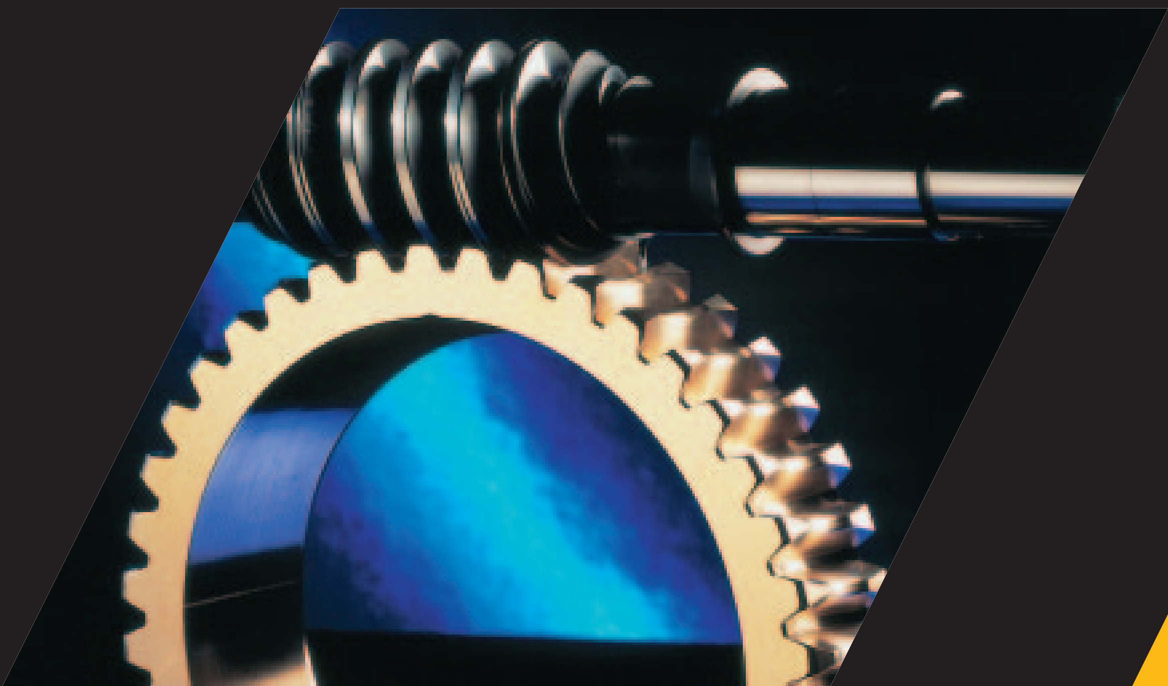
Lubrication of Gear Systems



From large to small stationary gear sets

Lubrication is our World

Front page: Worm gear
Photo courtesy of Rhein-Getriebe
40667 Meerbusch, Germany

Lubrication of Gear Systems

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1 Introduction

Gear wheels and simple gear units were already common ancient times and used in grain mills (*Fig. 1*), water pumping stations and wind mills.

Even today, many centuries after the gear wheel was invented, there is nothing better, more compact or efficient than a gear unit when it comes to transferring power, converting torques and adapting speed. It is therefore not surprising to find gears in all fields of technology, e.g. in turbo-generator drives with an output of 100 MW and more, tube mill and kiln drives, or small and miniature low-output drives used in mechanical equipment and computers.

The development of gear systems is characterized by the requirement for enhanced power and torque transfer by means of smaller and lighter gears on all performance levels.

This requirement resulted in very compact gears with a high power density. In other words, the power-weight ratio (kg/kW) of gears was reduced considerably in the past few years.

Increased efficiency is mainly achieved by means of gear materials that are more resistant to wear (case hardened, high alloy steel), improved flank machining methods and optimized tooth geometry.

The power-weight ratio of gear units can also be improved by means of torque division (planetary gears), light weight construction (light metal or plastic casings, hollow shafts), or by increasing the thermal limit which, as far as standard gears are concerned, often lies below their mechanical load limit (e.g. by using synthetic lubricating fluids).

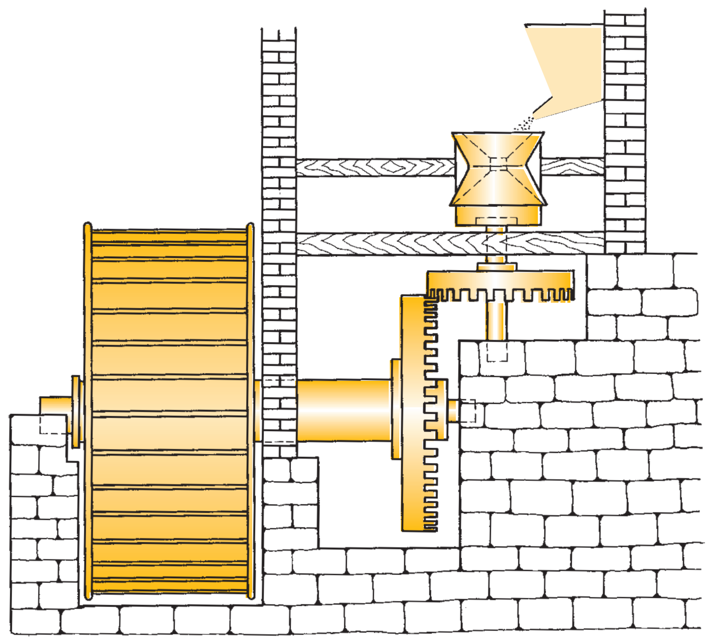

Fig.1: Roman corn mill with a right angle gear (acc. to Vitruv)

The requirements of lubricants have to meet in terms of anti-wear and anti-scuffing properties as well as high temperature resistance increase while the power-weight ratio of the gear units decreases. Excess heat must be dissipated from gears with a very small housing surface. This often results in higher operating temperatures, which has a negative effect on the service life of gears and lubricants.

Measures to decrease power losses and reduce temperatures are gaining more and more importance. They include the reduction of friction losses in gear components as well as gear cooling.

The use of synthetic lubricants with a low friction coefficient and high resistance to increased temperatures and oxidation turned out to be a cost effective way of decreasing power losses and reducing operating temperatures. In many cases complicated cooling installations are no longer necessary.

But these are not the only reasons why synthetic lubricants, especially synthetic lubricating oils, are so important in gear units. Apart from reducing gear temperatures, it is possible to ensure a substantially longer lubricant life (in case of worm gears this can even mean lifetime lubrication) or transfer more power by increasing the oil's operating temperature. Costs and energy will be saved in any case.

The importance of synthetic gear oils also increases in view of ecological concerns. Increased lubricant life (3 to 5 times longer than mineral oil) means a lot less consumption and less used oil to dispose, hence less damage to the environment.
Reduced maintenance costs due to less frequent oil changes is an additional spin-off.

This brochure will focus on synthetic gear oils and describes their advantages as compared to mineral hydrocarbon oils. The application-related advantages such as improved performance, extension of oil change intervals, reduced energy costs and high temperature lubrication will be explained in detail by means of practice related examples (see 7.2.2).

The selection of synthetic oils (7.2.2 and 8.3) and determination of viscosity based on the new Klüber selection method (8.2.2) will also be described in detail.

A special lubricant manufacturer, Klüber offers a comprehensive range of modern synthetic, high quality and high performance oils suitable for all gear lubrication purposes, including food grade lubricants for the food processing and pharmaceutical industries and rapidly biodegradable gear oils. Gear oils are very important for stationary industrial gears that are used to transfer power. Gear greases, on the other hand, are mainly applied in small and miniature gear motors used in all industrial sectors and, owing to automation and improved comfort, also in domestic equipment (household and garden appliances, hobby tools).

Small and miniature gears are often lubricated with greases because they may not be oiltight, be installed in inaccessible or hard-to-reach places, because it may not be possible to perform maintenance, the gear may not be stationary during operation, or because lifetime lubrication is required.

There is no such thing as a universal gear grease because there are many types of small gears whose components are made of most different materials (steel, nonferrous metals, plastics, composites).

Application, operating and ambient conditions also vary considerably.

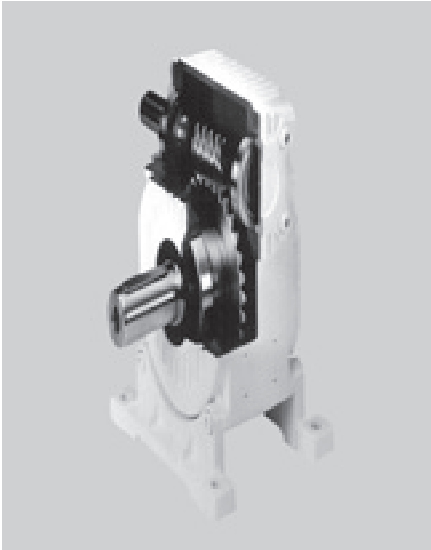

Fig. 2: High-performance worm gear, made by ZAE-Antriebssysteme, Hamburg
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Therefore a large number of different gear greases is required, and offered by Klüber, with a consistency varying between fluid to soft, with different base oils, additives suitable for the specific gears and loads, and with thickeners imparting special properties to the greases such as adhesiveness, resistance to media, noise damping, etc.

As compared to gear oils which are exclusively used in oiltight gears, the selection of a suitable gear grease is not that easy. Apart from the type of base oil and viscosity it is important to determine the grease's consistency and the required type of thickener.

As there is not very much written information available about gear greases, their application and selection, and because DIN standards are restricted to an absolute minimum, the selection of gear greases is treated in detail in this brochure.

This chapter, together with the survey of Klüber gear greases at the end of the brochure, will make it relatively easy even for the not-so-versed reader, to select a suitable product.

For the sake of completeness this brochure also briefly deals with the lubrication of large open girth gear drives and open gear stages. This special type of gear lubrication mainly relies on adhesive gear lubricants.

More detailed information about the lubrication of large gear drives with adhesive lubricants is contained in our brochure *Lubrication of large gear drives*, 9.2 e.

2 Gear systems

Being extremely versatile, gear units are widely used in all industrial sectors. Their task is to modify the torque and speed generated by a power source at the *input end* of the gear with maximum efficiency and in such a way that it suits the requirements of the *output end*. Gear units are speed/torque converters which also provide the possibility of reversing the direction of rotation. A simple gear unit comprises:

- two shafts
- two meshing gear wheels, and
- the required number of shaft bearings.

These components are either fully or semi-enclosed by a housing. In open gears the shafts of the driving and the driven gear wheel are supported by separate bearings (e.g. girth gear drives).

The quality and efficiency of a gear unit depend on the amount of power loss resulting from the tribo-system made up by the power transmitting tooth flanks (friction system).

2.1 Gear types

Gears are classified in three main groups depending on the position of the shafts relative to each other, the type of flank contact and the characteristic tooth traces:

- primarily rolling contact gears
- combined rolling and sliding gears
- primarily sliding contact gears

Table 1 is based on DIN 868 and the GfT worksheet 2.4.2 and provides a survey of gear types (GfT = Gesellschaft für Tribologie, German Society of Tribology).

A gear unit performs rolling and sliding movements on the power transmitting flanks of the meshing teeth. The load on the tooth flanks is a function of the tooth geometry and the forces generated by the sliding movement.

In gears mainly performing a rolling movement (spur and bevel gears) the load on the tooth flanks is generally lower than in gears mainly performing a sliding movement (worm and hypoid gears), see *table 1*, column "sliding percentage".

The higher the sliding percentage, the higher the wear load on the tooth flanks, the higher the requirements a lubricant has to meet.

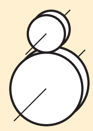
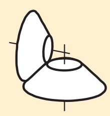
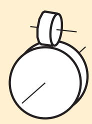
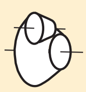
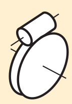
Types	Gears	Position of the shafts	Tooth flank contact	Gear components		Type of movement	Sliding percentage [%]
Rolling contact	Spur gears	parallel	line	cylinders		rolling and sliding	10 to 30
	Bevel gears	inter-secting	line	cones		rolling and sliding	20 to 40
Rolling / sliding contact	Crossed helical gears	crossing	point	cylinders		increased sliding	60 to 70
	Hypoid gears	crossing	line	cones		increased sliding	60 to 70
Sliding contact	Worm gears	crossing	line	cylindrical and globoid element		mainly sliding	70 to 100

Table 1: Types of gears

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2.1.1 Spur gears

Spur gears are characterized by the cylindrical shape of the gear wheels and the parallel shafts. A distinction is made between straight toothed, helical, double helical and herringbone gears depending on the tooth trace, and between internal or external gears depending on the relative position of the gear wheels. The contact of the intermeshing teeth is linear over the entire flank width.

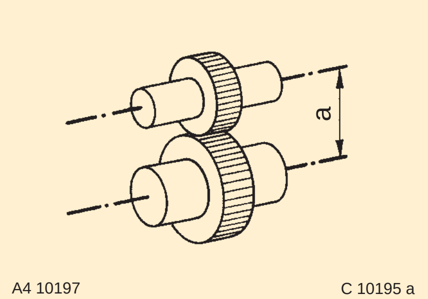


Fig. 3: Spur gear, external gear pair

Straight-toothed spur gears usually have one or two meshing pairs of teeth. As they have to transfer the full power, their load carrying capacity is relatively low and they generate a lot of noise.

Helical gears are better because their contact ratio is higher. Shocks during meshing and running noises are reduced as the flanks mesh gradually. The manufacturing costs of helical gears are higher, and the shaft bearing design is more complicated due to the axial forces that are generated.

Axial thrust can be eliminated by means of double helical or herringbone gears. The latter are very complicated in design and are therefore rarely used.

The combination of internal and external gear wheels provides the advantage of a relatively short center distance. Internal gears are often used as an "annulus" in planetary systems.

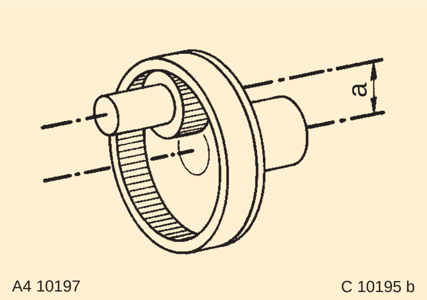


Fig. 4: Spur gear, internal gear pair

2.1.2 Planetary gears

Planetary gears have a driving and a driven shaft located on the same axis. Their main advantage as compared to conventional spur gears is the increased power density, i.e. they utilize the installation space much better. The input torque is split and transmitted to three or more planetary wheels.

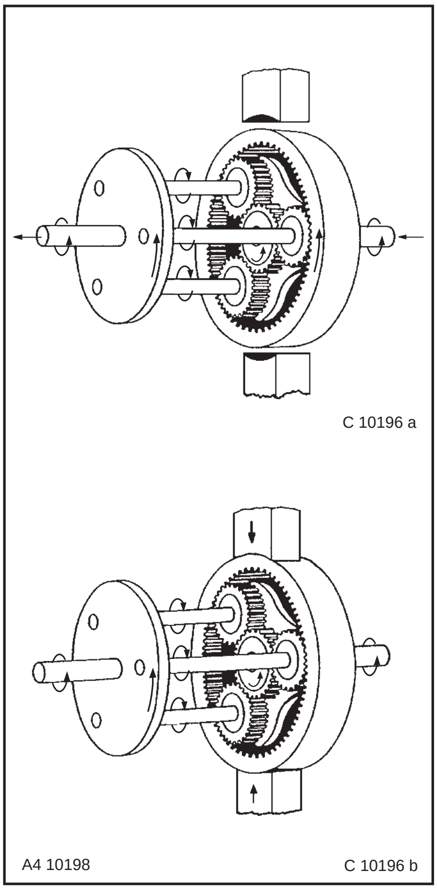


Fig. 5: Planetary gear, three wheels

By splitting it is possible to multiply the transferrable power under the same specific load.

Other advantages include low rolling and sliding speeds of the tooth flanks.

2.1.3 Bevel gears

Bevel gears are characterized by intersecting axes, most frequently at 90°, and cone-shaped gear components. The gear teeth mesh in a line contact.

The tooth trace can be straight, inclined, curved or helical and depends on the required power transmission and smooth operation.

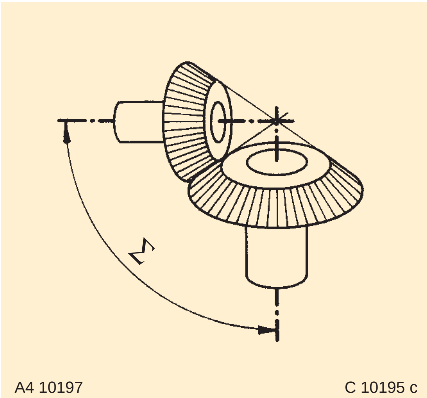


Fig. 6: Bevel gear pair

Straight-toothed bevel gears generate a lot of noise, whereas bevel gears with other types of teeth operate much more silently and permit a higher degree of power transfer. It is important to remember that bevel gears always generate axial thrust.

2.1.4 Crossed helical gears

These gears are characterized by crossing shafts and cylindrical rolling elements. The teeth are the same as in the case of helical spur gears. Owing to the offset axes the teeth only have a point contact, i.e.

when two flanks mesh there is no line as in the case of spur gears with parallel shafts.

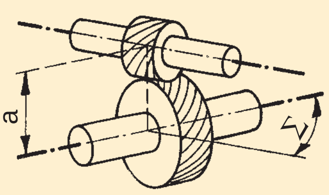


Fig. 7: Crossed helical gear pair

This results in a high sliding speed and, in consequence, high thermal load on the tooth flanks. Therefore cross helical gears are only suitable to transfer movements and can only carry low loads.

2.1.5 Hypoid gears

Hypoid gears are characterized by hyperbolic teeth, crossing shafts and a pinion axis offset to the center of the crown gear. The tooth traces are curved. Owing to the staggered axes the hypoid pinion has a larger diameter than that of a bevel gear with the same crown diameter. In consequence, hypoid gears have a better load carrying capacity while ensuring the same transmission ratio.

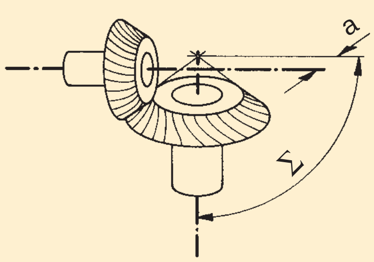


Fig. 8: Hypoid gear pair

When the teeth are meshing there is a point contact with an ellipsoid surface, which results in vertical sliding movements and an increased percentage in the horizontal direction.

Owing to the increased contact ratio and the additional sliding motion, hypoid gears operate much smoother than bevel gears.

Their disadvantage, however, is that the horizontal sliding movement reduces the scuffing load capacity the more the axes are offset, resulting in a lower gear efficiency.

Special hypoid gear oils can help. As compared to bevel gears of the same size, hypoid gears have a better noise behavior, higher transmission ratio and increased power transfer capacity. Their efficiency, in contrast, is worse.

2.1.6 Worm gears

These gears are characterized by shafts crossing at an angle of 90°. *Fig. 10* shows the main worm and wheel types and the respective geometrical components. Cylindrical worm gears (*Fig. 10 a*) are most common.

Having a high transmission ratio (i = 5 to 70), worm gears usually are of a single-stage design. They can transfer increased forces and are still small in size. As compared to all other gear units, they have the best noise behavior but also a very high sliding percentage which results in friction losses. Worm gears therefore operate hotter and have a lower efficiency.

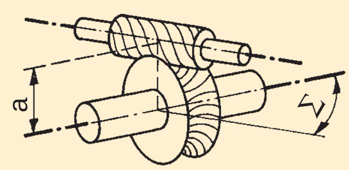


Fig. 9: Worm gear pair

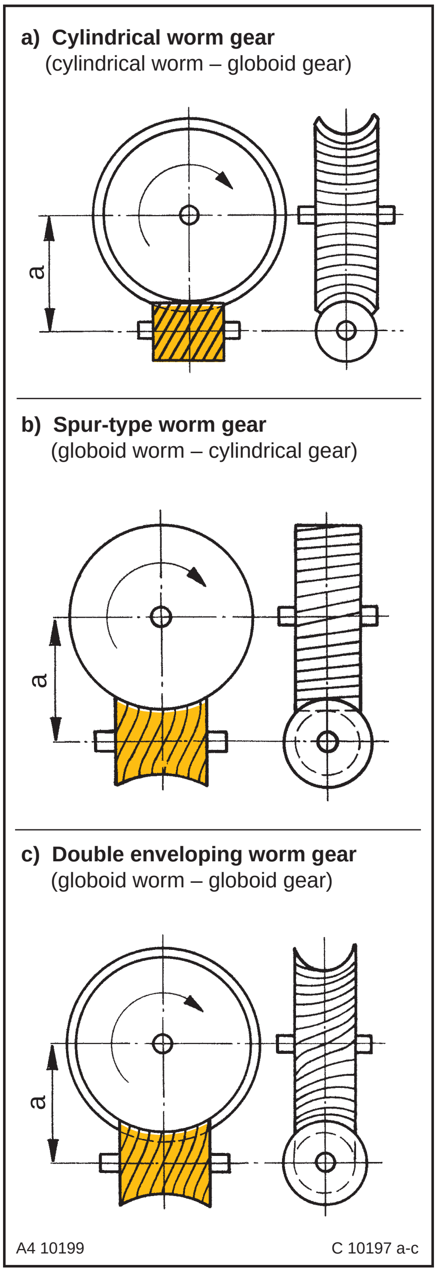


Fig. 10: Worm gears

3 Gear components

Gear systems are made up of wheels, shafts, bearings and housings. All components interact closely, and lubricants play an important role.

3.1 Gear wheels

A gear wheel is a machine element rotating around a shaft. It consists of a wheel body, contact surface and teeth. Depending on the position of the teeth relative to the wheel body, a distinction is made between internal and external gear wheels.

A gear pair consists of two gear wheels separated by the center distance *a*. The smaller wheel is called *pinion*, the larger *wheel*.

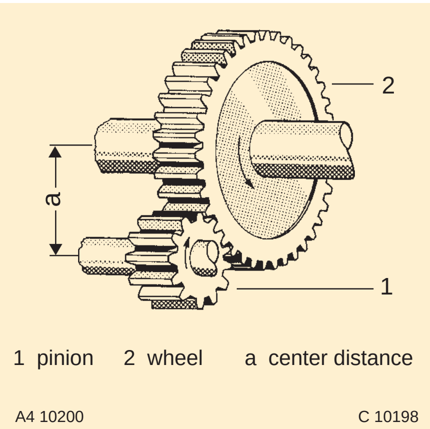


Fig. 11: Main components of a gear pair

The general terms relating to gear wheels, pairs and units are defined in DIN 868 which also explains the basic principles.

Other relevant standards include DIN 3960, 3971, 3975 and 3989.

3 .1.1 Tooth geometry

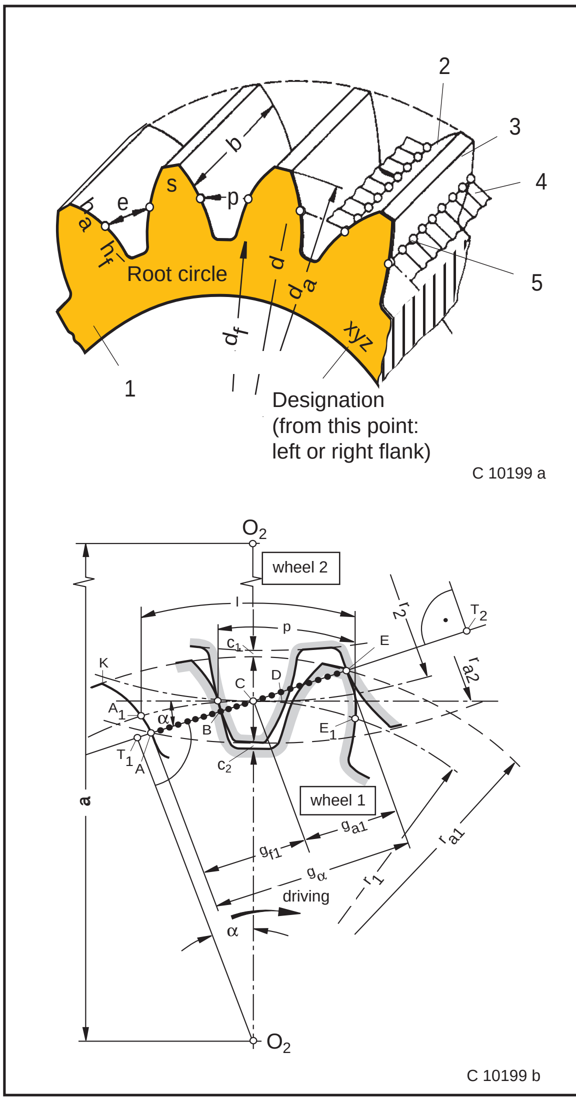
Involute gears are most widely used in machine construction. As compared to other gears, the tooth geometry has the following advantages:

- simple and precise manufacturing
- exchangeable when used in spur gears
- uniform transfer of movements even with center distance variations
- uniform direction and amount of normal force
- one tool with variable profiles can be used for various tooth geometries and center distances.

Positive and negative profile corrections are made in order to avoid undercutting when the number of teeth is low and to increase the root's load carrying capacity. They are also made to improve the flanks' load carrying capacity (larger curve radius) and reduce the sliding percentage to decrease power losses.

Other gears such as cycloid gears (e.g. for pinion stands, clocks) and lantern gear drives (e.g. live ring drives) are not used very often.

Even though cycloid gears operate more precisely than involute gears, they are much more sensitive to variations in the center distance, and are very expensive to manufacture.



1 Gear rim
2 Left flank
3 Right flank
4 Reference cylinder (operating pitch cylinder)
5 Right tooth trace

a Center distance
b Face width
c Bottom clearance
d Reference circle (pitch circle)
d_f Root circle
d_a Addendum circle
e Space width
g_α Length of action A-C-E
α Pressure angle
h_a Addendum
h_f Dedendum
l Length of action (path of rotation from start of action A1 to end of action E1)
p Pitch
s Tooth thickness in reference circle

A Start of action
B Internal point of action: the leading teeth have just ended action (E)
C Pitch point
D External point of action: the following teeth are just starting action. – B is the external point of action of wheel 2.

Fig. 12: Nomenclature and dimensions of involute gears (no profile offset)

3.1.2 Gear materials

In view of the ever increasing requirements for higher gear output, smaller dimensions and lower weight, the materials used for gears were refined to an extremely high degree and the flank machining methods were improved.

Table 2 illustrates how high grade materials, improved flank surfaces, improved hardening methods and the application of high performance lubricants made it possible to reduce the size, weight and manufacturing costs of gear units considerably.

Today high performance gear units are usually equipped with gear wheels of alloyed steel and case hardened teeth which are ground after heat treatment. Industrial worm gears subject to high loads generally have a worm of alloyed case hardened steel and a gear wheel made of a high grade bronze alloy material. Gears subject to low loads are usually made of alloyed heat treatable steel, nitriding steel, alloyed cast steel or spheroidal graphite iron.

Gear wheels made of synthetic materials are rarely used in industrial gears due to their inferior load carrying capacity and low thermal resistance. They are mainly found in small and miniature low capacity gears used to transfer movements and speeds. In these cases low vibrations and noises as well as emergency running properties are of greater importance, as the flanks are often lubricated only once.

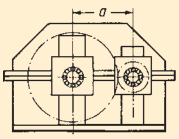
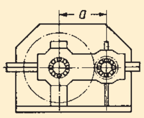
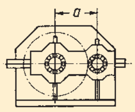
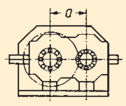
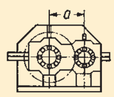
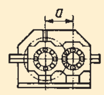
Material	Pinion and gear wheel **C 45**	Pinion and gear wheel **42 Cr Mo 4**	Pinion: **20 Mn Cr 5** Gear wheel: **42 Cr Mo 4**	Pinion and gear wheel **31 Cr Mo V 9**	Pinion and gear wheel **34 Cr Mo 4**	Pinion and gear wheel **20 Mn Cr 5**
Heat treatment	normalizing	hardening and tempering	pinion: case-hardening wheel: hardening and tempering	gas nitriding	induction hardening of flanks	case hardening
Machining method	hobbing	hobbing	pinion: grinding wheel: hobbing	fine milling	milling and lapping	grinding
Center distance *a*	830 mm	650 mm	585 mm	490 mm	470 mm	390 mm
Module *m*	10	10	10	10	14	10
Size (welded housing)						 C 10200 a-f
Weight of rolling bearings	95 kg	95 kg	95 kg	105 kg	105 kg	120 kg
Total weight	8 505 kg	4 860 kg	3 465 kg	2 620 kg	2 390 kg	1 581 kg
Total weight by percentage	174 %	**100 %**	71 %	54 %	49 %	33 %
Price by percentage	132 %	**100 %**	85 %	78 %	66 %	63 %
Reliability S_H	1.3	1.3	1.3	1.3	1.4	1.6
Reliability S_F	6.1	5.7	3.9	2.3	2.3	2.3

Table 2: Comparison of gear units equipped with gear wheels of different materials. Rated pinion moment 21 400 Nm ; n_1 = 500 rpm; i = 3; application factor K_A = 1.25; prevention of pittings: $S_{H\,min}$ = 1.3; root: $S_{F\,min}$ = 2.3; individual design

3.2 Bearings

3.2.1 Rolling bearings

Low to medium capacity gear units operating at speeds up to approx. 3000 rpm are mainly equipped with rolling bearings.

Advantages:

– low starting friction
– glow friction losses throughout the speed range
– the shaft material does not have an impact on the running properties
– easy lubrication; lifetime lubrication is possible
– good emergency running properties in case of bearing failure
– bearings are less wide and still accommodate the same loads
– international standardization

Disadvantages:

– limited suitability for high speeds (centrifugal forces)
– susceptible to contaminated lubricants
– limited service life

Rolling bearings can be lubricated with grease or oil. In case of oil lubrication the lubricant is usually applied by immersion or splashing, in some cases by means of a pressurized circulation system or by injection.

Lifetime lubrication is possible if a grease is used (depending on the intended service life of the gear).

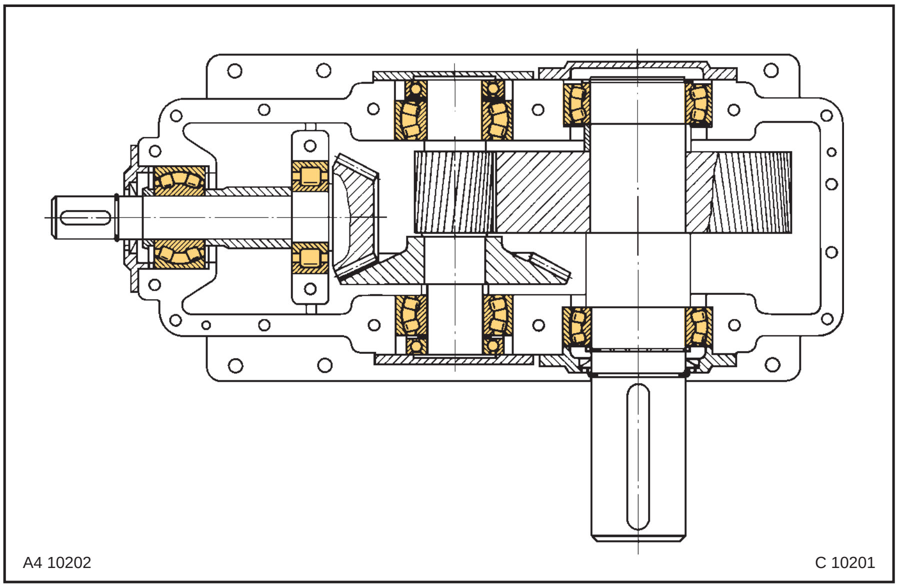


Fig. 13: Bevel / spur gear with rolling bearings

3.2.2 Plain bearings

Plain bearings are generally used wherever the performance limit of rolling bearings is exceeded (speed, load), for example in high speed turbo-gears (speeds between 3000 and 70,000 rpm) or large scale generators (power plants, ships).

They are also used in gears which have to operate especially smoothly.

Advantages:

– simple design
– unlimited life if lubricated adequately
– suitable for very high speeds
– not sensitive to the ingress of dust
– good vibration, shock and noise damping properties

Disadvantages:

– require complex lubrication and cooling systems
– plain hydrodynamic bearings: mixed friction occurs during starting and stopping
– hydrostatic jacking pumps required for large gears

Plain bearings are normally lubricated with oil by immersion, splashing or pressurized circulation. Bearings generating a low amount of heat and subject to low loads can also be lubricated with grease.

