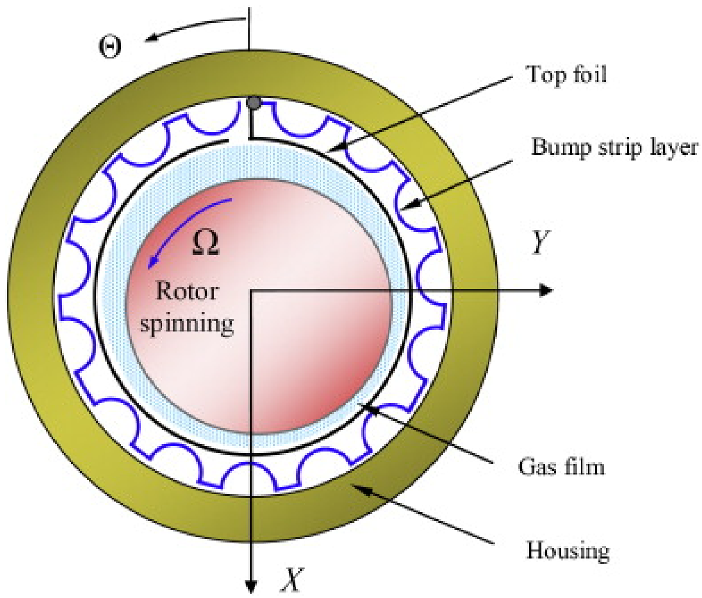

FIGURE 25: GAS BEARING

ROLLING CONTACT:

We are most familiar with rolling-contact bearings or "ball bearings". The basic definition of a rolling contact bearing is as follows: A rolling-element bearing, also known as a rolling bearing, is a bearing which carries a load by placing round elements between the two pieces. The relative motion of the pieces causes the round elements to roll with very little resistance and with little sliding. Little sliding is the key phrase here. The figure 26 below will indicate the variety of rolling contact types used every day by engineers who design dynamic machines.

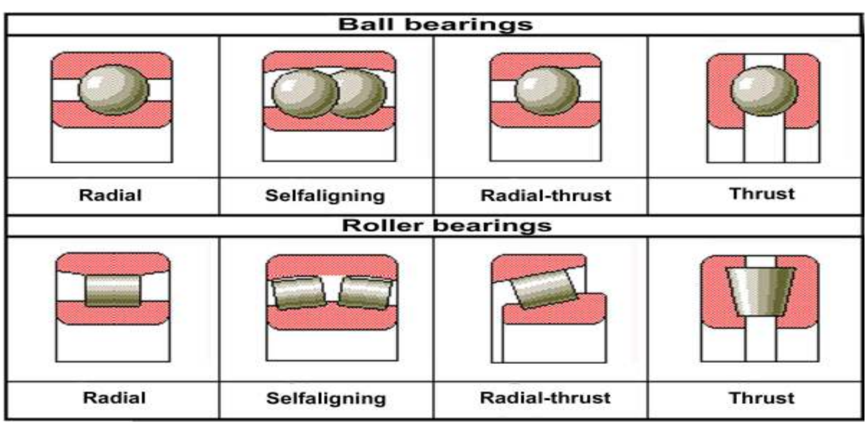

FIGURE 26: ROLLING ELEMENT BEARING

EMERGING FIELDS OF STUDY:

GREEN TRIBOLOGY

To mitigate friction and wear, we use various lubricants, the very first of which was animal fat. Over the centuries engineers and chemists have become much more ingenious, resulting in a huge variety of lubrication formulas and methods of application. Lubricants do their job so why not investigate making those lubricants less harmful to our environment—thus "green tribology".

"Green tribology" has been defined as 'the science and technology of the tribological aspects of ecological balance and of environmental and biological impacts'. It is basically involved with minimizing the following:

1.) Generation of pollution while using lubricants and lubrication techniques.
2.) Risk to human health and the environment while using lubrication to reduce wear and friction.

