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Lubrication of industrial gear drives

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National foreword

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TECHNICAL REPORT

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Lubrication of industrial gear drives

Lubrification des entraînements par engrenages industriels

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Foreword

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ISO/TR 18792 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

Introduction

Gear lubrication is important in all types of gear applications. Through adequate lubrication, gear design and selection of gear lubricant, the gear life can be extended and the gearbox efficiency improved. In order to focus on the available knowledge of gear lubrication, ISO/TC 60 decided to produce this Technical Report combining primary information about the design and use of lubricants for gearboxes.

Lubrication of industrial gear drives

1 Scope

This Technical Report is designed to provide currently available technical information with respect to the lubrication of industrial gear drives up to pitch line velocities of 30 m/s. It is intended to serve as a general guideline and source of information about the different types of gear, and lubricants, and their selection for gearbox design and service conditions. This Technical Report is addressed to gear manufacturers, gearbox users and gearbox service personnel, inclusive of manufacturers and distributors of lubricants.

This Technical Report is not applicable to gear drives for automotive transmissions.

2 Terms and definitions

For the purposes of this document, the following terms, definitions, symbols, indices and units apply.

Table 1 — Symbols, indices and units

Symbol, index	Term	Unit
A, B, C, D, E	points on the path of contact	—
b	face width	mm
C	cubic capacity of the oil pump	cm^3
d	diameter	mm
$d_{a1, 2}$	outside diameter pinion, wheel	mm
$d_{b1, 2}$	base circle diameter pinion, wheel	mm
$d_{w1, 2}$	operating pitch diameter pinion, wheel	mm
f_H	curvature factor	$N^{0,5}/mm^{1,5}$
f_L	load factor	—
F_{bt}	circumferential load at base circle	N
n_{shaft}	rotational speed of the oil pump driving shaft	rpm
p	pressure	bar
p_H	hertzian stress	N/mm^2
P	gear power	kW
P_{vz}	gear power loss	kW
P_{vzsum}	total gearbox power loss	kW
s	slip	—
t	time	sec

Table 1 (*continued*)

Symbol, index	Term	Unit
V	oil quantity	l
Q_e	oil flow	l/min
$Q_{bearings}$	oil flow through the bearings	l/min
Q_{gears}	oil flow through the gear mesh	l/min
Q_{pump}	oil pump flow	l/min
Q_{seals}	oil flow through the seals	l/min
v	pitch line velocity	m/s
$v_{1,2}$	surface velocity pinion, wheel	m/s
v_g	sliding velocity	m/s
v_t	pitch line velocity	m/s
v_{Σ}	sum velocity	m/s
V_{tank}	oil tank volume	l
z_1	number of pinion teeth	—
β	helix angle	degree
λ	relation between the calculated film thickness and the effective surface roughness	—

2.1
intermittent lubrication
intermittent common lubrication of gears which are not enclosed

NOTE Gears that are not enclosed are referred to as open gears.

2.2
manual lubrication
hand application
periodical application of lubricant by a user with a brush or spout can

2.3
centralized lubrication
intermittent lubrication of gears by means of a mechanical applicator in a centralized system

2.4
continuous lubrication
continuous application of lubricant to the gear mesh in service

2.5
splash lubrication
bath lubrication
immersion lubrication
dip lubrication
process, in an enclosed system, by which a rotating gear or an idler in mesh with one gear is allowed to dip into the lubricant and carry it to the mesh

2.6
oil stream lubrication
pressure-circulating lubrication
forced-circulation lubrication
continuous lubrication of gears and bearings using a pump system which collects the oil in a sump and recirculates it

2.7
drop lubrication
use of oil pump to siphon the lubricant directly onto the contact portion of the gears via a delivery pipe

2.8
spray lubrication
process in oil stream lubrication by which the oil is pumped under pressure to nozzles that deliver a stream or spray onto the gear tooth contact, and the excess oil is collected in the sump and then returned to the pump via a reservoir

2.9
spray lubrication for open gearing
continuous or intermittent application of lubricant using compressed air

2.10
oil mist lubrication
process by which oil mist, formed from the mixing of lubricant with compressed air, is sprayed against the contact region of the gears

NOTE It is especially suitable for high-speed gearing.

2.11
brush lubrication
process by which lubricant is continuously brushed onto the active tooth flanks of one gear

2.12
transfer lubrication
continuous transferral of lubricant onto the active tooth flanks of a gear by means of a special transfer pinion immersed in the lubricant or lubricated by a centralized lubrication system

3 Basics of gear lubrication and failure modes

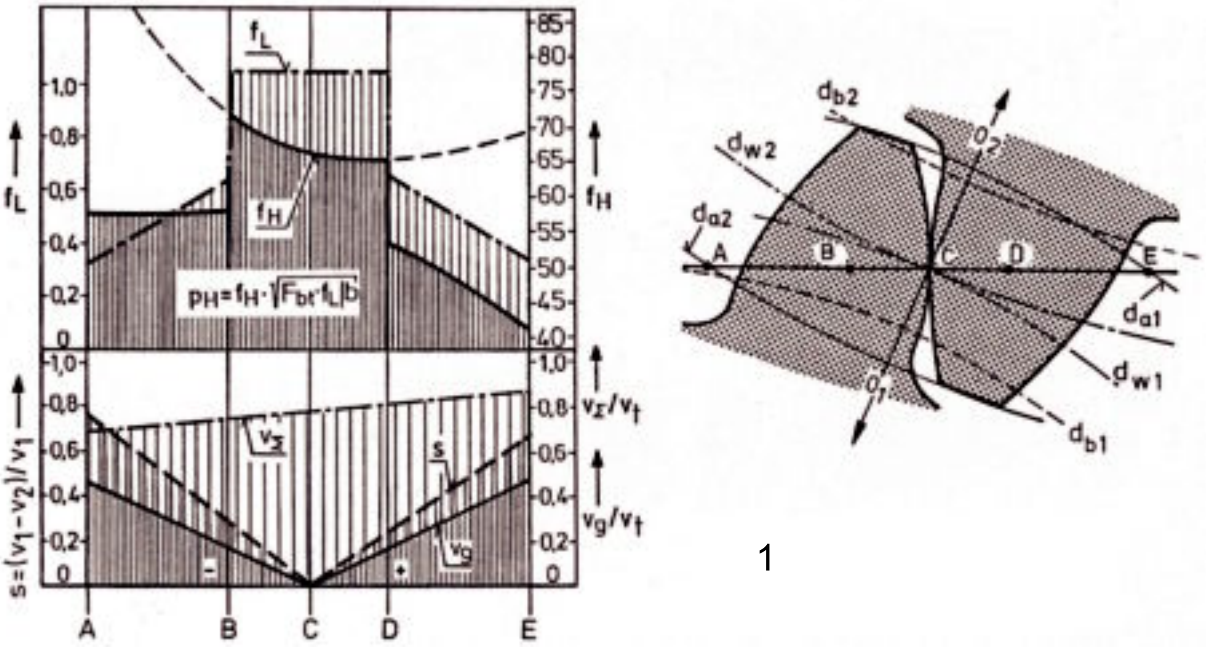
3.1 Tribo-technical parameters of gears

3.1.1 Gear types

There are different types of gear such as cylindrical, bevel and worm. The type of gear used depends on the application necessary. Cylindrical gears with parallel axes are manufactured as spur and helical gears. They typically have a line contact and sliding only in profile direction. Cylindrical gears with skewed axes have a point contact and additional sliding in the axial direction. Bevel gears with an arbitrary angle between their axes without gear offset have a point contact and sliding in profile direction. They generally have perpendicular axes and are manufactured as straight, helical or spiral bevel gears. Bevel gears with gear offset are called hypoid gears with point contact and sliding in profile and axial directions. Worm gears have crossed axes, line contact and sliding in profile and mainly axial direction.

3.1.2 Load and speed conditions

The main tribological parameters of a gear contact are load, pressure, and rolling and sliding speed. A static load distribution along the path of contact as shown in Figure 1 can be assumed for spur gears without profile modification. In the zone of single tooth contact the full load is transmitted by one tooth pair, in the zone of double tooth contact the load is shared between two tooth pairs in contact.



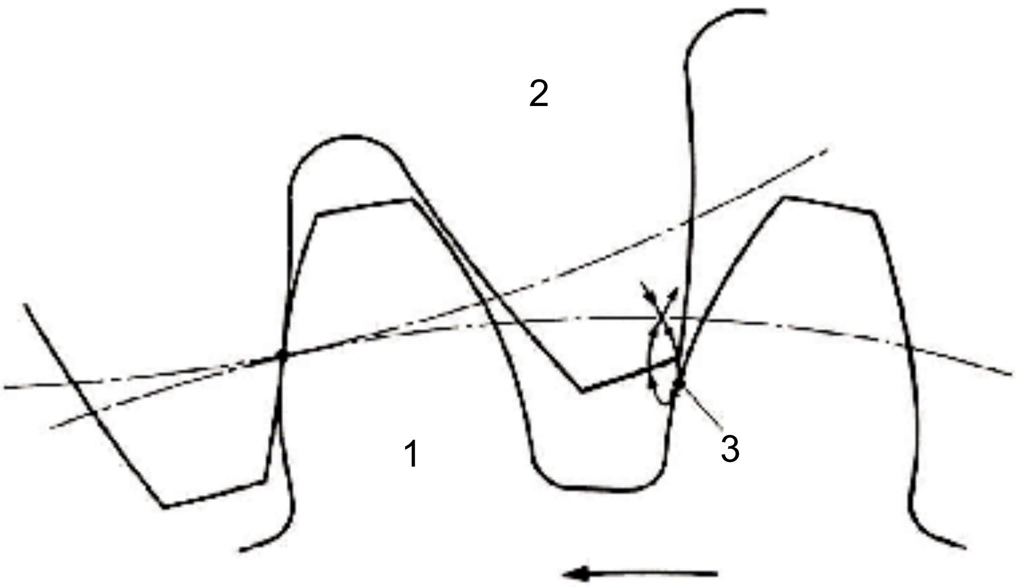
Key

1 spur gear without profile correction

Figure 1 — Load and speed distribution along the path of contact

The static load distribution along the path of contact can be modified through elasticity and profile modifications. Due to the vibrational system of the gear contact, dynamic loads occur as a function of the dynamic and natural frequency of the system. A local Hertzian stress for the unlubricated contact can be derived from the local load and the local radius of curvature (see Figure 1). When a separating lubricating film is present, the Hertzian pressure distribution in the contact is modified to an elastohydrodynamic pressure distribution with an inlet ramp, a region of Hertzian pressure distribution, possibly a pressure spike at the outlet and a steep decrease from the pressure maximum to the ambient.

The surface speed of the flanks changes continuously along the path of contact (see Figure 1). The sum of the surface speeds of pinion and wheel represents the hydrodynamically effective sum velocity; half of this value is known as entraining velocity. The difference of the flank speeds is the sliding velocity, which together with the frictional force results in a local power loss and contact heating. Rolling without sliding can only be found in the pitch point with its most favourable lubricating conditions. Unsteady conditions with changing pressure, sum and sliding velocity along the path of contact are the result. In addition, with each new tooth coming into contact, the elastohydrodynamic film must be formed anew under often unfavourable conditions of the scraping edge of the driven tooth (see Figure 2).



Key

1 pinion (driving)

2 wheel (driven)

3 first contact point

Figure 2 — Scraping edge at the ingoing mesh

3.2 Gear lubricants

3.2.1 Overview of lubrication

Regarding gear lubrication, the primary concern is usually the gears. In addition to the gears there are many other components that must also be served by the fluid in the gearbox. Consideration should also be given to the bearings, seals, and other ancillary equipment, e.g., pumps and heat exchangers, that can be affected by the choice of lubricant. In many open gear drives the bearings are lubricated independently of the gears, thus allowing for special fluid requirements should the need arise. However, most enclosed and semi-enclosed gear drives utilize a single lubricant and lubricant source of supply for the gears, bearings, seals, pumps, etc. Therefore, selecting the correct lubricant for a gear drive system includes addressing the lubrication needs of not only the gears but all other associated components in the system.

A lubricant is used in gear applications to control friction and wear between the intersecting surfaces, and in enclosed gear drive applications to transfer heat away from the contact area. They also serve as a medium to carry the additives that can be required for special functions. There are many different lubricants available to accomplish these tasks. The choice of an appropriate lubricant depends in part on matching its properties to the particular application. Lubricant properties can be quite varied depending on the source of the base stock(s), the type of additive(s), and any thickeners that might be used. The base stock and thickener components generally provide the foundation for the physical properties that define the lubricant, while the additives provide the chemical properties that are critical for certain performance needs. The overall performance of the lubricant is dependent on both the physical and chemical properties being in the correct balance for the application. The following clauses describe the more common types of base stocks, thickeners and additive chemicals used in gear lubricant formulations today.

3.2.2 Physical properties

The physical properties of a lubricant, such as viscosity and pour point, are largely derived from the base stock(s) from which they are produced. For example, the crude source, the fraction or cut, and the amount of refining, such as dewaxing, of a given mineral oil can significantly alter the way it will perform in service. While viscosity is the most common property associated with a lubricant, there are many other properties that contribute to the makeup and character of the finished product. The properties of finished gear lubricants are the result of a combination of base stock selection and additive technology.

3.3 Base fluid components

A key element of the finished fluid is the base oil. The base oil comes from two general sources: mineral; or, synthetic. The term mineral usually refers to base oils that have been refined from a crude oil source, whereas synthetics are usually the product of a chemical reaction of one or more selected starting materials. The finished fluids can also contain mixtures of one or more base oil types. Partial synthetic fluids contain mixtures of mineral and synthetic base oils. Full synthetic fluids can also be mixtures of two or more synthetic base oils. As a current example, mixtures of polyalphaolefins (PAO) and esters are commonly used in synthetic formulations. Mixtures are generally used to tailor the properties of the finished fluid to a specific application or need. An overview of the general characteristics of different base fluids is shown in Table 2. Additional information regarding base fluid characteristics is shown in the following sections.

Table 2 — General characteristics of base fluids

Characteristic	Mineral paraffinic	Polyalpha–olefins (PAO)	Ester	Poly-alkylene-glycol (PAG)	Phosphate esters
Viscosity – temperature relationship (typical viscosity index)	90 – 130	130 – 150	50 – 140	200 – 240	$<$100
Specific heat (relative)	1,0	1,3 – 1,5	1,1 – 1,3	1,1 – 1,3	1,0 – 1,2
Pressure-viscosity at 1 GPa (relative)	1,0	0,8	0,5	~1,0	1,0 – 1,1
Comparability solvency with mineral fluids	Excellent	Good	Excellent	Poor	Good
Comparability solvency with PAO fluids	Good	Excellent	Excellent	Poor	Good
Additive solvency	Good to Excellent	Good	Excellent	Limited	Good

3.3.1 Mineral-based fluids

Mineral-based gear oils have been successfully used for several years in many industrial gear drive systems. Mineral oil lubricants are petroleum-based fluids produced from crude oil through petroleum refining technology. Paraffinic mineral-based gear oils have viscosity indices (VI) that are commonly lower than most, but not all synthetic-based gear oils. This usually means that the low temperature properties of these mineral-based lubricants will not be as good as for a comparable grade synthetic fluid. If low ambient temperatures are involved with the operation of the equipment, this should be factored into the decision process. At high temperatures, mineral-based lubricants are more prone to oxidation than synthetics due in part to the amount of residual polar and unsaturated compounds in the base component. Mineral-based lubricants will generally provide a higher viscosity under pressure than most synthetics and therefore provide a thicker film at moderate temperatures. On the other hand, at higher temperatures, usually around 80 °C to 100 °C or more, the higher VI of synthetic fluids generally overcomes the disadvantage of having a lower pressure-viscosity coefficient. At these higher temperatures, the film thickness can be higher for PAOs compared to mineral oils. Probably the primary advantages of mineral-based oils over synthetic-based oils are their lower initial purchase cost and greater availability worldwide. If a mineral oil is preferred, some of the weaker properties, compared to a synthetic fluid, can be improved through the thickener and additive systems available today.