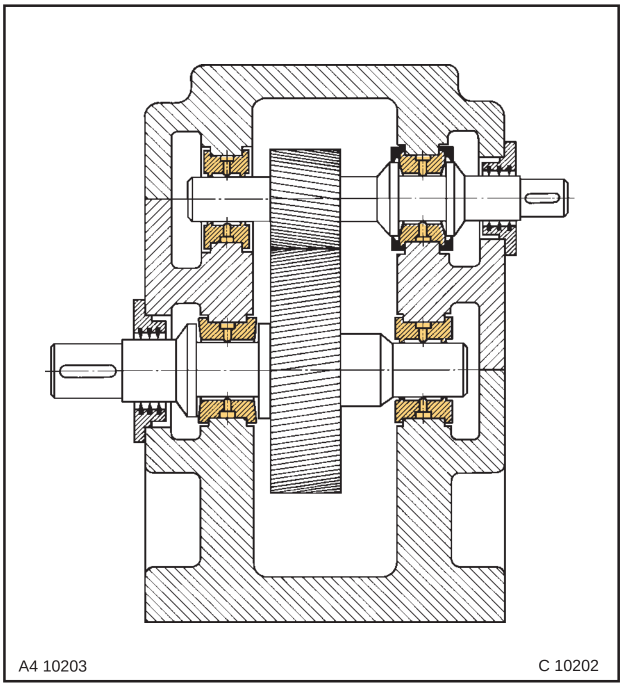


Fig. 14: Spur gear with plain bearings

Low-capacity small and miniature gears are often equipped with plain sintered metal bearings which are lubricated for life with special impregnating fluids and are suitable for high speeds.

Small bearings subject to low speeds and loads may also be equipped with plain bearings made of tribo-system materials (dry bearings).

Klüber offers semi-finished and ready-to-use bearing bushings made of tribo-system materials under the trade name *Klüberplast*.

3.3 Seals

A distinction is made between static and dynamic seals depending on whether the machine elements to be sealed have a relative movement or not.

Dynamic seals are divided into non-contact gap type seals, contact seals and sealing elements combining the functions of the first two types.

3.3.1 Static sealing elements

Flat seals

Flat seals are used to seal static gear components such as flanges, housing parts or oil trays. The sealing effect is generated by the fact that the two opposing surfaces exert pressure on the seal and thus compress irregularities into the seal surface.

Flat seals can be of the hard, composite or soft type, the latter being most common today.

Soft seals can be made of elastomers, rubber asbestos (IT materials in acc. with DIN 3754), cork and rubber compounds, artificial resin and thermoplastic materials.

Sealing masses, tapes, and putty

In contrast to flat seals, pressure is not required for these types of seals. They form a very thin sealing layer between the surfaces to be sealed without requiring any additional pressure. They are available in the form of removable rubber, kneadable and hardening masses and are preferred over flat seals if an impact on center distances or the concentricity of split bores is possible.

O-ring seals

Such seals are standardized under DIN 3770 (compression molded seals) and DIN 2693 (extrusion molded seals). They are made of elastomers and are used in gears for static sealing purposes, e.g. in bearing covers, inspection holes, threaded pipe connections, etc.

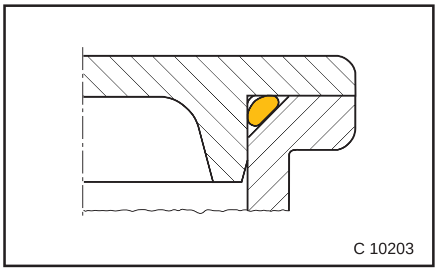


Fig. 15: O-Ring cover seal

3.3.2 Contact dynamic seals

are used on shafts and require precisely machined shaft surfaces as well as continuous lubrication to keep wear to a minimum. The use of such seals may be limited at high speeds due to the high friction temperatures that are generated.

Radial shaft seals

These sealing elements are used on rotating shafts. They are standardized in DIN 3760. The standard types are A and AS, i.e. radial shaft seals with a rubber coating without (A) and with (AS) an additional dust lip. For special cases of application there are also radial shaft seals with an outer metal case and seals with an outer case plus a (reinforcing) cap.

Radial shaft seals with a helix for reverse pumping action are used to improve the sealing effect in case of oil lubrication and to extend the service time, especially if elastomer seals (acrylate or fluorinated rubber) are used at increased temperatures.

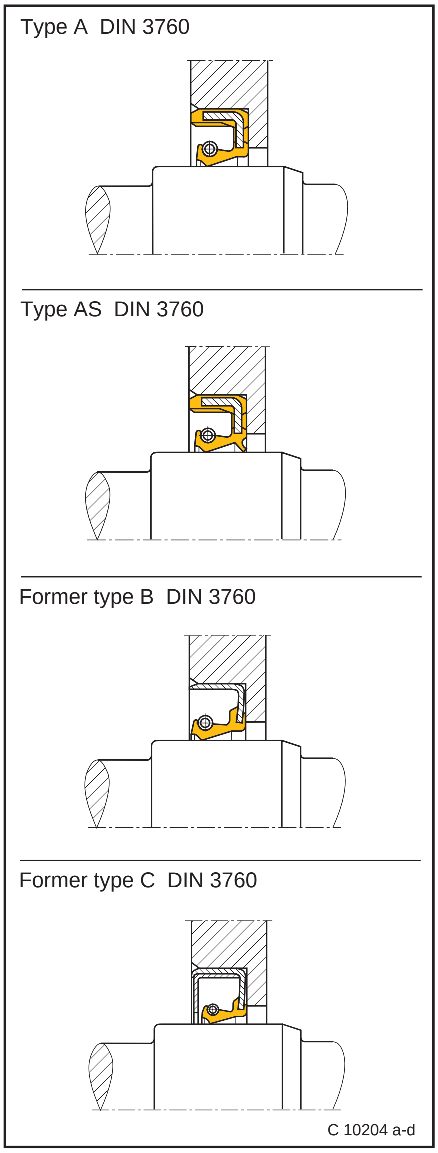


Fig. 16: Radial shaft seals

The speed limit of radial shaft seals depends, among other things, on the elastomer base material, the oil level in case of oil lubrication (dissipation of heat), the type of lubricant (oil, grease), the possibility of dissipating frictional heat (hollow or solid shaft, shaft diameter) and the surface roughness of the contact areas. If radial shaft seals are used in front of grease lubricated bearings (fig. 17) the space between the protective lip and the sealing lip must be completely filled with grease (NLGI 1 or 2) in order to prevent the seal from wearing and protect the shaft against corrosion. It is important to test the compatibility of the grease and the sealing material prior to application.

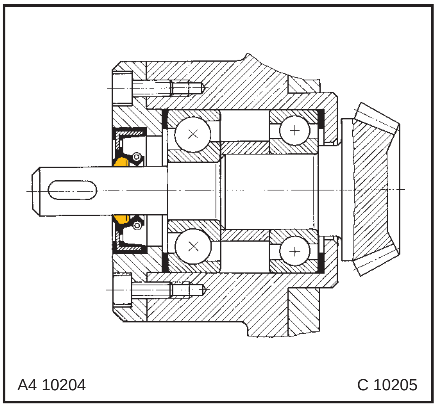


Fig. 17: Bevel gear with a radial shaft seal

The sealing and dust lips must never dry out and require greasing or oiling prior to installation. The same applies to the sealing surfaces on the shaft. In case of oil lubrication the oil must dissipate the frictional heat generated at the sealing lip.

Axial lip seals

The sealing effect is generated by exerting axial pressure on the elastomer sealing lip, thus pressing it against a contact surface that is perpendicular to the shaft axis (*Fig. 18 a*).

Axial sliding seals

These seals rotate together with the shaft. The sealing lip slides against a contact surface at a right angle to the shaft axis (*Fig. 18 b*).

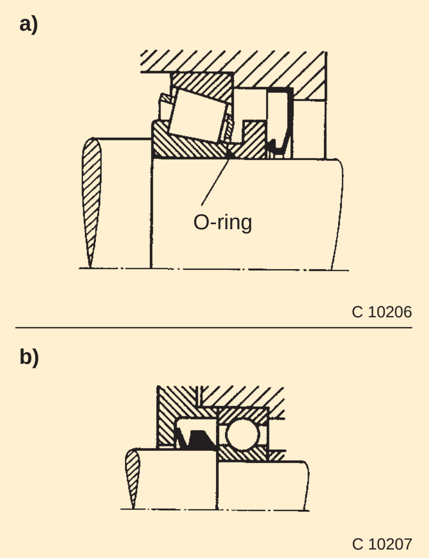


Fig. 18: a) Axial lip seal
b) Axial sliding seal

Rolling bearings with sealing disks

Sealing disks located at one or both sides of the bearing serve the purpose of retaining the initial fill of grease in the bearing and preventing the ingress of water and dust. These seals are not effective against fluids.

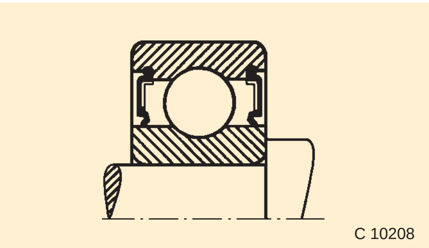


Fig. 19: Sealing disk– RS

3.3.3 Non-contact, dynamic seals

Advantages as compared to contact dynamic seals:

– no wear at high speeds
– negligible frictional heat
– no special requirements in terms of chemical and heat resistance of the seal material (temperature, medium).

A disadvantage is that these elements do not provide static sealing, i.e. they must never be in contact with the oil if the gear stands still in order to avoid leakage. Splash lubrication is therefore only possible to a limited extent because it is dependent on the oil level.

A basic distinction is made between seals with a smooth gap, (conveying) grooves, labyrinth gap, labyrinth and flinger.

Seals performing the function of a gap type seal after a certain period of running in are also counted among the non-contact seals. These include felt rings, resilient laminar rings and resilient sealing disks.

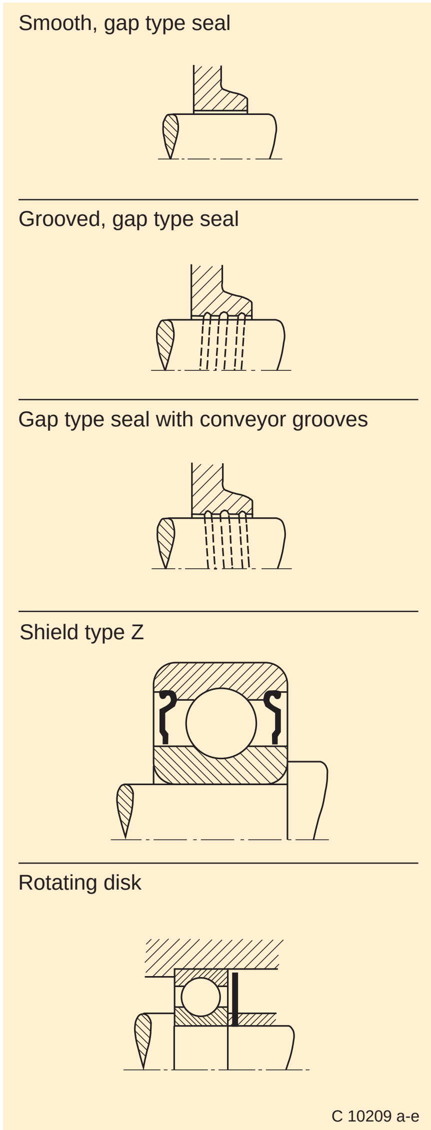


Fig. 20: Gap type seals

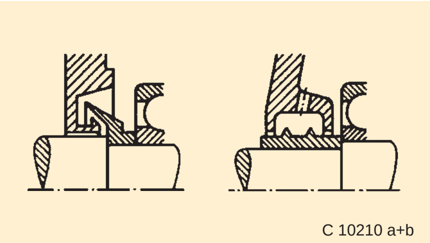


Fig. 21: Gap type seal with flinger

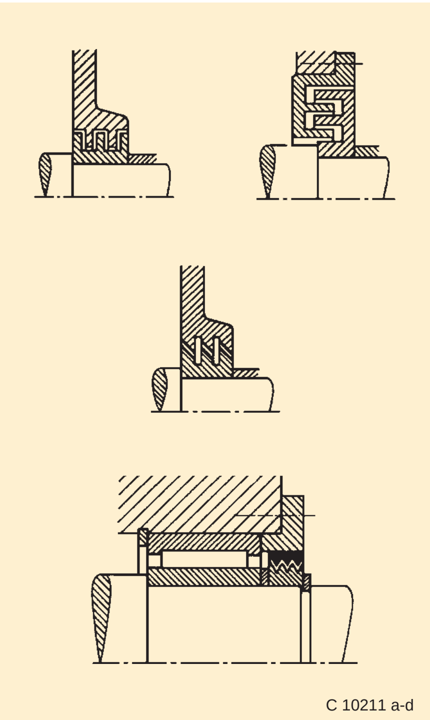


Fig. 22: Labyrinth seal

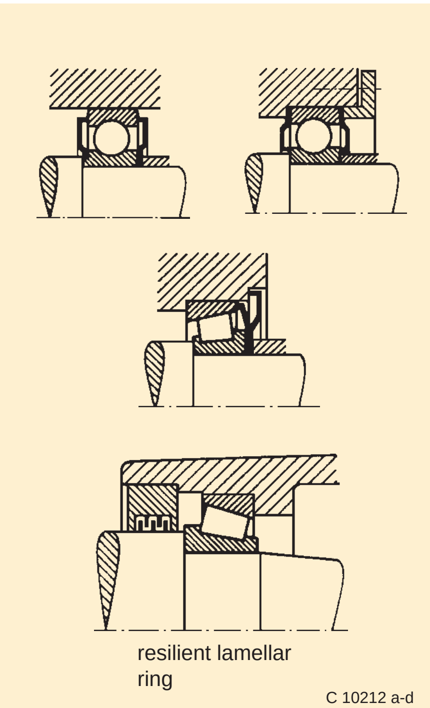


Fig. 23: Resilient sealing disk

4 Gear lubrication basics

4.1 Types of movement and speed

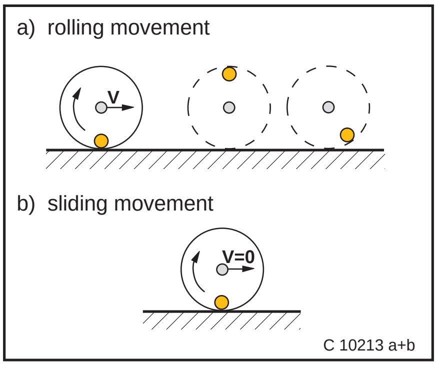


Fig. 24: Rolling and sliding movement

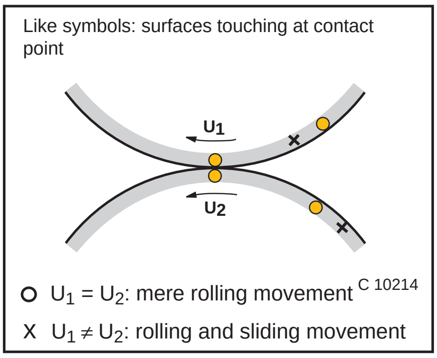


Fig. 25: Schematic drawing of a combined rolling and sliding movement

In all types of gears described under 2.1 the tooth flanks perform rolling and sliding movements while the teeth are meshing. *Table 1* on page 7 shows that the sliding percentage of the combined rolling/sliding movements varies depending on the type of gear. The ratio of the sliding and rolling speed is of vital importance for the load to which the lubricant is subject.

In gears where the rolling movement is predominant the tooth flanks and the lubricant are subject to less load than in gears with a predominant sliding movement (parallel/ helical gears, helical gears).

Fig. 26 illustrates the speed conditions in parallel gears and parallel/ helical gears. Point a) shows that in parallel gears the sliding speed is only vertical. It is zero in the pitch point (pure rolling friction) and increases in the direction of the tip of the tooth. Point b) illustrates that in parallel/helical gears there is also a sliding (spiral) movement in the horizontal direction and that there is even a certain percentage of sliding movement in the pitch point.

This is the reason why the ratio between the sliding and the rolling speed is relatively high in worm gears and hypoid gears with offset shafts. The direction of the sliding speed is not constant but changes in the vertical direction. The "wiping" movement considerably impairs the formation of a lubricant film under pressure.

The load on the tooth flanks and the lubricant increases as the sliding percentage grows.

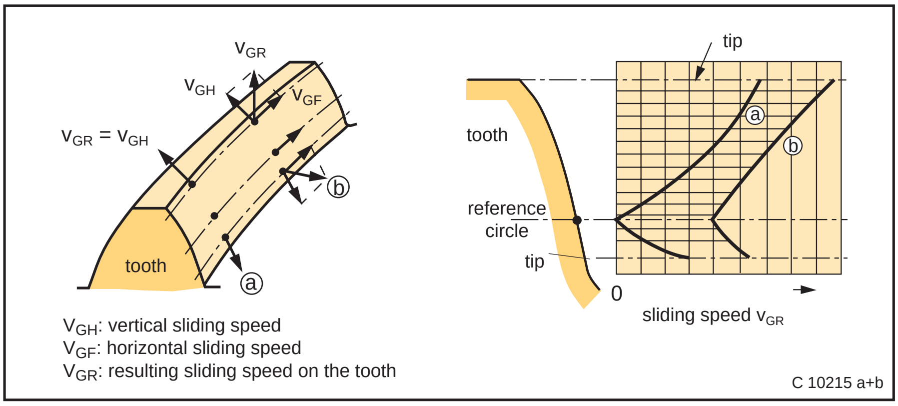


Fig. 26: Sliding speeds on tooth flanks of rolling spiral and helical gears

4.2 Lubrication condition

4.2.1 Full fluid film lubrication

Full fluid film lubrication would be the ideal condition of lubrication. The meshing tooth flanks would be completely separated by a lubricant film. Due to the gap geometry and the various movements, however, full fluid film lubrication on the meshing flanks is only achieved on sufficiently large flank areas under certain operating conditions. Mixed friction prevails on the other areas. The size of the full fluid film areas mainly depends on the tooth flank geometry and is largest in spur and bevel gears.

The formation of a load-carrying and separating lubricant film around the pitch point can be explained by means of the elasto-hydrodynamic lubrication theory (EHD), which is based on the following:

- High pressures up to 10,000 bar occur in the contact zones between the tooth flanks. They have an impact on the oil's viscosity since pressure increases viscosity.
- In the meshing phase the contact surfaces are deformed elastically (contact stress).

The high surface pressure between the tooth flanks results in a sudden increase in the oil film's viscosity making the lubricant so viscous that it cannot flow off. The oil film pressure deforms the tooth flanks elastically in the contact points (flatter and larger contact area), which leads to a greater lubricant gap and a thicker oil film.

The most favorable conditions for a separating film are present in spur gears due the high percentage of rolling movement and the line contact when the teeth mesh.

Film thickness increases as the pressure related viscosity and the peripheral speed rise. It decreases, however, as the pressure increases further and internal friction causes the oil film temperature to rise which, in turn, results in a decrease of viscosity.

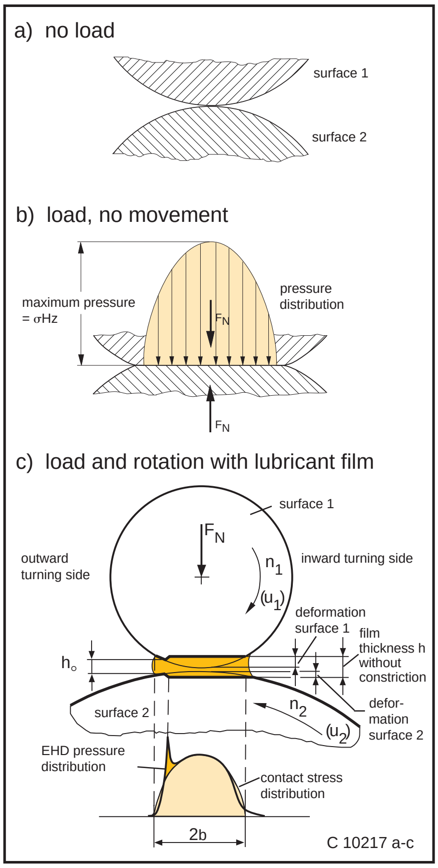


Fig. 28: Formation of a lubrication gap in case of elasto-hydrodynamic lubrication (EHD)

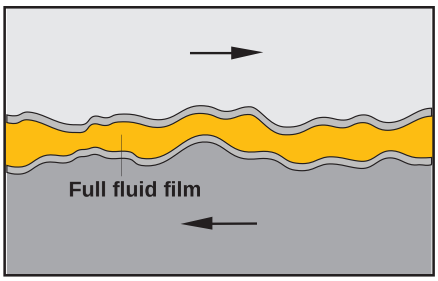


Fig. 27: Full fluid film lubrication C 10216

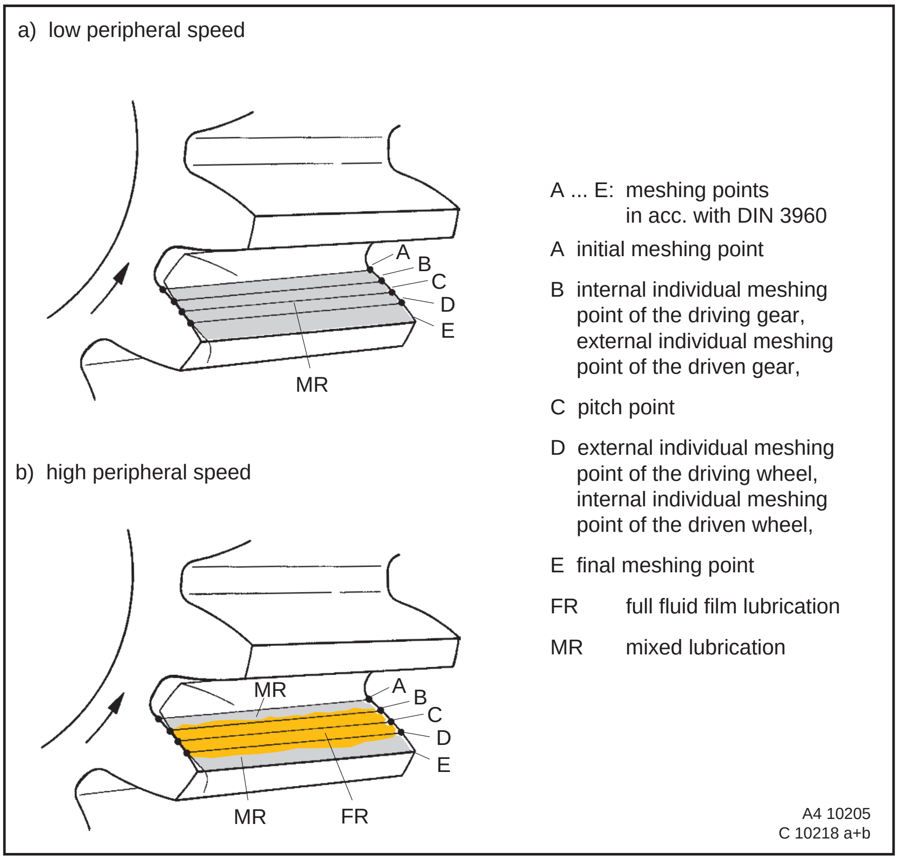
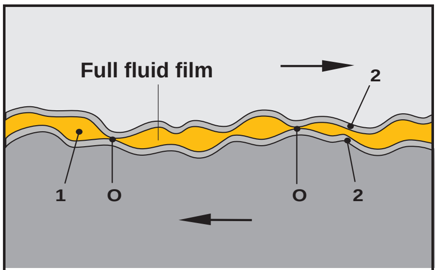


Fig. 29: Typical friction areas on tooth flanks

4.2.2 Mixed lubrication

Mixed lubrication is the lubrication condition that includes both fluid friction and dry friction.

The surfaces of the meshing teeth are not completely separated by a fluid film and are in direct contact in some parts.



O Surface layer friction
1 Lubricant
2 Surface reaction layer
C 10219

Fig. 30: Mixed friction

Mixed friction occurs in cylindrical and bevel gears operating at high peripheral speeds and subject to high specific load. At increased peripheral speeds mixed friction is replaced by fluid friction (*Fig. 30*).

Worm and hypoid gears usually operate under mixed friction conditions.

A lubricating oil with antiwear additives forms reaction layers (boundary layers) on the flank surfaces. This largely prevents metal/metal contact and avoids scoring and coarse wear. The most common extreme pressure additives are of the sulfur/phosphorous type. Chemical reactions with the tooth material produces iron sulfide and iron phosphorate layers.

The course and intensity of these tribo-chemical reactions depend on the following reaction parameters: pressure, temperature, type and quantity of the additives involved. The additives are usually combined in such a way that they become active with an increasing tooth flank temperature.

As compared to the base materials, the reaction layers have a low shear stability. This ensures a continuous process of wear and renewal of the boundary layers (layer wear).

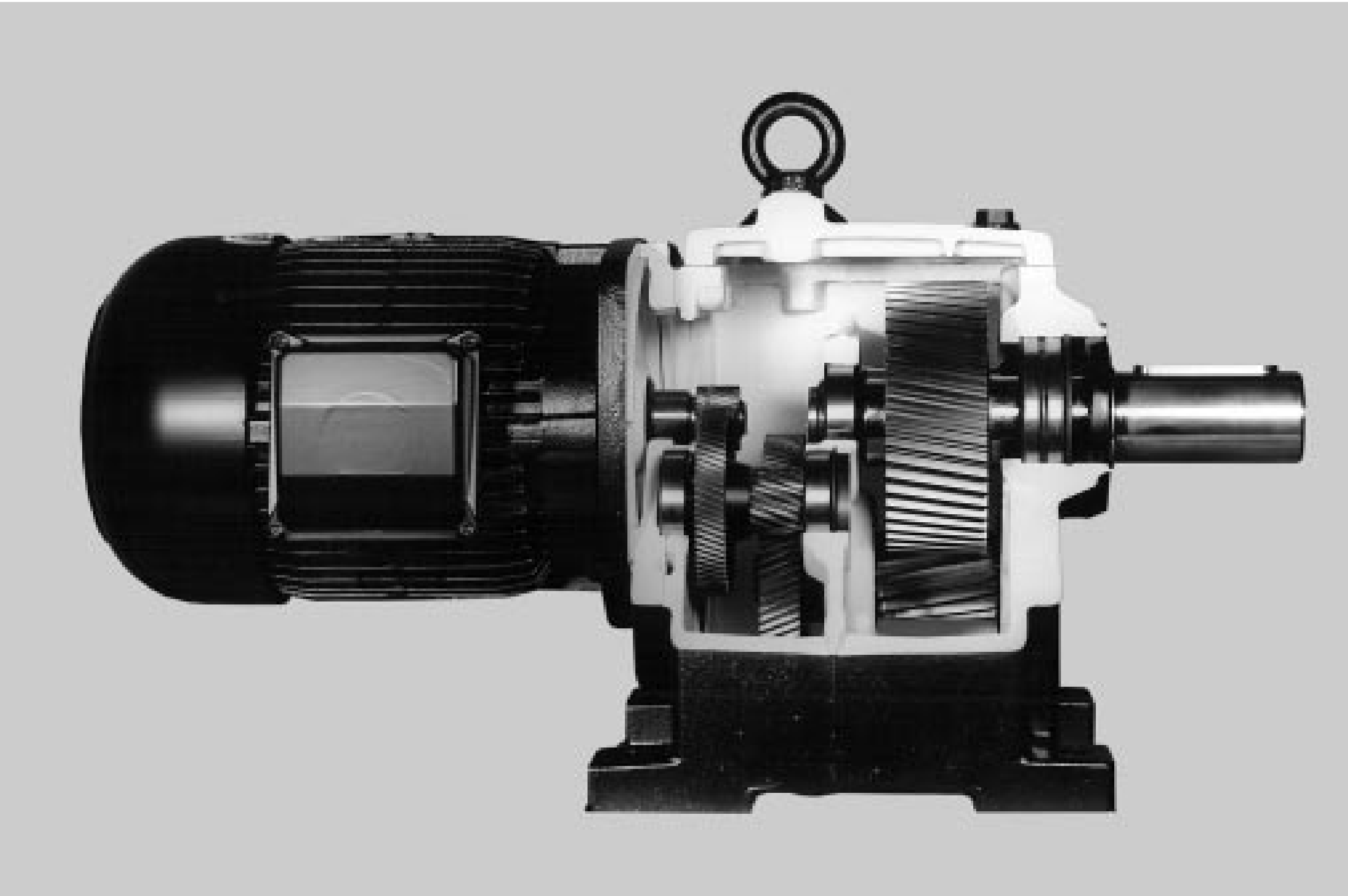


Fig. 31: Gear motor with a three-stage helical gear, made by Getriebebau Nord, D - 22941 Bargteheide, Germany F 10241

5 Types and methods of lubrication

There is a correlation between the type of gear and its peripheral speed, and the type and method of lubrication. See also *table 17* on *page 47*.

Depending on the type of lubricant, the application methods listed in *table 3* are most common for the lubrication of gear systems.

Lubricant type	Lubrication method
Oil	Splash lubrication Combined splash and circulation lubrication Force-fed lubrication (injection lubrication)
Fluid grease[1]	Splash lubrication
Grease [1]	Splash lubrication Splash lubrication with a gear housing almost completely filled with grease Spray lubrication (total loss lubrication) Lifetime lubrication of the tooth flanks
Adhesive lubricants [2]	Splash lubrication (if fluids are used) Spray lubrication (intermittent and quasi-continuous)

[1] See also table 17, page 47

[2] See also our brochure 9.2 e, "Lubrication of large gear drives"

Table 3: Types of lubricants and lubrication methods

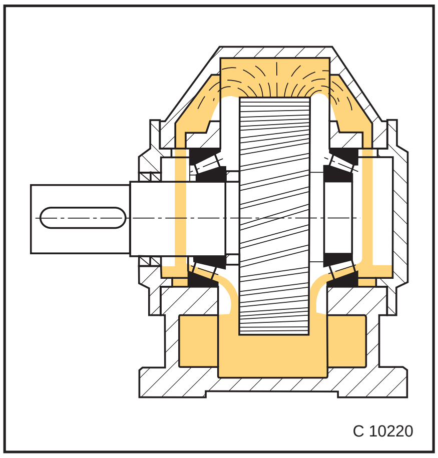


Fig. 32: Splash lubrication. Oil lubrication of rolling bearings via intermediate reservoirs

5.1 Oil lubrication

The majority of closed industrial gears are lubricated with an oil by means of splash lubrication.

Other application methods include splash/circulation lubrication (used if heat dissipation is important) and force-fed circulation lubrication (in large gears operating at high speeds).

Other lubrication methods are irrelevant in context with stationary gear drives and will not be dealt with in the following.

5.1.1 Splash lubrication

A lubricant depot in the gear housing into which the teeth are immersed is the easiest way of ensuring continuous lubrication. It is therefore the most common lubrication method for closed, oiltight gear systems.

It is especially economic, simple in design and reliable. Its cooling effect is sufficient for most applications and can be improved by means of special housing designs (e.g. cooling ribs) or auxiliary units such as cooling fans or water cooling lines.

In splash lubrication one wheel per gear stage usually plunges into the oil sump and carries some oil to the meshing zone. Oil dripping off the gears and shafts due to centrifugal forces either runs into special oil return lines and is carried to the bearings that also require lubrication, or runs directly back into the sump along the housing walls.

Splashing oil to a certain extent also lubricates the wheels that are not immersed. In some cases the shafts are equipped with additional oil splashers to intensify the splash effect.

Oil splash lubrication without any additional design measures is feasible up to a peripheral speed of approx. 20 m/s. Oil guides (baffle plates, oil pockets, etc.) are required for increased peripheral speeds.

Tests with single-stage spur gears have shown that splash lubrication is efficient up to a peripheral speed of 60 m/s.