In accomplishing the two goals as given above, there are twelve (12) principles that may be applied. These are:

- Minimize heat and energy dissipation
- Minimize wear
- Reduce or completely eliminate lubrication and self-lubrication
- Use natural lubrication if at all possible
- Use biodegradable lubrication when and if possible
- Use sustainable chemistry and green engineering principles
- Use biometric principles of engineering whenever possible
- Surface texturing should be applied to control surface properties. Conventional engineered surfaces have random roughness that makes it difficult to control friction and wear.
- Recognize environmental implication of coatings
- Design for the degradation of surfaces
- Apply real-time monitoring when hazardous substances are in use
- Apply sustainable energy methodology at all times.

These twelve principles, when applied, can greatly reduce the environmental impact lubricants can have when needed for any specific application.

NANO- TRIBOLOGY:

Nano-microtribology is a branch of tribology which studies friction phenomenon at the nanometer scale. The distinction between nano-microtribology and tribology is primarily due to the involvement of atomic forces in the determination of the final behavior of the system. Gears, bearings, and liquid lubricants can reduce friction in the macroscopic world, but the origins of friction for small devices such as micro-or nano-electromechanical systems (NEMS) require other solutions. The surface force apparatus (SFA), atomic force and friction force microscopes (AFM & FFM) are widely used in nano-tribological studies.

SUMMARY:

As we have seen, the study of tribology is not only important—it's fascinating and encompasses technologies that have been known for decades, if not centuries. Tribology is like any other technology in that most of the discoveries are evolutionary instead of revolutionary. As equipment needed to study the subject progresses, we see a definite refinement in our understanding. Nano-tribology is one example of that process.

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Glossary of Tribology Terms

Term	Description
Abrasion	Mechanical wear during sliding of two surfaces against each other.
Additives	Substances added in small amounts to lubricants to improve the performance.
Adhesion improvers/promoters	Additives to oils and greases to improve adhesion (e.g. polyisobutene).
Adhesive lubricants	Lubricants with adhesion-improving components, which are not thrown off by centrifugal forces.
AF coating	Means anti-friction coating, the most common and widely used type of dry solid lubrication of today. This group includes both air-dried and heat-cured materials. These formulations usually consist of a lubricating solid called the "pigment" and a bonding agent. See "Binder".
Ageing resistance	The resistivity against ageing which might occur due to oxidation, overheating, the presence of certain metals like copper, lead, silver etc. The resistance to ageing can be improved by certain additives (antioxidants).
ASTM	American Society for Testing Materials.
Base oil	Basic component of lubricating oils and greases.
Binder	An alternative term for non-volatile medium or vehicle and refers to the material which forms the varnish film and which in a paint or bonded coating binds the particles of solids (solid lubricants) together.
Bonded lubricant	See AF coating.
Break away torque	Effective leverage turned into rotating movement to loosen a bolted connection.
Chemically inert	(Lubricant) not reacting chemically with certain substances.

Coefficient of friction	Ratio of the frictional force between two surfaces sliding across one another to the force that is perpendicular to the surfaces.
Cold resistance	Guide values for oils are the cloud point, pour point and solidification point; for lubricating greases the Kesternich flow pressure and the low-temperature torque test.
Colloid	Small particles (10-5 to 10-7 cm) in liquid which behave like a solution (no settling of particles).
Complex greases	Lubricating greases with thickeners produced from metallic soaps with various acids. Particularly suitable for high temperatures and long-term applications.
Consistency	A measure of the condition of lubricating greases. It is measured as the un-worked and worked penetration and is indicated in accordance with the NLGI (National Lubricating Grease Institute). To simplify designation of the consistency of lubricating greases, the consistency range as a whole is divided into nine classes, measured as worked penetration, e.g. Consistency class Worked penetration (1/10 mm) 00 400 - 430 0 355 - 385 1 310 - 340 2 265 - 295
Density	The weight of a lubricant in grams per cm3 at 20 °C.
Detergent	Agent for loosening and removing residues and deposits from sliding surfaces.
Dispersion	Name given to two-substance systems in which one substance is contained in the other substance (liquid) in a dispersed form.
DN value	A guide to the grease which should be used in rolling-element bearings depending upon their speed of rotation. It represents the mean bearing