3.3.2 Synthetic-based fluids

Synthetic oils differ from petroleum-based oils in that they are not found in nature, but are manufactured chemically and have special properties that enhance performance or accommodate severe operating conditions. Because they are manufactured, many of their properties can be tailored to meet specific needs through the choice of starting materials and reaction processing. Many synthetic oils are stable at high operating temperatures, have high VI, i.e. smaller viscosity changes with temperature variations, and low pour points. This means that equipment filled with most commercially available synthetic gear oils can be started without difficulty at lower bulk oil temperatures than those using mineral oils. Another key advantage is that they are inherently more stable at higher temperatures against oxidative degradation than their mineral counterparts, again owing this advantageous property to the uniformity and composition of the fluid structure.

Each type of synthetic lubricant has unique characteristics and the limitations of each should be understood. Characteristics such as compatibility with other lubrication systems and mechanical components (seals, sealants, paints, backstops and clutches), behaviour in the presence of moisture, lubricating qualities and overall economics should be carefully analysed for each type of synthetic lubricant under consideration for a given application. In the absence of field experience in similar applications, the use of synthetic oil ought to be coordinated carefully between the user, the gear manufacturer and the lubricant supplier. Synthetic lubricants can improve gearbox efficiency and can operate cooler than mineral oils because of their viscosity-temperature characteristics and structure-influenced heat transfer properties. Decreasing the operating temperature of a gearbox lubricant is desirable. Lower lubricant temperatures increase the gear and bearing lives by increasing lubricant film thickness, and increase lubricant life by reducing oxidation.

There are several different types of synthetic base oils available today. Their compositions and properties result from the different chemicals that are combined in their manufacture. Some of the major types of synthetic base oils are described in the following clauses. The lubricant supplier is generally consulted for additional information on synthetics for a given application.

3.3.2.1 Polyalphaolefin-based oils

PAOs, or olefin oligomers, are paraffin-like liquid hydrocarbons which can be synthesized to achieve a unique combination of high viscosity-temperature characteristic, low volatility, excellent low temperature viscometrics and thermal stability, and a high degree of oxidation resistance with appropriate additive treatment along with a structure that can improve equipment efficiency. These characteristics result from the wax-free combination of moderately branched paraffinic hydrocarbon molecules of predetermined chain length. Compared to conventional mineral oils, some PAO lubricants have poorer solvency for additives and for sludge that can form as the oil ages. Lubricant formulators commonly add a higher solvency fluid, such as ester or alkylated aromatic fluids, in order to keep the additives in solution and to prevent sludge from being deposited on the gearbox components.

3.3.2.2 Synthetic ester lubricants

Esters are produced from the reaction of an alcohol with an organic acid. There are a wide variety of esters available that can be produced because of the numerous existing combinations of acids and alcohols. The principal advantage of many esters is their excellent thermal and oxidative stability. A primary weakness of some is poor hydrolytic stability. When in contact with water, esters can deteriorate through a reverse reaction and revert to an alcohol and organic acid. A secondary weakness with some esters is a VI lower than most paraffinic mineral-based oils. Some esters do, however, provide a VI higher than mineral or PAO lubricants. It is possible for some ester-based gear oils to be suitable in water protection areas since they can be biodegradable.

On the negative side, ester-based gear oils or lubricants containing esters can adversely affect filters, elastomeric seals, adhesives, sealants, paint, and other surface treatments such as layout lacquer used for contact pattern tests. Therefore, lubricants with esters should be tested for compatibility with all gearbox components before they are used in service. Another weakness of the ester class of lubricants is their poor film-forming capabilities. Esters tend to have very low pressure-viscosity coefficients which relate to the ability of the fluid’s film thickness in the contact region. This could lead to higher wear.

3.3.2.3 Polyalkyleneglycol (PAG) lubricants

PAG-based oils have a chemical structure that is distinctly different from both PAO and ester-based oils. PAGs are generally made from the reaction product of ethylene oxide and propylene oxide to form a polyether type structure. The properties of the structure are dependent on the molecular weight and the ratio of ethylene and propylene oxides used in the reaction mixture. PAG-based gear oils can have excellent thermal and oxidative stability and most have exceptionally high VI, many of which are greater than 200. However, many PAGs have poor corrosion properties in the presence of salt water. In standard distilled water corrosion tests, carefully selected additives can control rust.

The primary difficulties with PAG lubricants are that they can be very hygroscopic (tend to absorb water) and not very miscible with mineral or other synthetic fluid-type base fluids. The affinity for water and compatibility with more common fluids is a function of the ethylene oxide to propylene oxide ratio. Special flushing procedures are required when switching between a PAG and mineral or other synthetic fluid lubricant; the lubricant supplier is consulted for specific details. A secondary difficulty with PAG lubricants is that they can require different specifications for paints, seals, sealants, and filters. Also, special handling would be required for the disposal of PAG-type lubricants.

3.3.2.4 Phosphate esters

While there are many groups of phosphates, it is the trisubstituted, neutral esters of orthophosphoric acid that have found significant use as synthetic base stocks. The commercially significant derivatives used as synthetic base stocks are compounds in which all three substituents on the phosphorus molecule are alky, aryl, or alkyl-aryl moieties containing at least four carbon atoms plus hydrogen and oxygen. They are probably best known for their inherent fire-resistance and find wide use as fire-resistant industrial hydraulic fluids. Additionally, they can be used as gear lubricants in the gearboxes of gas and steam turbines.

The trisubstituted phosphate esters, being neutral, have demonstrated chemical stability through many years of practical industrial service over a wide temperature range. They generally do not react with most organic compounds and are excellent solvents for most commonly used lubricant additives. In addition, they have demonstrated excellent thermal and oxidative stability in various laboratory tests. When one thinks of synthetics, the most common characteristic is excellent viscosity-temperature relationships. This, however, is not the case for phosphate esters as they typically have viscosity indices (VI) below 100.

Consideration should also be given to phosphate ester-type fluids during service due to their affinity with water.

3.4 Thickeners

Thickeners, also known as viscosity modifiers (VM) or viscosity index improvers (VII), are not common in industrial gear oil formulations, but are used in some applications. Thickeners are generally polymers, which cause the oil to thicken to a much greater extent per unit volume of material than a conventional base stock, such as a bright stock or cylinder stock. At higher temperatures the molecule expands creating a thickening effect. As the temperature decreases the polymer molecule tends to contract minimizing the thickening effect. A schematic diagram of this principle is shown in Figure 3. The unique ability of these polymers to expand and contract as a function of temperature enables the finished blend to have much better viscosity-temperature characteristics, thus the terms VM and VII.

Polymers are merely a chemical combination of one starting unit, known as a monomer, into many repeating units. The properties of the polymer are a function of the relative molecular mass (M_r, the number of repeating units) and the chemical structure of the monomer. Some of the more common polymer types used as viscosity modifiers include poly-alpha-olefin, poly-isobutylene, poly-alkyl-acrylate and -methacrylate, and olefin copolymers.

In addition to altering the viscosity-temperature properties of the finished fluid the choice of polymer can also have an impact on the supporting film in the gear and bearing contact regions. The film formed in the contact will be a function of the temperature, pressure and velocity of the surfaces that come into contact with each other. On the negative side, polymers are subject to mechanical and thermal shearing which results in a temporary and/or permanent loss of viscosity. The rate of loss is directly proportional to the molecular weight (M_r) of the polymer, i.e., higher relative molecular mass polymers result in higher viscosity losses (see

Figure 3). Different polymer structures can also influence the response to pressure, temperature, and shear rate. Each of these parameters becomes important in the overall choice of a thickener.

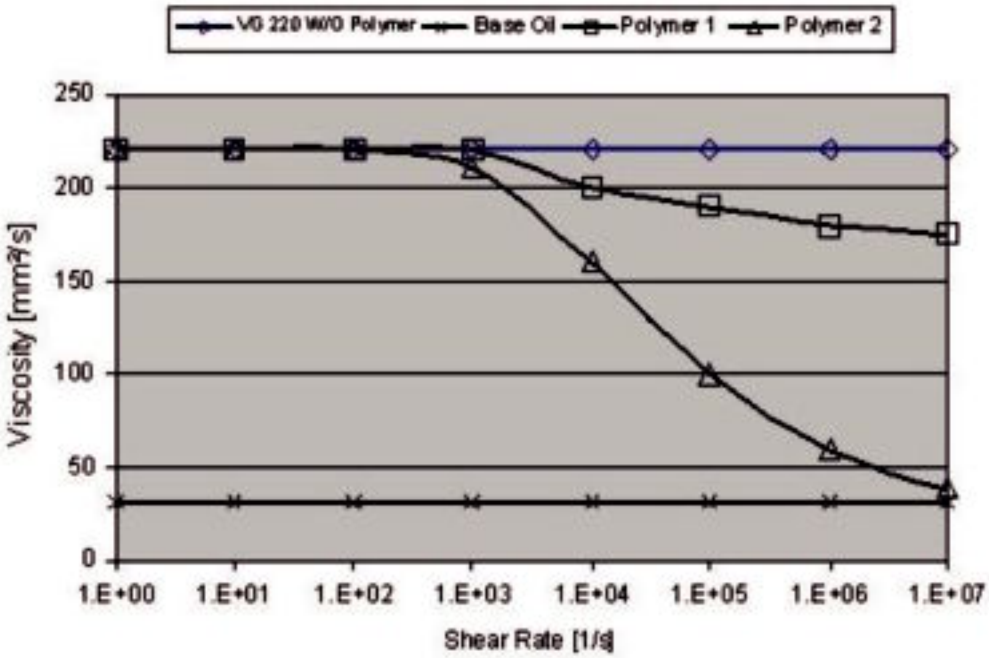


Figure 3 — Schematic diagram of shear effects on thickeners

3.5 Chemical properties of additives

Additives are typically a small (volume-wise), but critical part of the overall formulation. Additive is a broad term that encompasses many different chemicals, each providing performance or protection against certain types of damage or distress. These performance areas include, but are not limited to, antiwear (AW), extreme pressure (EP) or antiscuff, ferrous corrosion, non-ferrous corrosion, demulsibility, oxidation and foam inhibition. The chemicals impart or control these performance aspects in the application through reaction with the component surface or in the bulk oil phase. Most gear lubricants use a variety of chemicals in order to satisfy the many needs of the application. These chemicals must be selected properly not only for the desired performance function but for compatibility with the other chemicals in the package so that performance is not degraded.

Most commercial gear lubricants contain additives or chemicals that enable them to meet specific performance requirements. Typical additives include: rust inhibitor, oxidation inhibitor, defoamant, AW and antiscuff agents. Many of the chemicals used to form the additive “package” are single function, but some can provide benefit in multiple areas. For example, certain thiophosphorus compounds while primarily used for AW can also provide protection against scuffing or function as oxidation inhibitors. As a minimum base, oils are treated with some type of rust inhibitor and antioxidant; these are commonly known as R&O or circulating oils. These oils are not intended for applications where boundary lubrication is expected to occur. Blends containing AW and antiscuff agents are generally referred to as EP oils.

Additives alone, however, do not establish oil quality with respect to oxidation resistance, demulsibility, low temperature viscomentrics and viscosity index. Lubricant producers do not usually state which compounds are used to enhance the lubricant quality, but only specify the generic function such as AW, EP agents, or oxidation inhibitors. Furthermore, producers do not always use the same additive to accomplish a particular goal. Consequently, it is possible for any two brands selected for the same application not to be chemically identical. Users should be aware of these differences, which can have significant consequences when mixing different products. Another important consideration is incompatibility of lubricant types. Some oils, such as those used in turbine, hydraulic, and gear applications, are naturally acidic. Other oils, such as engine oils and some automotive driveline fluids, are alkaline. Acidic and alkaline lubricants are incompatible. Oils for similar applications but produced by different manufacturers can be incompatible owing to the additives used. When incompatible fluids are mixed, the additives can be consumed due to chemical reactions with one another. The resulting oil mixture can be deficient of essential additives and therefore unsuitable for the intended application. When fresh supplies of the oil in use are not available, the lubricant manufacturer should be consulted for a recommendation of a compatible oil. Whenever oil is added to a system, the oil and equipment should be checked frequently to ensure that there are no adverse reactions between the new and existing oil. Specific checks should include bearing temperatures and signs of foaming, rust or corrosion, and deposits.

Certain precautions must be observed with regard to lubricant additives. Some additives are consumed during use as part of their method of functioning. As these additives are consumed, lubricant performance for the specific application is reduced and equipment failure can result under continued use. Oil monitoring programmes should be implemented to periodically test oils and verify that the essential additives have not been depleted to unacceptable levels.

3.6 Solid lubricants

Solid lubricants have been used in many different ways over the years to provide additional functionality and performance to the application. They have been used as supplements to liquid lubricants and greases and as dry film coatings in specialized applications where liquid lubricants or greases could not be used. The solids can be grouped into a few classes, the most common being lamellar and polymer. Graphite and molybdenum disulfide (MoS_2) are the best known examples of lamellar solids and poly tetra-fluoroethylene (PTFE) or Teflon®[1)] is the most well-known polymer type. When these are applied to the surface they have a very great effect on the friction of the interacting surfaces. The key issue with many solid lubricants when used as additives in other liquid or grease formulations is enabling them to reach the surfaces and perform their function. Attention must also be given to the type of lubricating system in place on the equipment, i.e. splash or pressure-fed, pump tolerances, filtration levels, etc., as problems could arise that outweigh the potential benefit of the solid lubricant.

WARNING — In the case of gear oils, special attention is drawn to the fact that the effect of the solid lubricants is not impeded by the existing detergent/dispersant additive system. This is why highly concentrated solid lubricant suspensions should only be added to EP gear oils after consultation with the manufacturer of these oils.

3.7 Friction and temperature

The local coefficient of friction changes also with local parameters of load and speed. Mean values for the coefficient of friction along the path of contact can be recalculated from power loss measurements.

The balance between power loss in the components of a transmission and heat dissipation over the housing and the shafts results in a steady state oil temperature of a sump lubricated gearbox. For spray lubrication, part of the generated heat is removed in an external radiator. The oil bulk temperature is regarded as the decisive parameter for the thermal-oxidative behaviour of the lubricant throughout its service life. The allowable maximum oil sump temperature for a given application is dependent on the choice of base oil type and additive chemistry. Sump temperatures in excess of 95 °C can require special materials for non-metallic components such as oils, seals and shims.

Heat dissipation is related to ambient temperatures, typically in the range of –40 °C to +55 °C. The ambient temperature is defined as the dry bulb air temperature in the immediate vicinity of the installed gears. Specific oil type and viscosity grade will be determined, in part, by ambient temperature.

The mean gear tooth temperature determines the relevant viscosity of the lubricant transported into the contact and thus film thickness and together with surface roughness the lubricating regime of boundary, mixed or elasto-hydrodynamic lubrication. Film thickness is directly or indirectly correlated with wear, scuffing, micropitting and pitting performance of the gear pair.

