The Lubrication of Gears – Part 1

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Introduction

This is a three-part article explaining the principles of gear lubrication. It reviews current knowledge of the field of gear tribology and is intended for both gear designers and gear operators. Part 1 classifies gear tooth failures into five modes and explains the factors that a gear designer and operator must consider to avoid gear failures. It defines the nomenclature and gives a list of references for those interested in further research. It also contains an in-depth discussion of the gear tooth failure modes that are influenced by lubrication and gives methods for preventing gear tooth failures.

The second part gives an equation for calculating the lubricant film thickness which determines whether the gears operate in the boundary, elastohydrodynamic, or full-film lubrication regime. Also given is an equation for Blok's flash temperature, which is used for predicting the risk of scuffing. Finally, a recommendation for selecting lubricant type, viscosity, and application method and a case history, which demonstrates many of the principles of gear lubrication, are given.

Gear Tribology

Because gears are such common machine components, they may be taken for granted. Not generally appreciated is that they are complex systems requiring knowledge from all the engineering disciplines for their successful design. Gear design is a process of synthesis where gear geometry, materials, heat treatment, manufacturing methods, and lubrication are selected to meet the requirements of a given application. The designer must design the gearset with adequate strength, wear resistance, and scuffing resistance. To do this, he or she must consider gear tribology. The choice of lubricant and its application method is as important as the choice of steel alloy and heat treatment. The interrelationship of the following factors must be considered:

- Gear tooth geometry
- Gear tooth motion (kinematics)
- Gear tooth forces (static and dynamic)
- Gear tooth material and surface characteristics (physical and chemical)
- Lubricant characteristics (physical and chemical)
- Environment characteristics (physical and chemical)

Gear Tooth Failure Modes

To obtain optimum, minimum-weight gearsets the gear designer must be aware of the intricate details of many competing modes of failure. In its nomenclature publication,[1] the American Gear Manufacturers Association (AGMA) has classified 20 modes of gear failure under the broad categories of wear, surface fatigue, plastic flow, breakage, and associated gear failures. References 2 through 6 also give gear failure modes. For our purposes, the basic categories are overload, bending fatigue, Hertzian fatigue, wear, and scuffing.

Table 1 subdivides the five basic failure modes. Many gear failures are known by several names and/or qualifying terms, such as initial, moderate, destructive, etc. These names and terms are included in the lists in parentheses. The term "scoring" has been used in the past in the U.S.A., while the term "scuffing" is used in Europe to describe the severe form of adhesive wear which involves the welding and tearing of the surfaces of gear teeth. To agree with current usage, the term scuffing will be used in this article when referring to

TABLE 1

Basic Failure Modes	
<u>Overload</u>	<u>Hertzian Fatigue</u>
Brittle fracture	Pitting (initial, superficial, destructive, spalling)
Ductile fracture	Micropitting (frosting, grey staining, peeling)
Plastic deformation	Sub-case fatigue (case crushing)
cold flow	<u>Wear</u>
hot flow	Adhesion (normal, running-in, mild, moderate, severe, excessive)
indentation (rolling, bruising, peening, brinelling)	Abrasion (scoring, scratching, plowing, cutting, gouging)
rippling (fish scaling)	Corrosion
ridging	Fretting – corrosion
bending, yielding	Cavitation
tip-to-root interference	Electrical discharge damage
<u>Bending Fatigue</u>	Polishing (burnishing)
Low-cycle fatigue (<1000 cycles to failure)	<u>Scuffing</u>
High-cycle fatigue (<1000 cycles to failure)	Scuffing (scoring, galling, seizing, welding, smearing, initial, moderate, destructive)

this failure mode. The term scoring implies scratching, and it will be used to describe abrasive wear rather than scuffing.

Lubrication – Related Failure Modes

These articles are concerned with gear tooth failures that are influenced by lubrication. Pitting or scuffing may cause the gear teeth to deteriorate and generate dynamic forces, which in turn cause the gear teeth to fail by bending fatigue. In these cases the bending failure is secondary and not directly related to lubrication, while pitting or scuffing are the primary failure modes, and both are definitely influenced by lubrication. The failure analyst must discern the difference between primary and secondary failure modes because the wrong corrective action is likely to be recommended if a secondary failure mode is mistaken for the primary failure mode. For example, increasing the size of the gear teeth to prevent reoccurence of the above-mentioned bending failure would only make the situation worse by lowering the pitting and scuffing resistance. Godfrey[7] gives a good description of lubrication-related failure modes.