	diameter in mm multiplied by the speed in revolutions per minute.
Drop point	The drop point of grease is that temperature at which grease passes from a semisolid to a liquid state. It is a qualitative indication of the heat resistance of grease. The drop point temperature is determined when the first drop falls through the hole in the bottom of the cup during temperature increase.
Dynamic viscosity	A measure for inner friction during flowing of a lubricating oil (e. g. flowing through pipes or clearances).
EP additives	Chemical substances to improve the pressure absorption capacity and hence the wear resistance of oils and greases.
Emcor	The test for corrosion protection of lubricating greases in rolling-element bearings in the presence of water: A minimum of two grease-lubricated ball bearings run in water for about one week. The corrosion value of the rings ranges from 0-5 (0 = no corrosion, 5 = severe corrosion).
Ester oils	Compounds of acids and alcohols used for lubrication and the production of lubricating greases.
Flash point	The flash point is the lowest temperature at which during heating inflammable vapors are formed on the surface of the oil to be tested which shortly flare up in the presence of a flame.
Fluoro-silicones	Silicones which contain fluorine atoms in the molecule.
Freezing point	The freezing point of oil is the temperature in degrees Celsius at which the oil has just lost its ability to flow because of continuous cooling down. The solidifying of the oil is caused by the separation of paraffin crystals.
Fretting corrosion	Rust which occurs on seats. Better: frictional wear which occurs at fits and seats due to oscillations with very low amplitude and high frequency. Usually, the very small iron wear particles react to rust in combination with oxygen, which finally results in seizing of the seats. Another disadvantage of fretting corrosion is the rapid material fatigue of the steel, a fact which can easily lead to breaking. (Fretting corrosion can be prevented most effectively by the separation of both metal partners, e.g. by

	means of solid lubricants.)
Friction	Resistance against sliding of two surfaces against one another.
Grease	2-phase-system: thickener with fluid, lubricating medium.
Inhibitors	Additives for lubricants which reduce ageing and corrosion.
Lithium	Alkalimetal, the hydroxide of which is used together with organic acids to form lithium soaps as thickener for greases.
Lubricant	Medium to reduce friction and wear between two surfaces sliding against one another.
Measurement of viscosity	Viscosities can be measured in various viscosimeters. The dimension is mm2/s. An important factor for the measurement of the viscosity is the temperature, because the viscosity does significantly depend on the temperature. (Cold oils are more viscous, warm oils are less viscous.)
Molybdenum disulphide (MoS2)	A solid lubricant.
Oil separation	The. bleeding. of oil from lubricating greases during storage or as a result of mechanical/dynamic or temperature stress.
O.K. load	Indication of the pressure resistance of a lubricant. It is the very maximum load at which just no breakthrough of the lubricating film, and thus no welding of the test specimens, occurs (Newton).
Oxidation resistance	Resistance of hydrocarbons against a reaction with oxygen.
Pastes	Combination of solid lubricants with oil for easy application of thin lubricating film.
Penetration	Indicates the softness or hardness of grease. The depth of penetration of a standardized cone in a grease sample is measured. (The higher the