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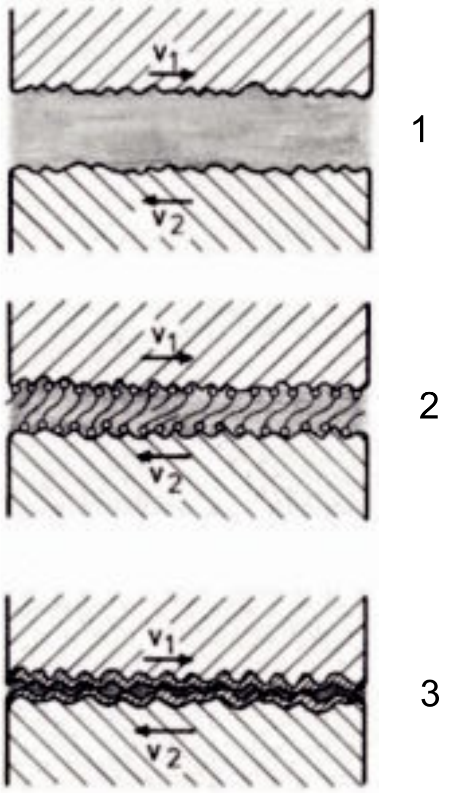
1) Poly tetra-fluoroethylene, known under the trade name Teflon®, is an example of a product available commercially. This information is given for the convenience of users of ISO/TR 18792 and does not constitute an endorsement by ISO of this product.

The gear bulk temperature, together with the local flash temperature in the mesh, govern the scuffing behaviour of the gear pair.

3.8 Lubricating regime

Using local values of load, speed and temperature, the local film thickness can be calculated along the path of contact. A good approximation for the estimation of the lubricating regime is to calculate the film thickness at the pitch point, introducing the viscosity, η, at gear bulk temperature, ϑ_M, and the pressure-viscosity coefficient, α, of the lubricant, and ϑ_M into the film thickness equation of Dowson and Higginson [54]. The relation λ between the calculated film thickness and the effective surface roughness determines the lubricating regime in the contact. A correlation between the specific film thickness, λ, and gear failures is shown by Wellauer and Holloway [55].

For λ values below 0,7 boundary lubrication, for λ between 0,7 and 2 mixed lubrication and for λ over 2, full film separation is expected. To prevent damage, the flank surfaces must be protected from direct metal-to-metal contact. For full film lubrication, the viscosity effect is sufficient, for smaller film thickness values, additives building up physically adsorbed or chemical reaction layers have to protect the flank surface (see Figure 4).



Key

1 elastohydrodynamic (hydrodynamic) lubrication

2 lubrication by physically adsorbed layers

3 boundary lubrication by chemically reacted layers

Figure 4 — Mechanisms of surface protection for oils with additives

3.9 Lubricant influence on gear failure

Definitions and pictures of individual failures are given in ISO 10825 [16]. The mechanism of gear failures influenced by lubricants is given in the following paragraphs.

3.9.1 Wear

Wear is a continuous removal of material from the flank surfaces. It occurs when there is asperity contact between the mating surfaces. As the film becomes thinner there is more asperity contact and, therefore, more wear. Wear rates can be affected by many factors such as lubricant, material properties and operating conditions. Some of the influential factors and their relative effects are shown in Table 3.

Table 3 — Example of influence factors on wear

		Life time		
Influence		**Lower**	**Same**	**Higher**
Material	**pairing of same hardness**			
	reference: case carburized			
	gas nitrided			xx
	through hardened	x		
	globular cast iron	x		
	hardness difference up to 50 HV	x		
	pairing of different hardness			
	reference: case carburized/case carburized			
	through hardened/through hardened		x	
	case carburized/through hardened	xx		
	case carburized/globular cast iron	x		
Lubricant	**reference: straight mineral oil ISO VG 220**			
	straight mineral oil ISO VG 460			x
	ISO VG 220 with AW additives			x
	ISO VG 220 with EP additives	x	x	x
	unlubricated, hard/hard	xx		
	unlubricated, soft/soft	xxx		
Geometry	**reference: module 4 mm, balanced sliding, without profile modification**			
	unbalanced sliding		x	
	module 8 mm			x
	adequate tip relief			x
x Lower, higher				
xx Much lower, higher				
xxx Very much lower, higher				

3.9.2 Scuffing

At high pressure and temperature without any surface protection the mating flanks weld together and due to the inherent energy and kinematics of the system are immediately torn apart again. Thus scuffing always occurs in corresponding areas of the mating tooth flanks, typically near the tooth root and tip with high sliding speeds. Scuffing is an instantaneous failure where one single and short overload can already cause catastrophic failure.

Newly manufactured surfaces have a higher probability of scuffing than well run-in surfaces. It has been shown that new surfaces can only carry 20 % of the load compared to run-in surfaces [55]. Lubricants with EP additives can improve resistance to scuffing, but other factors also influence performance. Some of the influential factors affecting scuffing are shown in Table 4.

Table 4 — Example of influence factors on scuffing load (transmittable torque)

		Torque capacity		
Influence		**Lower**	**Same**	**Higher**
Material surface	**reference: case carburized, martensitic structure, *Ra* = 0,5 µm, run-in surface**			
	Ra = 0,1 µm			x
	newly manufactured surface	xx		
	through hardened		x	
	gas nitrided			x
	phosphated			x
	copper plated			x
	30% retained austenite	x		
	stainless steel	xx		
Lubricant	**reference: straight mineral oil ISO VG 220**			
	straight mineral oil ISO VG 460			x
	ISO VG 220 with AW additives			x
	ISO VG 220 with EP additives			xx
Geometry operating conditions	**reference: spur gear module 4 mm, balanced sliding, without profile modification, medium speed**			
	unbalanced sliding			
	module 8mm	x		
	helical gear	x		
	adequate tip relief	x		x
	low speed			xx
x Lower, higher xx Much lower, higher xxx Very much lower, higher				

3.9.3 Micropitting

Gears running under mixed or boundary lubrication conditions can experience micropitting. Micropitting is a form of surface fatigue which occurs mainly, but not exclusively, in the dedendum of the gear flanks under negative sliding conditions. It is characterized visually as a grey, matte finish area on the tooth surface. It can progress to an unacceptable, material loss with increased dynamics and secondary failures. Such items as tooth modifications, surface roughness, viscosity, and the choice of additive have been shown to have an influence on the amount of micropitting. A summary of some of the key factors influencing micropitting are shown in Table 5.

Table 5 — Example of influence factors on micropitting (transmittable torque)

		Torque capacity		
Influence		**Lower**	**Same**	**Higher**
Material surface	**reference: case carburized, martensitic structure, *Ra* = 0,5 µm**			
	Ra = 0,1 µm			xx
	through hardened			x
	30% retained austenite			x
Lubricant	**reference: straight mineral oil ISO VG 220**			
	straight mineral oil ISO VG 460			x
	ISO VG 220 with AW/EP additives	x	x	x
	polyalphaolefin			x
Geometry operating conditions	**reference: balanced sliding, without profile modification, medium speed**			
	high negative sliding	x		
	adequate tip relief			x
	high speed			x
x Lower, higher				
xx Much lower, higher				
xxx Very much lower, higher				

3.9.4 Pitting

Pitting is a fatigue failure that occurs mainly in the dedendum of the gear flanks in the area of negative sliding under high Hertzian stress and surface shear conditions. There is generally a strong relationship between stress levels and cycles to failure. Again, key factors such as material properties, lubricant and operating conditions can have an effect on the pitting life of gears as shown in Table 6

Table 6 — Example of influence factors on pitting (transmittable torque)

		Torque capacity		
Influence		**Lower**	**Same**	**Higher**
Material surface	**reference: case carburized, martensitic structure, ground and run-in surface**			
	through hardened	xx		
	gas nitrided	x		
	hobbed	x		
	copper plated			x
	30 % retained austenite			x
Lubricant	**reference: straight mineral oil ISO VG 220**			
	straight mineral oil ISO VG 460			x
	ISO VG 220 with AW/EP additives	x	x	x
	polyalphaolefin, polyalkyleneglycole			x
Geometry	**reference: spur gear module 4 mm, pressure angle 20 °, without profile modification**			
	module 8 mm		x	
	pressure angle 28°			x
	high addendum teeth			x
	helical gear			x
	adequate tip relief			x
x Lower, higher				
xx Much lower, higher				
xxx Very much lower, higher				

4 Test methods for lubricants

4.1 Gear tests

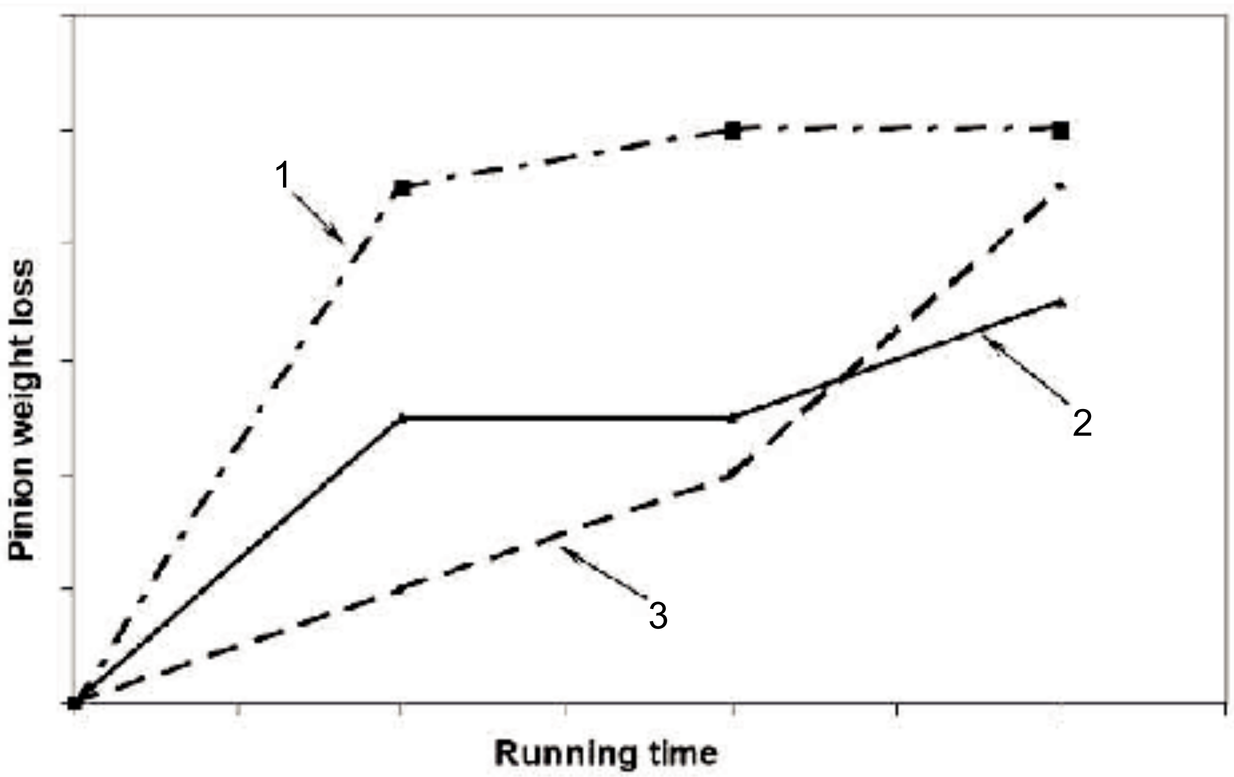
Experience has shown that correlation with practice is enhanced when gear tests with appropriate conditions are performed. Simple bench testing may be used for batch or identification controls when base oil, additive types and concentration are known [61].

4.1.1 Wear test

Wear performance of gear oils may be tested with the FZG test method as specified in ASTM D 4998 [38]. Test gears are run at a pitch line velocity of 0,57 m/s and an oil sump temperature of 121 °C under load for 20 h. After that, the weight losses of pinion and wheel are determined. Experience has shown advantages in rating the AW performance of industrial gear oils when carrying out the test a sufficient number of times to ensure repeatability (see Figure 5).

Another test method may be conducted at a pitch line velocity of 0,05 m/s with oil sump temperature of 90 °C in a first test sequence and 120 °C in a second sequence [45]. The result of the test can be expressed as a specific wear rate; this calculation is based on a method outlined by Plewe [62].

Gear lubricants are often used to lubricate bearings as well as gears. DIN 51819-3, a bearing wear test, is a common method for evaluating bearing wear performance of gear lubricants.



Key

1 oil A

2 oil B

3 oil C

Figure 5 — Examples of gear oil wear test results

4.1.2 Scuffing test

The scuffing load capacity of industrial gear oils may be evaluated with ISO 14635-1 [24]. This method is identical with national test methods as found in DIN 14635-1 [27], ASTM D 5182 [39], IP 334 [47] and CEC L-07-A-95 [44].

For lubricants of higher scuffing capacity as such used in manual gearboxes and final drives in automotive applications, modified procedures are used, e.g. ISO 14635-2 [25].

The results of the different scuffing tests can be introduced into the standard scuffing calculation specified in ISO/TR 13989 [20, 21].

4.1.3 Micropitting test

The micropitting capacity of lubricants may be tested using the test procedure described in Forschungvereinigung Antriebstechnik (FVA) Information sheet No. 54/I-IV [49].

Adjusted to the operating conditions of the gearbox, modifications of the test are performed at higher speed and/or lower temperature. For example, in wind turbine applications the oil temperature is reduced from 90 °C to 60 °C. In spite of better lubricating conditions with higher operating oil viscosity, sometimes lower micropitting capacity is observed due to different additive systems.

A micropitting capacity rating method is currently being investigated [57, 69].

4.1.4 Pitting test

The influence of lubricants on the pitting life of gears can be evaluated many ways. One method is described in FVA Information sheet No. 2/IV [50]. S-N curves can be developed from the data which can be useful when making comparisons or carrying out failure accumulation calculations [67]. Other operating conditions can be applied to meet the needs of different applications [49].

4.1.5 Efficiency test

The influence of lubricants on power loss and efficiency can be evaluated as specified in FVA Information sheet No. 345 [52]. Under these conditions load-independent and load-dependent losses in the area of boundary, mixed and full EHD conditions are measured and evaluated in comparison to a reference oil and as loss factors and mean values for the coefficient of friction.

The results of the test can be introduced into the calculation of power loss and expected gear oil temperature as per ISO/TR 14179-1 [22].

4.2 Other functional tests

4.2.1 Corrosion

There are several types of corrosion tests for petroleum products, depending upon the classification or the application of the lubricant. In order to examine the corrosion characteristics of a lubricant, tests are defined for conditions that approximate the conditions encountered in service. In general, properly formulated gear lubricants are not considered corrosive to steel or copper containing alloys. Most corrosion test methods are intended to measure the ability of a lubricant to prevent corrosion on an oil in contact with a metal surface. The methods are usually grouped, in keeping with the material to be protected, in general categories such as ferrous and non-ferrous. These tests indicate the tendency of the lubricant to prevent corrosion of the gears, bearings and other components while in service under normal operating conditions. If adverse conditions are expected such as high operating temperatures or high contamination levels or types, other considerations can be required to protect steel and cupric metal parts from corrosive attack.

The ISO 7120:1987 [14]/ASTM D 665-95 [32] test method evaluates the ability of an oil to prevent the rusting of ferrous parts in the event that water becomes mixed with the oil. The method consists of two parts: Procedure A using distilled water, and Procedure B using synthetic seawater. In this test method, 10 % water (distilled or synthetic seawater) is mixed in the oil and a polished 10180 grade carbon steel rod is immersed in the stirred mixture for 24 h at 60 °C (140 °F). If there is no rust on the specimen, the oil passes the test.