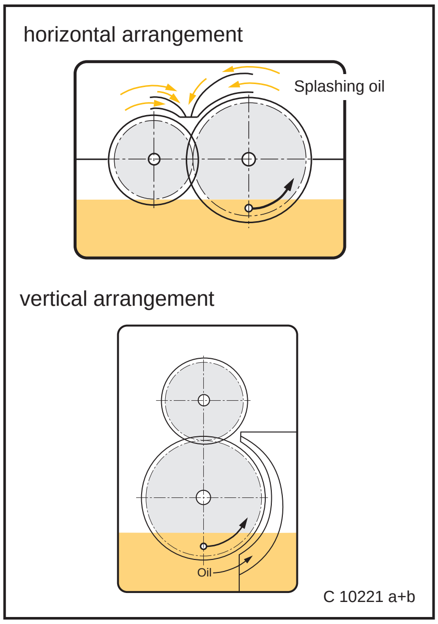


Fig. 33: Examples of oil guide plates

Oil level and immersion depth

With splash lubrication it is important that a certain oil level is retained at all times in order to prevent damage and ensure reliable operation.

If the oil level is too low, it may result in starved lubrication, inadequate heat dissipation and increased wear.

If the oil level is too high, churning losses may increase, resulting in higher temperatures. This, in turn, accelerates oil ageing, decreases the oil's service life, increases viscosity and thus reduces the lubricant's pressure absorption capacity in the meshing zone. It also results in the generation of foam and noises.

When the peripheral speed is increased, the depth of immersion is reduced in order to keep churning losses to a minimum. The following rule of thumb applies to spur gears:

Depth of immersion

3 to 5 times the module
up to 5 m/s

1 to 3 times the module
> 5 to 20 m/s

With higher peripheral speeds it becomes more difficult to wet the tooth flanks sufficiently, and the oil level is reduced due to oil splashing. In case of very high peripheral speeds it is therefore important to increase the depth of immersion of the gear wheels irrespective of potential churning losses. Gears with fully immersed pinions (e.g. bevel gears) are found quite often. A great difference in wheel diameters in the individual gear stages may also be a reason for deviating from the immersion depths listed in the following.

Type of gear	Operating conditions	Depth of immersion
Spur gears	Peripheral speed up to 5 m/s	3 to 5 times the module
	Peripheral speed > 5 ... 20 m/s	1 to 3 times the module
Bevel gears	–	Immersion of teeth over the entire width of the wheel
Worm gears	Worm on the top	Wheel immersed to approx. 1/3 of its diameter
	Worm at the bottom	Wheel immersed to approx. the center of the meshing zone
	Worm at the side	Wheel immersed to at least 1/2 of the worm height

Table 4: Recommended depth of immersion

5.1.2 Combined splash and circulation lubrication

If additional cooling ribs, fans or ducts in the housing are not enough to compensate for the power losses in a splash lubricated gear, heat can also be dissipated by using a combined splash and circulation lubrication system.

The easiest method is to install a pump driven by the gear which takes a certain oil quantity out of the oil sump and returns it to the gear via an oil cooler. Electrically operated oil pumps (main and backup pumps) are also used.

Thermostatically controlled coolers and filters installed in the oil circuit make it possible to extend the oil change intervals. The depth of wheel immersion is determined as described for splash lubrication.

The oil fill quantity has to be somewhat higher because a certain amount of oil is always circulating.

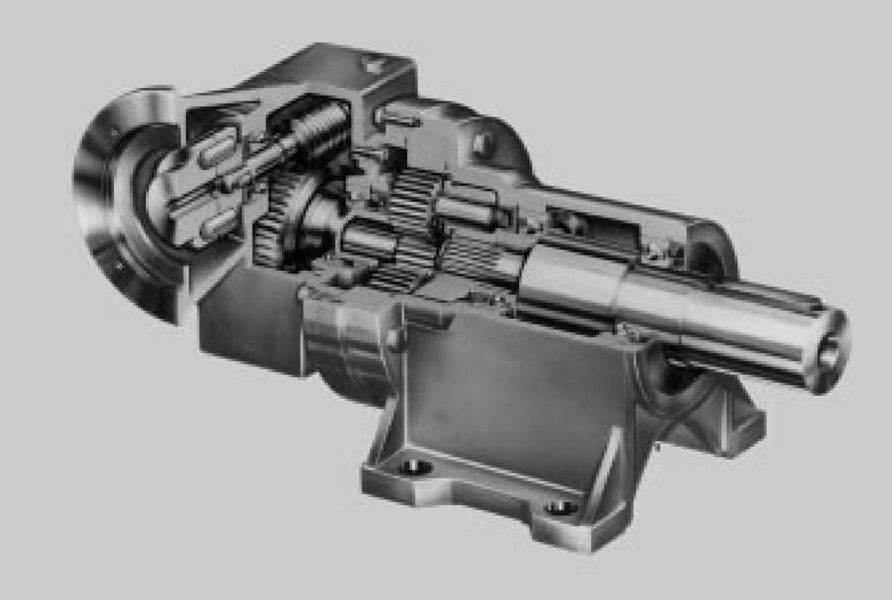


Fig. 34: Planetary worm gear, Rhein-Getriebe GmbH, D-40667 Meerbusch, Germany F 10242

5.1.3 Force-fed circulation lubrication
(injection lubrication)

Force-fed circulation lubrication is used at peripheral speeds too high for splash lubrication and in case of gears equipped with plain bearings. This lubrication method is suitable for even the highest peripheral speeds encountered in gear systems (approx. 250 m/s).

Oil is brought onto the tooth flanks via slotted or perforated nozzles. The oil is injected into the contact zone, either at the initial or the final meshing zone. It is assumed that oil injected into the initial meshing zone is more benefitial to the lubrication process, while oil injected into the final meshing zone intensifies the cooling effect.

The injection quantity depends on the amount of heat to be dissipated. As a rule of thumb we recommend 0.5 to 1.0 l / min per cm of tooth width.

The required oil circulation quantity is made up of the lubricant quantity required for the gear wheels plus the quantity required for the bearings.

Depending on the type of circulation, a distinction is made between wet and dry sump lubrication. In wet sump lubrication (*Fig. 35*) the oil reservoir is the oil sump in the housing from which the oils is brought to the friction points.

In dry sump lubrication (*Fig. 36*) the oil which returns from the friction points is collected in the housing and then transferred into a separate oil container from where it is brought back to the friction points.

Dry sump lubrication is used if the oil volume is too big to be accommodated in the gear housing.

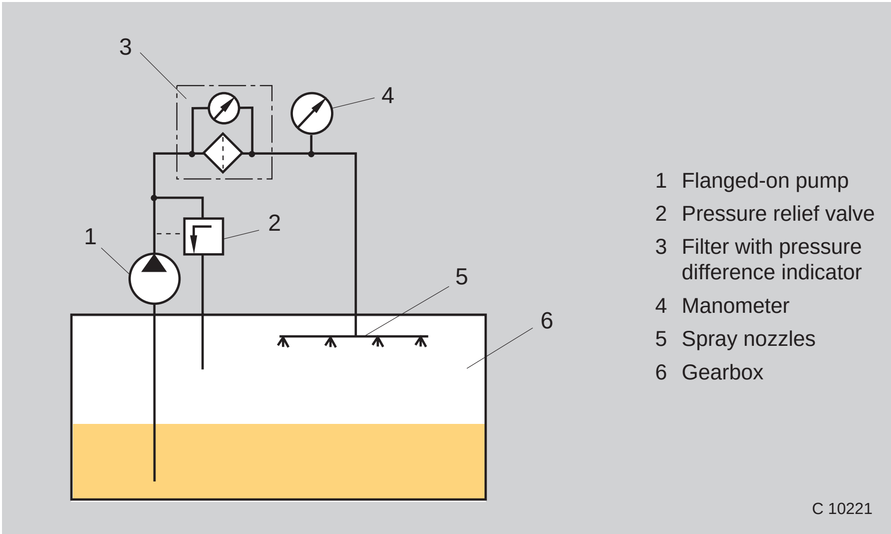


Fig. 35: Schematic drawing of a wet sump lubrication system

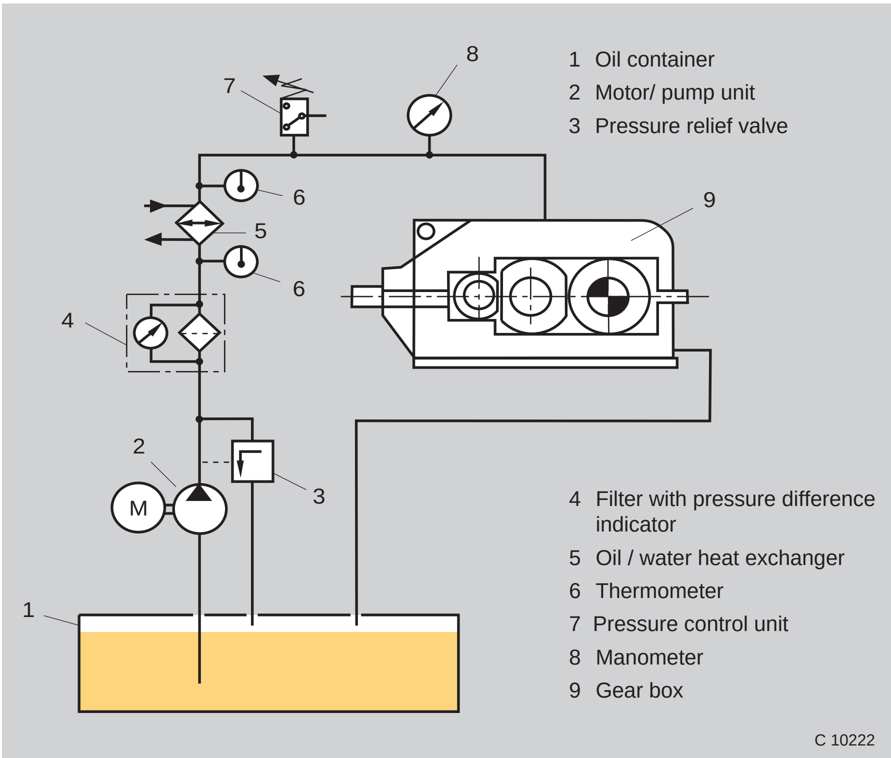


Fig. 36: Schematic drawing of a dry sump lubrication system

5.2 Grease lubrication

Grease lubrication is normally used for gears that are not oiltight or, for safety reasons (leakage), for oiltight gears installed in a position preventing maintenance and repair. Grease lubrication is also suitable

With grease lubrication it is important to take into account that exchanging the grease is very complicated because the gearbox usually has to be taken apart completely and cleaned thoroughly.

for gears with a limited life that are suitable for lifetime lubrication.

As grease cannot dissipate heat or only to a limited extent, grease lubrication is restricted to low and medium capacity gears operating in short or intermittent intervals.

Grease is mainly used on gear motors and in all types of small and miniature gears, e.g. power tools, household and office equipment, automotive servo drives, etc.

Splash lubrication is the common method of application. Gears with a low specific load that are mainly used to transfer movements over a limited period of time or gears designed for a certain operating period are usually lifetime lubricated. With this application method the peripheral speed is limited to 2 to 3 m/s.

5.2.1 Splash lubrication with gear greases

As already mentioned it is important to take into consideration that greases have a considerably lower heat dissipation capacity than lubricating oils.

Splash lubrication with gear greases is therefore restricted to low to medium capacity gears. As the development and dissipation of heat largely depend on the individual operating conditions (permanent or intermittent operation) and the casing design, it is not possible to indicate a specific capacity limit for grease lubrication.

Unless the gears have a very low capacity we recommend carrying out tests to establish the gear heating rate.

Gears with high peripheral speeds operating continuously and/or used to transfer power require a grease with a high base oil content to ensure heat dissipation: a fluid grease of consistency grade 000 or 00 or at least a very soft grease of grade 0.

In continuous operation the peripheral speed limit is as follows:

NLGI grade DIN 51 818	Peripheral speed [m/s]
000	6 to 8
00	4 to 5
0	2 to 3

Fluid gear greases have a less favorable flow behavior than oils. Excess grease thrown off the gear returns to the lubricant sump at a much lower speed. It is therefore important to fill the grease immersion bath to a higher level than in case of oil lubrication in order to prevent starved lubrication.

Depending on the grease's consistency the approximate depth of immersion should be between 1 and 3 times the tooth depth.

Splash lubrication with a gear grease at peripheral speeds above 8 to 10 m/s in continuous operation is possible if the following steps are taken:

a) Use a gear grease of consistency grade 0 or 00 and fill the casing to approx. 30 to 40 % of its volume.

b) Use a gear grease of consistency 1 or 2 and fill the casing almost completely.

Low capacity gears in short term or intermittent operation can be lubricated with a gear grease up to a peripheral speed of 25 m/s if method b) is applied.

NOTE: For detailed information about lubrication with gear greases refer to section 8.4, page 57, *Selection of gear greases*.

5.2.2 One-time lubrication of tooth flanks

See NOTE above.

5.3 Lubrication with adhesive lubricants

Adhesive lubricants are mainly used for the lubrication of open gears and gear systems of large to very large dimensions, e.g. drives of rotary kilns, tube mills, lifting cylinders, cranes and construction machines.

Depending on the operating mode and/or conditions and the drive design (with/without cover, with/without lubricant reservoir), the lubricant is applied

– once (or several times at very large intervals)
– continuously (long term lubrication)
– intermittently (total loss lubrication)

Fig. 37 on *page 22* shows the correlation between the types of lubrication and the application methods.

Peripheral speeds [m/s]	Application method
up to 2	manual application
up to approx. 8	splash or spray lubrication
> 8 to approx. 11	spray lubrication

Table 5: Correlation between the peripheral speed and the application method in case of lubrication with adhesive lubricants

The lubricant to be used determines the type of lubrication and the suitable application method.

Adhesive lubricants used on open gears can be fluid or semi-solid and therefore have an impact on the application method.

A description of all common methods of application would go beyond the scope of this brochure.

For details about lubrication with adhesive lubricants please refer to our brochure "Lubrication of large gear drives", 9.2 e.

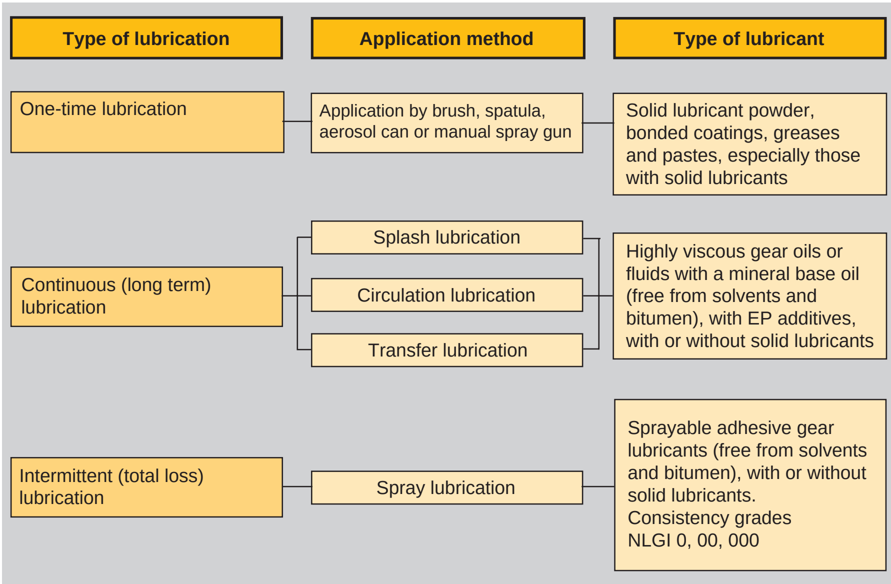


Fig. 37: Lubrication types and application methods common with adhesive lubricants

6 Energy losses and heating of gear systems

6.1 Efficiency of gear systems

The efficiency η o gear systems is defined as the ration of the input power P_b and the output power P_a.

$$\eta = \frac{P_b}{P_a} < 1$$

$P_b = P_a - P_v$ (P_v = power loss)

$$\eta = \frac{P_a - P_v}{P_a} = 1 - \frac{P_v}{P_a}$$

Gears with a low sliding percentage, such as spur and bevel gears, have a very high efficiency (up to 99 % in the case of single-stage spur gears), whereas gears with a high sliding percentage, e.g. hypoid and worm gears, have a relatively low efficiency. *Table 6* gives a general survey on gear efficiency.

It is important to note that efficiency decreases with an increasing number of gear stages.

The efficiency of worm gears depends on the worm speed, the transmission ratio and the overall size of the gear system. Efficiency increases with a higher worm speed (increasing sliding speed of the teeth), lower transmission ratio (increasing pitch angle) and larger center distance. Efficiency values stated in gear brochures usually refer to the gears' rated efficiency and apply to full load operation.

The efficiency of gear systems varies as a function of the load. It is at its maximum under full load and decreases under partial load.

Unless specified otherwise, the indicated efficiency applies to standard gears lubricated with a gear oil with a mineral hydrocarbon base.

Synthetic gear oils can improve a gear's efficiency up to 30 %, especially in the case of gears with a high sliding percentage, such as hypoid and worm gears. See also section 7.2.2, page 33.

Type of gear	Overall efficiency under nominal load approx. [%]
Spur gears	
single stage	up to 98.5
two stage	up to 97
three stage	up to 95.5
four stage	up to 94
Bevel gears	up to 98
Straight bevel gears	
two stage	up to 96.5
three stage	up to 95
four stage	up to 93.5
Planetary gears	
single stage	up to 97
Crossed helical gears	50 to 98.5
Hypoid gears	
high transmission ratio	50 to 90
low transmission ratio	85 to 96

Table 6: Efficiency of industrial gears

6.2 Power loss in gear systems

The total power loss P_V of a gear system is made up of various individual types of losses:

$$P_V = P_{VZ} + P_{VZ0} + P_{VB} + P_{VB0} + P_{VD} + P_{VX}$$

P_V = total power loss

P_{VZ} = tooth related power loss, *under load*

P_{VZ0} = tooth related power loss, *no-load condition*

P_{VB} = bearing related power loss, *under load*

P_{VB0} = bearing related power loss, *no-load condition*

P_{VD} = sealing related power loss, *irrespective of load*

P_{VX} = other losses *irrespective of load*

Tooth and bearing related power losses are divided into losses under load and under no-load conditions.

Fig. 38 shows that, with reference to the overall power loss (P_V), the losses under load decrease and the no-load losses increase with an increasing peripheral speed. The curve shown below applies to gears equipped with rolling bearings.

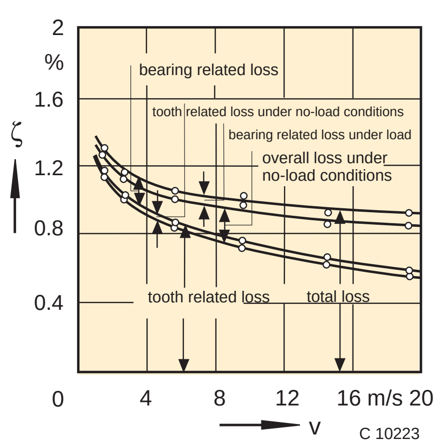


Fig. 38: Total loss in a gear system broken down by individual losses as a function of the peripheral speed (v) – (standard tooth load: 100 N per mm of tooth width)

6.2.1 Tooth-related power loss P_{vz}

This type of loss occurs due to the sliding movement of the meshing tooth flanks operating under load. The individual coefficient of friction on the tooth flanks is of decisive importance.
The tooth related power loss P_{vz} depends on the meshing position of the teeth.

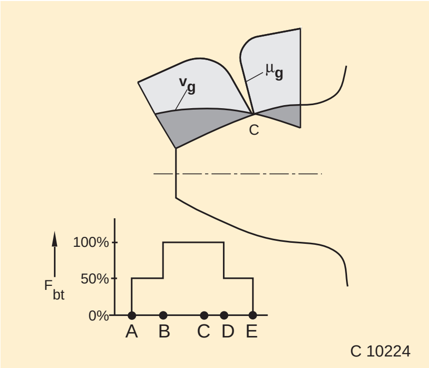


Fig. 39: Variation of the sliding friction coefficient μ_g, the sliding speed v_g and the standard tooth load F_{bt} on the tooth flank and the distance of action A - E

6.2.2 Tooth related power loss P_{VZ0}

In splash lubricated gears this type of power loss comprises the churning and squeezing losses, and in case of gears lubricated by oil injection it is equivalent to the injection loss.

Churning losses

Churning losses occur while the gear teeth plunge into the oil sump. It is determined by the following factors:

- width, diameter of the addendum circle and number of plunging gears
- shape of gear casing (clearance volume, juts, etc.)
- depth of immersion
- peripheral speed of the plunging gears
- operating viscosity of the lubricant

Type of gear	P_{VZ} % of P_a	P_V % of P_a
Industrial gears	0.2 to 0.5	1.0 to 2.8
Worm drive		
i = 5		4.0 to 9.0
i = 10		7.0 to 11
i = 70		24 to 35

Table 7: Average values of tooth related losses P_{VZ} and total losses P_V per gear stage. P_a = driving power

Churning losses in high speed gears may result in increased temperatures which, depending on the gear oil, may be excessive.

Squeezing losses

Squeezing losses occur when excess lubricant is forced out of the meshing zone in a vertical and/or horizontal direction. They are intensified with an increasing nominal viscosity of the lubricant.

Injection losses

Injection losses are generally lower than churning losses in case of splash lubrication. They occur when the oil is forced off the tooth flanks and the injected oil is accelerated and deflected. They depend on the injected oil quantity, the oil viscosity, the tooth width and the wheel diameter. Injection losses are intensified when these factors are increased, but are reduced when the module is increased.

6.2.3 Bearing related power loss P_{VB}

This type of loss mainly depends on the type of bearing, mounting position, lubrication, speed and load. Rolling bearings generally produce lower losses than plain bearings.

Rolling bearings

Bearing related power losses are made up of a load related portion (P_{VB}) and a portion not related to the load (P_{VB0}). Load related losses are caused by the elastic deformation of the rolling elements and the raceways and by sliding movements at the points of contact. Losses not related to the load occur due to the sliding friction between the cage, the rolling elements and the guideways, and because there are hydrodynamic losses in the lubricating oil. They depend on the lubricant's viscosity and quantity as well as the average peripheral speed of the gear.

It is known from experience that the power loss per rolling bearing amounts to approx. 0.1 % of the gear's efficiency.

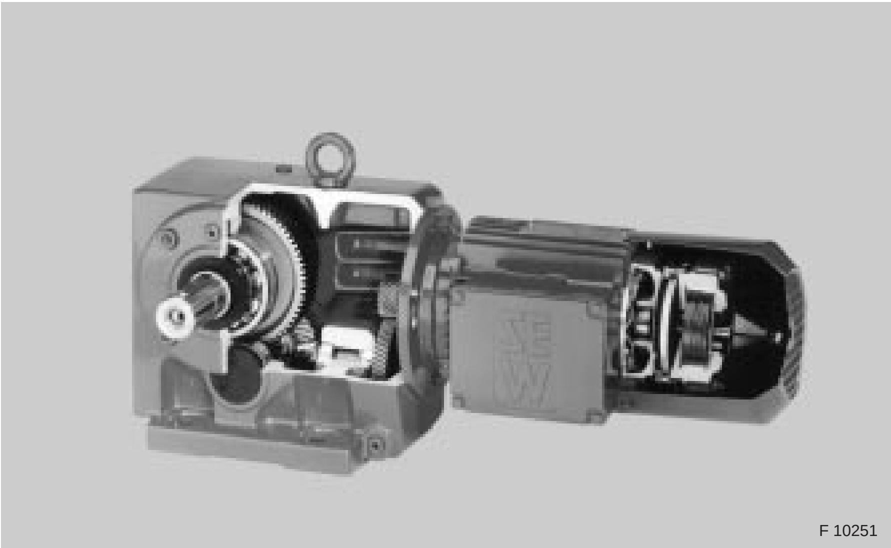


Fig. 40: Three-stage spur/bevel/spur gear motor, oil lubricated SEW-EURODRIVE GmbH & Co, D-76646 Bruchsal

Plain bearings

Losses in plain bearings are intensified with an increasing load.

Power losses are relatively high at low speeds. In the mixed friction regime they decrease with an increasing sliding speed, whereas under fluid friction conditions they increase with an increasing speed.

The average power loss of standard plain bearings is approx. 0.5 to 1.5 % of the nominal efficiency and in case of high performance bearings approx. 0.1 to 0.3 %.

6.2.4 Sealing related power loss P_{VD}

These losses mainly depend on the type of sealing. With respect to the total power losses they are relatively low in case of non-rubbing seals. However, they play a more important role in case of rubbing seals.
If radial shaft seals are used, sealing related losses increase with a larger shaft diameter and with a higher peripheral speed at the sealing lip. The average loss is between 0.01 and 0.04 kW per radial shaft seal.

6.2.5 Other losses P_{VX}

These include losses caused by freewheels, couplings, oil pumps and fans.

The average oil pump loss is between 0.15 % (large gears) and 1 % (small gears) in relation to the nominal output.

Fan losses usually are below 0.5 %.

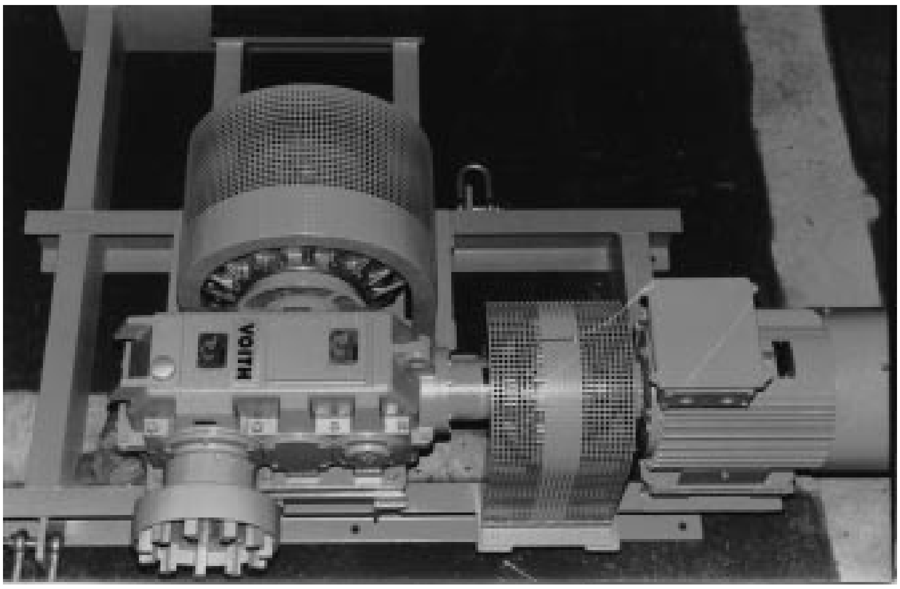


Fig. 41: Three-stage straight bevel gear driving a stone crusher, VOITH F 10244

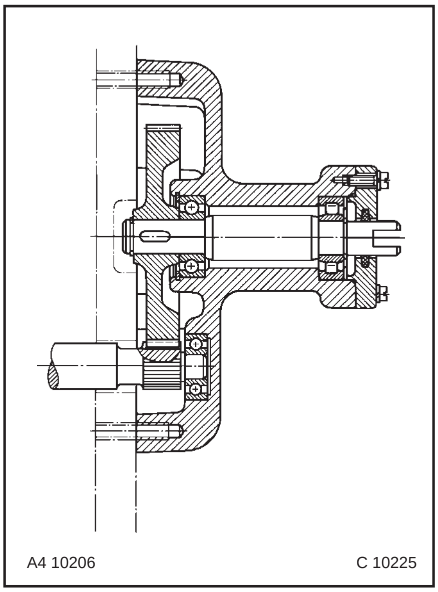


Fig. 42: Shaft-mounted spur gear

6.3 Heating of gear systems

The power losses produced in the friction points between the teeth, bearings and seals result in excessive heat that has an impact on the gears and the lubricating oil.

The gears may be heated additionally by the sun or heat dissipated from the motor. Heating occurs until enough heat is exchanged with the surrounding elements to reach a thermal balance. In most gear systems heat is only dissipated via the casing.

The following equation applies if there is a thermal balance between the gear system and the ambient elements:

$$P_V = \dot{Q}_V$$

Power loss = Thermal capacity

This means that the total power loss P_V is the same as the thermal loss $\dot{Q}_V$ dissipated via the casing surface to the environment.

6.3.1 Heat dissipation

Most of the heat generated by the power losses is dissipated from the friction points via the lubricating oil and transferred to the environment through the gear casing. Only a small share of the heat is dissipated through the connected shafts and foundations.

The gear casing dissipates heat to the environment by way of convection (the casing surface is cooled by air flowing past) and thermal radiation.

The thermal capacity of a gear system, i.e. to what extent waste heat can be dissipated by way of convection and thermal radiation, is determined by

the heat transmission coefficient α,

the casing surface area A,

and

the temperature difference between the oil bath and the ambient air ($t_{oil} – t_L$)

$$\dot{Q}_V = \alpha \cdot A\,(t_{oil} - t_L)$$

From this equation it can be deducted that the amount of heat ($\dot{Q}_V$) that can be dissipated from the gear can be increased by increasing the oil sump temperature. This also implies that it is possible to increase the total power loss (P_V) that can be balanced and in consequence also the transferrable power.

Whether the oil sump temperature can be increased depends on the type of oil and the temperature limit. Mineral hydrocarbon oils have a relatively low temperature limit and are therefore not suitable to increase a gear's thermal capacity by increasing the oil sump temperature.

Synthetic high temperature gear oils are suitable for oil sump temperatures between 100 and 150 °C and therefore make it possible to increase a gear's efficiency considerably by increasing its thermal capacity ($\dot{Q}_V$). See also section 7.2.2, page 36.

Example:
Temperature limit of a standard mineral oil: approx. 90 °C

Supposed ambient temperature: 22 °C

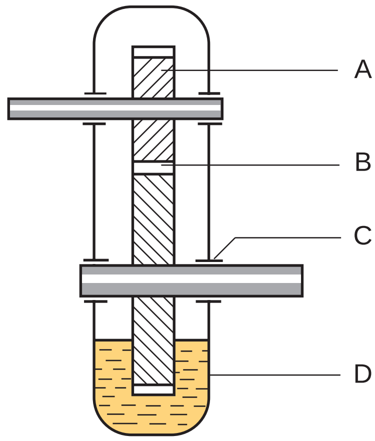
Increase of thermal capacity after conversion to a synthetic oil and increase of the oil sump temperature:

t_{oil} [°C]	$\dot{Q}_V$ increased by ≈ [%]
100	14.7
110	29.4
120	44.1

A gear's thermal capacity can also be increased by means of additional cooling (fan, fan + oil cooler) or by reducing the ambient temperature (ventilation).

6.4 Temperatures in gear systems

Heating of a gear system, individual gears, bearings and lubricants is one of the most important criteria to evaluate gear's performance. The existing temperatures are indicative of the power losses. *Fig. 43* shows the characteristic temperatures in a gear system.



A Temperature of the gear body (roughly corresponds to the temperature of the tooth center)

B Various tooth flank temperatures

C Bearing temperature

D Oil sump temperature

A4 10132

Fig. 43: Characteristic temperatures in a gear system

The various temperature levels have a decisive impact on

– the ratio between fluid friction and mixed friction,
– the formation of pittings on the tooth flanks,
– the degree of scuffing,
– the lubricant ageing rate,
– the bearing's service life, and
– the gear's efficiency.

It is important to ensure that the pertinent temperature limits are not exceeded when the individual gear components, the lubricant and the accessories (filter inserts, pumps, etc) are heated.

Synthetic gear oils, which have a considerably higher temperature limit than mineral oils, are suitable for high temperature lubrication (lubricant temperature > 100 °C) and make it possible to increase the thermal limit and, in consequence, the transferrable power.

The lubricant's operating temperature (defined by Klüber as the oil sump temperature or the temperature of the injected oil) is a decisive parameter when selecting a suitable viscosity.

Operating temperatures above average or temperature peaks indicate malfunctions or incipient damage.

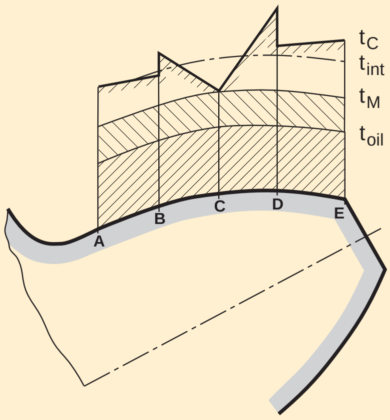


Fig. 44: Temperatures typically found on a tooth flank

6.4.1 Tooth flank temperatures

Fig. 44 illustrates the temperatures typically found on a tooth flank of a straight or helical spur or bevel gear.