With the above considerations, overload and bending fatigue are judged to be unrelated to lubrication and are eliminated from further discussion together with sub-case, Hertzian fatigue. Although corrosion, fretting-corrosion, cavitation, and electrical discharge damage are influenced by lubrication, they are not discussed because these failure modes occur relatively rarely in gear teeth. Hence, the following failure modes are included in the scope of this article: Hertzian fatigue, including pitting and micropitting; wear, including adhesion, abrasion, and polishing; and scuffing.

<u>Hertzian Fatigue – Pitting.</u> Pitting is a common failure mode for gear teeth because they are subjected to high Hertzian contact stresses and many stress cycles. For example, through-hardened gears are typically designed to withstand contact stresses of approximately 100,000 psi, while the contact stresses on carburized gears may reach 300,000 psi. In addition, a given tooth on a pinion that is revolving at 3600 rpm accumulates over 5 million stress cycles every 24 hours.

Pitting is a fatigue phenomenon[8] which occurs when a fatigue crack initiates either at the surface of the gear tooth or at a small depth below the surface. The crack usually propagates for a short distance in a direction roughly parallel to the tooth

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Nomenclature Table

Symbols

Symbol	Description	Units
B_M	– thermal contact coefficient	lbf/[in $s^{0.5}$ °F]
b_H	– semi-width of Hertzian contact band	in
c	– constant (See Table 3.)	hp/gpm
c_M	specific heat per unit mass	lbf in/[lb °F]
d	– operating pitch diameter of pinion	in
E_1, E_2	– modulus of elasticity (pinion, gear)	lbf/in^2
Er	– reduced modulus of elasticity	lbf/in^2
h_{min}	– minimum film thickness	in
L_{min}	minimum contact length	in
n	– pinion speed	rpm
P	– transmitted power	hp
q	– oil flow rate	gpm
S	– average surface roughness, rms	μin
Tb	– bulk temperature	°F
Tb, test	– bulk temperature of test gears	°F
Tc	– contact temperature	°F
Tf	– flash temperature	°F
Tf, test	– maximum flash temperature of test gears	°F
Ts	– scuffing temperature	°F
V	– operating pitch line velocity	ft/min
Ve	entraining velocity	in/s
Vr_1, Vr_2	– rolling velocity (pinion, gear)	in/s
W_{Nr}	– normal operating load	lbf
w_{Nr}	– normal unit load	lbf/in
X_W	– welding factor	– –
X_r	– load sharing factor	– –
α	– pressure-viscosity coefficient	in^2/lb^2
λ	– specific film thickness	– –
λ_M	– heat conductivity	lbf/[s °F]
μ_m	– mean coefficient of friction	– –
μ_o	– absolute viscosity	Reyns (lbs/in^2)
ν_1 ν_2	– Poisson's ratio (pinion, gear)	– –
ν_{40}	– kinematic viscosity of 40°C	cSt
ϱ_1 ϱ_2	– transverse radius of curvature (pinion, gear)	in
ϱ_M	– density	lb/in^3
ϱ_n	– normal relative radius of curvature	in
σ	– composite surface roughness, rms	μin
σ_1, σ_2	– surface roughness, rms (pinion, gear)	μin
ψ_b	– base helix angle	deg
ω_1, ω_2	– angular velocity (pinion, gear)	rad/s

surface before turning or branching to the surface. When the cracks have grown to the extent that they separate a piece of the surface material, a pit is formed. If several pits grow together, the resulting larger pit is often referred to as a "spall." There is no endurance limit for Hertzian fatigue, and pitting occurs even at low stresses if the gears are operated long enough. Because there is no endurance limit, gear teeth must be designed for a suitable, finite lifetime.