The ISO 2160 [5]/ASTM D 130 [30] copper corrosion test method measures the protective nature of lubricating oil on a copper strip that is immersed under static conditions in the oil. Sulfur containing compounds are the main sources of tarnishing or corroding of the copper and its metal alloys. The extent of the reactivity of the copper with the oil is classified by comparing the appearance to standard coupons. The method consists of placing a polished, cleaned copper strip in a test tube with the oil sample. The test is run for 3 h at 100 °C (212 °F). Discolouration of the copper is matched against reference standards and the oil is rated on a scale of increasing corrosivity from 1 to 4. An acceptable gear rear oil is required to a rating of 1a or 1b, which is considered slightly tarnished.

4.2.2 Oxidation resistance and thermal stability

Oxidation is a chemical process in which oxygen combines with the free radicals generated within a lubricant to produce organic acids that can corrode metals and produce higher molecular weight by-products which produce sludge and deposits in a lubricated system. Another product of oxidation is increased viscosity. Oxidation is enhanced by an elevated temperature and in the presence of a catalyst such as copper, water or foreign matter. Thermal stability is often, but inappropriately, interchanged with oxidation. Thermal stability is the property of a lubricant that characterizes its relative chemical stability in response to thermal stress.

A thermally unstable compound can decompose in response to heat alone, without the contribution of the oxidative processes. Thermal decomposition, like oxidation, can be catalyzed by metals, water, or other chemical compounds. Thermal breakdown products can themselves be reactive and promote oxidation, corrosion, or sludge formation.

There are many tests used to assess the thermal and oxidative stability of lubricants. Often these tests are metal catalyzed and it is possible for them to include the presence of water.

Oxidation resistance is an important measure of the functionality and useful service life of a lubricant. A lubricant's base oil and additive package are equally important determinants of its oxidation life. Operating temperature, however, is normally the most influential variable impacting the rate of oxidation. In any gear drive, localized heating (hotspots) must be taken into account in addition to a bulk lubricant operating temperature. These areas of localized heating can be sites where accelerated oxidative aging and thermal decomposition occur. Examples of localized heating include instantaneous frictional heat at the mesh point of the gear teeth (referred to as flash temperature), the point of highest load in support bearings and the surfaces of heating devices that come in direct contact with the lubricant.

4.2.3 Foaming

Foaming in a gear oil is detrimental to the performance and durability of the gear drive in which it is being used. It can also create housekeeping problems if it escapes the confines of the gear drive. Foaming in a lubricant can be controlled through the use of a foam inhibitor. This additive causes the foam to dissipate more rapidly by promoting the agglomeration of small bubbles into large bubbles which burst more easily. Foam inhibitors are commonly produced from silicon or polymeric compounds.

One method commonly used to measure the foaming tendency of oils is that of ISO 6247 [12]/ASTM D 892. This involves subjecting a fixed volume of the test oil in a graduated cylinder to air flowing through a fritted sparging device for 5 min and measuring the increase in volume of the liquid/foam mixture at the finish of the flowing air and then again after another 10 min of standing. This is done at 24 °C, 93.5 °C, and repeated at 24 °C with the same oil used in the 93,5 °C evaluation as specified in ISO 6247. The criteria for acceptance are defined by the user, consistent with the needs of the application.

4.2.4 Air entrainment

Air entrainment is also referred to as the air release property of a fluid. With industrial oils, this property is determined by establishing the density of the fluid in its natural state, aerating it, and measuring the time it takes to return to its original density. Viscosity and temperature will affect the rate at which a fluid will release entrained air. The ability of the bulk fluid to release entrained air is an inherent property of the base fluid. A base fluid with marginal air release capabilities in neat form can develop severe air entrainment with the use of the wrong combination of additives and/or the use of too high a concentration of additives. The same applies to base fluids with excellent inherent air release properties if too much additive is used. Therefore, the doping of gear oil with additional additives, especially foam inhibitors, should only be attempted under the careful guidance of the lubricant supplier. The addition of improper or too much additional additive(s) can lead to major gear drive operational problems and possible irreversible damage to the gears and/or bearings.

The only standardized test method designed to quantify air entrainment in oils is ISO 9120 [15], but this is limited to light viscosity turbine type oils. There are no specific requirements for air entrainment properties for high viscosity oils typically used in industrial gear applications. There are some specialized test methods for certain applications where air entrainment/release properties are important. An example is the Flender foam test which measures both the foaming tendency of the oil and its subsequent time to release the air entrained during the foam portion of the test.

4.2.5 Demulsibility

Demulsibility, also known as water separation, is the ability of a lubricating fluid to separate from water. The common demulsibility test method used for light and medium viscosity gear oils is ASTM D 1401 [34]. In this method, a 40 ml sample of both oil and distilled water are placed in a 100 ml graduated cylinder and placed in a temperature bath. The test is run at 54 °C (130 °F) for ISO 68 and lower oils and 82 °C (180°F) for oils that are ISO VG 100 and greater. After stirring (1 500 rpm) for 5 min at the appropriate test temperature, the time required for the oil and water to separate is measured. If separation does not occur in 1 h, the test is stopped and the volumes of water, oil and emulsion are recorded. For medium and high viscosity gear oils the typical test used is ASTM D 2711 [35], also known as the Wheeling Steel Demulsibility Test since it was developed by Wheeling Steel to measure the water separation properties of oils used to lubricate steel rolling mill stands. In the ASTM D 2711 [35] test method, 405 ml of oil and 45 ml of water are stirred together (4 500 rpm) for 5 min in a separatory funnel at 82 °C (180 °F). After settling for 5 h, a 50 ml sample is withdrawn from near the top and centrifuged to determine the “percentage of water in oil, volume, %”. The free water is drained from the bottom of the funnel and then a second volume of 100 ml of oil and water emulsion is withdrawn and centrifuged. The initial amount of free water drawn off plus the centrifuged water is recorded as “total free water”. The amount of water and oil remaining as emulsion after centrifuging is recorded as “Emulsion, mls”. This method was developed specifically for rust and oxidation inhibited oils, but it can be used to test high viscosity circulating oils and anti-scuff / EP gear oils. For these types of lubricants the method is usually modified to reduce the oil amount to 360 ml, increase the water to 90 ml and the stirrer slowed to 2 500 rpm.

4.2.6 Elastomer compatibility

The compatibility of lubricant with elastomer can be measured in a number of ways depending on the sealing system and its requirements. Two major types are static immersion testing and dynamic testing. Dynamic tests require special rigs and are often conducted to an equipment manufacturer's preferred duty cycle. A test can last 500 to 1 000 h or more. Dynamic testing is usually assessed by quantifying the amount of leakage that occurs during the course of the test. In addition, at the end of testing some require more in-depth analysis of the seal itself. This also requires specialized equipment, which is usually only available at the seal vendor’s laboratory.

Static immersion tests are popular and relatively simple to conduct. Static immersion test methods such as ISO 6072 [11] or ASTM D 5662 [41] are examples. These tests usually consist of suspending samples of the elastomer in a glass test tube containing the oil to be assessed. The test tube is placed in a controlled heated bath for a specified length of time. At the end of the specified time the elastomer samples are removed and rinsed with a hydrocarbon solvent to remove the oil. The elastomer is then evaluated for changes in volume, hardness, tensile strength and elongation.

Although ASTM D 5662 [41] specifies certain elastomer types and test conditions, these may be modified to accommodate the needs of specific end user applications. Regardless of the method chosen to determine elastomer compatibility, it is always recommended that the results are compared with a standard, or the results obtained with a reference oil, preferably one with a positive field service history.

4.2.7 Filterability

Oil with poor filterability characteristics will plug filters and can cause inadequate lubrication of vital machine components. It has been found that the poor filterability characteristics of some industrial lubricants are caused by the use of certain base stocks or additives, or contamination of the finished oil. Filterability has been assessed by several methods, with ISO 13357-1 [17] and ISO 13357-2 [18] the first to become widely accepted. However, these methods are designed to evaluate low viscosity turbine and hydraulic type oils. There are no International Standards available at this time for higher viscosity oils typically used in industrial gear applications. Equipment manufacturers have long been aware of the importance of filterability. Several have developed in-house filterability test methods. As with ISO 13357-2 [18], some of the methods determine the time to filter a quantity of water-free oil through a specified filter under prescribed conditions.

Since many types of filter media are adversely affected by the presence of water, some filterability test methods like ISO 13357-1 [17] will measure the filterability of a mixture of oil and water after it has been subjected to an ageing procedure. This test method is meant to simulate in-service conditions and to assess whether filtration efficiency is impaired after the oil has been in service for some time.

5 Lubricant viscosity selection

5.1 Guideline for lubricant selection for parallel and bevel gears (not hypoid)

For enclosed gearing, the following guidelines are for selection of a lubricant for cylindrical and bevel gearing.

In the absence of a recommendation from the gear manufacturer the following method is offered to select a viscosity for the application.

These guidelines are available for a tangential speed ranging from 1 m/s up to 30 m/s and for an oil temperature from 10 °C up to 100 °C.

In order to select the lubricant it is necessary to use the pitch line velocity of the lowest speed mesh. Consideration should be given to the following points.

— Viscosity requirements of the bearings.

— The viscosity calculated is compatible with its lubrication system (if present).

— The tangential velocity of the high-speed stage of the gearbox should be less than 35 m/s.

Tables 7, 8, 9 and 10 cover four representative viscosity index type fluids.

Table 7 — ISO Viscosity grade[1] at bulk oil operating temperature for oils having a viscosity index of 90[2]

Temp. °C	Pitch line velocity m/s [3] 1,0 – 2,5	2,5	5,0	10,0	15,0	20,0	25,0	30,0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	46	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1 000	680	320	220	150	100	100	68
75	1 500	680	460	320	220	150	150	100
80	2 200	1 000	680	460	220	220	220	150
85	3 200	1 500	1 000	460	320	220	220	150
90	3 200	2 200	1 000	680	460	320	320	220
95		3 200	1 500	1 000	460	460	320	220
100		3 200	2 200	1 000	680	460	460	320

NOTES:

1)

- (hatched) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade less than 32 or greater than 3 200 is indicated.
- (dark grey) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
- (white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
- (light grey) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine the pitch line velocity of all gear sets. Select the viscosity grade for the critical gear set taking into account cold start-up conditions.

Table 8 — ISO Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 120[2)]

Temp.	Pitch line velocity m/s [3)]							
°C	1,0 – 2,5	2,5	5,0	10,0	15,0	20,0	25,0	30,0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	100	100	68	46	46	46
55	320	220	150	100	68	68	46	46
60	460	320	220	150	68	68	68	46
65	680	460	320	150	100	100	100	68
70	1 000	460	320	220	150	150	100	68
75	1 000	680	460	220	150	150	150	100
80	1 500	1 000	460	320	220	220	150	100
85	2 200	1 000	680	460	220	220	220	100
90	2 200	1 500	1 000	460	320	320	220	150
95	3 200	2 200	1 000	680	320	320	320	220
100		2 200	1 500	680	460	460	320	220

NOTES:

1)

(hatched)	Consult gearbox, bearing and lubricant manufacturers if a viscosity grade less than 32 or greater than 3 200 is indicated.
(dark grey)	Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
(white)	Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
(light grey)	Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine the pitch line velocity of all gear sets. Select the viscosity grade for the critical gear set taking into account cold start-up conditions.

Table 9 — ISO Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 160[2)]

Temp. °C	Pitch line velocity m/s [3)] 1,0 – 2,5	2,5	5,0	10,0	15,0	20,0	25,0	30,0
10	32	32						
15	46	32	32					
20	68	46	32					
25	68	46	32	32				
30	100	68	46	32				
35	150	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	
50	220	150	100	68	46	46	46	32
55	320	220	150	100	68	68	46	32
60	460	220	150	100	68	68	68	46
65	460	320	220	150	100	100	68	46
70	680	460	220	150	100	100	100	68
75	680	460	320	220	150	150	100	68
80	1 000	680	320	220	150	150	150	100
85	1 500	680	460	320	220	220	150	100
90	1 500	1 000	680	320	220	220	220	150
95	2 200	1 500	680	460	320	220	220	150
100	3 200	1 500	1 000	460	320	320	220	150

NOTES:

1)

- (hatched) Consult gearbox, bearing and lubricant manufacturers if a viscosity grade less than 32 or greater than 3 200 is indicated.
- (dark grey) Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades
- (white) Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
- (light grey) Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine the pitch line velocity of all gear sets. Select the viscosity grade for the critical gear set taking into account cold start-up conditions.

Table 10 — ISO Viscosity grade[1)] at bulk oil operating temperature for oils having a viscosity index of 240[2)]

Temp. °C	Pitch line velocity m/s [3)] 1,0 – 2,5	2,5	5,0	10,0	15,0	20,0	25,0	30,0
10	46	46						
15	68	46	32					
20	68	68	32	32				
25	100	68	32	32				
30	100	68	32	32	32			
35	150	68	68	46	32	32		
40	150	100	68	46	32	32	32	
45	220	100	100	68	46	32	32	
50	220	100	100	68	46	46	46	32
55	320	150	150	68	68	46	46	32
60	320	150	150	100	68	68	46	46
65	460	220	150	100	100	68	68	46
70	460	320	220	150	100	68	68	46
75	680	320	220	150	100	100	68	68
80	680	460	220	150	100	100	100	68
85	1 000	460	320	220	150	100	100	68
90	1 000	680	320	220	150	150	100	100
95	1 000	680	460	320	220	150	150	100
100	1 500	1 000	460	320	220	150	150	100

NOTES:

1)

Hatched	Consult gearbox, bearing and lubricant manufacturers if a viscosity grade less than 32 or greater than 3 200 is indicated.
Dark grey	Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering these viscosity grades.
White	Select the viscosity grade that is most appropriate for the anticipated stabilized bulk oil operating temperature range.
Light grey	Baseline stabilized bulk oil operating temperature and bearing lubrication requirements.

2) This table assumes that the lubricant retains its viscosity characteristics over the expected oil change interval. Consult the lubricant manufacturer if this does not apply.

3) Determine the pitch line velocity of all gear sets. Select the viscosity grade for the critical gear set taking into account cold start-up conditions.

5.2 Guideline for lubricant selection for worm gears

The following guideline is available to select a lubricant for worm gears gearing.

In the absence of a recommendation from the gear manufacturer the following method, in Table 11, is offered to select a viscosity for the application.

Table 11 — ISO viscosity grade guidelines for enclosed cylindrical worm gear drives

Pitch line velocity of final reduction stage	**ISO viscosity grade for ambient temperature**, °C [a, b]		
	−40 to −10	−10 to +10	+10 to +55
Less than 2,25 m/s	220	460	680
Above 2,25 m/s	220	460	460

[a] Worm gear applications involving temperatures outside the limits shown above, or speeds exceeding 2 400 rpm or 10 m/s sliding velocity are addressed by the manufacturer. In general, for high speeds a pressurized lubrication system accompanies adjustments in the recommended viscosity grade.

[b] This table is relevant to lubricants with a viscosity index of 100 or less. For lubricants with a viscosity index greater than 100, wider temperature ranges are appropriate. The lubricant supplier is usually consulted.

5.3 Guideline for lubricant selection for open girth gears

5.3.1 Lubricant selection for open girth gear

Table 12 lists some of the advantages and disadvantages of various lubricants for open girth gears.