Contact temperature t_C

The contact temperature t_C is the flank temperature existing during the meshing phase at the individual contact points along the action distance E-A. In contact point C it is equal to the gear body temperature t_M.

Integral temperature t_{int}

The integral temperature t_{int} is the tooth flank temperature calculated from the contact temperature curve t_C. Scuffing occurs if a critical integral temperature (depending on the lubricant and the gear material) is exceeded.

Gear body temperature t_M

The gear body temperature t_M, also known as the mass or tooth center temperature, roughly corresponds to the tooth flank temperature before meshing occurs. It lies between the integral temperature t_{int} and the oil temperature t_{oil} in the casing (oil sump); $t_{oil} < t_M < t_{int}$.
The gear body temperature determines the temperature of the lubricating oil and its operating viscosity before reaching the lubrication gap, which, in turn, is decisive for the thickness of the oil film in the gap.

Oil temperature t_{oil}

The oil temperature t_{oil}, also known as the oil's operating temperature, is equal to the oil sump temperature in case of splash lubrication and to the injection temperature in case of oil injection lubrication.

The temperature along the action distance E-D in the lubrication gap between the meshing teeth roughly corresponds to the gear body temperature t_M. In the remaining action distance it is almost equal to the contact temperature t_C. This also applies to the oil film temperature.

The various tooth flank temperatures explained above are important for the calculation of the load carrying capacity of the flanks. GfT worksheet no. 2.4.2 illustrates the calculation principle to obtain the pertinent temperatures and scuffing resistance. This worksheet is based on DIN 3990 which describes how to calculate the load carrying capacity of spur and bevel gears.

6.4.2 Temperature limits

The temperature limit is the temperature which a (gear) component can sustain without suffering damage.

The temperature limit of a gear is determined by the component (including the lubricant) most susceptible to high temperatures. In mineral oil lubricated gears this usually is

Component	Temperature limit [°C]
Gear wheels	
Case hardened steel	180 to 300
Tempered steel	> 200
Bronze	> 200
Rolling bearings	
Standard bearings	120
High temperature bearings	300
Plain bearings	
Standard bearings	90
High temperature bearings	150
Shaft seals	
Nitrilorubber	100
Polyacrylate rubber	125
Fluorinated rubber	150
Lubricating oil (oil sump)	
Mineral oil	100
Synthetic oil	160

Table 8: Approximate temperature limits

the lubricating oil, which has a much lower temperature limit than the other gear components.

Gear lubrication with a synthetic oil makes it possible to increase gear temperatures by 20 to 30 °C while ensuring the same oil life than with a mineral oil.

This can increase the gear's thermal capacity and the transferrable power, provided the temperature limit of other components, materials and substances is not below the oil's operating temperature. In such a case it might be necessary to replace them with components with a higher thermal resistance.

This also applies to all components in contact with the gear oil but not listed in *table 8*, such as filter inserts, hoses, inspection glasses and couplings.

The temperature limit of the respective lubricating oil also has to be taken into account!

6.4.3 Oil temperatures (empirical values)

The permanent oil temperature in industrial gears is between 40 and 150 °C, depending on the type of gear, the application and the lubricant (mineral /synthetic oil).

Apart from design related influences, oil temperatures mainly depend on the existing operating temperatures. They rise with an increasing ambient temperature and if the oil is exposed to thermal radiation. They fall when the gear is operated under partial load conditions or intermittently.

A permanent oil sump temperature of 80 °C should not be exceeded if mineral gear oils are used.

Type of gear	Oil sump temperature °C
Splash lubrication	
Worm gears Spur and bevel gears	80 to 100
Force-fed lubrication	
Gears equipped with rolling bearings	50 to 80
Gears equipped with plain bearings	40 to 60

Table 9: Permanent oil sump temperatures (empirical values) if a mineral oil is used. The lower values pertain to an ambient temperature of approx. 20 °C, the higher values to high ambient or room temperatures.

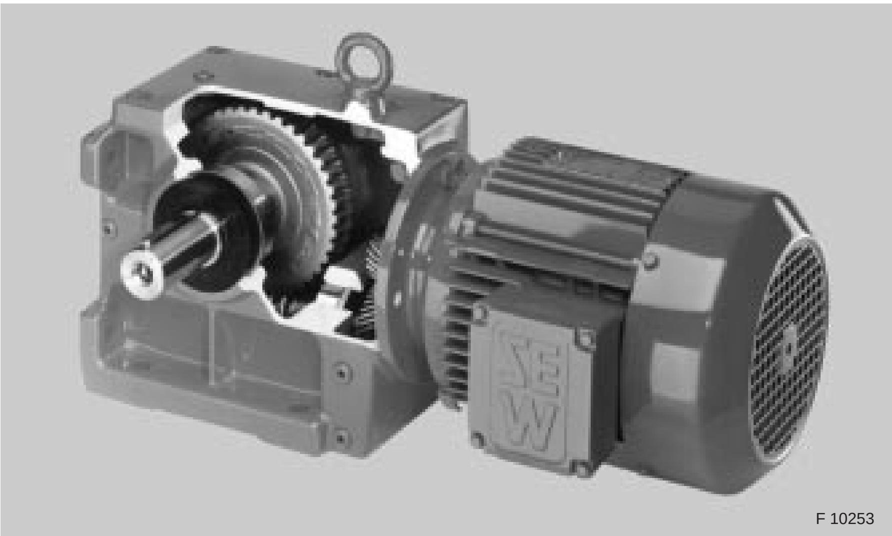


Fig. 45: Two-stage spur/worm gear motor, oil lubricated, SEW-EURODRIVE GmbH & Co, D-76646 Bruchsal, Germany

7 Lubricants for stationary gear systems

7.1 Tasks, requirements

Lubricants for gear systems have the following general tasks:

- transfer forces
- minimize wear
- reduce friction

Fluid lubricants also have to

- dissipate heat

  and
- remove abrasive particles

Depending on the type of gear and the operating conditions gear lubricants have to meet various requirements, some of them even conflicting. There is no universal gear lubricant which would meet all requirements. Instead there are different types of gear lubricants, such as gear oil, greases and adhesive lubricants, whose properties have to suit the individual operating conditions.

Gear oils may have to meet the following requirements:

- excellent resistance to ageing and oxidation
- low foaming tendency
- good air separation behaviour
- good load carrying capacity
- neutrality towards the materials involved (ferrous and nonferrous metals, seals, paints)
- suitability for high and/or low temperatures
- good viscosity-temperature behaviour

Gear greases, in contrast, may be required to ensure

- good adhesion
- low oil separation
- low starting torques
- compatibility with synthetic materials
- noise damping

Adhesive lubricants, e.g. for the lubrication of large open gear drives, have to meet the following requirements:

- excellent adhesion
- optimum separating and lubricating capacity under mixed friction conditions
- good emergency running properties in case of starved lubrication
- good pumping characteristics in automatic spraying equipment

The natural properties of the starting materials, mineral or synthetic oils or a mixture of both, usually are not sufficient to meet all the requirements listed above. The base oils are therefore equipped with additives improving their natural properties or imparting new properties.

The type and quantity of additives depends on the intended use and the type of lubricant. In case of gear greases not only the additives but also the thickening agents have an impact on the lubricant's properties.

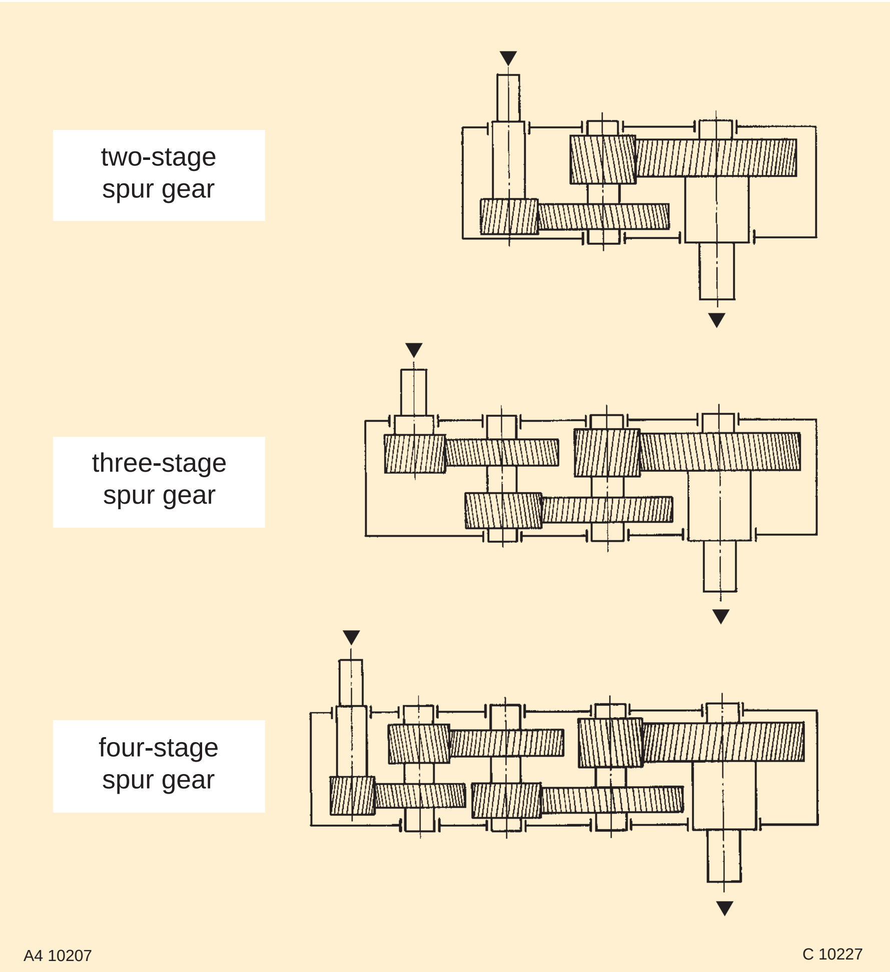


Fig. 46: Schematic drawing of multi-stage spur gears

7.2 Types of gear lubricants

7.2.1 Gear oil with a hydrocarbon base oil

Mineral oil lubricants are widely used in closed industrial gears.

Paraffin base solvent raffinates are often used as the standard base oil for the various lubricant types. These raffinates are called *plain* mineral oils or simply "**lubricating oils**".

Plain mineral oils (in acc. with DIN 51 517, Pt 1, lubricating oils "C") are no longer important for the lubrication of gear systems. They are only used if the requirements in terms of ageing resistance, corrosion and wear protection are very low.
Plain gear oils were largely replaced by *additive treated* oils, i.e. mineral oils containing oil soluble additives enhancing their properties and/or imparting new ones.

Depending on the additive types, a distinction is made between the following gear oils:

***CL lubricating oils** DIN 51 517*

These are mineral oils containing additives to improve corrosion protection and ageing resistance ("L"). In accordance with the above standard they are recommended if corrosion may occur, for example due to the impact of water, or if oils of the "C" type (plain mineral oils) would fail to achieve a sufficient service life at increased temperatures.

CL lubricating oils do not contain antiwear additives and are therefore mainly used in older gears subject to low loads, with a peripheral speed below 30 m/s, which are equipped with hardened and tempered gear wheels and are particularly resistant to fatigue.

Just like type "C" oils, lubricating oils type "CL" are not very important for gear lubrication.

***CLP lubricating oils** DIN 51 517*

These are mineral oils containing additives to improve corrosion protection and ageing resistance ("L") plus additives to minimize wear under mixed friction conditions ("P").

They are recommended if improved wear protection is required because the friction point (meshing zone) is subject to increased load and/or if tooth flank damage (scuffing) should be prevented in case of overloading. This type of oil is frequently used in industrial gear systems.

The pertinent antiwear additives are classified in two categories: additives improving lubricity and EP additives. Additives improving lubricity are also called "polar additives". They enhance the base oil’s wetting and adhesion properties by means of solid organic (animal or vegetable) or synthetic particles which separate the contact points by forming an adhesive, semi-solid, pressure resistant film, thus reducing friction and wear.

EP (**e**xtreme **p**ressure) additives are used for increased wear protection. They mainly consist of organic phosphorous and sulfur compounds having a chemical impact on the lubricant.

When subject to increased temperatures these compounds form a protective layer on the tooth flanks which reduces metal contacts between the flank surfaces, prevents welding and scoring and minimizes friction.

A combination of EP and lubricity enhancing additives is often used to make a lubricating oil suitable for many types of gears including gears with a high sliding percentage, such as worm gears. CLP lubricating oils are not suitable for bevel gears with offset axis (hypoid gears) unless they contain specific EP additives. Hypoid gears are usually lubricated with special oils with enhanced EP additives, so-called "HYP" oils (DIN 51 502).

DIN 51 517, Pt. 3 specifies the minimum requirements for lubricating oils containing antiwear additives. It stipulates, among others, a scuffing load capacity of 12 in the mechanical test on the FZG test rig in acc. with DIN 51 534, Pt. 2 (FZG = Forschungsstelle für Zahnrad- und Getriebebau, TU München – Technical Institute for the Study of Gears and Drive Systems of the Munich Technical University).

Multi-purpose oils

Such oils are not only suitable for the lubrication of gear systems but also for other machine components (clutches, backstops, etc.). They are often used as a working medium transferring power in turbines, electric current transformers and hydraulic systems.

Multi-purpose oils are used in machine tools equipped with only one circulation system for gear systems and hydraulic units, or for the lubrication and feedback control of steam and gas turbines while simultaneously supplying connected machines and gear systems.

They are also suitable for product streamlining purposes if, for example, one lubricating oil is to be used in various components in a larger system, provided it meets all the different quality and viscosity requirements.

Mineral multi-purpose oils include:

Lubricating and control oils L-TD
DIN 51 515

Turbine oils with a mineral base oil plus additives improving corrosion protection and ageing resistance. They are used for the lubrication and/or feedback control of steam turbines, stationary gas turbines, machines operated electrically or by steam turbines, e.g. generators, compressors, pumps and gear systems.

Hydraulic oils HL
DIN 51 524, Pt. 1

Pressure fluids based on mineral oil plus additives enhancing corrosion protection and ageing resistance. They are used to simultaneously supply hydraulic units and gear systems.

Hydraulic oils HLP
DIN 51 524, Pt. 2

Pressure fluids based on mineral oil plus additives improving corrosion protection and ageing resistance and reducing fretting corrosion under mixed friction conditions. They are used to simultaneously supply hydraulic units and gear systems subject to high wear.

Fig. 47: Shaft-mounted gear motor with three-stage spur gears, make Getriebebau Nord, D-22941 Bargteheide, Germany

7.2.2 Gear oils with a synthetic base oil

Synthetic gear oils are used whenever mineral gear oils have reached their performance limit and can no longer meet the requirements, e.g. at very low or high temperatures, extremely high loads, extraordinary ambient conditions, or if they fail to meet special requirements, e.g. in terms of flammability.

Even though many properties of mineral oils can be improved by means of additives, it is not possible to exert an unlimited influence on all their properties. This applies especially to material properties depending on the chemical structure, such as

- thermal resistance
- low temperature properties (fluidity, pour point)
- flash point
- evaporation losses

Synthetic oils provide a number of advantages. However, they do not necessarily outperform mineral oils in all respects and may even result in some drawbacks despite their advantages.

The **advantages** of synthetic lubricating oils (depending on the base oil) include:

- improved thermal and oxidation resistance
- improved viscosity-temperature behaviour, higher viscosity index
- improved low temperature properties
- lower evaporation losses
- reduced flammability
- improved lubricity
- lower tendency to form residues
- improved resistance to ambient media

Possible **disadvantages** include:

- higher price
- reaction in the presence of water (hydrolysis, corrosion)
- material compatibility problems (paints, elastomers, certain metals)
- limited miscibility with mineral oils

Application related advantages usually prevail, so that synthetic lubricants will be increasingly used for gear lubrication, especially under critical operating conditions.

Synthetic lubricants based on
synthetic hydrocarbon oils (SHCs),
polyglycols (PGs) and
ester oils (Es)
have proven particularly efficient in gear systems.

Lubricating oils based on synthetic hydrocarbon oils

They are similar to mineral hydrocarbons in their chemical structure.

They are equal to mineral oils as far as their compatibility with sealing materials, disposal, reprocessing and miscibility with mineral oils are concerned. Their main advantage is their excellent low temperature behaviour.

It is possible to manufacture food grade lubricants for the food processing and pharmaceutical industries with SHC base oils and special additives.

Lubricating oils based on polyglycols

These lubricants ensure especially low friction coefficients, which makes them suitable for gears with a high sliding percentage (worm and hypoid gears).

Containing appropriate additives, they have an excellent antiwear effect, for example in steel/bronze worm gears, and a very good pressure absorption capacity.

Polyglycol oils may have a negative impact on sealing materials and may dissolve paints. At operating temperatures above 100 °C only seals made of fluorinated rubber or PTFE are resistant, below that also seals made of NBR are resistant to PG oils.

This information is only a guideline. Before using PG oils in series applications it is important to test compatibility with paints, seals and inspection glass materials.

Miscibility with mineral oils is very limited; mixtures should therefore be avoided.

Polyglycols are neutral towards ferrous metals and almost all nonferrous metals. In the case of friction pairings with one component consisting of aluminum or aluminum alloys (e.g. rolling bearing cages containing aluminum) there may be increased wear under dynamic load (sliding movement and high load). In such cases we recommend carrying out compatibility tests.

Lubricating oils based on ester oils

Ester oils are the result of a reaction of acids and alcohols with water being split off. They exist in numerous structures, all of them having an impact on the chemical and physical properties of lubricants.

In the past these lubricating oils were mainly used in aviation technology for the lubrication of aircraft engines and turbines as well as gear systems in pumps, starters, etc.

Ester oils have a high thermal resistance and excellent low temperature behavior. In industrial applications *rapidly biodegradable* ester oils will gain importance because it seems possible to achieve the same efficiency as with polyglycol oils by selecting an appropriate ester base oil.

Application related advantages of synthetic lubricating oils

The following application related advantages result from the improved properties of synthetic lubricating oils as compared to mineral oils:

- improved efficiency due to reduced tooth related friction losses
- lower gearing losses due to reduced friction, thus less energy required
- oil change intervals 3 to 5 times as long as compared to mineral oils operating under the same temperature
- reduced operating temperatures under full load, thus increased component life; cooling devices may not be required.

Reduction of gearing losses and efficiency improvement

Owing to their special molecular structure, synthetic lubricating oils based on polyalphaolefins and polyglycols ensure that tooth related friction is considerably lower than with mineral oils. It may be up to 30 % lower than if a regular mineral gear oil with EP additives were used.

As the friction coefficient of synthetic oils is lower, tooth related friction is reduced to a large extent, thus increasing the gear's efficiency.

The efficiency of gears with a high sliding percentage, worm and hypoid gears, may be increased up to 15 % if a synthetic oil is used instead of a mineral oil. Even in case of spur and bevel gears, which already have a high degree of efficiency, it is possible to achieve an increase of up to 1 % by using a synthetic gear oil. This may not seem very much at first sight, but it may result in considerable cost savings depending on the nominal output of a gear or if several gears are concerned.

Table 10 shows to what extent synthetic gear oils can reduce losses, especially in gear systems with a high share of load dependent losses.

The data were gathered empirically in many tests conducted on Klüber's gear test rig. These results have been confirmed by gear manufacturers and operators.

Advantages of synthetic gear oils based on reduced friction

Increased gear efficiency

- Smaller gears with smaller motors sufficient for equivalent output
- Higher output with the same power input

Reduced oil temperatures

- Extension of the oil's life (5 times longer than mineral oils)
- Extended component life
- Cooling units may no longer be required

Reduced energy consumption

- Reduced costs for lost electric current due to lower total losses; 30 % and more in the case of worm gears
- Costs for electric power reduced up to 10 % due to improved efficiency

Type of gear / Effect	Worm gears, Hypoid gears	Spur gears Bevel gears with axis not offset
Reduction of total losses	30 % and more	20 % and more
Improved efficiency	15 % and more	up to 1 %
Reduction of operating (steady-state) temperature	20 °C and more	up to 12 °C

Table 10: Potential reduction of gearing losses and improvement of efficiency if a synthetic gear oil is used instead of a mineral oil.

The following test results and calculation examples illustrate the advantages of synthetic gear oils over mineral oils.

Improved efficiency and reduced wear when using synthetic oils

Synthetic gear oils can reduce power losses and operating temperatures in almost any gear type and thus increase gear efficiency. This applies particularly to gears with high or predominant sliding percentages (hypoid and worm gears) if oils on a polyglycol basis are used.

Previously it was assumed that the efficiency of spur and bevel gears could be increased up to 1 % and that of worm gears up to 10 % by using synthetic gear oils.

Tests comparing high grade CLP oils and modern synthetic gear oils, however, prove that the efficiency of spur, bevel and worm gears can be increased far beyond the values stated above.

Such tests were carried out with Klüber gear oils on our in-house test rig for worm gears.

The results show clearly that synthetic oils make the gears more efficient than mineral oils.

Klübersynth GH 6, a polyglycol lubricating oil, resulted in the highest degree of efficiency: 18 % more than with **Klüberoil GEM 1**, a high performance CLP oil. This proves once again that PG oils are particularly suitable for the lubrication of worm gears.

Klübersynth GEM 4 and **Klüberoil 4 UH 1**, two SHC gear oils, also made the test gears 8 to 9 % more efficient. The performance of Klüberoil 4 UH 1, which is authorized as a food grade lubricant in accordance with USDA-H1, is excellent. Food grade lubricants are often thought to be inferior to "*normal*" lubricants as far as their performance is concerned, an opinion which the test results definitely disprove.

The results are also confirmed by very positive practical experience in all sectors of the food processing and pharmaceutical industries.

Synthetic base oils already have an excellent wear protection behaviour which is even enhanced by appropriate antiwear additives contained in Klüber lubricants.

The diagrams in *Fig. 48* show that the antiwear behavior of synthetic lubricating oils is superior to that of mineral gear oils.

Wear was particularly low when **Klübersynth GH 6** was used.

All this is convincing proof that PG base gear oils containing specific additives are most suitable for the lubrication of gears with a high sliding percentage, especially worm gears.

Test oils

Klüberoil GEM 1 - 460 (mineral oil)

Klüberoil 4 UH1 - 460 (SHC oil)
Food grade lubricant

Klübersynth GEM 4 - 460 (SHC oil)

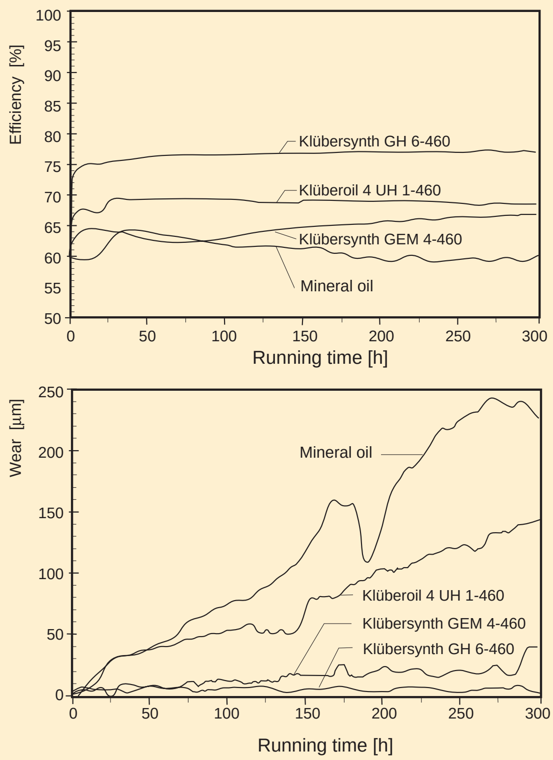
Klübersynth GH 6 - 460 (PG oil)

Test gear

Standard worm gear
i = 1 : 39 center distance a = 63 mm
Worm shaft: steel 16 Mn Cr S5
Worm wheel: GZ - Cu Sn12 Ni

Test conditions

Input speed:	350 rpm
Output speed:	300 Nm
Test duration:	300 h



a)

Test results: **Efficiency**

Klüberoil GEM 1 - 460 ...... 60 %
Klüberoil 4 UH 1 - 460 ...... 69 %
Klüberoil GEM 4 - 460 ...... 68 %
Klübersynth GH 6 - 460 ... **78 %**

b)

Test results: **Wear**

Klüberoil GEM 1 - 460 ... 225 µm
Klüberoil 4 UH 1 - 460 ... 142 µm
Klüberoil GEM 4 - 460 ..... 40 µm
Klübersynth GH 6 - 460 .... **2 µm**

C 10228 a+b

Fig. 48: Test diagrams a) Efficiency b) Wear

Extended oil change intervals using synthetic oils

Synthetic oils have a much better resistance to ageing and high temperatures and a longer service life than mineral oils.

Depending on the base oil (SHC or PG), the oil change intervals are 3 to 5 times longer at the same oil temperature.

Approximate oil change intervals at an oil temperature of 80 °C:

Mineral oil: 5 000 operating hours

SHC oil: 15 000 op.h. (extension factor 3)

PG oil: 25 000 op.h. (extension factor 5)

Synthetic oils have a lower gear related friction coefficient than mineral oils (up to 30 % lower) and a more favorable viscosity-temperature behaviour. This generally permits the use of an oil of a lower viscosity grade and makes it possible to reduce the oil temperature.

The extension factors for oil change intervals of synthetic oils are longer than the values stated above, which refer to an identical oil temperature.

The following comparison of test results illustrates this advantage.

Three high-performance Klüber lubricants were tested in a splash lubricated worm gear mounted on an in-house test rig.

Results and evaluation

The test records (*Fig. 50*) show the following oil sump temperatures after 300 operating hours:

Klüberoil GEM 1-460 ....110 °C
Klübersynth GEM 4-460 .... 90 °C
Klübersynth GH 6-460 .... **75 °C**

Based on these results the diagram in Fig. 49 indicates the following approximate oil change intervals:

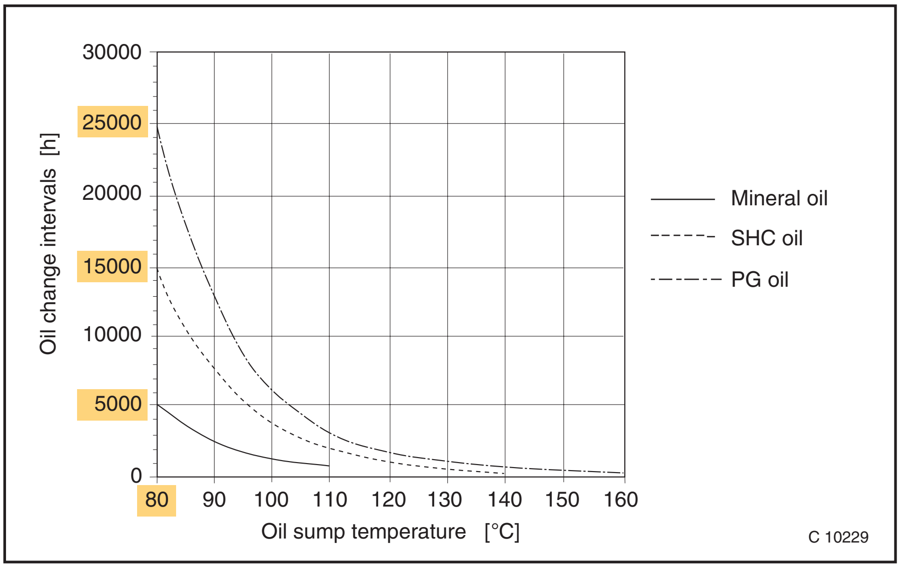


Fig. 49: Approximate oil change intervals in gears with oil sump lubrication. Comparison of synthetic oils Klübersynth GEM 4 (SHC oil) and Klübersynth GH 6 (PG oil) with Klüberoil GEM 1 (mineral oil)

Klüberoil GEM 1-460
800 op. hours

Klübersynth GEM 4-460
7,500 op. hours

Klübersynth GH 6-460
approx. 25,000 op. hours

The life extension factors of synthetic oils as compared to mineral oil are as follows:

GEM 4 (SHC oil) / GEM 1 (min. oil): approx. 9.4

GH 6 (PG oil) / GEM 1 (min. oil): **approx. 31**

These test results indicate quite clearly that a synthetic gear oil reduces the oil sump temperature drastically. It is also obvious that **Klübersynth GH 6**, a polyglycol gear oil, is particularly suitable for the lubrication of worm gears. This oil allows especially long oil change intervals, or even lifetime lubrication.

Klüber gear oils on a PG basis, e.g. SYNTHESO D...EP, are also suitable for the lubrication of hypoid gears which are mainly used in automotive transmissions. Hypoid gears are increasingly used in industrial applications because of their advantages over bevel gears (see section 2.1) and because lubrication is quite simple.

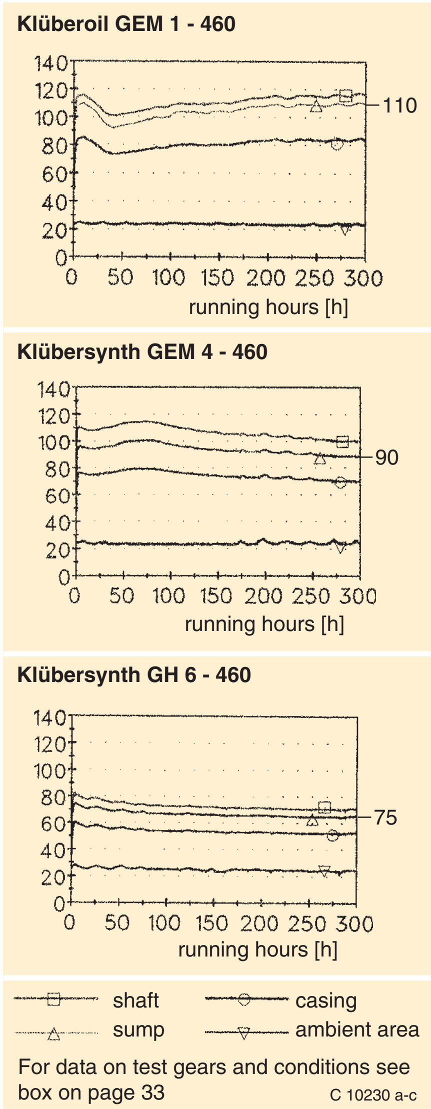


Fig. 50: Gear temperatures

How to save energy costs by using a synthetic lubricating oil, calculation example

A

Worm gear lubricated with a **mineral oil**

Power input P_a = 100 kW

Power output P_b = 70 kW

Power loss $P_v = P_a - P_b$ = 100 - 70 = 30 kW

B

Worm gear lubricated with a **synthetic oil**

Power input P_a = 100 kW

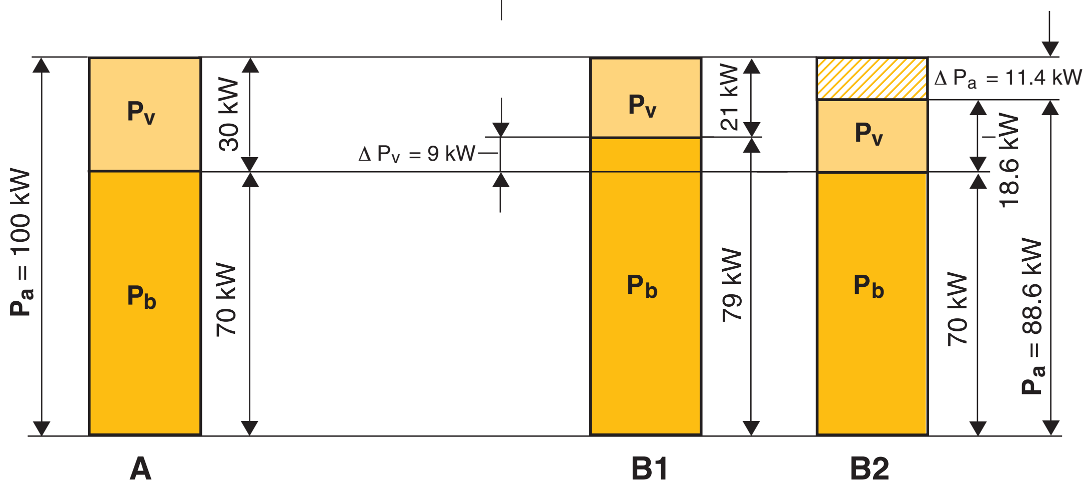
Power output P_b = 79 kW

Power loss $P_v = P_a - P_b$ = 100 - 79 = 21 kW

$$\text{Efficiency } \eta = \left(1 - \frac{P_v}{P_a}\right) \cdot 100\ [\%]$$

$$\eta_A = \left(1 - \frac{30}{100}\right) \cdot 100 = \mathbf{70\ \%}$$

$$\eta_B = \left(1 - \frac{21}{100}\right) \cdot 100 = \mathbf{79\ \%}$$



With a synthetic oil the reduction of friction results in power losses P_v of only 21 kW. In consequence:

Reduced overall power loss: $\Delta P_v = \left(1 - \frac{P_{v\,B}}{P_{v\,A}}\right) \cdot 100\ [\%] = \left(1 - \frac{21}{30}\right) \cdot 100 = \mathbf{30\ \%}$

Increased efficiency: $\Delta\eta = \eta_B - \eta_A = 79 - 70 = \mathbf{9\ \%}$

Advantages of a synthetic gear oil based on increased gear efficiency:

- **Higher power output P_b** – increase from 70 to 79 kW with the same input P_a = 100 kW.
  = increased efficiency of 13 % (see chart B1), or
- **Reduced power input P_a** with the predefined output of P_b = 70 kW

$P_a = \frac{P_b}{\eta} = \frac{70}{0.79} = 88.6$ kW $\Longrightarrow$ Reduction of power input of **11.4 kW** (see chart B2).