To extend the pitting life of a gearset, the designer must keep the contact stress low and the material strength and lubricant specific film thickness high. There are several geometric variables, such as diameter, face width, number of teeth, pressure angle, etc., that may be optimized to lower the contact stress. Material alloys and heat treatment are selected to obtain hard tooth surfaces with high strength. Maximum pitting resistance is obtained with carburized gear

Methods for Preventing Pitting

1. Reduce contact stresses by reducing loads or optimizing gear geometry.
2. Use clean steel, properly heat treated to high hardness, preferrably by carburizing.
3. Use smooth tooth surfaces produced by careful grinding or honing.
4. Use an adequate amount of cool, clean, and dry lubricant of adequate viscosity.

teeth because they have hard surfaces, and carburizing induces beneficial compressive residual stresses which effectively lower the load stresses. The drawbacks to using them are that they are relatively expensive to produce and that they must be finished by grinding. The details for obtaining high lubricant specific film thickness will be explained later when elastohydrodynamic (EHD) lubrication is discussed, but general recommendations are to use an adequate supply of cool, clean, and dry lubricant that has adequate viscosity and a high pressure-viscosity coefficient.

Pitting may initiate at the surface or at a subsurface defect, such as a nonmetallic inclusion. With gear teeth, pits are most often of the surface-initiated type because the lubricant film thickness is usually low, resulting in relatively high metal-to-metal contact. The interaction between asperities of contact at defects, such as nicks or furrows, creates surface-initiated, rather than subsurface-initiated cracks. For high-speed gears with smooth surface finishes, the film thickness is greater, and subsurface-initiated pitting, rather than surface-initiated, may predominate. In these cases, pitting usually starts at a subsurface inclusion, which acts as a point of stress concentration. Cleaner steels, such as those produced by vacuum melting, prolong the pitting life by reducing the number of inclusions.

Contamination from water in the lubricant is believed to promote pitting through hydrogen embrittlement of the metal, and abrasive particles in the lubricant cause pitting by indenting the tooth surfaces, causing stress concentrations and/or disrupting the lubricant film. At present, the influence of lubricant additives on pitting is unresolved.

Hertzian Fatigue – Micropitting. On relatively soft gear tooth surfaces, such as those of through-hardened gears, Hertzian fatigue forms large pits with dimensions on the order of millimeters. With surface-hardened gears, such as carburized, nitrided, induction-hardened, and flame-hardened, pitting may occur on a much smaller scale, typically only 10μm deep. To the naked eye, the areas where micropitting has occurred appear frosted, and "frosting" is a popular term for micropitting. Japanese researchers[9] have referred to the failure mode as "grey staining" because the light-scattering properties of micropitting gives the gear teeth a grey appearance. Under the scanning electron microscope (SEM) immediately evident is that micropitting proceeds by the same fatigue process as classical pitting, except the pits are extremely small.

Many times micropitting is not destructive to the gear tooth surface. It sometimes occurs only in patches and may stop after the tribological conditions have improved by running-in. The micropits may actually be removed by light polishing wear during running-in, in which case the micropitting is said to "heal." However, there have been examples[9-11] where micropitting has escalated into full-scale pitting, leading to the destruction of the gear teeth.

The specific film thickness is the most important parameter that influences micropitting. Damage seems to occur most readily on gear teeth with rough surfaces, especially when they are lubricated with low viscosity lubricants. Gears finished with special grinding wheels to a mirror-like finish[12] have effectively eliminated micropitting. Slow-speed gears are prone to micropitting because their film thickness is low.

Methods for Preventing Micropitting

1. Use smooth tooth surfaces produced by careful grinding or honing.
2. Use an adequate amount of cool, clean, and dry lubricant of the highest viscosity permissible.
3. Use high speeds if possible.
4. Use carburized steel with proper carbon content in the surface layers.

To prevent micropitting, the specific film thickness should be maximized by using smooth gear tooth surfaces, high-viscosity lubricants, and high speeds. Experiments[10] have shown that flame-hardened and induction-hardened gears have less resistance to micropitting than carburized gears of the same hardness. This is probably due to the lower carbon content of the surface layers of the flame-hardened and induction-hardened gears.

Wear – Adhesion. Adhesive wear is classified as "mild" if it is confined to the oxide layers of the gear tooth surfaces. If, however, the oxide layers are disrupted, and bare metal is exposed, the transition to severe adhesive wear usually occurs. Severe adhesive wear is termed scuffing and will be discussed later. Here we assume that scuffing has been avoided through proper design of the gears, selection of the lubricant, and control of the running-in process.

When new gear units are first operated, the contact between the gear teeth is not optimum because of unavoidable manufacturing inaccuracies. If the tribological conditions are favorable, mild adhesive wear occurs during running-in and usually subsides with time, resulting in a satisfactory lifetime for the gears. The wear that occurs during running-in is beneficial if it smooths the tooth surfaces, increasing the specific film thickness, and if it increases the area of contact by removing minor imperfections through local wear. To ensure that the wear rate remains under control, run in new gearsets by operating for at least the first ten hours at one-half load.