Table 12 — Advantages and disadvantages of various open girth gears lubricants

Product	Advantages	Disadvantages
Oil Either petroleum-based (R&O), EP, or synthetic oils with or without EP additives. These products also operate on the principle of an oil film separating the surfaces of the gear and the pinion. These oils are applied in much the same way as the residual compounds.	Diluents are not generally needed to aid flow of lubricant. High viscosity. No build-up tendency in tooth roots or on gear guard. Drains freely from gear guards. Very good inspection results.	Heat tracing and drum heaters might be required to obtain a proper spray pattern. Annual usage cost might be higher.
Greases Petroleum-based or synthetic oils to which soap thickeners or carriers are added. Friction modifiers (typically, graphite and molybdenum disulfide) and EP chemicals are usually added. Some have thixotropic properties where the viscosity of the lubricant changes with the pressure experienced during operation.	Diluents are not generally needed to aid flow of lubricant. Can have better low temperature pumping characteristics with 0 or 00 grades.	Necessitates use of run-in compounds. Application rates are more frequent, with less volume. Possible greater total usage than other products. Marginal film thicknesses. Lubricant builds up on gear guard sides. Annual usage costs can be higher. Base oil in the grease is the only source of viscosity Total loss lubricant. Difficult to see tooth surface at inspection.
Residual compounds A viscous mixture of petroleum-based compounds, also referred to as asphaltics. Most residual compounds use non-chlorinated diluents to provide pump ability. Most contain EP additives or friction modifiers (solid lubricants) such as graphite or molybdenum disulfide.	High viscosity. Diluents allow cleaner spray nozzles, aid flow, and allow lower temperature pumping. Newer base stocks no longer build up in the tooth roots and on the gear guard. Residuals provide extended lubrication film retention. Spent product drains freely from gear guards.	Replacement solvents have a lower flash point. More frequent application can wash off lube. Requires air purge of nozzles to prevent clogging. Difficult to see tooth surface at inspection. Compressed air required.
Compounds A synthetic or petroleum-based oil with friction modifiers and EP additives. Some contain diluents for pump ability. Many have polymer additives as viscosity enhancers. Some have thixotropic properties where the viscosity of the lubricant changes with the pressure experienced during operation. These products utilize friction modifiers to assist their thinner load carrying oil films. These products are applied in much the same way as the residual compounds and oils, above.	The friction modifiers can be viewed as a safety margin in addition to the oil film. Friction modifiers can provide protection at start-up or very slow speeds.	Thinner oil film. More difficult to pump. Marginal EHD oil film. Harder to drain from gear guard.
Compounded oils (high viscosity)	High viscosity, high lubricant film thickness, high temperature stability, high pressure stability, no deposits in the tooth fillet, lubricant depot in the tooth fillet, high lubricant film retention.	Heater needed for: good transport and delivery; good spray ability; good drainage.

5.3.2 Continuous lubrication

Minimum kinematical viscosity recommendations for continuous lubrication are given in Table 13. For compounds, the base oil viscosity is used.

Table 13 — Minimum Viscosity recommendation for continuous lubrication [mm^2/s at 40 °C]

Ambient temperature	Tooth temperature	Spray and splash	Idler immersion
−10°C to 0 °C	< 45 °C	220	1 000
	> 45 °C	460	1 000
0 °C to 20 °C	< 45 °C	460	1 000
	> 45 °C	680	1 500
20 °C to 40 °C	< 45 °C	680	4 200
	> 45 °C	1 000	5 000
> 40 °C		⩾ 1 500	⩾ 6 200

5.3.3 Intermittent lubrication

Minimum kinematical viscosity recommendations for intermittent lubrication are given in Table 14. For grease, compound or residual products (asphalted) the base oil viscosity is used.

Table 14 — Minimum base oil viscosity recommendation for intermittent lubrication [mm^2/s at 40 °C]

Ambient temperature	Tooth temperature	Grease	Compounds	Residual lubricants[a]
−10 °C to 0 °C	< 45 °C	460	3 000	420-650 [mm^2/s at 100 °C]
	> 45 °C	680	4 000	
0 °C to 20 °C	< 45 °C	680	5 000	
	> 45 °C	1 000	8 000	850-1 100 [mm^2/s at 100 °C]
20°C to 40°C	< 45°C	2 000	10 000	
	> 45°C	2 000	16 000	
> 40°C		⩾2 2 000	16 000	

[a] Defined in agreement with equipment and lubricant manufacturers.

6 Lubrication principles for gear units

The most important functions of a lubricant in gear units are:

— reduction of friction;

— discharge of particles and foreign matter;

— prevention of corrosion;

— reduction of wear;

— dissipation of heat;

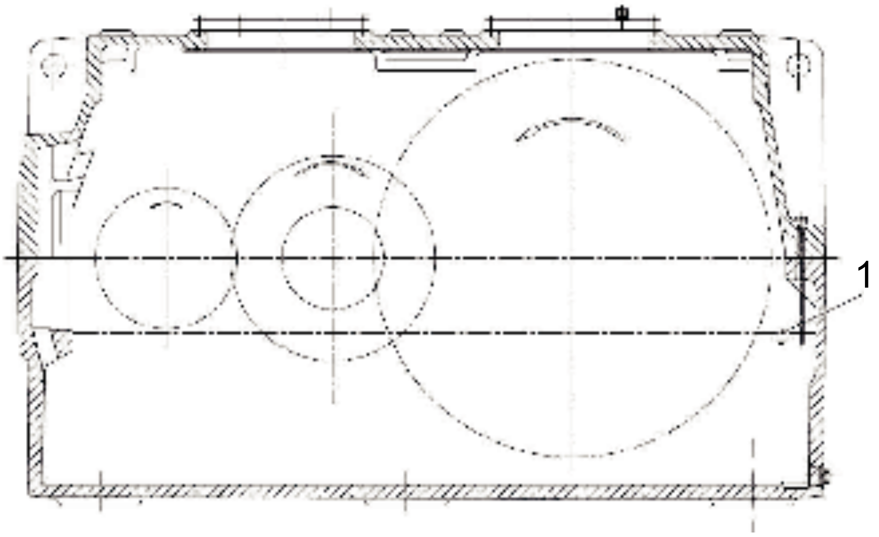
— medium for carrying additives.

To ensure that those tasks are fulfilled, an optimum quantity of lubricant has to be supplied to the proper location.

6.1 Enclosed gear units

6.1.1 Splash lubrication

The simplest method of supplying lubricant is splash lubrication, in which at least one of the gears of each gear stage is immersed in a lubricant bath as shown in Figure 6. Splash lubrication is standard for pitch line velocities up to 15 m/s (see Figure 6); with specific provisions it can reasonably be used up to 30 m/s and, in some special applications, even up to 100 m/s. The oil level in the gearbox has to be controlled by a dip stick or an oil sight glass, for instance. For adjusting the internal gearbox pressure due to heating and cooling, breathers are required. These can be simple vent holes or full systems including water and particle filters. Large oil inlet openings and smaller oil outlet holes are required. The oil outlet can be combined with a magnetic drain plug to prevent iron debris and wear particles from damaging the bearing races, in particular.



Key
1 oil level

Figure 6 — Immersion of gear wheels[2)]

Dependent on the actual gear design for pitch line velocities between 15 m/s and 30 m/s special shrouds with holes for limiting the oil access to the immersed gear wheel can be applied. Special baffles can be installed to direct a sufficient oil stream into the bearings and the gear mesh.

Sufficient immersion depth has to be maintained at any operating condition to provide lubricant to the bearings and gears for lubricant film formation and for heat removal from the mesh. Inclination of the gearbox has to be taken into account when immersion depth is defined (see Figure 7). Higher immersion depth causes, however, higher churning losses. Typical immersion depth of the gear wheel under operating conditions for horizontal centre distance is 2,5 times module, for vertical centre distance 6 times module. A minimum immersion depth

2) Source: A. Friedr. Flender AG.

of the gear wheel of 12 mm should be maintained in any case [66, 67]. The bearings have also to be immersed, either directly or with suitable aids.

The amount of lubricant in the gearbox has to be chosen considering air release and oil ageing phenomena. For industrial gearboxes a typical oil quantity, V, in litres, is given in Equation (1):

$$4P_{\text{vzsum}} \leqslant V \leqslant 12P_{\text{vzsum}} \tag{1}$$

where

P_{vz} is the estimated gear power loss for one stage, in kW, from Equation (2):

$$P_{\text{VZ}} \approx P \frac{0{,}1}{z_1 \cdot \cos\beta} + \frac{0{,}03}{v+2} \tag{2}$$

where

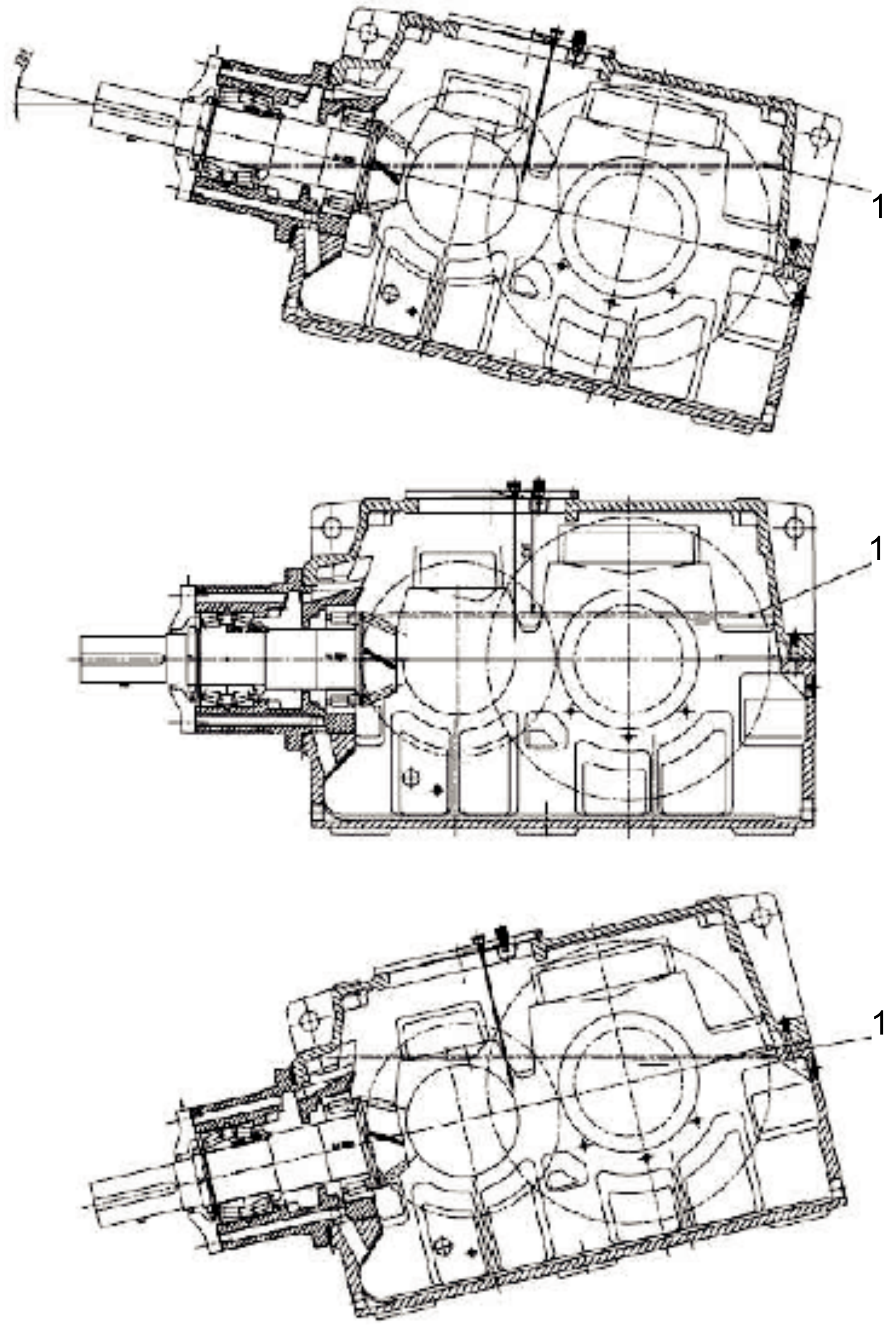
z_1 is the number of pinion teeth;

β is the helix angle;

v is the pitch line velocity, in m/s.

The oil volume in multistage gearboxes can be limited by adjusted design of the gearbox bottom (see Figure 8).

Splash lubrication is limited by the thermal capacity of the gearbox. The power loss from gears, bearings, seals etc. in the system has to be dissipated to the ambient at a reasonable oil temperature level. Typical maximum recommended continuous oil temperature in an industrial gearbox should not exceed 95 °C, independent of the oil type used. Thermal capacity calculation as the equilibrium between generated and dissipated heat is given in ISO/TR 14179-1 and ISO/TR 14179-2 [22, 23]. For too high calculated gearbox temperatures, the generated heat can be reduced by, for example, reducing churning losses with reduced oil viscosity or mesh losses by using synthetic low friction oils or the dissipated heat can be increased by additional cooling fins or an external fan, for instance. In case the steady state oil temperature cannot be reduced by these measures to an acceptable value, external oil cooling has to be applied.

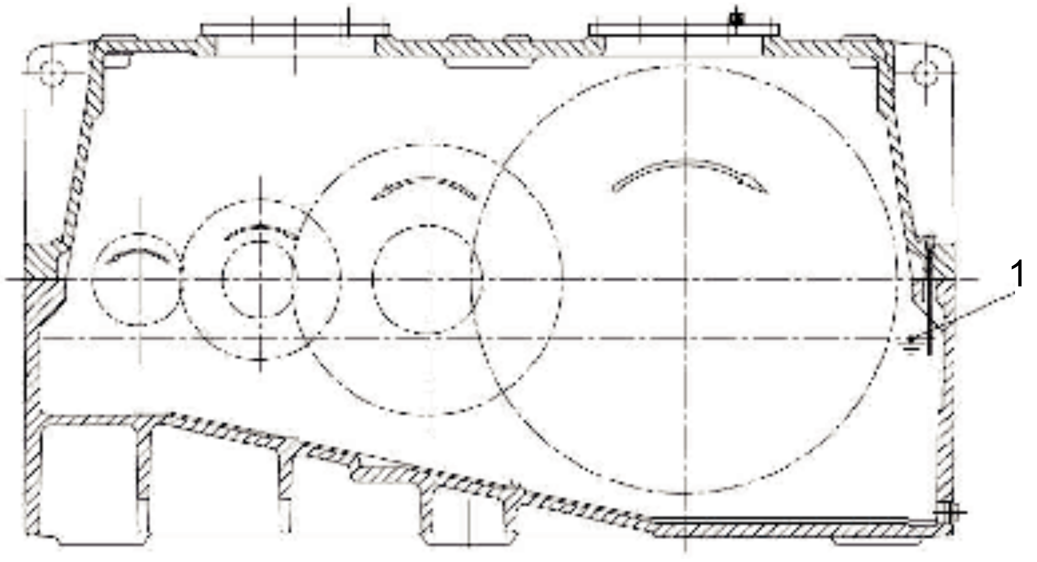


Key

1 oil level

Figure 7 — Immersion depth for different inclinations of the gearbox[3)]

3) Source: A. Friedr. Flender AG.



Key

1 oil level

Figure 8 — Immersion of gear wheels in a multistage gearbox[4)]

6.1.2 Spray lubrication

Spray lubrication is used for three main reasons:

— pitch line velocity too high to ensure a good immersion lubrication;

— necessity of a high oil quantity in order to ensure a cooling of the unit;

— use of a central lubrication unit with one tank for several units.

The number of nozzles depends on the face width. The oil spray should cover the complete active flank of the teeth.

Instead of circular nozzles, so called flat-spray jet nozzles are often used. With these the number of nozzles can be considerably reduced.