The reduced power input saves the following energy costs:

Assumption: 5 worm gears, 2-shift operation (4000 operating hours per year), electric energy cost DEM 0.23/kWh, reduced power input: 5 x 11.4 = 57 kW

Calculation: 57 kW x 4000 hrs x DEM 0.23/kWh = **DEM 52,440 less per year**

High temperature lubrication with synthetic oils

Synthetic gear oils are much more resistant to high temperatures and ageing than mineral gear oils, which makes it possible to increase the thermal load limit of gear systems and their overall performance.

The term "thermal load limit" refers to a gear's power output at a temperature at which even the least stressed component and the lubricant will not suffer any damage. Many gear systems are still lubricated with a mineral oil, a fact which limits the maximum temperature and thus a gear's output. *Table 11* shows that mineral oil has one of the lowest temperature limits of all gear components.

The temperature limit of synthetic gear oils is much higher than that of mineral oils. The gear temperature can therefore be increased by 20 to 30 % while still achieving the same oil life. The thermal load limit and the transferrable power are increased accordingly.

This, however, is only feasible if the temperature limit of all other elements and materials used in a gear system is not below the limit of the pertinent synthetic oil.

If a mineral oil is used, the mechanical capacity (nominal output) of splash lubricated standard gears is generally higher than its thermal load limit. This difference indicates to what extent the gear's performance could be enhanced by increasing the thermal load limit.

The thermal limit can be increased by installing additional cooling units (fans, internal coolers, etc.) or by applying a synthetic, high temperature lubricating oil (permanent oil sump temperatures between 100 and 150 °C), which in most cases is the less expensive solution, especially for gears that are already operating.

The following practice related example shows how a synthetic, high temperature oil can increase a gear's thermal capacity while ensuring an extended lubricant life.

The synthetic oils of the **Klübersynth GEM 4** and **Klübersynth GH 6** series are particularly suitable for high temperature lubrication of gear systems.

Example

A standard single-stage splash lubricated spur gear is required to drive an axial fan without any additional cooling units.

Operating conditions:

Electric motor	P_a = 100 kW
Driven machine (axial fan)	P_e = 96 kW
Motor speed	1500 rpm
Fan speed	425 rpm
Nominal transmission ratio	i_N = 3.55
Operating time	24 h / day
Starts per hour	5
Ambient temperature	20 °C

Component/Lubricant	Temperature limit [°C]
Gear wheels	
case-hardened steel	180 / 300
tempered steel	> 200
bronze	> 200
Rolling bearings	
standard bearings	120
heat stabilized bearings	300
Plain bearings	
standard bearings	90
heat stabilized bearings	150
Shaft seals	
nitrile rubber	100
polyacrylate rubber	125
fluorinated rubber	150
Lubricating oil (oil bath)	
mineral oil	100
synthetic oil	160

Table 11: Temperature limit of components and lubricants used in gears

1. *Determination of the gear's nominal capacity*

$P_N = P_e \times c$

(c = machine output correction factor, to be determined as specified by the gear manufacturer)

$P_N = 96 \times 1.44 =$ **138 kW**

2. *Determination of the gear's thermal capacity during operation without additional cooling*

$P_{th} = P_e \times c_w$

(c_w = thermal factor depending on the ambient temperature and the operating time per hour, to be determined as specified by the gear manufacturer)

$P_{th} = 96 \times 1.0 =$ **96 kW**

3. *Selection of a suitable gear from a standard brochure*

Based to the values P_N und P_{th} calculated above, a gear is required that has a nominal capacity of 295 kW and, without additional cooling, a thermal load limit of 105 kW when lubricated with a CLP gear oil which would be sufficient in view of the required thermal limit of 96 kW *(fig. 51)*.

However, the difference between the nominal capacity of the standard gear (295 kW) and the required capacity (138 kW) is too big, i.e. the gear is too large for the intended use.

A gear of the next smaller size would be sufficient as far as the nominal output (185 kW) is concerned, but its thermal capacity of 86 kW without additional cooling is too low.

As additional cooling will not be provided, this problem can only be solved by means of a synthetic, high temperature lubricant with a high thermal capacity (permanent oil sump temperature between 100 and 150 °C). The thermal capacity of standard gears with a mineral oil lubricant is generally limited to 100 °C (thermal limit of the oil) and cannot be increased without additional cooling.

4. *Determination of the anticipated oil sump temperature and lubricant life when increasing the thermal load limit from 86 kW to 96 kW by using a synthetic oil*

4.1 *Anticipated oil sump temperature*

The possible increase of the thermal limit and the transferrable power can be deducted from the thermal capacity equation $\dot{Q}_v$ (see section 6.3.1, p. 25).

$$\dot{Q}_v = \alpha \cdot A \cdot (t_{oil} - t_L)$$

whereby:

α = thermal transfer coefficient $[W/m^2 \cdot K]$

A = housing surface $[m^2]$

t_{oil} = oil bath temperature [°C]

t_L = ambient air temperature [°C]

To achieve the required thermal limit of 96 kW it is necessary to increase the gear's thermal limit (86 kW) by 11.62 %, i.e. the thermal capacity ($\dot{Q}_v$) must also be increased by the same amount.

With an oil bath temperature (t_{oil}) of 100 °C and an ambient air temperature (t_L) of 20 °C the gear's thermal capacity is as follows:

$$\dot{Q}_{v1} = 23 \cdot 0.85 \cdot (100 - 20) \left[\frac{W \cdot m^2 \cdot °C}{m^2 \cdot K}\right]$$

$$= 1564 \text{ W}$$

$\dot{Q}_{v1}$ increased by 11.62 %

$\dot{Q}_{v2}$ = 1746 W

By rearranging the $\dot{Q}_v$ equation it is now possible to calculate the oil bath temperature expected after increasing the thermal limit from 86 to 96 kW.

$$t_{oil} = \frac{\dot{Q}_{v2}}{(\alpha \cdot A)} + t_L = \frac{1746}{19.55} + 20$$

$$= \text{approx. } \mathbf{110\ °C}$$

This calculation did not take into account that synthetic lubricating oils reduce the friction losses occurring in gear systems up to 25 %, which means that the permanent oil bath temperature is expected to remain under 110 °C during operation.

4.2 *Determination of the lubricant's service life*

The diagram in Fig. 49, page 34, shows that the service life of a synthetic gear oil at an oil bath temperature of 100 °C is twice (Klübersynth GEM 4) or approx. three times (Klübersynth GH 6) as long as the life of a mineral oil at an oil sump temperature of 100 °C.

Assuming an identical service life, the oil sump temperature could be increased to approx. 123 °C when Klübersynth GH 6 is used instead of a mineral oil.

It would therefore be possible to increase the thermal limit of the selected gear by approx. 29 % from 86 kW to about 111 kW.

Ratio i_N	Speeds (rpm) n_1	n_2	Gear unit size 014	016	018	020	022	025	028	032	036
			Nom. power rating P_N (kW)								
1.12	1500	1340	280	410	500	730	1000 *	1400 *	* Additional injection lubrication required		
	1000	890	210	300	400	500	680	1000			
3.55	1500	425	125	185	295	390	510	700	920	1250	1700
	1000	280	90	135	200	280	340	480	620	860	1150
	750	210	70	110	150	220	270	380	550	700	920
4	1500	375	110	160	230	320	500	650	870	1230	1550
	1000	250	75	110	160	220	330	450	600	850	1100
	750	187	60	85	115	175	260	350	490	660	830
4.5	1500	335	85	140	200	270	380	560	820	1100	1400
	1000	220	60	95	140	185	260	380	580	800	1020
	750	166	45	75	105	150	200	290	460	620	780
5	1500	300	75	125	160	220	370	480	680	1000	1300
	1000	200	50	86	106	145	260	340	480	720	880
	750	150	40	65	85	110	195	260	375	550	700
5.6	1500	270	70	105	140	200	300	420	580	900	1150
	1000	180	45	75	90	140	205	290	410	610	800
	750	134	35	55	75	105	160	220	310	460	620

Thermal limit capacities P_{th} (kW)

Gear unit size		014	016	018	020	022	025	028	032	036
without cooling	P_{th1}	63	86	105	129	162	201	251	329	411
with fan	P_{th2}	100	131	167	206	259	321	402	536	657
with built-in cooler	P_{th3}	162	208	245	288	366	437	516	737	879
with fan and built-in cooler	P_{th4}	199	253	307	365	436	557	667	934	1125

Cooling required

C 10231

Fig. 51: Excerpt from a gear brochure, VOITH TURBO

Notes on high temperature lubrication

High-temperature lubrication, which is possible with synthetic gear oils, is only feasible if the temperature limits of all other gear components, gear wheels, bearings, seals, filters inserts, internal coatings, are not exceeded. It may be required to install new seals or replace other heat sensitive components.

If standard gears are converted from a mineral to a synthetic oil, it is indispensable to coordinate this process with the gear manufacturer and the lubricant supplier.

Advantages of high temperature lubrication with synthetic gear oils

- The thermal limit (P_{th})of the gear system can be increased by up to 50 %.
- Due to the increased thermal limit (P_{th}) a gear with a smaller nominal performance (P_N) can be selected from a gear brochure to achieve the same output (see example calculation above), provided $P_{th} < P_{mech}$.
- In many cases additional cooling units are not required.
- Future gears can be designed with an increased performance (smaller gears with a higher power transfer).

Synthetic oils help to save maintenance and disposal costs

As compared to mineral oils, the oil change intervals of synthetic oils are usually five times longer under the same thermal conditions.

Despite the fact that the purchase price and the costs of disposal of a synthetic oil are higher than that of a mineral oil, the extended oil change intervals save purchase and disposal costs when taking into account the gear's service life.

Table 12 shows a comparison of the costs for mineral and synthetic oils which proves that it is possible to save a lot of money by using a synthetic oil while at the same time conserving resources and reducing environmental impacts by reducing lubricant consumption.

	Mineral oil (Klüberoil GEM1 - 220)	Synthetic oil (Klübersynth GH6 - 220)
Filling quantity (l)	70	70
Oil change interval	annually	every 5 years
Gear life (years)	15	15
Amount of lubricant required (l)	1 050	210
The above data is the basis of the following calculation:		
Purchase price / l *	DEM 7.30	DEM 11.90
Total cost of lubricant	DEM 7,665.00	DEM 2,499.00
Disposal costs	DEM 157.50 (DEM 15.00 /100 l)	DEM 210.00 (DEM 100.00 /100 l)
Maintenance costs (2 hours à DEM 100.00 per cycle)	DEM 2,800.00 (14 x DEM 200.00)	DEM 400.00 (2 x DEM 200.00)
Overall costs	DEM 10,622.50	DEM 3,109.00
Cost savings with a synthetic oil	**DEM 7,513.00**	
* Price per liter if a 200 l drum is purchased, as of May 1994		

Table 12: Comparison of costs incurred when using a mineral and a synthetic oil

7.2.3 Gear greases

Lubricating greases are used in gear systems instead of lubricating oils if they are more suitable for technical or economical reasons, or if it is not possible to use a lubricating oil, for example in open gears, closed gears that are not oiltight, for safety reasons (leakages) in gears that are difficult to access, and wherever it is not possible to perform maintenance work or exchange the lubricant.

Fluid greases of NLGI 00 and 000 (DIN 51 818) are typically used in gear systems, mainly for splash lubrication in gear motors.

Gear greases are widely used in small and miniature gears, for example in office machines, domestic appliances, DIY machines, power tools, automobiles, model toys, etc.

Greases with a higher consistency (NLGI 0, 1 and 2) are mainly used in such gears because quite often they are not oiltight or the tooth flanks are only lubricated once.

Gear greases are selected in accordance with DIN 51 509, Pt. 2. The minimum requirements these greases have to fulfill are listed in DIN 51 826, "lubricating greases type G".

Gear greases, like lubricating greases in general, consist of a mineral or synthetic base oil or a mixture of both plus a thickening agent.

The most common thickening agents include metal soaps such as aluminum, barium, calcium and lithium soaps, but also non-soap thickeners such as gels, polyurea and bentonite. *Fig. 70* on page 63 shows the general structure of a lubricating grease.

The oils incorporated in the thickeners are similar to those in gear oils: raffinates, hydrocracking oils, synthetic hydrocarbons or polyglycols. Depending on the type and the consistency of the grease, the thickener share is between 5 and 25 %, i.e. the oil share prevails.

Synthetic base gear greases have the same advantages as synthetic gear oils: reduction of friction losses, good low and high temperature behavior, high resistance to ageing, suitability for lifetime lubrication.

Additives are incorporated in the greases in order to obtain specific properties such as resistance to ageing, corrosion protection, increased pressure absorption capacity, wear protection, etc. Their share is up to 5 %.

The theory that soaps contained in lubricating greases are nothing but a carrier medium and have no lubricating effect has been disproved. The base oil and the thickener both have a lubricating function.

7.2.4 Adhesive lubricants

This type of lubricant is used on large open gears, e.g. the drives of rotary kilns, cement mills, lifting cylinders, cranes, construction machinery, etc. They are called "gear greases type OG" in accordance with DIN 51 509, Pt. 2, and are divided into two categories: *adhesive lubricants without bitumen* and *adhesive lubricants containing bitumen*.

Klüber only manufactures high quality adhesive lubricants free from bitumen and solvents which are suitable for many applications.

These lubricants are available for all types of lubrication and application methods. Modern adhesive *fluids* are used for splash and circulation lubrication, and grease type adhesive spray lubricants of NLGI 0 and 00 are suitable for application through automatic spray systems.

Modern adhesive lubricants contain EP additives to ensure optimum lubrication under mixed friction conditions (which are often found in large open drives), adhesion improvers to optimize adhesion on the tooth flanks and solid lubricant particles to increase the load carrying capacity and improve emergency lubrication properties.

See also our brochure "Lubrication of large gear drives", 9.2 e.

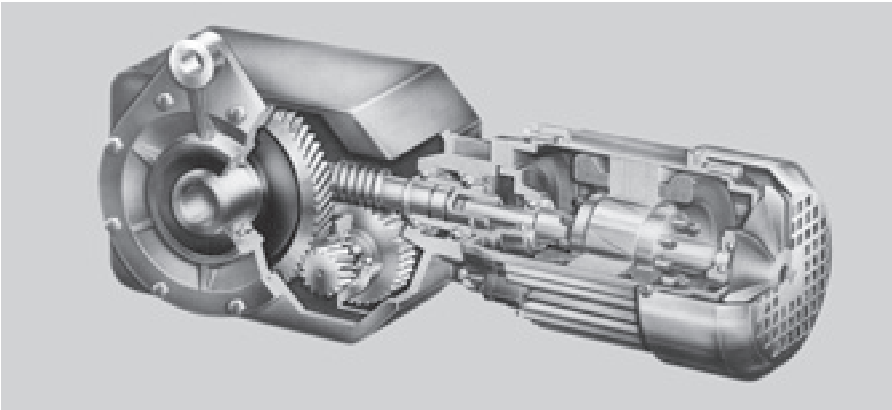


Fig. 52: Combined worm and spur gear unit with IEC-Motor, Rhein-Getriebe, Meerbusch, Germany

7.3 Standard requirements for gear lubricants

Gear lubricants have to fulfill a number of standard requirements in terms of application, composition, and lubrication properties which are checked for compliance with the pertinent standard.

The standards' purpose is to make lubricants of the same type interchangeable.

DIN standards only indicate the minimum requirements for gear lubricants. Modern products, especially synthetic gear oils and greases and most of the gear lubricants offered by Klüber, exceed these requirements by far.

Similar to new technologies, new types of lubricants are only included in standards if they have been used successfully for some time and have gained acceptance on a wide basis.

State of the art lubrication technology proves that lubricants have kept pace with the general technological development and are well ahead of standardization.

Table 13 lists all gear lubricants meeting DIN requirements.

Standardized lubricants	Requirements	Type of lubricant	Application	Klüber lubricants
CLP lubricating oils	DIN 51 517, Pt 3	Mineral oils with additives enhancing resistance to corrosion and ageing, and reducing wear under mixed friction conditions. FZG scuffing load stage (A/8.3/90): at least 12	Gears with high flank loads and/or a high sliding percentage; permanent oil bath temperature up to 100 °C	**Klüberoil® GEM 1**
L - TD lubricating and control oils	DIN 51 515, Pt 1	Mineral oil base turbine oils with additives enhancing resistance against corrosion and ageing	Gears in steam and gas turbines	
HL hydraulic oils	DIN 51 524, Pt 1	Pressure fluids made of mineral oils containing additives to enhance resistance against corrosion and ageing	Combined application in hydraulic systems and gears	
HLP hydraulic oils	DIN 51 524, Pt 2	Pressure fluids made of mineral oils containing additives to enhance resistance against corrosion and ageing and to reduce wear. FZG scuffing load stage (A/8.3/90): at least 10	Combined application in hydraulic systems and gears subject to a high degree of wear	**LAMORA® HLP**
Lubricating greases type G	DIN 51 526	Fluid to very soft greases of NLGI grade 000 to 1, based on mineral oil and/or synthetic oil plus thickener	Splash lubrication of closed gears, e.g. gear motors, drum motors, actuators	See product survey p. 75-77: Greases for industrial gears p. 78-82: Greases for small gears

Table 13: Standard requirements for common gear greases

7.4 Properties of gear oils

Gear oil properties are determined by the base oil (mixture) and the additives and are described in standards, classifications and specifications.

Individual properties are divided into selection parameters and quality parameters (see *table 14*).

Selection parameters include those properties which are important for the individual case of application but do not provide any information about the oil's quality or operational properties (e.g. viscosity).

Quality parameters, in contrast, are indicative of the service properties of a particular product.

The distinction between primary and secondary properties does not indicate a qualitative graduation.

Instead, primary properties describe the gear related requirements such as antiwear and antiscuffing behaviour.

Secondary properties, on the other hand, are important from a general point of view. For example, it is important that the base oil and additives contained in a specific gear oil do not attack the sealing material.

7.4.1 Viscosity

Gear oils change their flow behaviour depending on temperature and pressure. Their viscosity[1)] decreases with a rising temperature and increases with a rising pressure.

This crucial behaviour of gear oils is therefore of primary importance when determining the required viscosity and selecting a suitable type of oil (mineral, synthetic).

The impact of an oil's viscosity on gear lubrication can be summarized as follows:

Increased viscosity results in a thicker lubricant film, thus improving antiwear and damping behaviour and, to a certain extent, the oil's scuffing load capacity.

If the **viscosity** is **too high**, increased churning and squeezing losses result in excessive heat, especially at an increased peripheral speed. The operating viscosity will finally decrease.
Decreased viscosity improves the flow behaviour at low temperatures, the air separation properties and idling losses.

If the **viscosity** is **too low**, mixed friction conditions prevail and will result in increased wear.

Viscosity-temperature behaviour

An oil's viscosity-temperature (VT) behaviour describes the fact that viscosity changes as a function of the existing temperature. It increases when the temperature rises.

The degree to which viscosity changes with the temperature varies from oil to oil. It depends on the base oil (mineral or synthetic) and the additives influencing (improving) the oil's VT behaviour.

An oil's VT behaviour is generally depicted on an *Ubbelohde* VT chart. The coordinates were defined in a way to make it possible to describe the VT characteristics of mineral lubricating oils as a straight line.

Synthetic hydrocarbon base oils (SHC) also show a straight VT line, whereas PG (polyglycol) base oils result in a curve.

Fig. 53 on page 42 shows the general viscosity temperature characteristics of gear oils with different base oils.

The flatter the VT line, the less viscosity depends on temperature. The viscosity index (VI) generally describes the temperature-related change of viscosity. It is calculated according to DIN ISO 2909 and is a dimensionless number.

Selection parameters	**Quality parameters** – Primary properties	**Quality parameters** – Secondary properties
Viscosity Service temperature range	Friction behaviour Viscosity/temperature behaviour Antiwear behaviour Antiscuffing behaviour Running-in behaviour High temperature behaviour Low temperature behaviour	Chemical behaviour (corrosion, attack on nonferrous metals) Thermal resistance (oxidation) Foaming properties

Table 14: Gear oil properties

1) Viscosity is the property of a fluid that enables it to resist the laminar shearing forces (deformation) of two adjacent layers (internal friction, shear stress).
DIN 1342, DIN 51 550, DIN ISO 3104

The higher a gear oil's viscosity index, the smaller the change in viscosity depending on temperature, i.e. the wider its service temperature range.

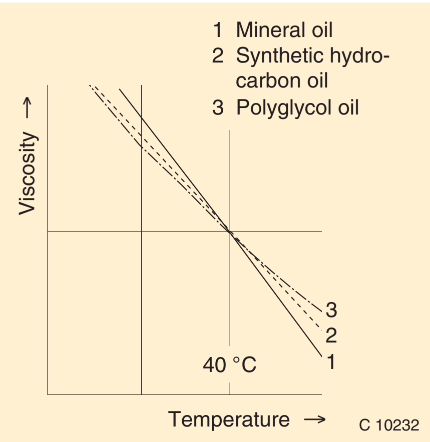


Fig. 53: Viscosity-temperature curves of different types of gear oils

Comparison of VIs:

Gear oils on the basis of

Mineral oil	VI approx. 85 to 100
Synth. hydrocarbons	VI approx. 130 to 160
Polyglycols	VI approx. 150 to 260

In practice a high viscosity index means that a gear will start smoothly under low temperature conditions and have only minimum power losses, or that a load bearing lubricant film is formed at increased temperatures that provides good protection against wear.

This comparison above shows that synthetic gear oils have a clearly higher viscosity index than mineral gear oils. In consequence, they also have a wider service temperature range. It is between approx. -50 and 160 °C, depending on the base oil type.

The service temperature of mineral gear oil ranges between approx. -20 and 100 °C.

Viscosity-pressure behaviour

A lubricating oil's viscosity increases when the pressure rises. Elasto-hydrodynamic processes occurring in the lubrication gap between the intermeshing flanks produce pressures up to 10,000 bar or more. This results in an increase of viscosity that may be some powers of ten higher than the initial viscosity of the oil under atmospheric pressure (*Fig. 54*).

Viscosity measurement

Viscosity is measured by means of a viscometer. Viscometer types include:

- capillary viscometer (kinematic viscosity)
- drop-ball viscometer (dynamic viscosity)
- rotational viscometer (dynamic viscosity)

Capillary and drop-ball viscometers are used for measurements above 0 °C, rotational viscometers below 0 °C.
If the lubricant flows out of the measuring equipment only due to its own gravity, like in a capillary viscometer, it is possible to determine the kinematic viscosity ν using the SI unit [$mm^2 \cdot s^{-1}$].

The gear manufacturers' instructions, the lubricant manufacturers' product information leaflets and technical brochures, and the DIN standards almost exclusively contain information about a lubricants' kinematic viscosity.

If viscosity is measured by means of additional parameters, e.g. dropping ball, pressure, torque, it is possible to determine the dynamic viscosity η using the SI unit [MPa · s].

A lubricating oil's kinematic viscosity ν is obtained by dividing its dynamic viscosity η by its density ρ.

$$\nu = \eta / \rho$$

Viscosity units:

Kinematic viscosity

SI unit: $m^2 \cdot s^{-1}$

Derived SI unit: $mm^2 \cdot s^{-1}$

Conversion factor: $1\ m^2 \cdot s^{-1} = 10^6\ mm^2 \cdot s^{-1}$

Dynamic viscosity

SI unit: $N \cdot s \cdot m^{-2} = Pa \cdot s$ (Pascal second)

Derived SI unit: MPa · s

Conversion factor: $1\ Pa \cdot s = 10^3\ MPa \cdot s$

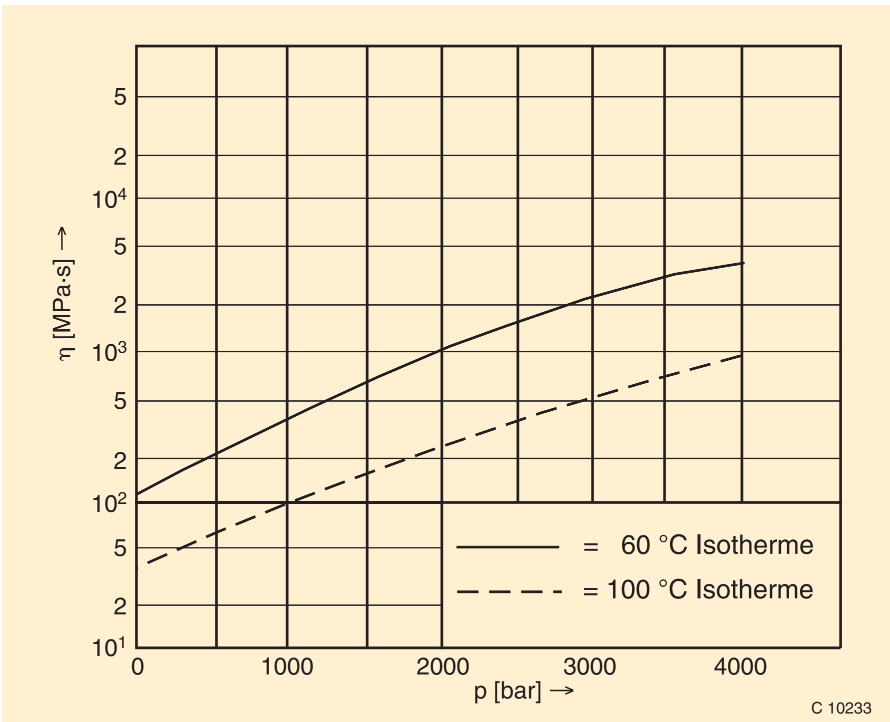


Fig. 54: Pressure-temperature behaviour of a synthetic oil ISO VG 220 at 60 °C and 100 °C

7.4.2 Ageing behaviour

External impacts continuously change an oil's chemical structure, thus causing it to age. Changes mainly occur under high temperatures, when the oil mixes with air, and when it is in contact with metal catalysts (e.g. copper, iron). The speed of the ageing process primarily depends on the oil's structure and the amount and duration of heat to which the oil is subject. The ageing process can be retarded by means of additives.

Contaminants like water, rust or dust also have a negative impact on oil ageing. Ageing is indicated by oil discolorations, increased viscosity, formation of acids enhancing corrosion, as well as residues in the form of lacquer, gum or sludge clogging oil lines and filters.

Ageing also has a negative impact on the oil's demulsifying and antifoaming capacity, its anticorrosion and antiwear properties, and, to a certain extent, its air separation capacity.

When using a mineral oil it must be ensured that the permanent oil bath temperature never exceeds 80 °C because the ageing speed doubles with every temperature increase by 10 K. As compared to mineral oils, synthetic gear oils in similar applications are much more resistant to ageing. Depending on the oil type, oil change intervals can be 5 times as long and service temperatures can be increased (*Fig. 49*).

Even though there are a number of ageing tests, there is no generally accepted test method. One of the methods used to determine the ageing properties of gear oils is laid down in DIN 51 586 (CLP lubricating oils).

7.4.3 Behaviour in the mixed friction regime

Under mixed friction conditions, fluid and dry friction occurs on the intermeshing tooth flanks. In order to minimize the effects of dry friction, scuffing and abrasive wear, it is important that the lubricating oils contain antiwear additives preventing direct metal contact between the tooth flanks (boundary lubrication).

The effects of the additives are described in section 7.2.1, page 29, "CL lubricating oils".

7.4.4 Anti-corrosion properties

Anti-corrosion properties are evaluated from two points of view:

– corrosion protection on steel (rust prevention)

– corrosion protection on copper (compatibility with nonferrous metals)

Rust prevention

If there is water in a gear system, either leakage or condensation water, it will combine with ambient oxygen and lead to rust forming on inadequately protected steel surfaces.

Wear is the result of corrosion caused by rust particles contained in the oil that are returned to the meshing zone and the bearings, where they have an abrasive effect.

Rust also has a negative impact on an oil's ageing resistance and demulsification (water separation) properties, and may result in the formation of sludge.
To enhance their rust prevention properties, gear oils contain polar rust inhibitors forming a compact and protective, water repelling layer. Other additives neutralize the corrosive decomposition products generated during ageing.

DIN 51 355 and DIN 51 385 describe standardized tests to determine an oil's rust prevention ability.

Compatibility with nonferrous metals

Gear oils containing EP additives (to form boundary layers) must not react with nonferrous metals, i.e. they must not lead to corrosion on such materials, especially on copper or copper alloys (bronze, brass), throughout the entire service life. Components made of nonferrous metals are found, for example, in worm gears (bronze wheels), shut-off valves, cooling units (copper alloys).

According to the requirements specified in DIN 51 517 Pt 3, CLP lubrication oils must not be corrosive on copper. The corrosion behaviour of lubricating oils on copper is tested in accordance with DIN 51 759 (copper strip test).

All CLP and synthetic gear oils offered by Klüber comply with this requirement and are not corrosive on copper.

7.4.5 Behaviour towards air

Gear oils should be able to separate rapidly from dispersed air and prevent the formation of stable surface foam. Foam is generated by air bubbles rising to the surface. The bubbles should burst as quickly as possible to keep the foam to a minimum.
In case of splash lubrication at medium to high peripheral speeds, the oil has a pronounced foaming tendency due to the air that is constantly introduced. Contaminants such as water, dust, corrosion particles and ageing residues may even increase the foaming tendency. Foaming has a high negative impact

on the lubricant's properties, such as oxidation, pressure absorption capacity, etc. Excessive foaming may cause the foam to be forced out of the breather vent; in case of pressurized circulation lubrication there is the danger of foam being sucked into the oil pump.

The foaming tendency can be reduced by means of anti-foam (AF) additives. However, a too high concentration may have a negative impact on the air separation capacity.

The foaming tendency of a lubrication oil is determined in accordance with DIN E 51 566.

7.4.6 Cold flow behaviour

Depending on the base oil type, lubricating oils solidify at low temperatures due to their increased viscosity (synthetic oils) or wax crystallization (paraffin mineral oils). An oil's pour point is indicative of its cold flow behaviour. The pour point is the lowest temperature at which the oil still flows if it is cooled down under specified test conditions. It is determined in accordance with DIN ISO 3016.

In order to ensure rapid and sufficient lubricant supply during a cold start, the lowest temperature occurring in a gear (starting temperature) should always be several degrees above the pour point.

Synthetic gear oils show a much more favorable cold flow behaviour than mineral oils. Their pour point is much lower, sometimes even below -50 °C. Due to their high viscosity index, synthetic oils are less viscous at lower temperatures than mineral oils with the same nominal viscosity. Their cold starting behaviour is therefore particularly good at very low temperatures. Friction points in the gear are supplied with lubricant more rapidly, which makes oil pre-heating elements unnecessary. In addition, friction losses are reduced during the heating phase which is often very long.