The amount of wear considered tolerable depends on the expected lifetime for the gears and requirements for control of noise and vibration.

Methods for Preventing Adhesive Wear

1. Use smooth tooth surfaces.
2. If possible, run in new gearsets by operating the first ten hours at one-half load.
3. Use high speeds if possible. Otherwise, recognize that highly loaded, slow-speed gears are boundary lubricated and are especially prone to excessive wear. For these conditions, specify nitrided gears and the highest permissible lubricant viscosity.
4. For very slow-speed gears (<10 fpm), avoid using lubricants with sulphur-phosphorous additives.
5. Use an adequate amount of cool, clean, and dry lubricant of the highest viscosity permissible.

Wear is considered excessive when the tooth profiles wear to the extent that high dynamic loads occur or the tooth thickness is reduced to the extent that bending fatigue becomes possible.

Many gears, because of practical limits on lubricant viscosity, speed, and temperature, must operate under boundary-lubricated conditions where some wear is inevitable. Highly loaded, slow speed (<100 fpm), boundary-lubricated gears are especially prone to excessive wear. Tests with slow-speed gears[10] have shown that nitrided gears have good wear resistance, while carburized and through-hardened gears have similar, but lower wear resistance. Reference 10 concluded that lubricant viscosity has the greatest influence on slow-speed, adhesive wear, and that high – viscosity lubricants reduce the wear rate significantly. The same authors found that sulphur-phosphorous additives can be detrimental with slow-speed (<10 fpm) gears, giving very high wear rates.

A few gear units operate under ideal conditions with smooth tooth surfaces, high pitch line speed, and high lubricant film thickness. For example, turbine gears that operated almost continuously at 30,000 fpm pitch line speed still had the original machining marks on their teeth, even after operating for 20 years. Most gears, however, operate between the boundary and full-film lubrication regimes, under elastohydrodynamic (EHD) conditions. In the EHD regime, with the proper type and viscosity of lubricant, the wear rate usually reduces during running-in and adhesive wear virtually ceases once running-in is completed. If the lubricant is properly maintained (cool, clean, and dry), the gearset should not suffer an adhesive wear failure.

Wear – Abrasion. Abrasive wear on gear teeth is usually caused by contamination of the lubricant by hard, sharp-edged particles. Contamination enters gearboxes by being built-in, internally generated, ingested through breathers and seals, or inadvertently added during maintenance.

Many gear manufacturers do not fully appreciate the significance of clean assembly; it is not uncommon to find sand, machining chips, grinding dust, weld splatter, or other debris in new gear boxes. To remove built-in contamination, drain and flush the gearbox lubricant before start-up and again after the first 50 hours of operation, refill with the recommended lubricant, and install a new oil filter.

Methods for Preventing Abrasive Wear

1. Remove built-in contamination from new gearboxes by draining and flushing the lubricant before start-up and again after the first 50 hours of operation. Refill with the recommended lubricant and install a new filter.
2. Minimize internally generated wear debris by using surface-hardened gear teeth, smooth tooth surfaces, and high-viscosity lubricants.
3. Minimize ingested contamination by maintaining oil-tight seals and using filtered breather vents located in clean, non-pressurized areas.
4. Minimize contamination that is added during maintenance by using good housekeeping procedures.
5. For circulating oil systems, use fine filtration.
6. For oil bath systems, change the lubricant at least every 2500 hours or every six months.
7. Monitor the lubricant with spectrographic and ferrographic analysis together with analysis of acid number, viscosity, and water content.

Internally generated particles are usually wear debris from gears or bearings due to Hertzian fatigue pitting or adhesive and abrasive wear. The wear particles are especially abrasive because they become work-hardened when they are trapped between the gear teeth. Internally generated wear debris can be minimized by using accurate, surface-hardened gear teeth (with high pitting resistance), smooth surfaces, and high viscosity lubricants.

Breather vents are used on gearboxes to vent internal pressure, which may occur when air enters through seals, or when air within the gearbox expands (or contracts) during the normal heating and cooling of the gear unit. The breather vent should be located in a clean, non-pressurized area and should have a filter to prevent ingress of airborne contaminants. In especially harsh environments, the gearbox can be completely sealed, and the pressure variation can be accommodated by an expansion chamber with a flexible diaphragm.