	penetration, the softer is the grease.)
Pitting	Crater-like metal cavities in the pitch circle of gear wheels, caused by material fatigue.
Polyalpha-olefin	Synthetic hydrocarbon with a defined molecular structure. Low-temperature, high-temperature and viscosity/temperature characteristics are better than with mineral oil.
Pour point	Lowest temperature at which lubricating oil remains free-flowing.
Running-in	Surface asperities of new sliding surfaces are modified during the running-in period.
Salt-water spray test	The corrosion of steel is measured under the influence of saline fog. Sheet steel is coated with a lubricant and exposed to saline fog in a closed chamber. After the test, the number of hours are measured which have passed until a certain grade of corrosion was reached.
Scoring	Trench-shaped marks in metal, caused by machining or by scuffing.
Scuffing	Damage to material surface through inadequate supply of lubricant, or as a result of overloading. The lubricating film is broken.
Self-ignition point	The temperature at which oil ignites by itself, i.e. without the presence of a flame.
Service temperature range	The range in which the lubricant meets requirements and an acceptable lubrication interval is achieved.
Silicones	Polymers with good temperature and oxidation resistance. Also used as high and low temperature lubricants.
Soap in lubricating grease	Combination of a fatty acid and a metal hydroxide. Through the proper selection of the fatty acid and the metal hydroxide (calcium, lithium, aluminum) the properties of the soap can be changed as to water resistance

	and temperature resistance.
Solid lubricants	Solid substances which are applied between sliding surfaces to reduce friction and wear and prevent scoring.
Solvent	A liquid which will dissolve a material and yield a homogeneous product.
Specialty lubricants	Lubricants with particular properties/characteristics for special applications.
Specific weight	See density.
Stick-slip	Jerky relative movements of two bearing surfaces, caused by the difference in coefficient of friction between hydrodynamic and boundary lubrication.
Stress cracks	Cracks in materials caused by corrosive changes of the surface structure after penetration of undesirable elements.
Suspension	A uniform dispersion of the fine particles of a solid in a liquid which does not dissolve them.
Swelling	Under the action of lubricants, vapors or gases, sealing materials made from rubber, elastomer, etc., can be negatively affected by swelling.
Synthetic oils	In contrast to mineral oils, these are artificially produced oils. Synthetic oils usually have a good viscosity temperature behavior, low tendency to carbonize, deep freezing point, high temperature stability, and good chemical resistance.
Thickeners	Thickeners usually are metal soaps (soap-thickened) but also organic or inorganic thickening agents (not soap-thickened as e. g. silica, bentone, urea, PTFE etc.).
Tightening torque	Effective leverage turned into rotating movement to tighten a screw connection.
Tribology	Science of scientific research and technical application of the relation

	between friction, wear and lubrication, including lubricants.
Un-worked penetration	The consistency of a grease or paste in the state of rest, i.e. in the state of material as supplied.
Viscosity	The viscosity of a liquid is the resistance of molecules against pressure from outside. This resistance is described as inner friction.
Water resistance of a grease	The behavior of lubricating greases in the presence of water is of great importance for their applicability as antifriction bearing greases. For this application, either a water-repellent (water resistant) or water-absorbent (emulsifiable) lubricating grease is required.
Wear	Caused by friction and contact between bearing surfaces after break-through of the lubricating film.
Weld load	The ability of a lubricant to absorb pressure, measured in Newton (N), the load at which the lubricating film breaks, during sliding of test specimens against each other, and at which both test specimens weld together.
Worked penetration	Under mechanical shear, lubricating greases often change their consistency. Therefore, it is more reasonable to indicate the worked penetration. It is the consistency of worked grease.

MATERIAL 1	MATERIAL 2	Coefficient Of Friction			
		Dry		Greasy	
		Static	Sliding	Static	Sliding
Aluminum	Aluminum	1.05-1.35	1.4	0.3	

Aluminum	Mild Steel	0.61	0.47		
Brake Material	Cast Iron	0.4			
Brake Material	Cast Iron (Wet)	0.2			
Brass	Cast Iron		0.3		
Brick	Wood	0.6			
Bronze	Cast Iron		0.22		
Bronze	Steel			0.16	
Cadmium	Cadmium	0.5		0.05	
Cadmium	Mild Steel		0.46		
Cast Iron	Cast Iron	1.1	0.15		0.07
Cast Iron	Oak		0.49		0.075
Chromium	Chromium	0.41		0.34	
Copper	Cast Iron	1.05	0.29		
Copper	Copper	1.0		0.08	
Copper	Mild Steel	0.53	0.36		0.18
Copper-Lead Alloy	Steel	0.22		-	
Diamond	Diamond	0.1		0.05 - 0.1	
Diamond	Metal	0.1 -0.15		0.1	
Glass	Glass	0.9 - 1.0	0.4	0.1 - 0.6	0.09-0.12