Product	ISO VG DIN 51 519	Viscosity index DIN ISO 29 09 approx.	Pour point DIN ISO 30 16 [°C]
Klüberoil GEM 1-220 (mineral oil)	220	95	< – 10
Klübersynth GEM 4-220 (Synthetic hydrocarbon/ester oil)	220	> 150	< – 40
Klübersynth GH 6-220 (PG oil)	220	> 220	< – 30

Table 15: Comparison of approximate low-temperature values of a mineral gear oil and synthetic gears oils (Klüber gear oils)

7.4.7 Compatibility with elastomers (seals)

The term "elastomers" refers to a number of different synthetic materials used to manufacture rotary shaft lip seals. When exposed to operating temperatures, the gear oil and its additives must neither embrittle nor soften the seals or impair their sealing effect. A certain amount of swelling is desired in order to compensate for minimum wear of the sealing lips.

Gear oils must be compatible with shaft seals and all other nonmetallic components installed in a gear system or in a connected oil circulation system. These components include flat, O-ring and molded seals, sealing putty, rolling bearing cages, and filter inserts.

The nonmetallic materials must not release any particles into the gear oil which would change the oil's properties or have a negative effect on the gear's performance.

The impact of a gear oil on a seal's swelling behavior and Shore-A hardness are determined in accordance with DIN 53 521 and DIN 53 505.

The extent to which a gear oil has an impact on elastomers depends on the oil's composition and its viscosity. Low viscosity promotes swelling, which, in turn, can result in excessive seal wear. Sulfur additives may lead to embrittlement and impair the sealing effect.

With increasing oil temperatures the interaction between the lubricating oil and the seals intensifies. *Table 16* gives a general survey of the compatibility of gear oils with various sealing materials.

Converting a gear from a mineral to a synthetic oil is often done to improve performance at higher operating temperatures.

Compabitility with the sealing material, however, must also be taken into account. This pertains especially to the base oil type (SHC/PG oil) and the temperatures occurring at the sealing points.

Sealing material			Lubricating oil		
Abbreviation	Type	Thermal resistance	Mineral gear oils for industrial gears	**SHC** base oils	**PG** base oils
NBR	Acrylonitrile-butadiene-rubber, e.g. Buna-N	to 100 °C	●	●	●
ACM	Acrylate rubber	to 125 °C	●	●	⊗
VQM	Silicone rubber	to 125 °C	Compatible with all gear oils, but impact on air separation capacity		
FKM	Fluorinated rubber, e.g. Viton	to 150 °C	○	●	⊗
PTFE	Polytetrafluoroethylene	to 150 °C	○	●	●

Legend:

● = compatible

⊗ = no or limited compatibility

○ = mineral oils suitable for sealing point only up to 125 °C

Table 16: Compatibility of gear oils with sealing materials, e.g. radial shaft lip seals

7.4.8 Compatibility with interior coatings

Gear housings made of grey cast iron or sheet steel are usually coated to protect them against corrosion during storage, transport or extended periods of standstill. Artificial resin primers are generally resistant to mineral gear oils up to a temperature of 100 °C. However, they are not resistant at higher temperatures (> 100 °C) or to synthetic gear oils, especially those with a PG base oil. The coatings may get soft, dissolve or form blisters and chip off, thus causing gear malfunctions or damage by clogging oil lines, filters and deaeration holes.

We recommend to have the paint manufacturer carry out compatibility tests prior to series applications.

7.4.9 Miscibility of different types of gear oils

Different mineral gear oils are miscible with each other. However, we recommend to mix only such oils which meet the gear manufacturer's instructions in terms of quality and viscosity (observe collective lubricant charts).

Lubricants of different manufacturers generally vary in their composition (base oil, additives) and properties. Mixing such oils may therefore result in quality impairment, for example, a reduction of antiwear properties and/or viscosity-temperature behaviour. If possible, the same type of oil should be used for refilling.

Mineral and synthetic gear oils are not miscible or only to a certain extent. Synthetic oils with a different base oil (PG/SHC) must not be mixed. The following is important when changing from a mineral to a synthetic gear oil:

SHC gear oils

These gear oils (e.g. Klübersynth series GEM 4) are miscible with mineral oil residues (up to approx. 5 %) which cannot be removed by normal draining and remain in the gear system (including the oil container and filter).

PG gear oils

These gear oils (e.g. Klübersynth series GH 6) are neither miscible with mineral oils nor synthetic HC oils. Before changing over to a PG oil the previous oil has to be removed completely by flushing the system with the new product. Filter inserts have to be changed. The PG oil used for flushing must **not** be used for lubrication purposes. It can, however, be re-used for flushing purposes. The gear should then be filled with fresh oil.

Residual amounts of mineral or SHC oil (up to 5 %) in polyglycol are not problematic.

8 Selection of gear lubricants

Maximum operational security throughout a gear's service life can only be ensured if lubricants are not only considered necessary ingredients but are taken into account during all design phases as integral structural elements.

Lubricants should already be selected in the initial design phase. Based on the gear's performance, speed, ambient influences and operating conditions (e.g. connection to an oil circulation system), the following parameters are important:

- type of lubricant
- application method
- viscosity and consistency
- lubricant quality

8.1 Selection of lubricant type and application method

The selection of a suitable lubricant type and application method depends on the type of gear and its peripheral speed.

Table 17 shows correlations and makes it possible to determine appropriate lubricants and application methods.

The indicated peripheral speeds are guide values that can be exceeded if appropriate design measures are taken, e.g. oil pockets, depots, baffle plates or cooling units. The decisive selection parameter is the gear stage with the highest peripheral speed.

8.1.1 Notes on lubricant selection

Oil lubrication

Oil lubrication is preferred in closed gears because a fluid lubricant not only lubricates the meshing teeth but also dissipates heat from the friction points.

Another decisive edge: All other friction points in a gear system, such as rolling and plain bearings, seals, couplings, backstops, etc. can be lubricated with only one product. In individual cases, however, additional grease lubrication may be required for the rolling bearings.

Lubrication with gear greases

Gear greases are only used in exceptional cases and under certain design conditions, for example if it is not possible to change the lubricant on a regular basis (long-term or lifetime lubrication), or in closed gears that are not oiltight. Gear greases are frequently used in small closed gears whose mounting position is unknown or whose position changes during operation, because such gears are often not completely tight. Gear greases have a lower heat dissipation capacity than oils, which makes them only suitable for low-performance gears.

The general peripheral speed limit is 4 m/s. It is, however, possible to use fluid greases in gear motors up to 20 m/s.

Fluid greases are only applied by way of splash lubrication supplying both the gear wheels and the bearing.

Fluid greases of NLGI grade 00 and 000 are particularly suitable to lubricate the various stages of gear motors.

Lubricating greases of NLGI 0, 1, and 2 are often used in small and miniature gears in office machines, domestic appliances, automotive gear motors, power tools, etc.

Lubrication with adhesive lubricants

Adhesive lubricants are mainly used on the tooth flanks of open or encased gears (girth gear drives, open gear stages). State-of-the-art adhesive lubricants are high-performance specialty lubricants free from bitumen and solvents. They are available in the form of fluids (for splash and circulation lubrication) or grease-like adhesive spray lubricants for application through automatic spray systems.

The main characteristics of modern adhesive lubricants are determined by excellent EP additives, adhesion improvers and solid lubricants enhancing the load-carrying and emergency lubrication capacity.

Being a specialty lubricant manufacturer, Klüber has developed a systematic lubrication method for large girth gear drives (tube mills, rotary kilns, etc.) known and applied worldwide under the name "**A-B-C lubrication**".

Lubrication of Gear Systems

Type of gear system ▶	Closed gear, oiltight		Closed gear, not oiltight		Open gear	
Type of lubricant ▶	**Lubricating oil**		**Gear grease (fluid grease)**		**Adhesive lubricant (grease, fluid)**	
Type of gear ▼	**Peripheral speed** [m · s $^{-1}$]	**Application method**	**Peripheral speed** [m · s $^{-1}$]	**Application method**	**Peripheral speed** [m · s $^{-1}$]	**Application method**
Spur and bevel gears	to ≈ 20 from ≈ 20 to 250	immersion spraying	to 4	immersion	to 2 to ≈ 8 from ≈ 8 to 11	manual immersion or spraying spraying
Worm gears (immersing worm)	to ≈ 12 from ≈ 12	immersion spraying in direction of action	to 4	immersion	–	–
Worm gears (immersing wheel)	to ≈ 8 from ≈ 8	immersion spraying in direction of action	to 1	immersion	–	–

Table 17: Guide values to determine the lubricant type and application method depending on the type of gear and its peripheral speed.

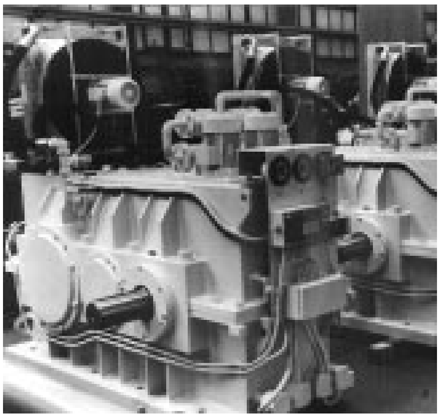


Fig. 55: Closed oiltight gear F 10248

Fig. 56: Closed gear, not oiltight (exposed miniature gear) F 10249

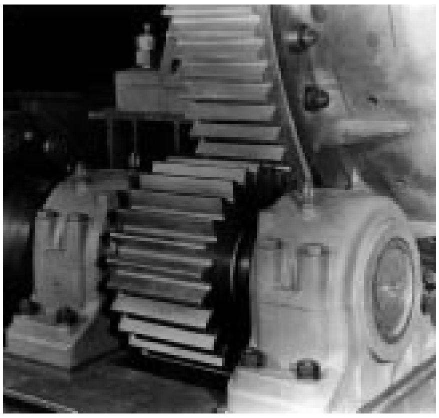


Fig. 57: Open gear (girth gear drive) F 10250

8.2 Determination of the required viscosity

8.2.1 Determination of the viscosity of mineral lubricating oils DIN 51 509

Due to the different kinematic conditions it is important to distinguish between spur and bevel gears on the one hand and worm gears on the other hand.

Depending on the type of gear, the nominal kinematic viscosity is determined in $mm^2 \bullet s^{-1}$ at 40 °C (mid-point viscosity) as a function of the force-speed factor (k_s/v) by means of the curves shown in *fig. 58* and *59*.

They are based on practical gear lubrication experience. The force-speed factor (k_s/v) takes into consideration the specific load on the gear. The indicated viscosities are guide values referring to an ambient temperature of approx. 20 °C and an oil service temperature of 70 °C.

Viscosity values will differ in case of other ambient temperatures and special operating conditions.

See also notes on page 50/51.

The power-speed factor is calculated in accordance with the following equations:

Calculation of the force-speed factor

for spur and bevel gears

$$k_s/v = \left[\frac{F_t}{b \cdot d_1} \cdot \frac{u+1}{u} \cdot Z_H^2 \cdot Z_\epsilon^2 \cdot K_A\right] / v$$

k_s/v = Force-speed factor [N · s/mm² · m = MPa·s·m⁻¹]
v = Peripheral speed on the reference circle [m·s⁻¹]
k_s = Stribeck's rolling pressure [N/mm²]
F_t = (Nominal) Peripheral force [N]
b = Tooth width [mm]
d_1 = (Pitch) Reference circle diameter [mm]
u = Gear ratio z_2 / z_1 ($z_2 > z_1$)
Z_H = Zone factor *
Z_ϵ = Contact ratio factor *
K_A = Application factor **

for worm gears

$$k_s / v = \frac{T_2}{a^3 \cdot n_s} \cdot K_A$$

k_s/v = Force-speed factor [N·min/m²]
v = Peripheral speed [$m \cdot s^{-1}$]
T_2 = Initial torque [Nm]
a = Center distance [m]
n_s = Worm speed [min^{-1}]
K_A = Application factor **

* Z_H and Z_ϵ are determined in accordance with DIN 3990 Pt 2. The following equation can be taken as the basis of a rough estimate: $Z_H^2 \cdot Z_\epsilon^2 \approx 3$

** Approximate values of the application factor K_A are listed in table 21. This table as well as tables 22, 23 and 24 were taken from DIN 3990 Pt6.

Calculation examples

Example 1:

Determination of the nominal kinematic viscosity required for a single-stage spur gear driving a fan

Operating conditions

Peripheral speed on the reference circle	v	=	$4\ m \cdot s^{-1}$
(Nominal) Peripheral force	F_t	=	3 000 N
Tooth width	b	=	25 mm
(Pitch) Reference circle diameter	d_1	=	230 mm
Gear ration z_2/z_1	u	=	2.5
Application factor	K_A	=	1 *(Table 21)*
Drive motor	Electric motor		

Calculation of Stribeck's rolling pressure k_s

$$k_s = \frac{F_t}{b \cdot d_1} \cdot \frac{u+1}{u} \cdot Z_H^2 \cdot Z_\epsilon^2 \cdot K_A \ [N/mm^2]$$

For a rough calculation of Stribeck's rolling pressure (except for hypoid gears) it is possible to use the equation

$Z_\epsilon \quad Z_H^2 \cdot Z_\epsilon^2 = \mathbf{3}$ as a base.

$$k_S = \frac{3000}{25 \cdot 230} \cdot \frac{2.5 + 1}{2.5} \cdot 3 \cdot 1 \frac{N}{mm \cdot mm}$$

$$= \mathbf{2.2} \frac{N}{mm^2}$$

Force-speed factor k_S / v

$$\frac{k_S}{v} = \frac{2.2}{4} \frac{N \cdot s}{mm^2 \cdot m} = \mathbf{0.55}\ MPa \cdot s \cdot m^{-1}$$

Having calculated the force-speed factor, it is possible to determine the required nominal kinematic viscosity in *Fig. 58*.
In our example the nominal kinematic viscosity at 40 °C = **150** $mm^2 \cdot s^{-1}$.
According to *table 18*, page 50, the oil suitable for this application would be ISO VG **150**.

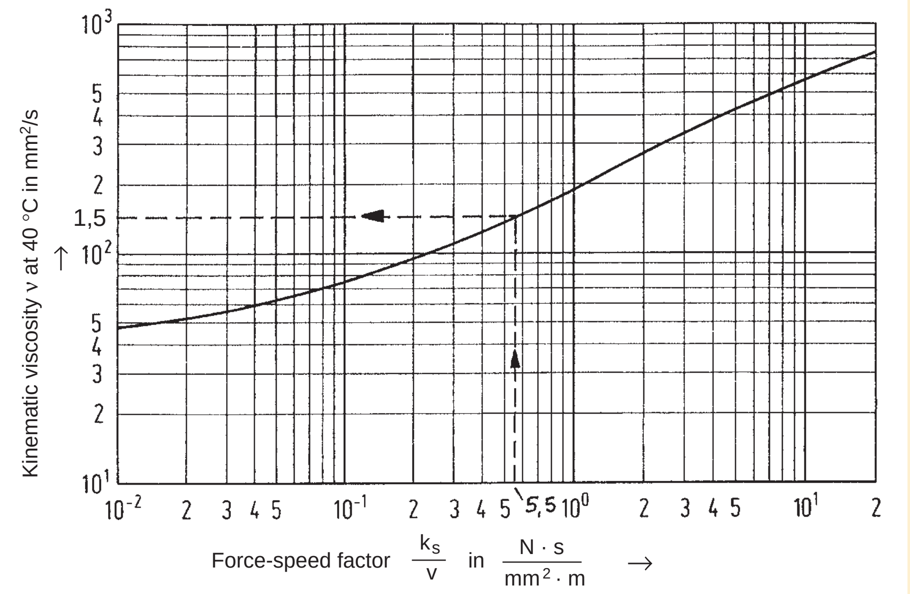


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Fig. 58: Determination of the required viscosity for spur and bevel gears (except for hypoid gears)

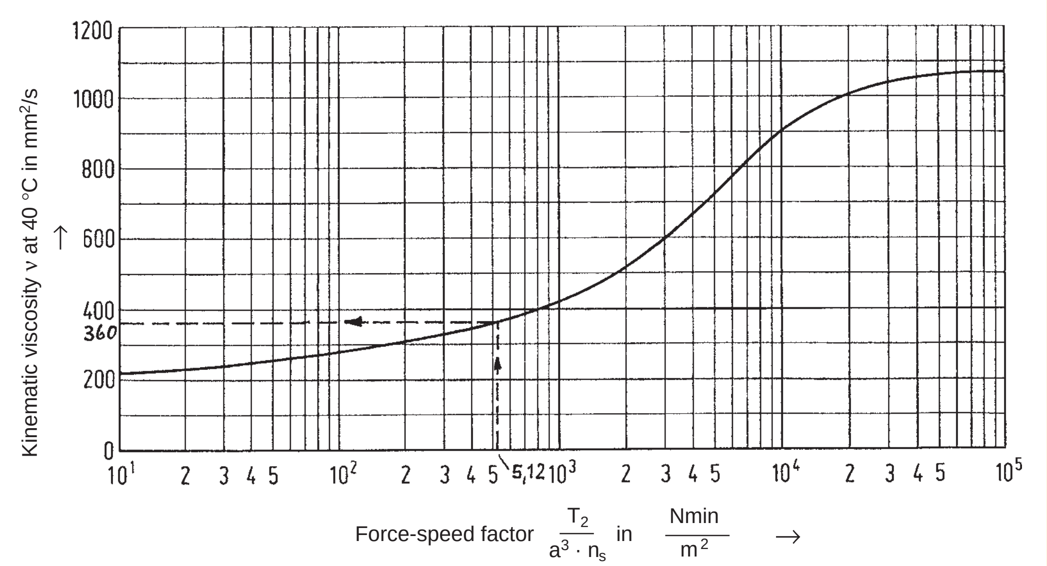

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Fig. 59: Determination of the required viscosity for worm gears

Example 2:

Determination of the nominal kinematic viscosity required for a worm gear driving a light-weight elevator

Operating conditions

Initial torque	T_2	= 1000 Nm
Center distance	a	= 0.125 m
Worm speed	n_s	= 1000 min^{-1}
Application factor	K_A	= 1 *(Table 21)*
Drive motor	Electric motor	

Force-speed factor k_s/v

$$\frac{k_s}{v} = \frac{T_2}{a^3 \cdot n_s} = \frac{1000}{0.125^3 \cdot 1000} \cdot 1 \; \frac{Nm}{m^3 \cdot min^{-1}}$$

$$= \mathbf{512} \; \frac{N \cdot min}{m^2}$$

On the basis of this force-speed factor it is possible to determine the required nominal kinematic viscosity in *Fig. 59*.

In our example the nominal kinematic viscosity at 40 °C = **360** $mm^2 \cdot s^{-1}$.

As the determined value exceeds the limits of ISO VG 320 (see *table 18*) an oil of the next higher viscosity – in this case ISO VG **460** – has to be selected.

Viscosity grade ISO	Mid-point viscosity at 40 °C [$mm^2 \cdot s^{-1}$]	Kinematic viscosity limits at 40 °C [$mm^2 \cdot s^{-1}$]	
		min.	max.
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.48
ISO VG 10	10	9.00	11.0
ISO VG 15	15	13.5	16.5
ISO VG 22	22	19.8	24.2
ISO VG 32	32	28.8	35.2
ISO VG 46	46	41.4	50.6
ISO VG 68	68	61.2	74.8
ISO VG 100	100	90.0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650

Table 18: ISO viscosity grades for industrial lubricating oils acc. to DIN 51 519

NOTE

When determining the force-speed factor (K_s/v) for the viscosity calculations, it is important to take into account that in multi-stage gears the individual gear steps are subject to different operating conditions.

In case of two-stage gears the operating conditions existing at the final stage are important for the calculations.

In case of triple-stage gears the force-speed factor (k_s/v) has to be determined for the second and the third stage. The average of these two values is required to calculate the nominal kinematic viscosity for the entire gear system. The calculation process for gears with more than three stages is the same.

Correction of the nominal kinematic viscosity

The nominal kinematic viscosity values determined in *Fig. 58* and *59* are valid for normal operating conditions and an average ambient temperature of approx. 20 °C (tolerance range: 10 °C to 25 °C).

If the ambient temperature is permanently outside the tolerance range it is necessary to correct the nominal viscosity in order to ensure that the operating viscosity remains the same.

Apart from temperature related corrections, it may also be necessary to correct the nominal viscosity due to loads, materials involved or tooth-related reasons.

Conditions for ***increased*** *viscosity values*

- If the ambient temperature is permanently above 25 °C, the nominal kinematic viscosity has to be increased by approx. 10 % for every 10 °C.
- If the gear pairs consist of similar steels or CrNi steel, the nominal kinematic viscosity has to be increased by 35 %. This does not refer to surface-hardened or nitride steel.
- Gear pairs susceptible to scuffing (no or inadequate elasto-hydrodynamic lubricating film) also require an increased nominal kinematic viscosity. The same applies to lubricants not containing antiwear additives.

*Conditions for **decreased** viscosity values*

- If the ambient temperature is permanently below 10 °C, the nominal kinematic viscosity has to be decreased by approx. 10 % for every 3 °C.
- The nominal kinematic viscosity can be decreased up to 25 % if the tooth flanks are phosphated or copper-coated.

If temperature-related viscosity corrections are required because the actual ambient temperatures differ from the normal temperature range (10 °C to 25 °C), it is possible to use the viscosity-temperature diagram shown in *Fig. 60* to determine a suitable viscosity. The indicated example shows how to proceed.

In case of doubt always select the next higher viscosity grade. At low temperatures it is important to observe the lubricant's flow limit.

IMPORTANT

After having determined a viscosity grade suitable for your gear, we recommend to calculate the scuffing resistance, for instance by means of the GfT Worksheet 2.4.2 (GfT = German Tribological Society).
In addition, it is important to take into account the viscosity requirements of the other friction points in the gear system which will also be lubricated with the gear oil, e.g. rolling or plain bearings, mechanical or hydraulic couplings, oil pumps and connected hydraulic systems.

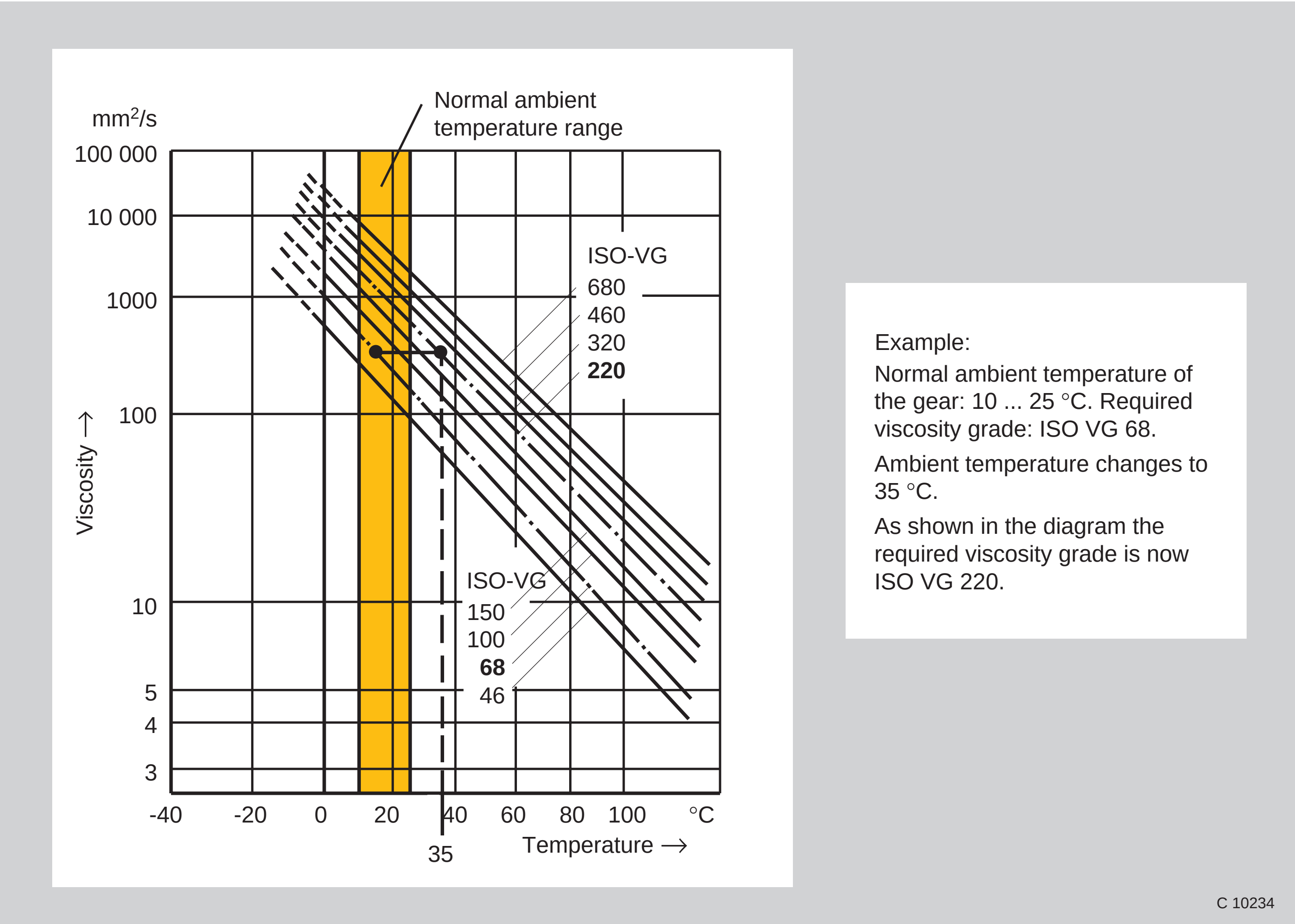


Fig. 60: Viscosity-temperature diagram / Example of oil selection at ambient temperatures outside the normal range

8.2.2 Determination of the viscosity of synthetic Klüber oils

Viscosity calculations in accordance with DIN 51 509 are only valid for mineral oils with a viscosity index between 90 and 95. These calculations only take into account a specific viscosity-temperature behaviour. In case of synthetic lubricating oils, whose viscosity is not determined by way of calculation (e.g. on the basis of the EHD theory), it is important to take into account the different viscosity-temperature behaviour of synthetic oils as well as their temperature-related pressure-viscosity behaviour.

As there are no standardized procedures to determine the required viscosity of synthetic oils and the methods currently used are quite complicated, we have developed special worksheets for our new synthetic gear oils (Klübersynth GEM 4 and Klübersynth GH 6 series). These worksheets make it quite easy to select an appropriate ISO VG grade.

The selection process is partly based on DIN 51 509. Calculate the force-speed factor (k_s/v) – taking into account the type of gear – and use the result to determine the Klüber viscosity number. With this viscosity number and the expected oil operating temperature it is possible to determine the required ISO VG grade in the selection diagram (*Fig. 61 and 62*) pertaining to the respective Klüber oil series. The selection diagrams take into account the specific viscosity-temperature and viscosity-pressure behaviour of synthetic oils.

NOTE

Klüber defines the oil operating temperature as the oil sump temperature or the temperature of the injected oil.

The expected oil operating temperature is determined on the basis of the gear's thermal economy (considering losses) or, in case of installed gears, is simply measured.

Examples for the determination of the viscosity according to the Klüber selection process:

Example 1:

Determination of the viscosity grade required for a single-stage spur gear driving a fan

Operating conditions

Peripheral speed on the reference circle	$v = 4\ m \cdot s^{-1}$
Nominal peripheral force	$F_t = 3\,000\ N$
Tooth width	$b = 25\ mm$
Reference circle diameter	$d_1 = 230\ mm$
Gear ratio z_2/z_1	$u = 2.5$
Application factor	$K_A = 1$ *(Table 21)*
Drive motor	Electric motor
Expected oil temperature	approx. 90 °C
Selected lubricant:	Klübersynth GEM 4 (SHC oil)

Determination of the force-speed factor k_s/v

Calculation corresponding to page 48, "*Determination of the nominal kinematic viscosity required for a single-stage spur gear ...* "

$$\frac{k_s}{v} = \mathbf{0.55}\ MPa \cdot s \cdot m^{-1}$$

Determination of the Klüber viscosity number (KVZ)

The Klüber viscosity number is determined in *Table 19*.
The example calculation results in a force-speed factor of 0.55 MPa · s · m^{-1}.
The corresponding Klüber viscosity number is **4**.

Force-speed factor k_s/v [MPa · s · m^{-1}]	Klüber viscosity number KVZ
≤ 0.02	1
> 0.02 to 0.08	2
> 0.08 to 0.3	3
→ > 0.3 to 0.8	→ 4
> 0.8 to 1.8	5
> 1.8 to 3.5	6
> 3.5 to 7.0	7
> 7.0	8

Table 19: Klüber viscosity number as a function of the force-speed factor (k_s/v) for spur and bevel gears (except for hypoid gears)

Determination of the viscosity grade

The required viscosity grade is determined on the basis of the viscosity selection diagram (*Fig. 61*).

Assuming a Klüber viscosity number (KVZ) of 4 and an expected oil service temperature of approx. 90 °C, the diagram indicates a suitable viscosity grade of ISO VG 220. In other words:
Klübersynth GEM 4-220 is the suitable lubricant.

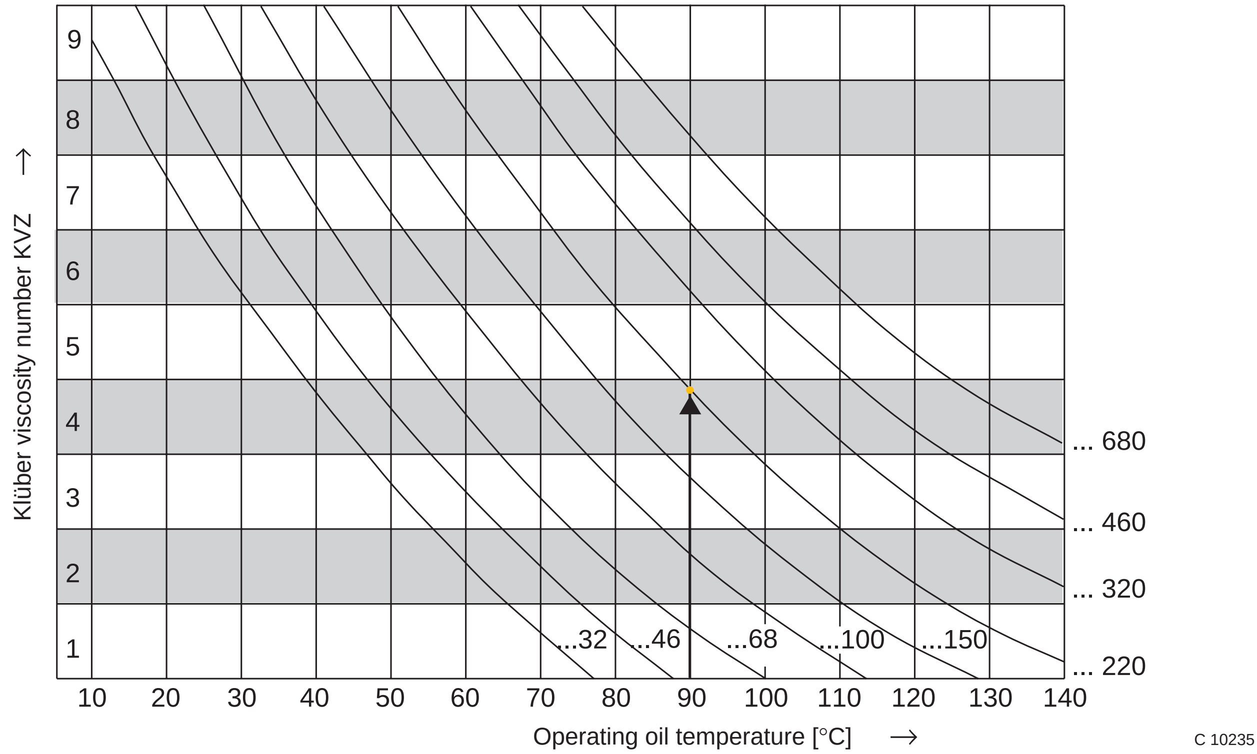


Fig. 61: Viscosity selection diagram for ***Klübersynth GEM 4 - 32 ... 680*** *lubricating oils (see notes on page 54 on how to use the diagram)*

Example 2:

Determination of the viscosity grade required for the worm gear stage of a gear motor driving a circular conveyor

Operating conditions

Output torque	T_2 = 300 Nm
Worm speed	n_S = 350 min^{-1}
Center distance	a = 0.063 m
Application factor	K_A = 1 *(Table 21)*
Expected oil temperature	approx. 85 °C
Drive motor	Electric motor
Selected lubricant:	Klübersynth GH 6 (PG oil)

Determination of the force-speed factor k_S/v

$$\frac{k_S}{v} = \frac{T_2}{a^3 \cdot n_S} \cdot K_A = \frac{300}{0.063^3 \cdot 350} \cdot 1$$

$$= \mathbf{3428.6} \ \frac{N \cdot min}{m^2}$$

Determination of the Klüber viscosity number (KVZ)

The Klüber viscosity number is determined in *Table 20*.
The example calculation results in a force-speed factor of 3428.6 N · min / m². The corresponding Klüber viscosity number is **8**.