6.1.2.1 Determination of the spray parameters

6.1.2.1.1 Position of the point of injection

Common industrial gear units have pitch line velocities of the gears up to 30 m/s. The oil spray should be directed into the gear mesh, ingoing or outgoing. In particular cases with very big gears and adequate large power losses, it would be possible to spray into the ingoing and into the outgoing gear mesh. This variant is used especially for cooling.

6.1.2.1.2 Oil flow on the gear

Only a very small quantity of oil is necessary in order to ensure a correct lubrication of the meshing gears. The main quantity is required to ensure the cooling of the contact.

a) an oil quantity of 0,05 l/min up to 0,1 l/min per mm of gear facewidth is recommended;

4) Source: A. Friedr. Flender AG.

b) spray nozzle should be equally spaced along the facewidth of the gear;

c) consideration is to be given, however, to the amount of heat that needs to be dissipated.

6.1.2.2 Typical spray unit configuration

There are two typical types of spray lubrication:

— wet sump lubrication, where the gearbox housing is used as a lubricant tank;

— dry sump lubrication, where an external tank is used.

A typical spray unit will have the following equipment:

— tank;

— pump;

— pressure limiter;

— filter.

On this minimum configuration some equipment can be added, such as:

— cooler in order to inject oil with an optimum temperature;

— heater in order to permit the pumpability of the lubricant at low temperatures;

— off line oil filtering system.

Other monitoring devices such as flow, pressure temperature sensors, filter clogging sensor, level control and particle counter can be added to the circuit.

6.1.2.3 Description of the elements

6.1.2.3.1 Oil tank

The oil tank performs the following functions:

— cooling of the oil;

— sedimentation of foreign particles;

— deaeration of oil.

The oil tank can be equipped with the following elements:

— oil level checking device,

NOTE This device can be visual or electrical.

— breather, in order to enable air dilatation in the tank;

— drain, in order to change oil or to take oil sample for analysis.

The oil tank volume must be defined in order to have a sufficient dwell time of the oil to allow the deposit of pollution and the oil deaeration.

The oil tank volume is given by Equation (3):

$$V_{\text{tank}} = Q_{\text{pump}} \cdot t \tag{3}$$

where

Q_{pump} is the oil flow rate, in l/min, of the pump;

t is the minimum dwelling time, in minutes.

NOTE The dwelling time is usually between 4 min and 10 minutes.

The longer the dwelling time, the better will be the sedimentation of foreign particles and oil deaeration.

6.1.2.3.2 Pump

Oil pumps, which are mostly volumetric, should be chosen using the following parameters.

a) Cubic capacity: in order to determine the cubic capacity of the pump it is necessary to define the pump flow:

The pump flow, Q_{pump}, is the total flow needed to perform the gearbox lubrication, given in Equation (4):

$$Q_{\text{pump}} = \sum Q_{\text{gears}} + \sum Q_{\text{bearings}} + \sum Q_{\text{seals}} \tag{4}$$

The cubic capacity is given by Equation (5):

$$C = 1\,000 \frac{Q_{\text{pump}}}{n_{\text{shaft}}} \tag{5}$$

where

C is the cubic capacity, in cm^3;

Q_{pump} is the total flow needed, in l/min;

n_{shaft} is the rotational speed of the pump drive shaft, in rpm.

b) Pump pressure: the oil pressure at pump outlet is the sum of the pressure needed at injection point and of the pressure loss in pipes and accessories (valves, filters, coolers…).

NOTE The pump would be able to handle the lubricant at the lowest temperature defined. If not, heaters can be added.

6.1.2.3.3 Pressure limiter

A pressure limiter should be installed in order to protect the lubrication unit from high pressure that might occur. It should be adjusted in order to limit the pressure at a value below the pressure capability of the weakest element of the unit.

6.1.2.3.4 Filter

A filter should be installed in the lubrication unit in order to prevent wear. The selected filter mesh size may range from 5 to 60µm. This is mainly dependent on two parameters:

— the filtration needed in order to have the optimal component life;

— the pressure drop caused in the filter system.

6.1.2.3.5 Pipes

In lubrication systems there are normally three types of pipe:

— suction pipes from the tank or directly from the gear housing to the pump;

— pressure pipes from the pump to the spray point(s);

— return pipes from the gear unit to the tank.

The design criteria for the minimum diameter of these pipes are the permitted flow velocities and the pressure losses.

Typical maximum values for the flow velocities are given in Table 15.

Table 15 — Typical maximum oil flow velocities

Type of pipe	Typical oil flow velocity [m/s]
Suction pipe	1
Pressure pipe	3
Return pipe	2 - 3

The pressure losses in the pipes depend on the oil flow, the oil viscosity, the pipe diameter, the pipe length and the pipe bends. For the acceptable pressure loss in the different pipes one has to consider the geodetic altitude difference of the tank and the gear unit too.

The following points have to be considered.

a) The type and dimension of the suction pipes should be appropriate to the pump size used.

b) The size of the pressure pipes is governed by the maximum pressure loss allowed for the system. The allowable pressure loss is limited by the pressure rating of the pipe and the setting of safety valves. Consideration should be given to the oil viscosity at low temperatures.

c) Return pipes should guarantee the free run back of the depressurized oil. They should be designated to be no more than one-half full during normal operation.

7.1.2.3.6 Cooler

There are two main types of coolers, oil-air and oil-water coolers. The selection of the cooler depends on its technology and the service and operation conditions should be performed by the manufacturer.

7.1.2.3.7 Heater

In case the lubricant has to be preheated in order to allow the starting of the lubrication unit at low temperature, a heating system should be provided in the tank. The surface power at the heating elements in contact with the oil should, typically, not exceed 0,7 W/cm^2, in order to avoid the risk of lubricant decomposition.

In case of long pipe distance between the gearbox and the pump, and low ambient temperature, a thermal insulation and heat tracing of the pipes might also be necessary.

Figure 9 presents an example of a gearbox lubrication unit.

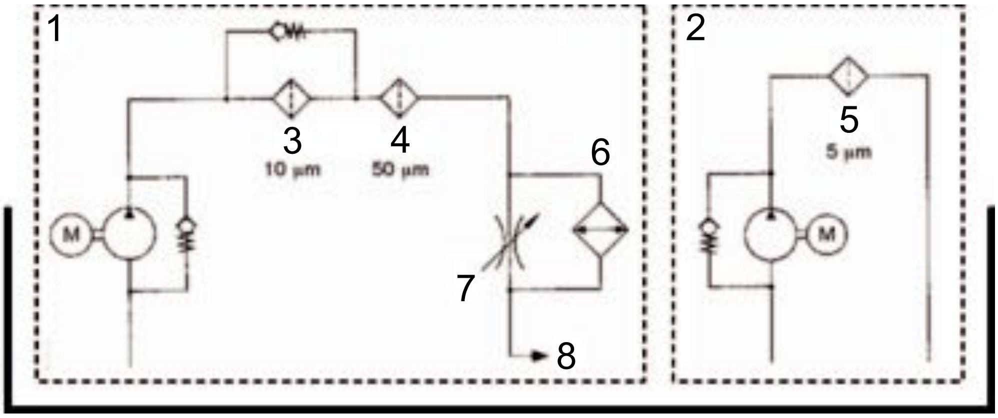


Figure 9 — Examples of circuit design, combination of filtration and cooling systems

6.2 Open gearing

Large girth gear lubrication is as important for open gears as in other enclosed gearboxes. Optimum lubrication depends on the lubricant type and its method of application on the tooth. Indeed, the majority of gear surface distress problems occur (such as wear, scuffing or pitting) due to lubrication related issues. The correct type and application of lubricant is critical to the long-term service life of gearing. The lubricant type and delivery method must be compatible. Issues include:

— oil lubrication is preferable to grease lubrication;

— continuous lubrication is preferable to intermittent lubrication;

— recirculating lubrication is preferable to loss lubrication;

Base rules for application system selection

— recirculating lubrication is preferable to sump lubrication;

NOTE Recirculation lubrication is preferable to spray lubrication.

— contamination, airborne, external contamination in lubricant;
— quantity;
— frequency;
— application;
— drainage;

Base rules for lubricant selection

— lubricant type and grade;
— air temperature.

The gear manufacturer’s recommendations should be followed.

6.2.1 Open gear lubricant types

In some applications running-in lubricants have been used to improve the surface roughness and the contact pattern. This type of lubricant must be used under the supervision of the lubricant supplier.

6.2.1.1 Residual compounds

Residual compounds are a viscous mixture of petroleum-based compounds, also referred to as asphaltics or bituminous.

Most residual compounds use non-chlorinated diluent to provide pumpability. Most contain EP additives or solid lubricants such as graphite, or molybdenum disulfide, or both.

The use of residual compounds can be restricted by local authorities due to ecological and health reasons.

6.2.1.2 Greases

Greases are petroleum-based or synthetic oils to which soap thickeners or carriers are added.

Solid lubricants (typically, graphite and molybdenum disulphide) and EP additives are usually added. Some have thixotropic properties where the viscosity of the lubricant changes with the shear rate experienced during operation. This type of grease should have high adhesion properties. The advantages and disadvantages of greases are given in Table 16.

Table 16 — Advantages and disadvantages of greases

Advantages	Disadvantages
Diluents not needed to aid flow of lubricant	Marginal film thickness
	Lubricant builds up on gear guard sides
No leakage problems	Loss lubrication system
	Heat dissipation

6.2.1.3 Oils

Oils are either mineral-based or synthetic, with EP additives.

These are gear oil type lubricants. Advantages and disadvantages of oils are given in Table 17.

Table 17 — Advantages and disadvantages of oils

Advantages	Disadvantages
High Viscosity	Heat tracing and heaters can be required to obtain proper spray pattern
No diluents are used	
No build up tendency in tooth roots or on gear guard	Leakage problems in cases of insufficient seals
Drain freely from gear guards	
High temperature capability (synthetic only)	

6.2.1.4 Lubricating compounds

Lubricating compounds are synthetic or mineral-based oil with or without solid lubricants and EP additives.

Some contain a diluent for better pumpability. Many have polymer additives as viscosity enhancers. Some have thixotropic properties where the viscosity of the lubricant changes with the flow experienced during operation. Advantages and disadvantages of lubricating compounds are given in Table 18.

Table 18 — Advantages and disadvantages of lubricating compounds

Advantages	Disadvantages
High viscosity	More difficult to pump Heat dissipation
No build up tendency in tooth roots or on gear guard	
High adhesion properties	
Re-lubrication effect due to lubricant depot in the tooth roots – long film life time	Leakage problems in cases of insufficient seals

6.2.2 Method of lubrication – application

The application method is to be selected considering the following parameters:

- type of lubricant;
- lubrication environment (temperature, contamination);
- pitch line velocity;
- level of monitoring expected.

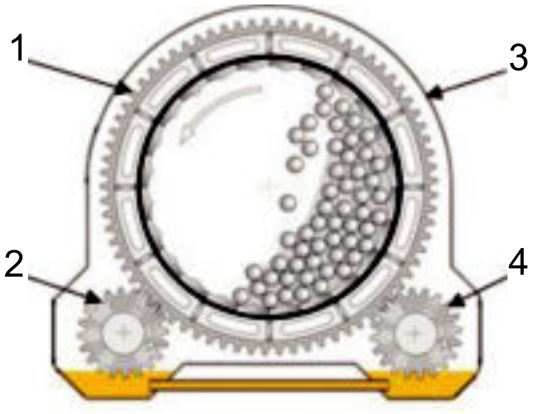
6.2.2.1 Continuous lubrication

Continuous lubrication means that the lubricant is distributed in the meshing area without interruption.

6.2.2.2 Immersion lubrication

This method consists in filling the housing with lubricant so that the pinion(s) or gear dips into the lubricant. For single pinion drive the lubricant is taken up by the pinion or by the girth gear, for dual pinions the lubricant is taken-up by both pinions as shown in Figure 10. This method requires that the lubrication bath remains filled and the gear housing is sufficiently sealed in order to avoid leaks and contamination of the lubricant.

To ensure reliable and safe operation of the gear, the lubricant has to have good backflow behaviour to avoid channelling, good viscosity/temperature behaviour to exclude heating or cooling device and low evaporation losses. In case the lubricant cannot undergo extreme cold temperatures a heating device must be installed in order to avoid channelling of the lubricant.



Key

1 gear rim
2 inward turning pinion
3 girth gear cover
4 outward turning pinion

Figure 10 — Immersion lubrication[5)]

6.2.2.3 Transfer lubrication

This type of lubrication consists in using paddle wheels that plunge into the oil tank and then transfer the lubricant to the driving pinion as shown in Figure 11.

To ensure reliable and safe operation of the gear, the lubricant has to have good backflow behaviour to avoid channelling, good viscosity/temperature behaviour to exclude heating or cooling device and low evaporation losses. In case the lubricant cannot withstand extreme cold temperatures a heating device must be installed in order to avoid channelling of the lubricant.

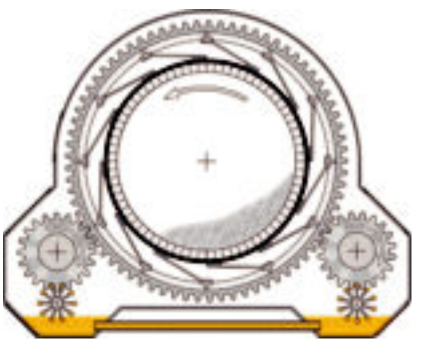


Figure 11 — Transfer lubrication[6)]

6.2.2.4 Circulation lubrication

In this system a flow of lubricant is distributed on the pinion(s) working tooth flanks through a pipe by the means of an external pump as shown in Figure 12. For double pinions a lubrication pipe for each one is required. Its main advantage is that the lubricant can be mechanically cleaned by filters. If needed, the system can be equipped with a heating system, a cooler and a monitoring system (pressure, flow, temperature).

NOTE Technically the favourite type of lubrication is circulation lubrication by using an oil of a kinematic viscosity of 680 mm^2/s – 1 500 mm^2/s or a very soft type of grease. Attention should be drawn to the selection of an adequate sealing system.

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5) Source: Klüber Lubrication München KG.

6) Source: Klüber Lubrication München KG.

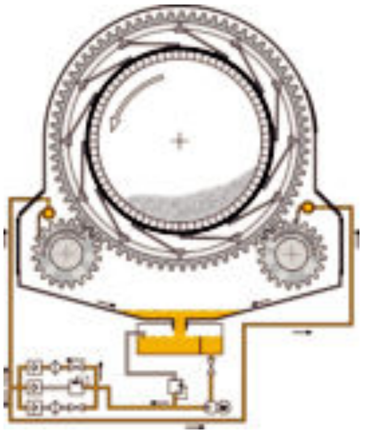

Figure 12 — Circulation lubrication[7)]

6.2.2.5 Intermittent lubrication

Intermittent lubrication means that a certain quantity of lubricant is applied at intervals.

6.2.2.6 Automatic spraying lubrication

This system consists in spraying a suitable amount of lubricant on the gear flanks. The spraying system is driven by pressurized air as shown in Figure 13.

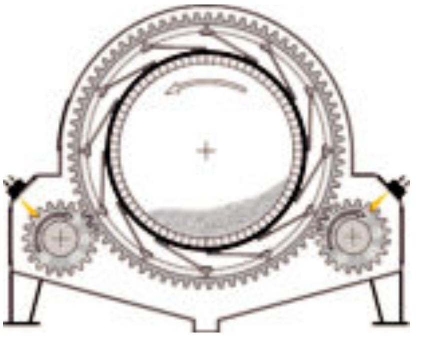

Figure 13 — Automatic spraying lubrication[8)]

Generally the lubricants are applied on the pinion(s). For double pinions, spray equipment for each one is required. For gears of larger modules the lubricants are applied on the pinion(s) as well as onto the girth gear. More frequent application of small quantities of lubricant is preferred. For some residual compounds the lubricant is applied to the girth gear.