Glass	Metal	0.5 - 0.7		0.2 - 0.3	
Glass	Nickel	0.78	0.56		
Graphite	Graphite	0.1		0.1	
Graphite	Steel	0.1		0.1	
Graphite (In vacuum)	Graphite (In vacuum)	0.5 - 0.8			
Hard Carbon	Hard Carbon	0.16		0.12 - 0.14	
Hard Carbon	Steel	0.14		0.11 - 0.14	
Iron	Iron	1.0		0.15 - 0.2	
Lead	Cast Iron		0.43		
Leather	Wood	0.3 - 0.4			
Leather	Metal(Clean)	0.6		0.2	
Leather	Metal(Wet)	0.4			
Leather	Oak (Parallel grain)	0.61	0.52		
Magnesium	Magnesium	0.6		0.08	
Nickel	Nickel	0.7-1.1	0.53	0.28	0.12
Nickel	Mild Steel		0.64;		0.178
Nylon	Nylon	0.15 - 0.25			
Oak	Oak (parallel grain)	0.62	0.48		
Oak	Oak (cross grain)	0.54	0.32		0.072

Platinum	Platinum	1.2		0.25	
Plexiglas	Plexiglas	0.8		0.8	
Plexiglas	Steel	0.4 - 0.5		0.4 - 0.5	
Polystyrene	Polystyrene	0.5		0.5	
Polystyrene	Steel	0.3-0.35		0.3-0.35	
Polythene	Steel	0.2		0.2	
Rubber	Asphalt (Dry)		0.5-0.8		
Rubber	Asphalt (Wet)		0.25-0.0.75		
Rubber	Concrete (Dry)		0.6-0.85		
Rubber	Concrete (Wet)		0.45-0.75		
Saphire	Saphire	0.2		0.2	
Silver	Silver	1.4		0.55	
Sintered Bronze	Steel	-		0.13	
Solids	Rubber	1.0 - 4.0		--	
Steel	Aluminum Bros	0.45			
Steel	Brass	0.35		0.19	
Steel(Mild)	Brass	0.51	0.44		
Steel (Mild)	Cast Iron		0.23	0.183	0.133
Steel	Cast Iron	0.4		0.21	

Steel	Copper Lead Alloy	0.22		0.16	0.145
Steel (Hard)	Graphite	0.21		0.09	
Steel	Graphite	0.1		0.1	
Steel (Mild)	Lead	0.95	0.95	0.5	0.3
Steel (Mild)	Phos. Bros		0.34		0.173
Steel	Phos Bros	0.35			
Steel(Hard)	Polythened	0.2		0.2	
Steel(Hard)	Polystyrene	0.3-0.35		0.3-0.35	
Steel (Mild)	Steel (Mild)	0.74	0.57		0.09-0.19
Steel(Hard)	Steel (Hard)	0.78	0.42	0.05 -0.11	0.029-.12
Steel	Zinc (Plated on steel)	0.5	0.45	-	-
Teflon	Steel	0.04		0.04	0.04
Teflon	Teflon	0.04		0.04	0.04
Tin	Cast Iron		.32		
Tungsten Carbide	Tungsten Carbide	0.2-0.25		0.12	
Tungsten Carbide	Steel	0.4 - 0.6		0.08 - 0.2	
Tungsten Carbide	Copper	0.35			
Tungsten Carbide	Iron	0.8			
Wood	Wood(clean)	0.25 - 0.5			

Wood	Wood (Wet)	0.2			
Wood	Metals(Clean)	0.2-0.6			
Wood	Metals (Wet)	0.2			
Wood	Brick	0.6			
Wood	Concrete	0.62			
Zinc	Zinc	0.6		0.04	
Zinc	Cast Iron	0.85	0.21		

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