Force-speed factor k_S/v [N · min / m²]	Klüber viscosity number KVZ
≤ 60	5
> 60 to 400	6
> 400 to 1800	7
→ > 1800 to 6000	→ 8
> 6000	9

Table 20: Klüber viscosity number (KSV) as a function of the force-speed factor (k_S/v) for worm gears

Determination of the viscosity grade

The required viscosity grade is determined on the basis of the viscosity selection diagram *(Fig. 62).*

Assuming a Klüber viscosity number (KVZ) of 8 and an expected oil service temperature of approx. 85 °C the diagram indicates a suitable viscosity grade of ISO VG 460. In other words:
Klübersynth GH 6 - 460 is the suitable lubricant.

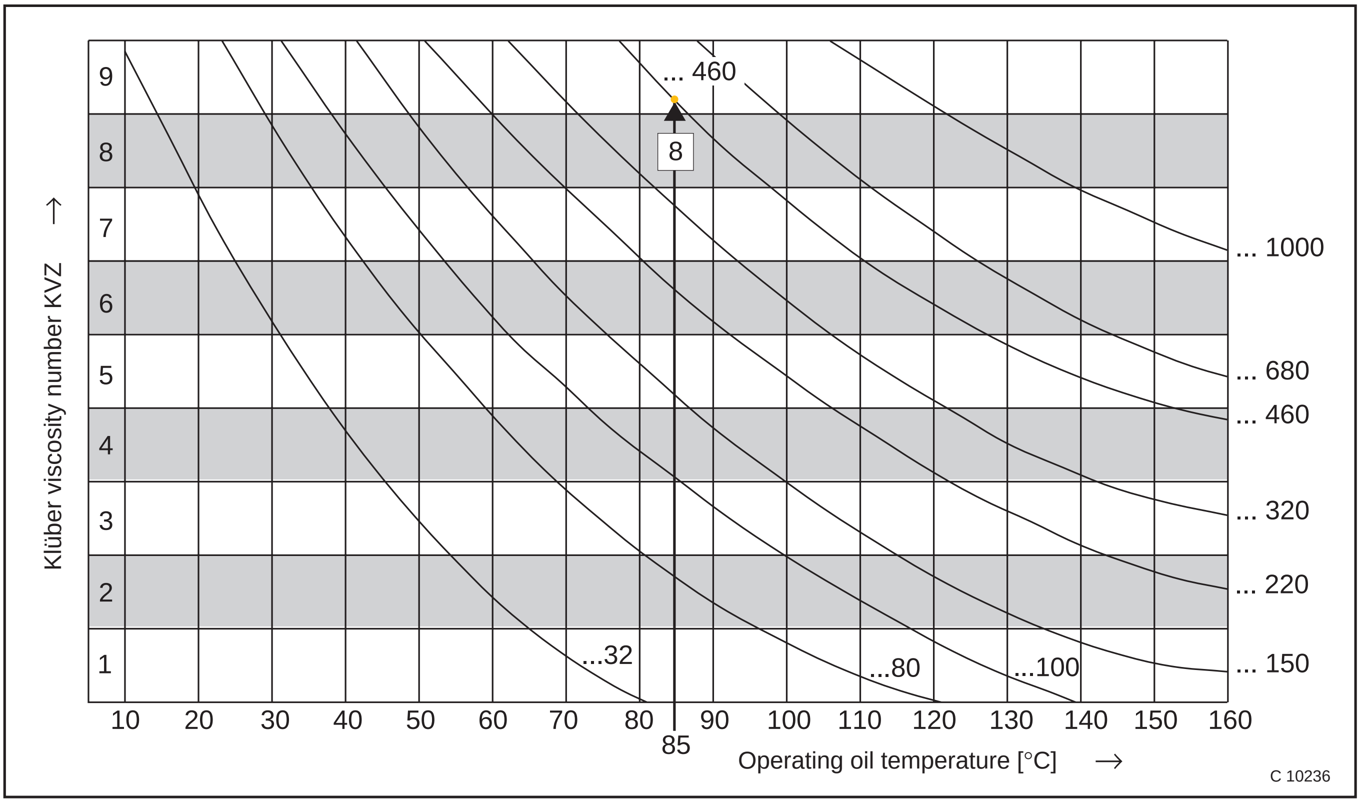


*Fig. 62: Viscosity selection diagram for **Klübersynth GH 6 - 32 ... 1000** lubricating oils (see notes below on how to use the diagram)*

NOTES

– In example 1 the viscosity grade can be determined quite easily because the coordinates of the Klüber viscosity number and the oil service temperature (90 °C) intersect exactly on the VG 220 curve.
  If the point of intersection is located between two curves (see example 2: between VG 320 and VG 460), always select the higher viscosity grade (in this case VG 460).

– Viscosity selection as shown in example 1 and 2 is only valid for **one** gear stage. In case of multi-stage gears each stage has to be considered individually. From all the determined viscosity grades select the one which is the best compromise to ensure reliable lubrication of the gear stages and the other gear components involved.

– In order to ensure adequate lubrication during cold start and at low ambient temperatures it may be necessary to select a lower viscosity grade. Check the viscosities at the pertinent starting temperature (especially in case of oil circulation lubrication) and test the involved components under the expected starting temperatures (especially in case of splash lubrication).

– When determining a suitable viscosity grade in accordance with the Klüber selection process, take into account that the selection diagrams only pertain to **one** Klüber oil series. The different types of lubricating oils have different synthetic base oils and vary considerably in their viscosity-temperature behaviour. Before determining a suitable viscosity grade it is therefore important to know the type of oil to be used.

The viscosity selection diagrams for *Klübersynth GEM 4* and *Klübersynth GH 6* products are shown in *Fig. 61 and 62*.

Note on page 55:

Tables 21 - 24 were taken from DIN 3990 Pt 1/12.87, Appendix A.

Operation of driving unit	Operation of driven unit			
	uniform	moderate shocks	medium shocks	heavy shocks
Uniform	1.00	1.25	1.50	1.75
Light shocks	1.10	1.35	1.60	1.85
Moderate shocks	1.25	1.50	1.75	2.0
Heavy shocks	1.50	1.75	2.0	2.25 or more
The values pertain to the nominal torque of the driving unit, alternatively to the nominal torque of the drive motor if it corresponds to the torque required by the driven unit. They are only valid for machines not operating in the resonance range and only if they have a uniform power requirement. In applications with exceptional loads, motors with a high starting moment, intermittent operation, extreme and recurring shock loads, the gears have to be checked for static and fatigue strength. For examples see DIN 3990 Pt 6, page 9.				

Table 21: Application factor K_A

Mode of operation	Driving unit
Uniform	Electric motor (e.g. direct current), steam or gas turbine operating uniformly *) (low, infrequent starting torques) **)
Light shocks	Steam or gas turbines, hydraulic or electric motor (high, frequent starting torques) **)
Moderate shocks	Multi-cylinder combustion engine
Heavy shocks	Single-cylinder combustion engine
*) As determined in frequency tests or as known by experience **) Comparative life curves Z_{NT}; Y_{NT} of the material in DIN 3990 Pt 2 and 3 takes into account short-term overload torques	

Table 22: Examples of driving units with different modes of operation

Mode of operation	Driven unit
Uniform	Power generators, uniformly fed belt conveyors or apron feeders, light-weight elevators, packaging machines, feed drives of machine tools, fans, light-weight centrifuges, rotary pumps, agitators and mixers for light fluids or substances of a uniform density, cutters, presses, punches [1]); rotary units, drive units [2]).
Moderate shocks	Intermittently fed belt conveyors or apron feeders, main drive of machine tools, heavy elevators, rotary units of cranes, industrial and mining fan systems, heavy centrifuges, rotary pumps, agitators and mixers for viscous fluids or substances of varying density, multi-cylinder piston pumps, feeding pumps, extruders in general, calenders, rotary kilns, rolling mills [3]) (continuous zinc and aluminum belt rolling mills, wire and rod mills).
Medium shocks	Rubber extruders, intermittently operating mixers for rubber and synthetic materials, light-weight ball mills, woodworking machines (automatic saws, lathes), blooming mills [3]), [4]); lifting units, single-cylinder piston pumps
Heavy shocks	Excavators, bucket wheel and chain drives, screen drives, dredging shovels, rubber kneaders, stone and ore crushers, mining machinery, heavy feed pumps, rotary drilling installations, brick presses, debarking drums, peeling machines, cold belt rolling mills [3]), [5]); briquetting presses; edge mills
[1]) Nominal torque = maximum cutting, pressing, punching torque [2]) Nominal torque = maximum starting torque [3]) Nominal torque = maximum rolling torque [4]) Torque based on power limit [5]) K_A up to 2.0 due to frequent belt breakage	

Table 23: Industrial gears, examples of operating modes of driven units

Mode of operation	Driven unit
Uniform	Radial compressor in a/c systems (for process gas), power test rig, generator and exciter for base or permanent load, main drive of paper machines
Moderate shocks	Radial compressor for air or pipelines, axial compressor, centrifugal fan, generator and exciter for peak loads, all types of rotary pumps except rotary axial-flow pumps, gear pumps, paper industry: Jordan refiner, secondary drives of paper machines, pulp compactors
Medium shocks	Rotary cam blower, rotary radial-flow cam compressor, piston compressor (3 or more cylinders), suction fans for industrial or mining applications, (large units with frequent starts), rotary boiler feed pump, rotary cam pump, piston pump (3 or more cylinders)
Heavy shocks	Double-cylinder piston compressor; rotary pump (surge tank); slurry pump; double-cylinder piston pump

Table 24: High-speed gears, examples of operating modes of driven units

8.3 Selection of synthetic gear oils

Lubricating oils with a synthetic hydrocarbon (SHC), polyalkylene glycol (PG) or ester base oil have proven particularly efficient in the lubrication of gear systems. Lubricants with a PG base oil are usually referred to as polyglycol oils.

Synthetic gear oils have the following advantages over mineral lubricating oils:

- better resistance to temperatures and ageing
- lower tooth-related friction coefficient
- better cold flow properties
- more favorable viscosity-temperature behaviour

Table 25 shows a comparison of the properties of high quality mineral oil base gear oils and synthetic oil base lubricating oils.

Properties \ Type of oil	High-quality EP gear oils with a mineral base oil Klüberoil GEM 1 series	Gear oils with a SHC base oil Klübersynth GEM 4 series	Gear oils with a PG base oil Klübersynth GH 6 series	Gear oils with a ester base oil Klüberbio series
Viscosity-temperature behaviour	2	**1**	**1**	**1**
Low-temperature properties	2 - 3	**1**	1 - 2	2
Wear protection	2	2	**1**	2
Friction behaviour	2	**1/2**	**1**	**1**
Oxidation resistance	2	**1**	**1**	**1**
Water separation capacity	**1**	**1**	3	2
Corrosion protection	**1**	**1**	**1**	2
Miscibility with mineral oil		2	4	2
Behaviour towards paints	**1**	**1**	3	3
Compatibility with sealing materials	**1**	**1**	2	3
Low evaporation loss	2	**1**	**1**	**1**

Table 25: Comparison of the properties of mineral and synthetic gear oils (Klüber products). 1 = excellent, 2 = good, 3 = sufficient, 4 = poor

A direct comparison of the properties of SHC, PG, and ester oil base lubricating oils gives the following hints on lubricants selection:

	SHC base lubricating oils	**PG base lubricating oils**	**Ester oil base lubricating oils**
Advantages	• excellent low-temperature behaviour • miscible with mineral oil residues • compatibility with paints and sealing materials equal to that of mineral oils • very low evaporation losses up to 140 °C (also in case of low viscosity) • disposal or treatment same as for mineral oil • SHC suitable as a base oil for food grade lubricants	• low friction coefficient, therefore especially suitable for the lubrication of gears with a high sliding percentage (worm and hypoid gears) • excellent viscosity-temperature behaviour (high VI) • excellent antiwear behaviour • very good pressure absorption capacity	• good low-temperature behaviour • low friction coefficients possible • high viscosity index (VI) • high resistance to oxidation • rapidly biodegradable
Disadvantages	• antiwear behaviour inferior to that of PG oils • less favorable lubricating behaviour than PG oils	• not miscible with mineral oils • may change sealing materials and dissolve paints; only resistant to fluorinated rubber or PTFE seals as well as epoxy resin coatings	• may change sealing materials and dissolve paints; compatibility tests required • antiwear properties not as good as that of PG oils

8.4 Selection of gear greases

In DIN 51 509 Pt 2 lubricating greases for gear systems are divided into two groups:

Gear greases type **G**
Lubricating greases for closed gears and splash lubrication, where grease lubrication is preferred over oil lubrication due to the sealing or operating conditions.

Gear greases type **OG**
Adhesive lubricants free from bitumen or bituminous adhesive lubricants for open gear systems.

Gear greases type G

These are lubricating greases of NLGI grade 000 to 1, DIN 51 818, which are very soft or fluid at ambient temperatures between 18 and 28 °C according to DIN 51 014. They contain an increased amount of mineral and/or synthetic oil as well as a thickener (typically a metal soap).

They may also contain additives reducing friction and wear and/or increasing the load-carrying capacity.

The minimum requirements for such a grease are specified in DIN 51 826. According to this standard the application range of these greases is very limited because it only mentions *closed* gears with splash lubrication, e.g. gear motors, axial cylinder motors, actuators and gear couplings.

The standard does not refer to small and miniature gears, which are also lubricated with gear greases type G of NLGI grade 0 and 1 as well as lubricating greases of NLGI grade 2. Gear greases type OG are not suitable for such gears.

Gear greases type OG

These are used for the lubrication of open gear systems or girth gears of large or very large dimensions with steel/steel gear teeth, e.g. drives of rotary kilns, tube mills, lifting cylinders, cranes, construction machinery.

They are not suitable for the lubrication of small open gears!

Today large open drives are mainly lubricated with special adhesive lubricants free from bitumen and solvents, such as the products of Klüber's GRAFLOSCON or Klüberfluid series.

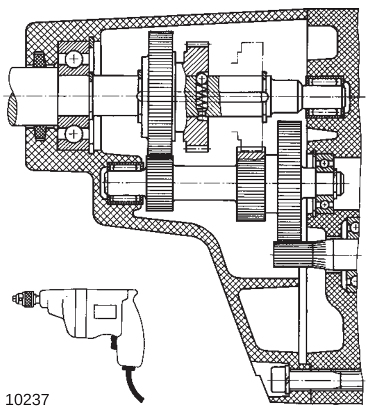
Depending on the type of lubrication and the application method, these lubricants are available as consistent spray lubricants of NLGI grade 00, 0 and 1 or in the form of fluids classified in NLGI grade 000.

Selection criteria for gear greases

To select a suitable gear grease it is important to take into consideration the type of gear, lubricant application method, gear performance, operating conditions and ambient influences. In addition, the following parameters have to be determined:

- suitable grease consistency
- required base oil viscosity
- type of base oil
- thickener properties

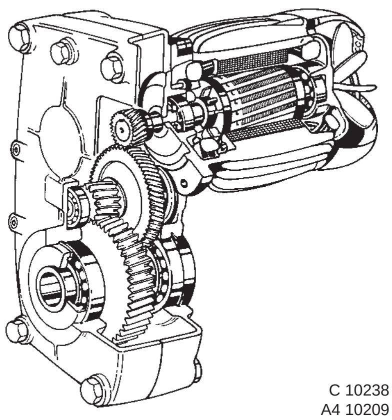
Power hand drill gear system



C 10237
A4 10208

Fig. 63: Power tools – main field of application of gear greases type G of NLGI 1 and 2

Two-stage spur gear motor



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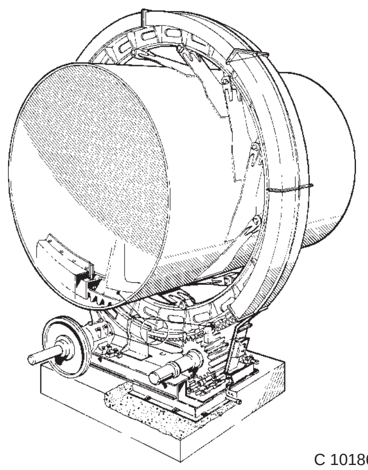
A4 10209

Fig. 64: Gear motors – classical field of application of gear greases type G of NLGI 00 and 000

Rotary kiln with girth gear drive



C 10186

Fig. 65: Open gears and gear stages – application of gear greases type OG in the form of adhesive sprays (NLGI 00, 0 and 1) or fluids (NLGI 000)

NLGI grade DIN 51 818	Worked penetration DIN ISO 21 37 Units 1)	Grease texture	Suitability for gear types	Peripheral speed $[m \cdot s^{-1}]$ Permanent operation	Short-term or intermittent operation	Application method
000	445 to 475	**fluid**	closed, oiltight	6 to 8	up to 10	immersion
00	400 to 430	**almost fluid**	closed, oiltight	4 to 5	up to 7	immersion
0	355 to 385	**extremely soft**	closed, not oiltight 5)	2 to 3	up to 5	immersion
			semi-open, open	up to 8	up to 11	spraying (total loss lubrication) 2)
				–	up to 4	one-time greasing 3)
1	310 to 340	**very soft**	closed, not oiltight	up to 10	20 to 25	immersion with almost complete filling of the gear housing 4)
			semi-open, open	–	2 to 3	one-time greasing 3)
2	285 to 295	**soft**	closed, not oiltight	up to 10	20 to 25	immersion with almost complete filling of the gear housing 4)
			semi-open, open	–	2 to 3	one-time greasing 3)

1) One unit $\hat{=}$ 0.1 mm

2) Under permanent operation and if power has to be transferred it is important that the lubricant film, which is continuously destroyed by the friction components (tooth flanks), is rebuilt regularly in order to maintain a minimum film thickness that prevents scuffing due to occasional starved lubrication. Relubrication is most reliable if carried out by means of an automatic spraying equipment which supplies the required lubricant quantity to the tooth flanks intermittently. Other relubrication methods: manual application by brush or spatula (when gear is not operating) or with a pneumatic spraying unit – e.g. Klübermatic LB – while the gear is operating up to a maximum peripheral speed of 2 $m \cdot s^{-1}$.

IMPORTANT: Only if heat dissipation is not mandatory

3) One-time greasing of the tooth flanks sufficient for the entire service life is only suitable for gears used to convert torques and/or transfer movements in short-time or intermittent operation.

4) Lubrication method ensuring reliable gear lubrication at increased peripheral speeds.

Approx. 10 % of the grease quantity are used for lubrication, the rest serves as a barrier to prevent the lubricant from leaving the meshing zone. This type of immersion lubrication is only suitable for small gears transmitting power in short-time or intermittent operation, in order to avoid the destruction of the grease due to excessive heating.

5) This also includes so-called semi-open and open gears provided they are encased and equipped with a lubricant reservoir making them suitable for immersion lubrication.

Table 26: Determination of the required grease consistency as a function of the gear type, peripheral speed and lubrication method

8.4.1 Determination of grease consistency

Consistency is the resistance of a grease to deformation, similar to an oil's viscosity. In order to ensure reliable gear lubrication, gear greases must have a certain degree of fluidity, which is why mainly soft to fluid grease are applied in gear systems.

Table 26 gives a survey of grease consistencies and indicates all details on gear types, peripheral speeds and lubrication methods that are important to determine a suitable consistency.

The table is based on DIN 51 528 but also includes NLGI grade 2. These greases are also used for gear lubrication, especially on tooth flanks in small gears or for forced-feed lubrication of gears operating at high peripheral speeds (20 to 25 m/s) with a complete fill of the gear housing.

Table 26 only lists the main selection criteria for consistency determination. Other criteria include:

- extent of gear sealing
- heat dissipation capacity
- adhesion properties.

Gear sealing

The extent of gear sealing is of special importance if gear greases with a consistency of NLGI 000 and 00 are to be used, so-called fluid greases with a base oil content of 90 to 95 %. In such a case a gear should be as oiltight as possible, or greases with a higher consistency should be applied.

Especially splash-lubricated small gears, whose position may constantly change (e.g. in power hand tools, machines for craftsmen, gears in production robots), there is the danger of oil leakage. In these cases we recommend at least an NLGI 0 grease, if possible even a grease of consistency 1 or 1-2.

As lubricating greases of such a consistency are no longer fluid, a quasi-complete filling of the gear housing is required to ensure reliable gear lubrication in all operating positions.

NOTE

If NLGI 1 and 2 greases are used for immersion lubrication, it has to be taken into consideration that they have a channeling tendency, i.e. upon immersion into the lubricant sump, the gear wheels leave a channel in the grease. Greases of this consistency have no backflow behaviour, and gear wheels or worms are subject to starved lubrication and increased wear after a very short period of time.

Channeling can be avoided by means of grease scrapers where the grease which is removed from the sump accumulates until it falls back into the sump by force of gravity.

Another way to prevent channeling is to fill the gear housing completely with grease. This also makes it possible to operate the gear at a higher peripheral speed (up to 25 m/s) because the grease is not thrown off.

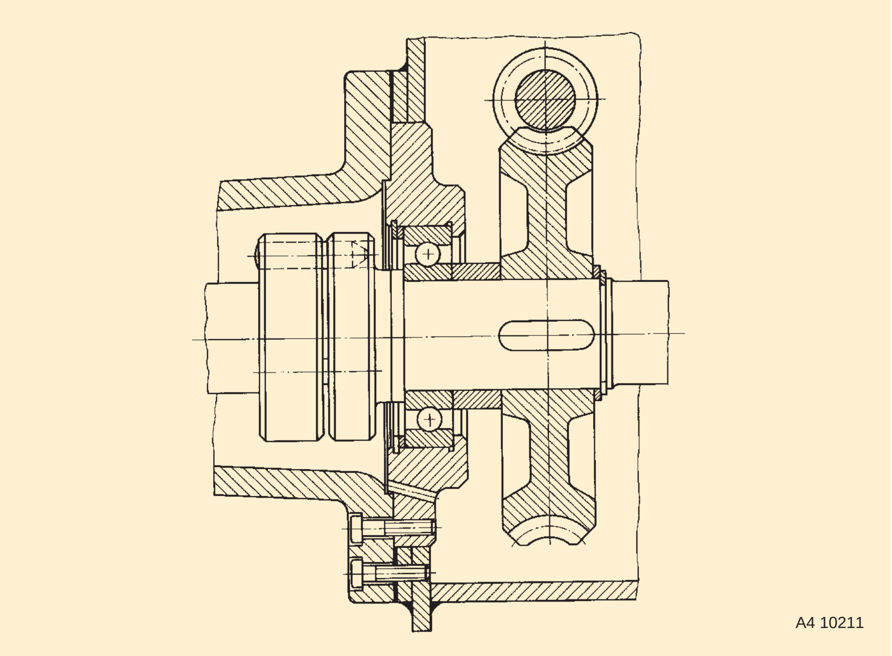


Fig. 66: Cylinder-worm gear

However, this type of immersion lubrication is only suitable for gears transmitting power in intermittent operation in order to avoid grease destruction due to excessive heating.

Heat dissipation capacity

Greases, in contrast to lubricating oils, only have a low heat dissipation capacity. Gears operating permanently at increased peripheral speeds and/or used for power transfer purposes require greases with a high base oil content for heat dissipation, i.e. fluid greases of NLGI 000 and 00. The only suitable lubrication method is splash lubrication, and the gears have to be oiltight.

Adhesion capacity

Gears mainly used to convert torques or transfer movements and low power are often lubricated for life. For this type of lubrication it is important for the grease to remain in the meshing zone throughout the gear’s service life. It must not be thrown off nor lose its lubricity due to oil separation.

Gear greases with an increased consistency, i.e. of NLGI grade 1 and 2, are most suitable to meet these requirements.

Greases of NLGI grade 0 are also suitable for tooth flank lubrication provided they have special adhesion properties. These greases are adhesive greases without solid lubricant particles and must not be confused with gear greases of the OG type.

8.4.2 Determination of the required base oil viscosity

The required base oil viscosity depends on the operating conditions and the intended use of the gear.

In case of gears used to transfer power the required base oil viscosity (nominal kinematic viscosity at 40 °C) should always be based on concrete calculations in order to prevent pittings and wear.

The required nominal viscosity can be determined in accordance with DIN 51 509 Pt 1 (section 8.2.1, page 48), no matter whether the base oil is of the synthetic or mineral hydrocarbon type.

Temperature or load-related viscosity corrections can only be effected to a limited extent in case of gear greases because pertinent data is only available for the most common base oil viscosities. When in doubt always select the lubricating grease – taking into consideration a suitable consistency – whose base oil viscosity is above the calculated nominal viscosity.

When determining the required base oil viscosity for grease lubricated gears that are used to convert torques or transfer movements – mainly low-performance small or miniature gears – the requirements in terms of smooth operation, low starting torques, vibration and noise damping are most important.

Smooth operation and low starting torques are ensured by means of *dynamically light* greases, i.e. greases with a very low base oil viscosity. Most suitable greases have a nominal kinematic base oil viscosity of 30 $mm^2 \cdot s^{-1}$ and lower at 40 °C.

Gear greases with a higher base oil viscosity should be used (between 200 and 1000 $mm^2 \cdot s^{-1}$ depending on the peripheral speed and load) if good vibration and noise damping is required. High base oil viscosities result in a thicker lubricant film on the meshing points, which increases vibration and noise damping.

In case of immersion lubricated gears with modules < 3 mm the base oil viscosity should not be higher than 460 mm^2/s in order to ensure that the grease flows back into the space between the teeth. Higher base oil viscosities are possible with modules > 3 mm.

8.4.3 Determination of the base oil type

Lubricating greases generally consist of a base oil and a thickening agent. Depending on the type and consistency of the grease the thickener share is between approx. 5 and 25 %, i.e. the oil is the main constituent. Greases are nothing but oils prevented from flowing by means of a thickening agent. When selecting a gear grease the base oil type is therefore just as important as in the case of a gear oil.

Base oils used for gear greases include mineral, synthetic ester, synthetic hydrocarbon and polyglycol oils. To obtain certain characteristics it is also possible to use a mixture of these base oil types, e.g. a mineral oil plus a synthetic hydrocarbon oil, or ester oil / paraffin oil / synthetic hydrocarbon oil.

Gear greases with a synthetic base oil have the same advantages as synthetic gear oils:

- increased resistance to high temperatures and ageing
- reduced tooth-related friction coefficient
- improved behaviour at low temperatures
- more favorable viscosity/ temperature behaviour

Gear greases with a polyglycol base oil are especially suitable for small gears subject to high loads (also in case of an increased peripheral speed), e.g. worm gears with a worm/wheel combination made of steel/bronze, because they have an excellent pressure absorption capacity, hardly change their viscosity at varying service temperatures and have a very low friction coefficient.

Polyglycol gear greases are not suitable for the lubrication of gears consisting of aluminum alloys. Their suitability for plastic gears has to be checked on a case-to-case basis.

Gear greases with synthetic hydrocarbon oils have proven effective in the lubrication of plastic/plastic and

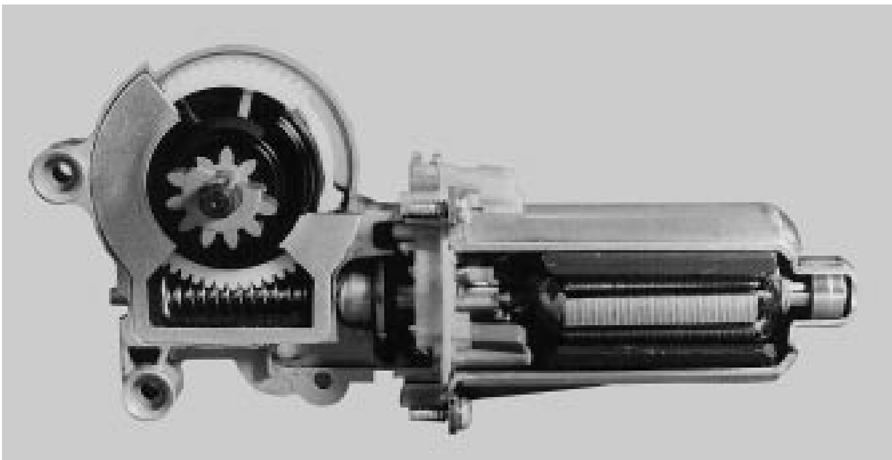


Fig. 67: Worm gear motor, servo drive for car seat adjustment F 10252

steel/plastic gears because of their compatibility with many synthetic materials.

In addition, SHC gear greases have a very good low-temperature behaviour and are very suitable for lifetime lubrication of tooth flanks due to their low evaporation loss.

Gear greases with a mineral base oil are the most inexpensive type of grease. They are mainly used as fluid greases for splash lubrication of gear motors subject to normal loads and operating permanently at almost the same service temperature. Their service temperature range is considerably smaller than that of synthetic gear greases.

In permanent operation the grease sump temperature should not exceed 60 °C in order to avoid premature grease ageing and bearing failure due to a diminishing lubricating effect.

Gear greases with a base oil mixture, e.g. mineral / SHC oil, permit an extended service temperature range.

8.4.4 Properties of the thickening agent

Thickening agents used in gear greases – mainly single or complex metal soaps (Li, Ca, Al and Ba soaps) – do more than perform the function of a base oil carrier. They also have a lubricating effect and impart certain characteristics to the grease.

They are decisive for the service temperature range, the pressure absorption capacity and the grease's resistance to ambient media.

Depending on the thickening agent, metal soap greases have the following properties:

G PG 00 K - 30

- lower service temperature in °C
- upper service temperature and behaviour towards water 3
- NLGI grade (see table 26, page 58)
- letter(s) identifying base oil and additives 2
- lubricating grease for: 1

1

Lubricating grease for:	
G	Closed gears DIN 51 826
OG	Open gears
K	Rolling and plain bearings, sliding surfaces DIN 52 825
M	Plain bearings and seals (lower requirements than K)

2

Additional letters identifying	
Base oil types	
E	Ester oils
FK	Fluorinated hydrocarbons
HC	Synthetic hydrocarbons
PG	Polyglycol
SI	Silicone oils
Additives	
P	Extreme pressure additives
F	Solid lubricant particles, e.g. MoS_2

3

Upper service temperature and behaviour towards water under test temperature conditions *)			
	Temperature limit	**Behaviour towards water**	**Test temperature**
C D	+60°C	0 or 1 2 or 3	40 °C
E F	+80 °C	0 or 1 2 or 3	40 °C
G H	+100 °C	0 or 1 2 or 3	90 °C
K M	+120 °C	0 or 1 2 or 3	90 °C
N	+140 °C	As required	
P	+160 °C		
R	+180 °C		
S	+200 °C		
T	+220 °C		
U	> +220 °C		
*) DIN 51 807, Pt 1, static test		0 = no change 1 = minor change 2 = moderate change 3 = major change	

Fig. 68: Denomination of lubricating greases in acc. with DIN 51 502

Aluminum complex soap greases

Greases of this type show good adherence to metal surfaces, are very resistant to water and have a good wetting behaviour due to their special rheological behaviour (they become softer under mechanical load). For this reason they are especially suitable for the lubrication of tooth flanks of large open gears with steel/steel gear wheels subject to increased loads. They are also suitable for splash lubrication.

Lithium soap greases

Lithium soap greases are water repellent, have a high drop point and, depending on the type and consistency of the base oil, very good low-temperature properties.

Li soap greases of NLGI grade 1 or 2 with a synthetic hydrocarbon base oil or a mixture of mineral and synthetic hydrocarbon oils are particularly suitable for lifetime lubrication of small gears with steel/plastic and plastic/plastic gear wheels, especially for gears with a high centrifugal acceleration, where oil/thickener separation must not occur. Greases with a low base oil viscosity permit very low starting torques. Li soap greases with a polyglycol base oil of low consistency are preferred for splash lubrication of small gears and gear motors subject to low loads, also for increased peripheral and sliding speeds. They are also suitable for the lubrication of worm gears with steel/bronze material pairings.