All maintenance procedures which involve opening any part of the gearbox or lubrication system must be carefully performed to prevent contamination of the gearbox system.

Abrasive wear due to foreign contaminants, such as sand or internally generated wear debris, called three-body abrasion, is a common occurrence. Two-body abrasion also occurs when hard particles or asperities on one gear tooth abrade the opposing tooth surface. Unless the tooth surfaces of a surface-hardened gear are smoothly finished, they will act like files if the mating gear is appreciably softer. This is the reason that a worm pinion is polished after grinding before it is run with a bronze worm wheel. Manufacturers of computer disk drives have found that stainless steel pinions mated with anodized aluminum racks have excessively high wear rates. The anodized layer of the aluminum rack is extremely thin and brittle, and it breaks up and impregnates the relatively soft stainless steel pinion. The aluminum oxide particles then act like emery paper and wear the teeth of the rack very quickly.

The lubrication system should be carefully maintained and monitored to ensure that the gears receive an adequate amount of cool, clean, and dry lubricant. For circulating-oil systems, fine filtration removes contamination. Filters as fine as 3 μm have significantly increased gear life. For oil-bath gearboxes, the lubricant should be changed frequently to remove contamination. Under normal operating conditions the lubricant should be changed at least every 2500 operating hours or six months, whichever occurs first. For critical gearboxes a regular program of lubricant monitoring can help prevent gear failures by showing when maintenance is required. The lubricant monitoring should include spectrographic and ferrographic analysis of contamination, along with analysis of acid number, viscosity, and water content.

Polishing wear. If the extreme pressure (EP) additives in the lubricant are too chemically reactive, they may cause polishing of the gear tooth surfaces until they attain a bright, mirror finish. Although the polished gear teeth may look good, polishing wear is undesirable because it generally reduces gear accuracy by wearing the tooth profiles away from their ideal form. EP additives used in lubricants to prevent scuffing, such as sulfer and phosphorous, will be covered when scuffing is discussed. They function by forming iron sulfide and iron phosphate films on areas of the gear teeth where high temperatures occur. Ideally, the additives should react only at temperatures where there is a danger of welding. If the rate of reaction is too high, and there is a continuous removal of the surface films caused by very fine abrasives in the lubricant, the polishing wear may become excessive.

Polishing wear can be prevented by using less chemically active additives. As an alternative to sulfur-phosphorous additives, anti-scuff lubricants are available with dispersions of potassium borate[13] which deposit EP films without chemically reacting with the metal. Removing the abrasives in the lubricant by using fine filtration or frequent oil changes is helpful.

Methods of Preventing Polishing Wear

1. Use less chemically active anti-scuff additives, such as borate.
2. Remove abrasives from the lubricant by using fine filtration or frequent oil changes.

Scuffing. Scuffing is defined as localized damage caused by solid-phase welding between sliding surfaces. It is accompanied by transfer of metal from one surface to another due to welding and tearing. It may occur in any sliding and rolling contact where the oil film is not thick enough to separate the surfaces. The symptoms are microscopically rough, matte, torn surfaces. Surface analysis that shows transfer of metal from one surface to the other is proof of scuffing.

Scuffing can occur in gear teeth when they operate in the boundary lubrication regime. If the lubricant film is insufficient to prevent significant metal-to-metal contact, the oxide layers that normally protect the gear tooth surfaces may be broken through, and the bare metal surfaces may weld together. The sliding that occurs between gear teeth results in tearing of the welded junctions, metal transfer, and catastrophic damage.

In contrast to pitting and bending fatigue, which only occur after a period of running time, scuffing may occur immediately upon start-up. In fact, gears are most vulnerable to scuffing when they are new, and their tooth surfaces have not yet been smoothed by running in. For this reason, it is wise to run in a new gearbox under one-half load for at least ten hours to reduce the surface roughness of the teeth before applying full load. The gear teeth can be coated with iron manganese phosphate or plated with copper or silver to protect them from scuffing during the critical running-in period.