For residual compounds the intended intervals between applications must be sufficient to permit complete diluent evaporation. The position of spraying nozzles has to be checked to obtain a complete covering of the tooth width of the working flank.

This is done to allow sufficient time between application and entry into mesh for complete diluent evaporation.

7) Source: Klüber Lubrication München KG.

8) Source: Klüber Lubrication München KG.

6.2.3 System selection

Table 19 gives the useable lubrication system in keeping with the pitch line velocity of the drive. Table 20 gives the options based on the type of lubricant.

Table 19 — Lubrication system selection based on pitch line velocity

Tangential speed (m/s)	0	1	2	3	4	5	6	7
Transfer	X	X						
Immersion	X	X	X	X				
Circulation	X	X	X	X	X	X	X	X
Automatic spraying	X	X	X	X	X	X	X	X
X = ok								

Table 20 — Lubrication system selection based on the type of lubricant

	Residual compounds	Oil	Compound	Grease
Transfer		X	X	
Immersion		X	X	
Circulation		X	X	
Automatic spraying	X	X	X	X
X = ok				

7 Gearbox service information

7.1 Initial lubricant fill and initial lubricant change period

The lubricant cleanliness in service is essential for maximizing gear drive and component life. In order to minimize the risk of damage due to small, hard particles in the lubricant, a filtration system using an appropriate filter pore size should be used. This is especially important in fill-for-life applications. An appropriate filter pore size depends on system design and lubricant viscosity. Therefore, it is recommended to contact the equipment manufacturer and lubricant supplier for guidance.

The initial start-up and operating oil of a new gear drive should be thoroughly drained after a period between 250 and 500 operating hours. The importance of a thorough gear case cleaning with flushing oil to remove particles during the first lubricant change cannot be overemphasized.

7.2 Subsequent lubricant change interval

Recommendations for subsequent lubricant changes are given in the gearbox manual. The basis for setting change periods are, generally, the worst case known by the gearbox manufacturer. With active support from the lubricant supplier, optimized change periods can be defined based on the working conditions and results from used oil analysis. Under normal operating conditions the lubricant should be changed in keeping with the recommendations of the gear manufacturer or responsible party, e.g., lubricant supplier. In a clean operating environment and supported by the aid of lubricant analysis the service period may be extended, but should be done so only after consulting the responsible party for the equipment. Conversely, under more severe operating conditions, more frequent change periods should be considered.

Higher operational temperatures and more severe operating conditions can require a more frequent lubricant change.

Typical recommendations for an adequate lubricant service are given in Table 21.

Table 21 — Typical recommended lubricant service

Task	Inspection interval
Oil temperature Oil pressure Oil level Oil filters Oil leakages Gearbox sound (noise)	Daily to weekly
Gear oil condition: visual inspection sample examination	 Weekly to monthly Once to twice per year
Tooth flank inspection Inner condition of the gear housing Function of the oil circulating system	At every oil change

For larger gear drives, computer controlled online oil condition monitoring systems are available on the market. These systems are very valuable only if the information generated by different sensors is combined adequately. A system based on only one or two types of sensor may support the manual oil service, but cannot be used as a full information system regarding the lubricant service condition.

In Table 22 information and recommendations regarding the use of online lubricant condition monitoring systems are given. As this is rapidly changing technology, an adequate market search should be done and the lubricant supplier should be consulted before installation.

Table 22 — Examples for an online oil condition monitoring system

Task	Remarks
Oil temperature	Plot trend over time
Oil pressure	Set alarm points for minimum and maximum
Oil level	Set alarm points for minimum and maximum
Oil flow	Plot trend over time
Oil filters	Plot pressure difference trend over time
Oil cleanliness (particles)	Plot trend over time
Water content	Plot trend over time
Oil viscosity	Set alarm points for minimum and maximum

7.3 Recommendations for best practice for lubricant changes

WARNING — It has been assumed by the compilers of this document that anyone using the following recommendations will either be fully trained and familiar with all normal engineering and laboratory practice, or will be under the direct supervision of such a person.

There are some recommendations for best practice.

a) Drain oil when still warm from service, just after shut down of the drive. Take care that as much of the used oil as possible is drained from the system.

b) Clean the magnetic plug and the oil level control stick (if available).

c) Replace the filters.

d) Flush gearbox with same type of lubricant as used for service, alternatively with a non-additivated oil. The flushing oil might be chosen of lower viscosity (maximum two viscosity grades) to improve flushing efficiency. No load is applied to the drive during flushing.

e) Take care that the flushing process removes all deposits in corners and bearing housings.

f) Avoid flushing with special cleaning fluids containing detergents, dispersants and other cleaning agents. Even small amounts of residual fluid remaining in the system can affect the service lubricant fill negatively.

g) Replace the filters after the flushing process. (If needed, flushing can be continued until the filters remain clean.)

h) Refill the system with the same type of lubricant as used before. Use a filter of adequate filter pore size on the inlet line to ensure clean fluid.

i) Avoid mixtures of lubricants of different base oil and/or different additive technology. Consult the lubricant supplier if lubricant compatibility questions arise.

7.4 Used gear lubricant sample analysis

The importance of obtaining a representative lubricant sample cannot be overemphasized. However, the interpretation of results of the sample analysis onsite and in the laboratory can only be as valid as the sample is representative of the bulk oil in the lubrication system. For the analysis results to be of value, samples must be drawn into clean containers, from a representative stream of oil from the lubrication system at its operating temperature. Proper technique in obtaining an oil sample is critical, especially if the sample will be analysed for coarse metal particles. Larger heavy particles settle out quickly and are more readily removed in the filter. Therefore their distribution in a machine’s oil system is not always uniform. Obtaining the sample from a bulk oil source in the same way each time is required to attain reliable trending results. The selection of the sampling point and method should be made with input from the gear drive manufacturer and lubricant supplier. To accurately monitor a gear drive using performance and condition analysis, it is essential to establish, as soon as possible, alarm or out of control levels, above which some form of remedial action is required to prevent irreparable drive damage or failure. Alarm levels in the initial stages of monitoring are typically determined from experience. A single sample of lubricant when analysed can reveal interesting information regarding the condition of the complete drive or a component part from which it was drawn. However, to obtain maximum benefit from a used lubricant analysis programme, the results must be trended. Graphical representation of the data can be advantageous. Spurious data, especially from incorrectly drawn oil samples, often leads to unappreciated stoppages in production, only to discover no abnormal condition exists.

7.4.1 Sampling techniques for oil

WARNING — Properly used oil sampling procedures involves working with hot lubricating oil that can cause injury to unprotected, exposed human tissues. Always wear appropriate personal protective equipment and follow appropriate safety practices when performing used oil sampling procedures.

Obtaining a representative sample containing the full range of particle sizes of wear debris and other contaminants is of vital importance. Wear particles and contaminants are a separate phase in the oil and most will eventually settle out of the oil sample. Most particles that are very small, in the order of 5 µm or less, will usually stay suspended in an oil sample. These can be the only particles the laboratory detects if the sample is not representative of the “working” oil in the system. Lubricant analytical laboratories typically cannot test for metal particles larger than 5 µm unless they are equipped to perform Particle Quantification (PQ), Ferrography or XRF Spectroscopy. Thus, it becomes especially important to obtain a representative sample in order for the analytical laboratory to measure coarse metals. Without a coarse metal analysis, severe wear modes will likely be undetected. There are three generally accepted methods for properly obtaining representative oil samples.

a) From the drain plug of the compartment.

Two precautions should be observed if this method is used. First the entire area around the plug must be wiped or washed clean before the plug is removed. Secondly, after the drain plug has been removed, a litre or more of the used oil must be allowed to drain until it is at the bulk oil temperature before the sample is drawn into the sample bottle. Do not over or under fill the sample bottle. 85 % to 95 % of the bottle capacity is usually satisfactory. (The drained oil can be placed back in the gearbox or oil reservoir.)

b) Through a sample probe fitting, or sampling valve installed on the return line to the reservoir.

Ensure that the sample probe fitting or sampling valve is prior to any in-line filters. Do not over or under fill the sample bottle. 85 % to 95 % of the bottle capacity is usually satisfactory. Follow the same cleaning precautions as described in a).

c) With a suction pump, to which is attached a flexible plastic tube.

NOTE Fresh, new flexible sampling tubing must be used when obtaining each representative lubricant sample using this technique.

When obtaining samples, the clean flexible sampling tube may be inserted through the reservoir filler tube, a modified air breather connection, dipstick tube, or the filler/level plug on a gear case or transmission. The sample is drawn into the sample container that is mounted directly on the pump. Ensure that the sample is drawn from approximately midpoint of the working level of the oil and not near or at the bottom of the reservoir or gear case where debris and sludge typically accumulates. Do not over or under fill the sample bottle. 85 % to 95 % of the bottle capacity is usually satisfactory.

7.4.2 Sample containers

It is of utmost importance that sample containers be both meticulously clean and free of moisture before oil samples are placed in them. The typical nominal 100 ml capacity sample bottles supplied by lubricant analysis laboratories as part of a kit are generally of an acceptable cleanliness standard, providing the cap has not been displaced prior to use. If foaming and/or air release properties of the oil samples are to be determined by the analytical laboratory, sample volumes of 500 ml to 1 000 ml are usually required. Sample containers are usually also available from lubricant suppliers. Samples and the appropriate submission forms should be delivered to the analytical laboratory immediately for valid results.

7.4.3 Sample identification

Every oil sample taken should be correctly and consistently identified with the following information, usually on a card, form or label supplied. Lack of the required information lessens the validity and usefulness of the report issued.

- company name;
- location site;
- machine plant no.;
- machine make;
- machine type;
- compartment;

— lube brand;

— lube grade;

— date sample taken;

— total machine hours;

— oil hours;

— oil changed: yes/no;

— date/hours oil changed;

— comments.

7.4.4 Frequency of sampling

The physical properties of lubricants are subject to change during service. The speed of these changes and how they happen depend very much on the type of lubricant, the service conditions, such as load, speed, temperature, and the environment, e.g. dust and humidity.

Regular checks on the lubricant conditions by sample analysis are recommended. A combination of simple sample analysis onsite and sample analysis in a specialized lubricant laboratory can lead to a significant lubricant life extension, if properly done.

The frequency of sampling is dependent on the operating criticality of the machine. In the first months of operation, the monitoring interval should be short so that a database of information can be created for each machine component. Whenever abnormal condition reports are received, the frequency of sampling should be increased until the machine health condition is once again under control. A typical sampling frequency for general industrial equipment would be every month for the first six months, then reviewed and adjusted. Whenever abnormal condition reports are received, the frequency of sampling should be increased until the machine health condition is once again under control. If reliable trend line reporting is required for prediction of machine condition, then a strict programme of sampling is absolutely essential. Some typical frequencies for a range of machines are shown below:

— gearboxes, high speed/duty 500 h or monthly;

— gearboxes, low speed duty 1 000 h or every two months.

Computer-based software is available for the management of sampling intervals based on condition analysis results obtained. The software makes use of cumulative distribution function, and probability density function, suitably adjusted for time intervals, condition status, and cost/risk potential. As a further step, when combined with the costs of inspection and the costs of failure, the software also takes this forward to the determination of the economically optimum point at which the inspection should be undertaken.

7.4.5 Simple lubricant sample analysis onsite

Simple sample analysis onsite gives first impressions of the lubricant service condition. This type of analysis can be carried out by skilled service personnel and should be done as a comparison test of the fresh, unused lubricant. Sophisticated laboratory equipment is not needed for these tests. The main recommended tests are of appearance, colour, odour, crackle and sedimentation.

7.4.5.1 Appearance test

This is useful to identify potential problems with gross contamination or oxidation. Place a sample of the lubricant in a clean, glass bottle (a tall, narrow bottle is best). Compare the sample from the equipment to a new oil sample in the same type of container. The oil should appear clear and bright. A hazy, cloudy, or milky appearance suggests the presence of water; if so, run the “crackle” test. A darkened colour can indicate oxidation or contamination with very fine wear particles. Tilting the bottles (new and used oil samples) simultaneously will give an indication of changes in viscosity which could be related to oxidation or shear losses. Look for sediment in the bottom of the sample bottle; if present, run the sedimentation test.

7.4.5.2 Colour test

The colour of any lubricant changes in service. Generally, lubricants darken in service, due to oxidation. Some synthetic lubricants can change their colour to pink, red or even violet. This is normal for these lubricants and indicates that the antioxidant incorporated in the formulation works. The speed of the colour change might be an indication regarding the thermal and oxidative stress the lubricant has to sustain in service.

The colour of lubricants should be classified as identified by the ASTM D 1500 colour numbers. A simple description of these colour numbers might be:

— colourless to light yellow ASTM colour 1 to 3

— light brown to dark brown ASTM colour 4 to 6

— very dark brown ASTM colour 7 to 8

— black ASTM colour darker then 8

Carry out the colour test together with the appearance test. The colour should be similar to that of the new oil sample. A significantly darkened colour, of more then two ASTM colour numbers, might indicate oxidation or contamination.

7.4.5.3 Odour test

CAUTION — Vapours from used lubricant samples could be harmful to sensitive human tissues. Exercise extreme caution when performing this test.

This test should be performed in a well-ventilated area. Carefully sniff the oil sample by wafting air directly over the top surface by hand towards the nose. Compare the odour to that of a sample of new oil. Oils that have oxidized noticeably will typically have a burnt odour or smell acrid, sour or pungent.

7.4.5.4 Crackle test

CAUTION — Use appropriate personal protection equipment when performing a crackle test, e.g., face shield and gloves, for protection from hot oil splatter.

If the presence of water is suspected, the following simple test can be used to confirm its presence. Place a drop of the oil in question onto a hot plate that has been warmed to 130 °C to 150 °C. If the sample bubbles, it is possible that water is present in excess of 0,05 % (500 ppm). If the sample bubbles and crackles, the water level could be in excess of 0,1 % (1 000 ppm). If water is detected, the results should be confirmed with the laboratory analysis as the crackle test is a simple method and might not have the precision level required for a detailed analysis.

7.4.5.5 Sedimentation test

If sediment is noted during the appearance test, the following test should be performed to supplement or confirm this. Place a sample of the oil in a clean, white plastic cup (but no cups made of polystyrene thermal insulation material) and allow it to stand covered for two days. The cup should be covered or stored in a clean, dust-free area to prevent external contaminants from the environment influencing this test. Carefully pour off all but a few millilitres of the oil. If any particles are visible at the bottom of the cup, contaminants are present. If the particles respond to a magnet moved under the cup then these contain ferrous debris. If there is no response from the magnet and the solids feel gritty then they are likely sand, dirt or non-ferrous debris.

7.4.6 Laboratory analysis

The number and type of laboratory analysis to be carried out should be selected taking into account the specific application, the type of lubricant, ambient and operating conditions, and, if available, the results of an on-site sample analysis. Typical for gear lubricants are the following analytical tests:

— appearance;

— kinematic viscosity at 40 °C or at 40 °C and 100 °C;

— acid number (TAN);

— water content;

— solid impurities;

— additive and wear elements.

For some applications, additional tests might be required. In such applications, contact with the gear manufacturer and the lubricant supplier is recommended, before taking the lubricant sample.

7.4.6.1 Appearance test

The simplest test is visual appearance. Often the test will disclose problems such as gross contamination or oxidation.