The basic mechanism of scuffing is not understood clearly, but by general agreement, it is believed to be caused by intense frictional heating generated by the combination of high sliding velocity and intense surface pressure. Blok's[14] critical temperature theory is believed to be the best criterion for predicting scuffing. It states that scuffing will occur in gear teeth that are sliding under boundary-lubricated conditions, when the maximum contact temperature of the gear teeth reaches a critical magnitude. For mineral oils without anti-scuff/EP additives, each combination of oil and rubbing materials has a critical scuffing temperature which is constant, regardless of the operating conditions.[15] The critical scuffing temperatures are not constant for synthetic lubricants and lubricants with anti-scuff additives; they must be determined from tests which closely simulate the operating conditions of the gears.

Today, most anti-scuff additives are sulfur-phosphorus compounds, which form boundary lubricating films by chemically reacting with the metal surfaces of the gear teeth at local points of high temperature. Anti-scuff films help prevent scuffing by forming solid films on the gear tooth surfaces and inhibiting true metal-to-metal contact. The films of iron sulfide and iron phosphate have

Methods for Preventing Scuffing

1. Use smooth tooth surfaces produced by careful grinding or honing.
2. Protect the gear teeth during the critical running-in period by coating them with iron manganese phosphate or plating them with copper or silver. Run in new gearsets by operating the first ten hours at one-half load.
3. Use high viscosity lubricants with anti-scuff additives, such as sulfur, phosphorous, or borate.
4. Cool the gear teeth by supplying an adequate amount of cool lubricant. For circulating-oil systems, use a heat exchanger to cool the lubricant.
5. Optimize the gear tooth geometry by using small teeth, addendum, and profile modification.
6. Use accurate gear teeth, rigid gear mountings, and good helix alignment.
7. Use nitrided steels for maximum scuffing resistance. Do not use stainless steel or aluminum for gears.

high melting points, allowing them to remain as solids on the gear tooth surfaces even at high contact temperatures. The rate of reaction of the anti-scuff additives is greatest where the gear tooth contact temperatures are highest. Because of the rubbing action of the gear teeth, the surface films are repeatedly scraped off and reformed. In effect, scuffing is prevented by substituting mild corrosion in its place. Occasionally, anti-scuff additives, such as sulfer, are too chemically active, causing polishing wear and necessitating a change to less aggressive additives. Lubricants with anti-scuff additives of potassium borate do not cause polishing wear because they deposit glass-like boundary films without reacting with the metal.

For mineral oils without anti-scuff additives, the critical scuffing temperature increases with increasing viscosity, and ranges from 150°C to 300°C. The increased scuffing resistance of high viscosity lubricants is believed to be due to differences in chemical composition rather than increases in viscosity. However, a viscosity increase also helps to reduce the risk of scuffing by increasing the lubricant film thickness and reducing the contact temperature generated by metal-to-metal contact.

Scuffing is controlled by the total contact temperature Tc, which consists of the sum of the gear bulk temperature Tb, and the flash temperature Tf; i.e.,

$$Tc = Tb + Tf$$

The bulk temperature is the equilibrium temperature of the surface of the gear teeth before they enter the meshing zone. The flash temperature is the local and instantaneous temperature rise that occurs on the gear teeth due to the frictional heating as they pass through the meshing zone.

Anything that reduces either the bulk temperature or the flash temperature will reduce the total contact temperature and lessen the risk of scuffing. Higher viscosity lubricants or smoother tooth surfaces help by increasing the specific film thickness, which in turn reduces the frictional heat and, therefore, the flash temperature. Also, the lubri-

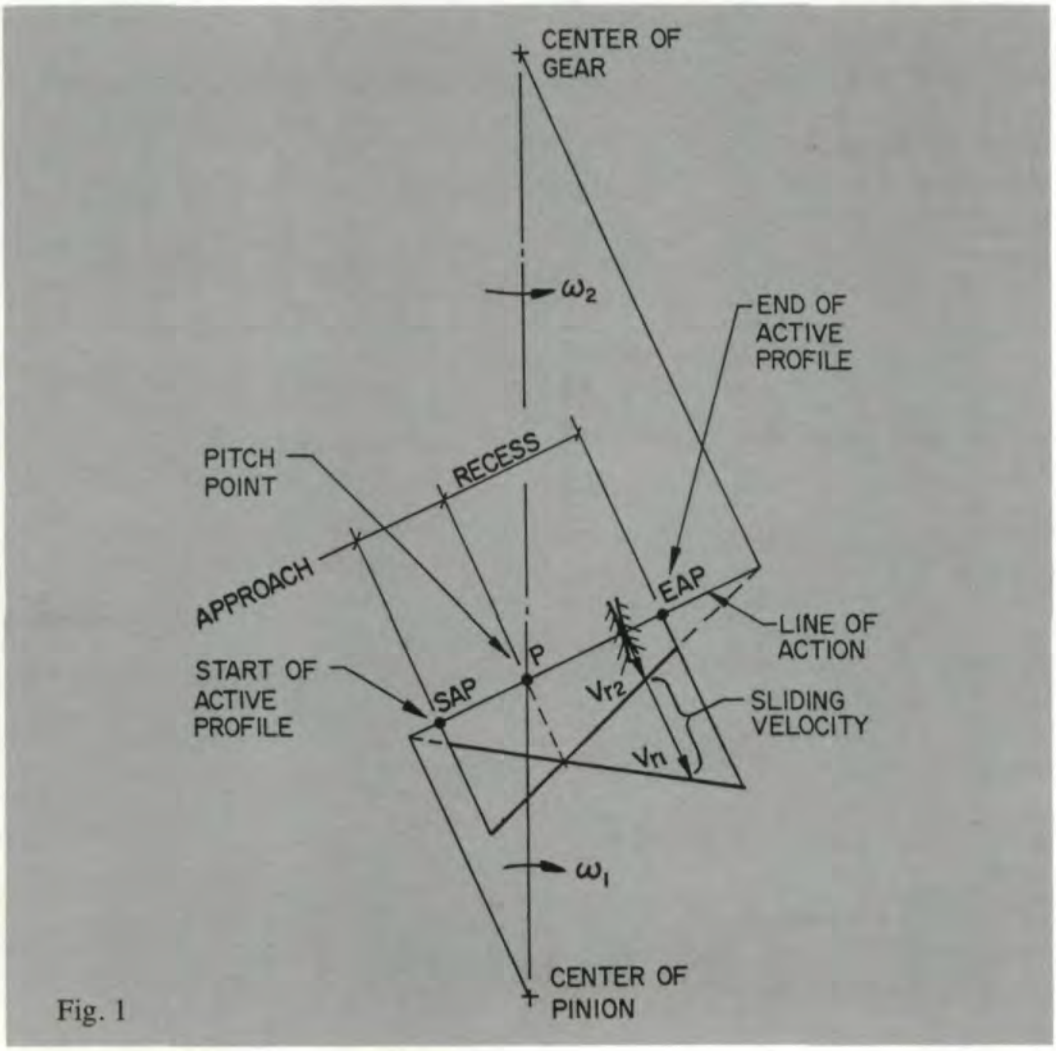

Fig. 1

cant performs the important function of removing heat from the gear teeth. It must be supplied to the gear teeth in such a way that it removes heat rapidly and keeps a low bulk temperature. A heat exchanger can be used with a circulating oil system to cool the lubricant before it is sprayed at the gears. The gear designer can maximize scuffing resistance by optimizing the gear geometry so that the gear teeth are as small as possible, consistent with bending strength requirements, to reduce the temperature rise caused by sliding. Fig. 1 shows that the rolling velocities, Vr_1 and Vr_2, linearly increase from zero at the interference points to a maximum at each end of the path of contact. The sliding velocity is represented by the distance between the Vr_1 and Vr_2 lines. The amount of sliding is proportional to the distance from the pitch point, P, and is zero when the gear teeth contact at the pitch point, and largest at the ends of the path. Addendem modification can be used to balance and minimize the temperature rise that occurs in the addendum and dedendum of the gear teeth. The temperature rise may also be reduced by modifying the tooth profiles with slight tip and/or root relief to ease the load at the start and end of the engagement path where the sliding velocities are the greatest. Also, the gear teeth must be accurate and held rigidly in good alignment to minimize tooth loading and, therefore, the temperature rise.

Gear materials should be chosen with their scuffing resistance in mind. Nitrided steels, such as Nitralloy 135M, are generally found to have the highest resistance to scuffing, while stainless steels are liable to scuff even under near-zero loads. The thin oxide layer on stainless steel is hard and brittle, and it breaks up easily under sliding loads, exposing the bare metal, thus promoting scuffing. Anodized aluminum has a low scuffing resistance similar to stainless steel. Hardness does not seem to be a reliable indication of scuffing resistance.

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