See 7.4.5.1.

7.4.6.2 Viscosity

The ISO 3104 [7] test is an accurate, widely accepted method for determining kinematic viscosities of lubricants. It measures the time for a fixed volume of oil to flow through a capillary viscometer under an accurately reproducible head at closely controlled temperatures. Viscosities are then calculated from the measured flow time and the calibration constant of the viscometer. Units for kinematic viscosity are mm^2/s, but they are commonly referred to as centistokes, cSt.

Viscosity is usually measured at 40 °C and 100 °C. Viscosities at other temperatures can be determined by plotting the two points on special log paper (ASTM D 341 [31]). Viscosity index, VI, is a means of expressing the variation of viscosity with temperature. VI is calculated from the measured viscosity at 40 °C and 100 °C using ISO 2909 [6].

An increase in viscosity over that of fresh oil can be caused by oxidation or by contamination with dirt or water. A decrease in viscosity can be caused by contamination with a solvent or fluid of lower viscosity, or from a mechanical shearing action on polymeric components that can be used in the formulation.

7.4.6.3 Acid number

The standard test for acid number, AN, is ISO 6619:1988 [13]. The test uses potassium hydroxide, KOH, to neutralize the acidic constituents in the oil. It yields a single number that represents the amount of KOH used for a given sample of oil in units of mg of KOH/g of oil.

When tested, most new, unused oils will have an acid number because the KOH reacts with additives in the oil. Depending on the additives, the new oil baseline AN can vary widely. Therefore, new oil should be tested to establish a baseline AN. With the new oil baseline AN known, any change in acidity from the new oil baseline can be monitored.

The AN is a measure of the acidity of an oil sample. The higher the AN, the more acidic constituents are present. The acids usually form when high temperatures cause the oil to oxidize. The oxidation can be promoted by contaminants such as water, or wear debris such as iron and copper, that act as catalysts.

Oil oxidation is detrimental because it can increase viscosity, change colour and odour, cause residues and sludge, and create acids that promote corrosion.

7.4.6.4 Water content

7.4.6.4.1 Test methods

There are several tests for determining water content in lubricants. They are listed below in order of increasing accuracy.

— Crackle test. A simple test for water contamination is described in 7.4.5.4.

— Distillation test, ISO 3733 [8]. The distillation test is usually used on oils that prove to be positive by the crackle test and require a more accurate determination of water content. The test is a simple distillation of the oil and separation of the water. It detects water at levels of 0,1 % (1 000 ppm) or greater with reasonable precision.

— Infrared analysis. Infrared spectroscopy is sometimes used when water is present at levels above 0,05 % (500 ppm). A baseline new oil reference is recommended for comparisons. The presence of additives depending on their chemical functionality can limit accurate interpretation of the results.

— Karl Fischer test (ASTM D 6304 [43]). The Karl Fischer titration test determines water content from the chemical reaction between a reagent and the water in the oil sample. It detects water as low as 0,001 % (10 ppm). It is commonly used because it is accurate and relatively inexpensive. However, sulfur AW or antiscuff additives can interfere with the test and give erroneous values for water content.

7.4.6.5 Spectrochemical analysis

7.4.6.5.1 Description

This test detects microscopic size metal particles in an oil sample. The typical spectrometer is capable of identifying about 20 metals, the source of which can be wear debris, contaminants, or inorganic additives in the lubricant. Knowing the type and quantity of metals can help diagnose wear problems or disclose sources of contamination. For example, a high concentration of iron, chromium, manganese, molybdenum or nickel could indicate wear debris from gear teeth or bearings.

Spectrochemical analysis is rapid and inexpensive. The oil sample is burned and the light emitted is separated by diffraction into distinct wavelengths. Because each metal has its own characteristic wavelength, specific metals in the oil sample can be identified.

Table 23 gives a general list of elements that can be detected by spectrochemical analysis. It lists typical sources for each element.

Table 23 — Sources of metallic elements

Element	Symbol	Typical source
Aluminium	Al	Dirt, labyrinth seals
Antimony	Sb	Journal bearings, grease, additives
Arsenic	As	Journal bearings
Barium	Ba	Water, grease, additives
Bismuth	Bi	Journal bearings
Boron	B	Additives
Cadmium	Cd	Journal bearings, plating
Calcium	Ca	Additives, water, grease, dirt
Chromium	Cr	Gears, bearings, shafts
Cobalt	Co	----
Copper	Cu	Bearings, coolers, additives
Indium	In	Solder
Iron	Fe	Gears, shafts, bearings, rust
Lead	Pb	Journal bearings, solder, grease, paint
Magnesium	Mg	Dirt, additives
Manganese	Mn	Gears, bearings, shafts
Molybdenum	Mo	Gears, bearings, shafts, additives
Nickel	Ni	Gears, shafts, bearings
Phosphorus	P	Additives
Potassium	K	Dirt
Silicon	Si	Additives, dirt, sealants
Sodium	Na	Additives, dirt
Tin	Sn	Bearings, solder, coolers
Titanium	Ti	Paint, dirt
Vanadium	V	----
Zinc	Zn	Additives, coolers, brass components

7.4.6.5.2 Limitations

Emission spectroscopy works well for detecting metal particles up to a few micrometres. Low results are obtained for particles greater than a few micrometres because it is incapable of completely and consistently burning large particles. Therefore, it readily detects particles from mild adhesive wear, polishing and micropitting because wear debris from these wear modes are within the detectable range. However, failure modes such as severe abrasion, macropitting or scuffing usually generate particles that are larger than 10 µm. In such situations, ferrography, particle counting or analysis with a ferrous debris analyser can be superior monitoring techniques.

Emission spectroscopy does not distinguish between particles of free metal and particles of metal oxides or other compounds of metal. For example, rust particles can show up as increased iron content, but emission spectroscopy cannot identify whether the iron is in the form of wear debris, iron oxide, or iron sulfide. High concentrations of silicon or aluminum can indicate contaminants such as sand, dust or dirt. However, there are other sources for silicon, such as silicone antifoam additives or silicone gasket sealants. It is therefore important to analyse samples of fresh oil from new oil drums to establish a baseline level of silicon to help distinguish between contaminants and lubricant additives.

Some spectroscopy does not detect sulfur and certain other elements. Specific elements of interest should be requested from the laboratory processing the test.

It is helpful to plot the results of spectrochemical analyses over time. The graphs will indicate the normal test variability and will help with following any trends in test results. An accelerating wear problem is most easily predicted from a trend line that is increasing.

7.4.6.6 Solid impurities

7.4.6.6.1 Automatic particle counting

Particle counters are a common method used to determine lubricant cleanliness. They monitor the number of particles of a given size range in a given volume of oil sample or a given oil flow. A common method for determining lubricant cleanliness, particle counting, detects all particles regardless of their composition and is capable of detecting particle sizes in the range of 0,5 µm to 100 µm, or greater.

Particle counters can use, for example, light-interruption, laser-scanning, induction sensors or conductivity measurements. These methods provide analyses that are rapid and inexpensive. Light-interruption and laser–scanning techniques detect all particles regardless of their composition, whereas induction and conductivity sensors allow discrimination between ferrous and non-ferrous particles. Theoretically, all methods are capable of detecting particles in the range of 0,5 µm to 100 µm, or greater. In practical applications however, the sensitivity can be reduced due to high oil viscosity, oil discolouration (additives, oxidation), opaqueness (water) or air bubbles.

With light-interruption particle counters, the lubricant flows through a small passage while a light beam scans the oil through a window. Particles in the oil that are within a set size range momentarily interrupt the light beam. The output from a detector that senses the interruption of the light beam is related to the time of interruption and hence the size of the particle.

Laser-scanning particle counters operate on a principle that is similar to that of the light-interruption type, except the oil sample remains stationary in a clear glass container while it is scanned by a revolving laser beam. The particle size range is selectable, as it is with the light-interruption method.

7.4.6.6.2 Limitations of automatic particle counting

While particle counting detects all particles, it gives no information on the shape or composition of the particles. It is susceptible to incorrect particle counts caused by bubbles of air or water. The oil sample must not be opaque. For accurate results, the oil sample should be well agitated to produce a uniform suspension of particulates, and the concentration of particles should be low enough to avoid counting two or more particles as one. Some lubricants can contain additives which contribute to particle counting. That can lead to misleading results.

7.4.6.6.3 ISO solid contamination code

The International Organization for Standardization, ISO, Solid Contamination Code, ISO 406 [10] has been universally accepted as the simplest and best means for expressing cleanliness levels.

The ISO 4406 [10] code changed in 1999. The revised system uses three code numbers, corresponding to concentrations of particles larger than 4 µm, 6 µm, and 14 µm. The new 6 µm and 14 µm sizes were chosen so code numbers would not change significantly from the older system based on sizes of 5 µm and 15 µm. For higher viscosity oils such as gear oils, the number of particles which are smaller than or equal to 5 µm is generally not reported, the value being substituted with a "–", for example –/15/12. Some companies use a three-digit form of the ISO 4406:1987 code, representing 2 µm-, 5 µm- and 15 µm-sized particles. This three-digit code can be upgraded to the new ISO 4406:1999 [10] system by increasing the first digit by one while keeping the last two digits the same. For example, 17/15/12 under ISO 4406:1987 becomes 18/15/12 under ISO 4406:1999 [10].

There are two ways of assigning the ISO code. In the first method, the range numbers are selected from a table of range numbers versus particle concentration (see Table 24) for the number of particles greater than 4 µm/ml, 6 µm/ml and 14 µm/ml. If a particle count falls between adjacent particle concentrations, the ISO range number is found opposite the higher concentration. In the second method, the particle counts are plotted on graph paper and the range numbers are determined where the line crosses the 4 µm, 6 µm and 14 µm vertical lines (see Table 25).

Table 24 — What the ISO codes mean

ISO number	Number of particles per millilitre of fluid
25	160 000 to 320 000
24	80 000 to 160 000
23	40 000 to 80 000
22	20 000 to 40 000
21	10 000 to 20 000
20	5 000 to 10 000
19	2 500 to 5 000
18	1 300 to 2 500
17	640 to 1 300
16	320 to 640
15	160 to 320
14	80 to 160
13	40 to 80
12	20 to 40
11	10 to 20
10	5 to 10
9	2,5 to 5
8	1,3 to 2,5
ISO code examples: --/21/18 Dirty system --/17/14 New oil --/16/13 Average system (inline filter) --/13/10 Clean system (offline filtration)	

Table 25 — Example of particle size and counts

Particle size, µm(c)	**Particles per millilitre**
>4	1 617
>6	78
>14	17
>50	3
>100	0

Since there are 1 617 particles per millilitre greater than 4 µm, the first range number given by Table 24 is 18. The 78 particles per millilitre greater than 6 µm give a range number of 13. The 17 particles greater than 14 µm give a range number of 11. Therefore, the ISO code number for the sample is 18/13/11.

7.4.6.6.4 Ferrographic analysis

Ferrographic analysis separates wear debris and contaminants from a lubricant sample by magnetic precipitation. It is capable of precipitating particles that range from less than 1 µm to 100 µm, or greater. Ferrography provides two types of analysis, a relatively inexpensive direct reading (DR) ferrograph, and a more expensive analytical ferrograph.

In the DR ferrograph, a diluted sample of lubricant is siphoned through a precipitation tube that resides in a powerful magnetic field. The combination of magnetic force and the viscous forces exerted by the lubricant cause the particles to be separated by size. The large particles, greater than 5 µm, are deposited first, near the entry of the tube, then the smaller particles, 1 µm to 2 µm, are deposited farther down the tube. Two light beams pass through the precipitation tube, one at the entry deposit and one several millimetres further down the tube where small particles deposit. Light attenuation at the two locations along the tube is used to quantify the relative amount of large to small particles. The results are reported as two scalar readings, direct large, DL, and direct small, DS.

In the analytical ferrograph, a diluted sample of lubricant is pumped across a microscope slide that is mounted at an angle above a magnet so that the field varies along the length of the slide. The particles are subjected to a continuously increasing magnetic force as they flow along the slide. Consequently the particles precipitate, distributed by size, along a narrow band about 50 mm long. Ferrous particles line up in strings that follow the magnetic lines. Non-ferrous particles and contaminants travel down the field in a random pattern. The slide is rinsed with a fixative that washes away the oil, locks the particles in place, and causes other material to float away.

The ferrogram (slide upon which particles have been deposited) is examined in a bichromatic microscope equipped with a camera. The microscope uses both transmitted green light projected from the bottom of the ferrogram and red light reflected from the top of the ferrogram to distinguish the size, shape, texture and composition of both metallic and nonmetallic particles. The particles have characteristics that help determine the wear mechanism and identify the source of the particles. Table 26 classifies the types of particles.

7.4.6.6.5 Wear particle analyser

In the ferrous debris analyser, a diluted sample of lubricant is drawn through a filter that is in a strong magnetic field. The filter has a matrix of fine, ferromagnetic fibers that become magnetized in the magnetic field. The fibres capture small particles magnetically, and physically capture particles larger than the spacing between the fibres.

A flux sensor determines the change in the magnetic field due to the presence of the particles, and displays the magnetic equivalent of the captured particles, in micrograms, of iron metal. The reading, known as the magnetic iron content, MIC, is independent of particle size.

The ferrous debris analyser is capable of capturing particles 1 µm and larger with an efficiency of 95 % or greater.

The filter can be back washed with solvent to recover the particles for microscopic examination and other diagnostic analyses.

Table 26 — Characteristics of particles

Wear particles	
Rubbing	Flat platelets <15 µm long, <1 µm thick. Found in lubricants of all machines, and are indicators of normal wear.
Sliding	Generally >15µm with length to thickness ratio between 5 and 30. Surfaces have parallel striations and might have temper colours. They are evidence of severe sliding.
Cutting	Long, curled chips resembling lathe cuttings. They are evidence of abrasive wear. Can be caused by contamination of lubricant by hard, sharp edged particles.
Fatigue	Generally >5µm with length to thickness ratio <5. Surfaces are rough and particles are shaped like chunks of coal, rather than platelets.
Laminar	Thin, bright particles often with holes in their surfaces and edges that are split. Length to thickness ratio >30. Typical of gear and rolling bearing wear particles that pass between contacting surfaces.
Spherical	Generally <5 µm diameter with smooth surfaces. They are a precursor to rolling bearing fatigue failure.
Ferrous oxides	
Red oxide	Red/orange particles magnetically aligned. Appear thick, rounded and translucent. They are severe sliding wear particles that have oxidized.
Dark metallo-oxide	Resemble red oxide sliding wear particles, except they are not translucent. Often show flecks of free metal on their surfaces. They are caused by heat and can be evidence of lubricant starvation or severe wear.
Black oxide	Dark grey/black particles magnetically aligned. Shaped like pebbles. They are evidence of inadequate lubrication and represent a more severe condition than red oxide particles.
Corrosion	Fine deposit of <1 µm size particles at the exit of ferrogram. They are formed by corrosive attack of metal surfaces and depletion of lubricant additives.
Contaminants	
Friction polymer	Amorphous, translucent material of no particular size. Indicates lubricant polymerization under extreme conditions.
Sand, dirt	Generally >5 µm crystalline particles, not magnetically aligned.
Fibres	Translucent, fibrous particles not magnetically aligned. Typical fibres include hair, cotton, wood, glass, minerals, nylon and cellulose.
Spheres	Generally >5 µm with rough surfaces. Can be contaminants from grinding, welding or shot blasting.
Other	Contaminants such as paper, paint, varnish, glue, gasket or seal materials, or lubricant additives such as molybdenum disulfide or graphite.

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