Calcium soap greases

Compared to other metal soap greases, Ca greases have a better flow behaviour and are more resistant to working. They also have a good pressure absorption capacity. Ca greases with a synthetic hydrocarbon base oil are suitable for lifetime lubrication of small gears with steel/plastic and plastic/plastic gear wheels subject to increased specific loads.

Due to their good flow properties Ca greases of increased consistency are suitable for splash lubrication of gears that are not oiltight with little free housing space and a quasi-complete fill.

Greases with low-viscosity base oils have good low-temperature properties and ensure low starting torques.

Barium complex soap greases

Lubricating greases with barium complex soap as a thickening agent are very resistant to working, have an excellent resistance to water, a high load-carrying capacity and very good wetting properties.

The thickener share is very high, which gives these greases anti-noise and anti-vibration properties. As their backflow behaviour is not very good, they are mainly used for lifetime lubrication of tooth flanks.

Such greases usually have a low-viscosity base oil in order to improve gear starting properties, especially in case of low input performance. Barium complex soap greases are particularly suitable for the lubrication of small gears with steel/steel components.

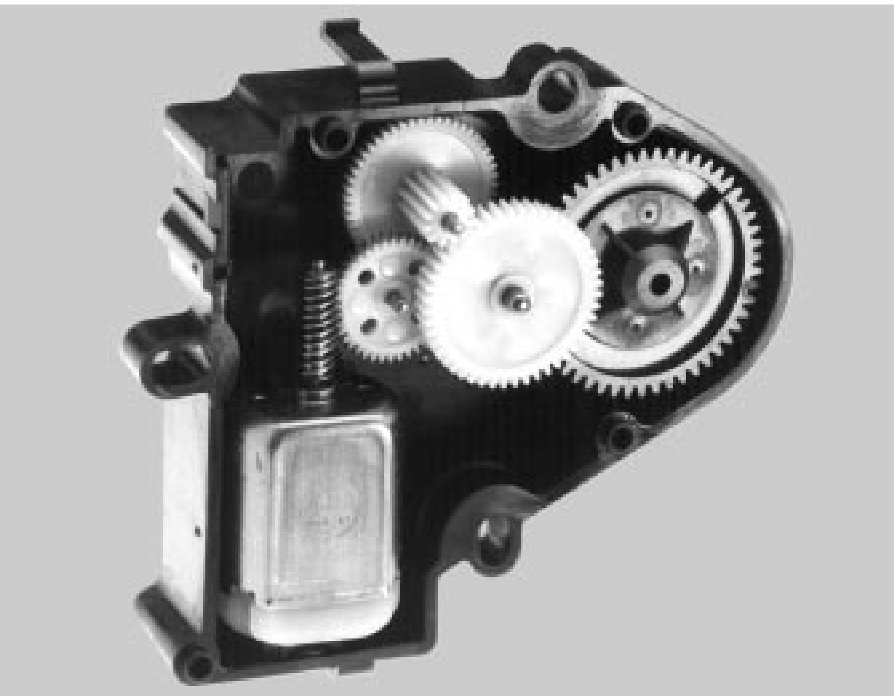


Fig. 69: Small gear motor (worm/spur gear) for flap adjustment in automotive a/c systems, manufactured by Bühler Nachfolger GmbH, Nürnberg, Germany F 10255

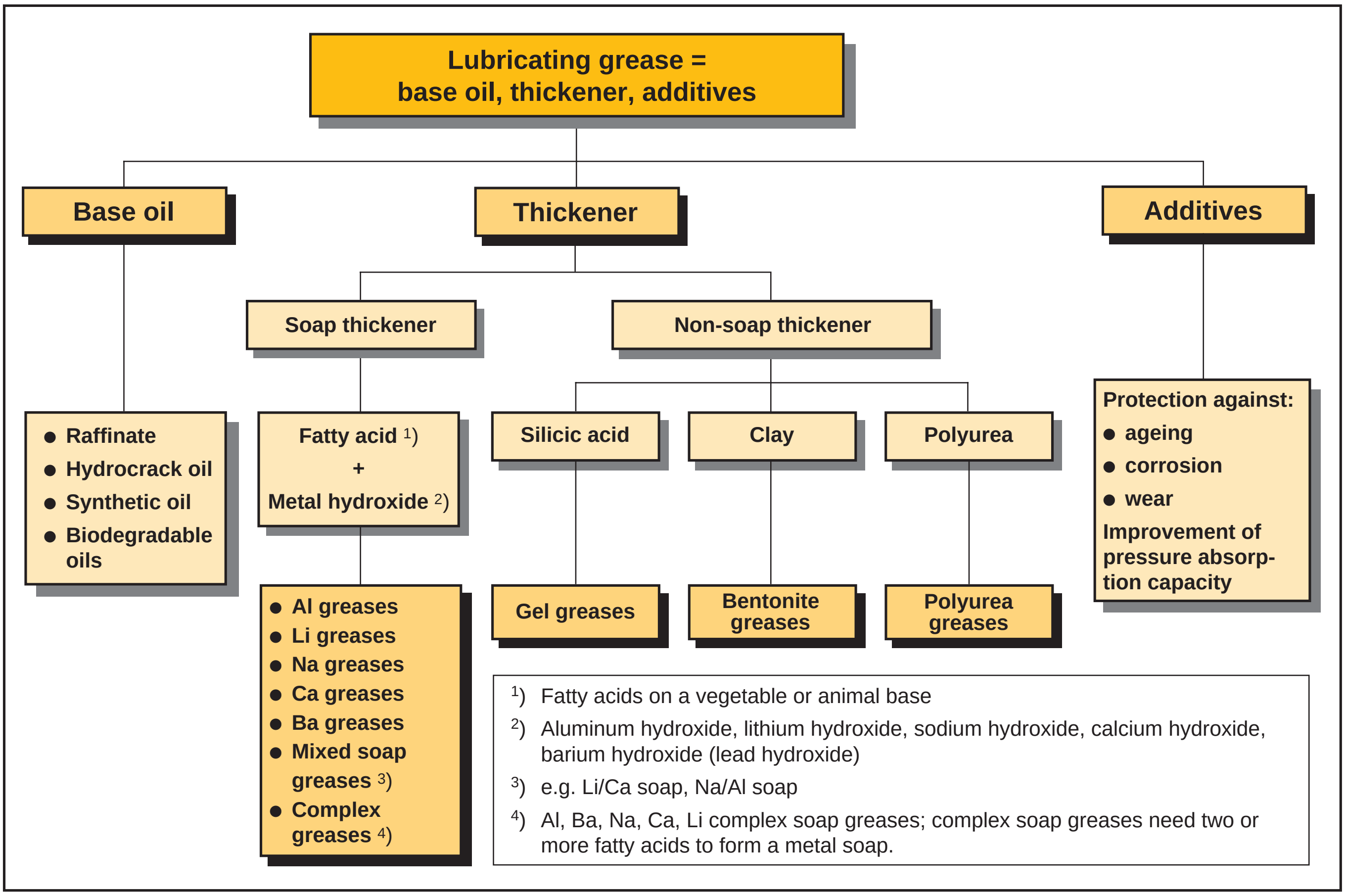


Fig. 70: Structure and manufacturing method of lubricating greases

8.4.5 Behaviour of lubricating greases towards synthetic materials

Lubricating greases are mainly used for long-term or lifetime lubrication of small and miniature gears produced in large series. In these gears synthetic materials are increasingly replacing metals in gear wheel, bearing and housing components.

However, long-term lubrication of gears with plastic components is only possible if the affected synthetic parts are lubricated as reliably as if they were made of metal.

As the interaction between plastics and greases is different than that between metals and greases, it is necessary to perform component tests under service conditions before starting production in order to ensure that the synthetic materials are sufficiently compatible with the lubricants. These practice-related tests are required because conventional compatibility tests are only carried out under static conditions with standardized test rods, disregarding

– that additives only become effective when the lubricant is subjected to tribological loads, and

– that oil ageing under service conditions is much faster than under static loads due to the formation of aggressive oil products under mechano-dynamical loads.

There are no standardized or quasi-standardized tests or guidelines to identify mechano-dynamical loads.

Describing the problems of compatibility of plastic materials and lubricants would go beyond the scope of this brochure. For more information, please refer to our technical brochure "Lubrication of synthetic materials" 9.19 e, where you will find everything you need to know about the interaction between lubricants and plastics, and which will help you select a suitable lubricant.

Table 27 on page 64 provides a general survey of the compatibility of various thermoplastic materials with specialty lubricating greases.

The data indicated in this table is only intended as a guideline and does in no way substitute specific tests!

Abbre-viation	Thermoplastics	Grease groups A	Grease groups B	Grease groups C
ABS	ABS-copolymer	not resistant	not resistant	resistant
CA	Cellulose acetate	not resistant	not resistant	resistant
PA	Polyamide	resistant	resistant	resistant
PC	Polycarbonate	not resistant	not resistant	resistant
PE	Polyethylene			
	high-density	resistant	resistant	resistant
	low-density	partly resistant	partly resistant	resistant
POM	Polyoxymethylene	mostly resistant	mostly resistant	mostly resistant
PP	Polypropylene	partly resistant	partly resistant	resistant
PPO	Polyphenylene oxide	not resistant	not resistant	resistant
PS	Polystyrene	not resistant	not resistant	resistant
PTFE	Polytetrafluoro-ethylene	resistant	resistant	resistant
PUR	Polyurethane	not resistant	not resistant	resistant
PVC	Polyvinyl chloride	not resistant	not resistant	resistant

A = Special ester oil greases with
- alkaline earth complex soap
- lithium soap
- inorganic thickener

B = Special ester oil greases with lithium soap

C = Synthetic hydrocarbon oil greases with
- alkaline earth complex soaps and other metal soap thickener

Table 27: Behaviour of special lubricating greases towards thermoplastic materials

8.5 Notes on the lubrication of small gears

There is no general definition of the term "small gear". In order to distinguish them from so-called "industrial gears", Klüber defines small gears as gears with an output torque of less than 100 Nm.

Gears with an output torque below 10 Nm are called "miniature gears".

Basically there is no difference between the lubrication of industrial and small gears.

The type of lubrication (with an oil, fluid grease or grease) and the application method (immersion, spraying, manual application) are not dependent on the size of the gear but on the type of gear and the peripheral speed of gear stage operating at the highest speed.

Type of gear	Output torque [Nm]
Industrial gear	> 100
Small gear	< 100 to 10
Miniature gear	< 10

Table 28: Classification of gears in terms of output torque (Klüber)

The same applies to the selection of a suitable lubricant, which does not depend on the gear size but on the operating and ambient conditions, the gear materials and the type of gear.

For small gears this means that a gear stage with an output torque of less than 100 Nm used for permanent power transfer (maximum peripheral speed: 18 m · s^{-1}) can only be lubricated with an oil, and not with a grease. On the one hand it is required to dissipate excessive heat, on the other hand a fluid grease of NLGI 000 is only suitable for a peripheral speed up to 8 m · s^{-1} when used in permanent operation (see *tables 17* and *26* on page 47 and 58).

The gear housing must be oiltight and the gear motor installed in a way to allow oil changes.

For the following reasons small and miniature gears are mainly grease lubricated:

– it is often not possible to make gear housings or seals absolutely oiltight,

– they are often not accessible or difficult to access, which makes it impossible to change the lubricant (long-term or lifetime lubrication required),

– they are of the open or semi-open type

– the gear position during operation varies

– lifetime lubrication is required.

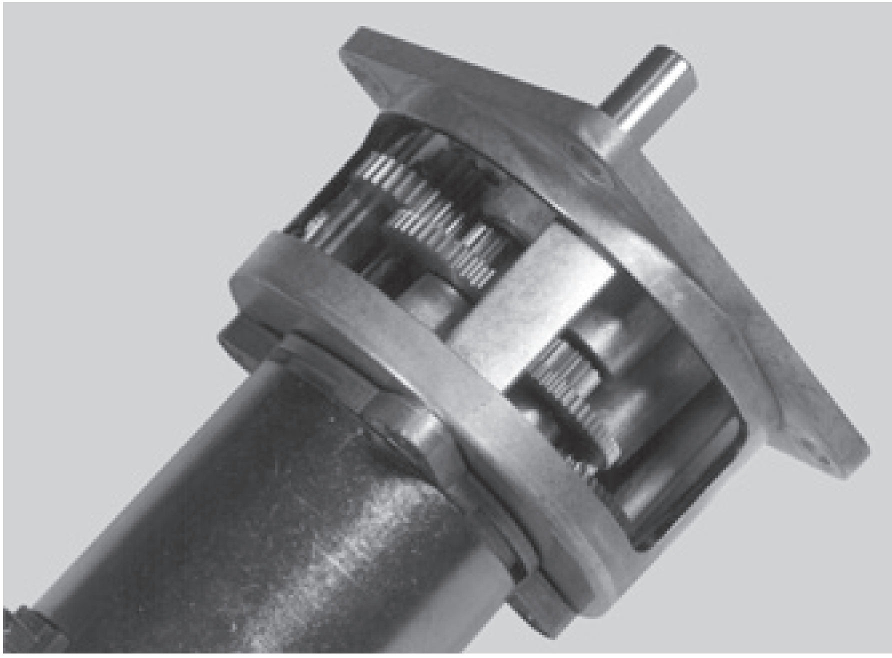


Fig. 71: Spur gear stage of a miniature gear motor for universal application (copying machines, vending machines, equipment of all types), manufactured by Bühler Nachfolger GmbH, Nürnberg, Germany F 10254

9 Tooth flank damage

More than 50 % of the damage in industrial gears occurs on the gear teeth. *Table 29* gives a survey of the different types of tooth flank damage and their causes. Damage mainly due to deficient lubrication is listed at the bottom of the table. *Fig. 72* indicates the load limits and the pertinent tooth flank damage. Tooth flank damage can manifest itself in various forms.

Based on the cause of the damage a distinction is made between

- abrasive wear
- scuffing wear
- pitting.

DIN 3979 describes the damage effects in detail.

Abrasive and scuffing wear are mainly caused by deficient lubrication, whereas the formation of pittings can only be influenced by the lubricant to a limited extent.

The lubricant does not have an immediate impact on surface fatigue. However, in case of high peripheral speeds and high lubricant viscosity in the lubricating gap, there may be a damping effect on the additional dynamic forces, thus exerting indirect influence on surface fatigue.

9.1 Abrasive wear

In the context of gear damage "abrasive wear" is a generic term including grinding, scratching and scoring wear. Abrasive tooth flank wear is caused by

- furrowing effects of surface roughness

Damage caused by	Type of damage	Tooth breakage		Tooth flank damage																							
				Wear						Chipping			Cracks				Deformation				Corrosion		Other damage				
		Overload breakage	Fatigue breakage	Normal wear	Abrasive wear	Interference wear	Scratching	Scoring	Scuffing	Pitting	Spalling	Flaking	Grinding cracks	Quenching cracks	Material cracks	Fatigue cracks	Indentations	Rippling	Hot flowing	Cold flowing	Chemical corrosion	Fretting corrosion	Cavitation	Erosion	Electric discharge	Scaling	Overheating
Material defects	Slag inclusion		●								●			●	●												
	Forging marks													●	●												
	Non-metal inclusions														●												
	Inadequate material pairing									●	●								●	●							
Deficient design	Wrong dimensions		●						●	●						●				●							
	Wrong tooth geometry					●			●																		
	Meshing interference					●																					
	Wrong backlash								●																		●
Deficient production	Deficient forging		●																								
	Excessive heat during mechanical treatment										●		●														
	Inadequate heat treatment										●		●	●						●						●	
	Inadequate surface quality			●				●		●																	
Faulty as-sembly	Mis-alignment			●					●	●	●					●											
	Insufficient insulation																								●		
Operating conditions	Frequent load alternations		●							●																	
	Shocks, vibrations	●	●							●		●						●		●		●	●				
	Overloading	●						●	●	●		●				●		●		●							●
	Inadequate running in								●	●																	
	Peripheral speed too high / low							●	●	●										●							
Deficient lubrication	Insufficient lubricant quantity			●														●	●	●		●					●
	Wrong viscosity			●				●	●	●								●				●					●
	Insufficient quality			●				●	●	●								●			●	●					
	Solid / fluid contaminants				●		●	●									●				●		●	●			
	Insufficient surface quality																							●			

Table 29: Causes and types of gear damage

- solid contaminants in the lubricant (dust, rust, scale, metal wear particles, molding sand residues, etc.) which reach the lubricating gap and result in grooves in the tooth flanks.

The degree of contamination of a lubricant and the extent of surface roughness of the gear material are decisive for the speed of wear. Wear usually occurs most rapidly on the tooth tip and the base.

9.2 Scuffing (adhesive) wear

Scuffing wear occurs if the lubricant film is destroyed by excessive temperatures or loads, causing the metal tooth flank surfaces to touch. Gears are susceptible to scuffing wear if they operate at high peripheral speeds (hot scuffing) or at low peripheral speeds and high flank pressure (cold scuffing)

Hot scuffing

In case of high friction temperatures and sliding speeds the lubricant film fails due to excessive heating.

Cold scuffing

The lubricant film fails due to high local flank pressure at low peripheral speeds (up to approx. 4 m · s^{-1}). Cold scuffing mainly occurs in heat-treated gears with a rough tooth surface.

Scuffing is characterized by local seizures which are torn apart by the relative movement of the tooth flanks, causing rough spots of varying widths and depths (individual spots or over the entire flank width) on the tooth flanks (*Fig. 74*). The particles removed from the tooth surface are carried along and deposited on other spots, where they are subject to plastic deformation. Scuffing wear is especially pronounced at the tooth tip and base because this is where the sliding speed is highest.

Practical experience shows that a gear's scuffing load capacity depends on four main parameters which have an impact on its performance limit:

1. Lubricant
2. Tooth geometry
3. Surface, material
4. Operating conditions

Table 30 shows the range of variation of the transferable torque (hot scuffing limit) as a function of these parameters. Contrary to pittings and tooth breakage, there is no *fatigue strength* as far scuffing wear is concerned. Even a short period of overloading can initiate scuffing wear and damage the tooth flanks.

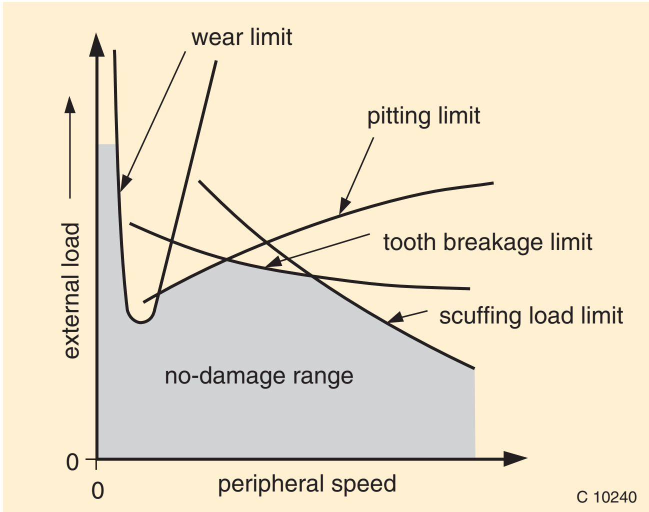


Fig. 72: Schematic drawing of load limits of hardened steel gears. The location of the curves relative to each other and their gradients are exemplary and do not pertain to a certain type of gear

It is therefore important to take into account a gear's scuffing resistance in the pertinent design calculations.

DIN 3990, ISO DP 6336 provides information on how to calculate a gear's scuffing load limit and scuffing wear resistance.

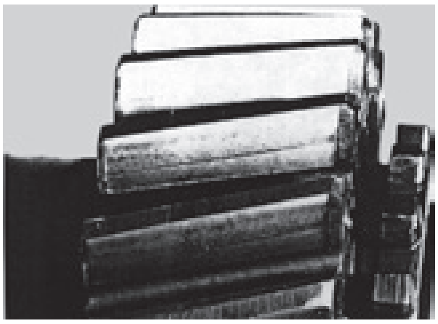


Fig. 73: Abrasive wear C 10241

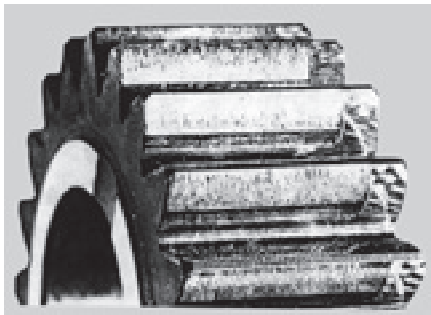


Fig. 74: Scuffing (adhesive wear) C 10242

9.3 Pitting

Pitting is a form of tooth flank material fatigue. It occurs if a material's permanent rolling resistance is exceeded by loads applied locally or over the entire tooth width.

The damage manifests itself in the form of flat, crater-like indentations called "pittings". Their size varies from 0.01 mm (micro-pitting) to several mm. In spur gears they mainly occur in the area below the pitch circle. Lubricants can have a limited effect on the formation of pittings, but an increased operating viscosity retards their development.

For measures to increases a gear's pitting load capacity see *Fig. 76*.

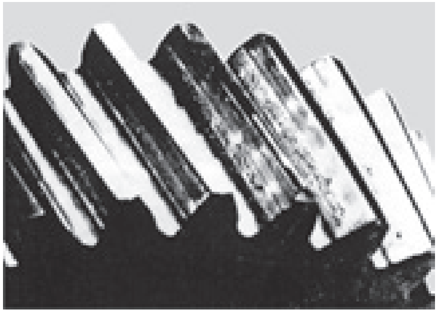


Fig. 75: Pittings C 10243

	Parameters	Range of variation of the transferable torque
1	Additives, same base oil viscosity	**1 : 5**
	Oil viscosity, e.g. doubled	**up to 1 : 1.5**
	Service oil temperature, e.g. reduced by 20 °C	**1 : 1.2**
2	Tooth geometry	**1 : 1.6**
	Tip relief	**1 : 1.5**
3	Surface roughness, e.g. reduction of flank roughness to 1/16	**1 : 2**
	Phosphatized flanks	**1 : 1.4**
	Nitriding	**1 : 1.8**
4	Peripheral speed	**1 : 2.5**

*Table 30: Range of variation of the transferable torque (hot scuffing limit) .
Change of main parameters: 1 = lubricant, 2 = teeth, 3 = material surface
4 = operating conditions
The individual parameters cannot be combined to obtain synergistic effects.*

9.4 Micro-pitting

Micro-pitting is a lubricant related damage manifesting itself in the form of dull spots on the tooth flanks of surface-hardened gear wheels.
It is characterized as follows:

- dull areas consisting of numerous chippings of microscopic size (depth approx. 20 µm),
- direction and development of cracks depending on sliding/rolling conditions (as in the case of "conventional" pittings), i.e. the cracks start on the surface and, at a flat angle, develop in the direction of the frictional force,
- occurs mainly at medium peripheral speeds when low-viscosity oils are used on the tooth flanks of hardened gear wheels,
- the operating time until micro-pitting occurs may be quite short and the loads may be significantly below the pitting load capacity,
- the incipient damage is wear-like, the progressive damage is fatigue-like,

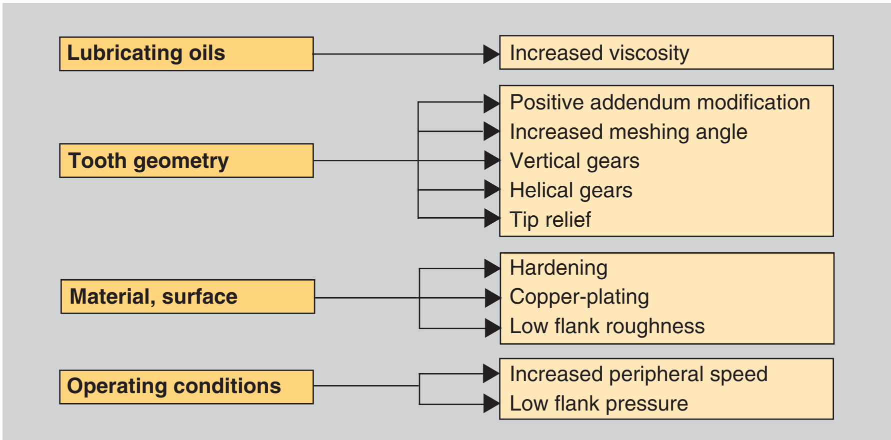


Fig. 76: Measures to increase a gear's pitting load capacity

- lubricant viscosity and additives have a great impact on micro-pitting,
- lubricants with a low micro-pitting resistance may lead to erosion in the micro-pitting area, resulting in increased internal dynamic forces and gear noises as well as a reduced pitting load capacity.

The impact of a lubricant on micro-pitting is determined in a modified FZG test based on DIN 51 354 with gear wheels less susceptible to scuffing. The load is increased in

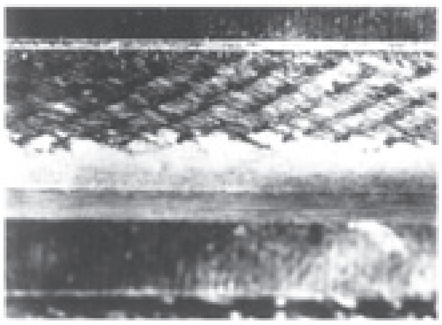


Fig. 77: Micro-pitting of case-hardened gear C 10244

steps until the dimensional change of the flank is twice as high as in the beginning. If this damage criterion is only reached in the 8th or an even higher load stage, the oil generally provides sufficient resistance to micro-pitting.

To confirm the result and obtain information of the course of the damage (which may be degressive) over extended periods of operation, the progressive load test is followed by an endurance test with 10 ... 50 · 10^6 load alternations.

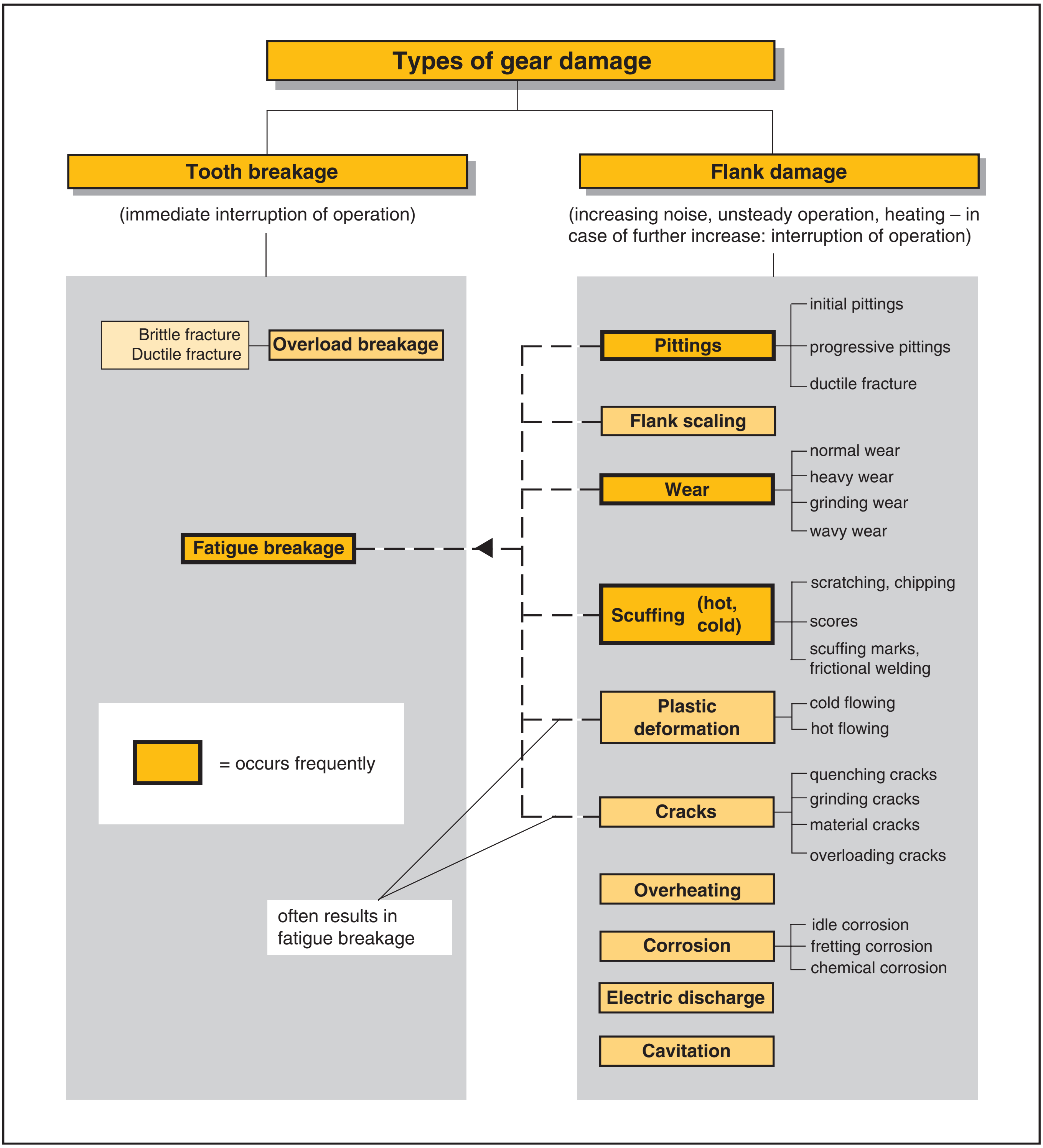


Fig. 78: Types of gear damage

Additional literature:

(1) Bartz W.J.: Getriebeschmierung, Expert publishing house, 1989

(2) Dubbel: Taschenbuch für den Maschinenbau, Springer publishing house, 16th edition, 1987

(3) Klüber Lubrication: Lubrication of large girth gear drives, edition 07.1993

(4) Niemann G., Winter H.: Maschinenelemente, Vol. II, Springer publishing house, 1985

(5) Wollhofen G.P. and 17 co-authors: Getriebeschmierung in der Anlagentechnik, Expert publishing house, 1990

Sources of photographs (numbers in parentheses refer to the additional literature indicated above):

Fig. 1: Private photo/Klüber
Fig. 3, 4: from (4), page 4
Fig. 5: Source unknown
Fig. 6,7,8,9: from (4), page 4
Fig. 10: from (2), G 128
Fig. 11: (1), page 9
Fig. 12: (4), page 37
Fig. 13: from (2), G 138
Fig. 14: Sartorius
Fig. 15: Freudenberg
Fig. 16: Freudenberg
Fig. 17: Klüber
Fig. 18,12,20,21,22,12: GfT Worksheet 2.42.
Fig. 24: from (1), page 15
Fig. 25: from (1), page 16
Fig. 26: from (1), page 17
Fig. 27: Klüber
Fig. 28: Schmiertechnik und Tribologie, 1973, No. 5, page 148
Fig. 29: Klüber
Fig. 30: Klüber
Fig. 32: acc. to P. Walter
Fig. 35, 36: Klüber
Fig. 38: acc. to Ohlendorf
Fig. 39: GfT Worksheet 2.4.2
Fig. 42: Klüber
Fig. 43: acc. to GfT Worksheet 2.4.2
Fig. 44: acc. to GfT Worksheet 2.4.2
Fig. 46: Klüber
Fig. 48: Klüber
Fig. 49: Klüber
Fig. 50: Klüber
Fig. 51: VOITH TURBO gear brochure
Fig. 53: Klüber
Fig. 54: Klüber
Fig. 58,59,60: GfT Worksheet 2.4.2
Fig. 61: Klüber
Fig. 62: Klüber
Fig. 63: Klüber
Fig. 64: Bauer Getriebe, Esslingen/Klüber
Fig. 65: FLS
Fig. 66: Klüber
Fig. 70: ÖMV brochure: Schmierstoffe für Kraftfahrzeuge, 1993
Fig. 72,73,74,75: GfT Worksheet 2.4.2
Fig. 76: Klüber
Fig. 77: Forschungsstelle für Zahnräder und Getriebebau (FZG), TU München
Fig. 78: acc. to Bartz

Product survey

Gear oils

Gear greases for industrial gears

Gear greases for small gears

Lubricant / Parameters	ISO VG DIN 51 519	Service temperature range* °C, approx. in case of splash lubrication	Type of oil	Properties, Application notes (for detailed information, please see product information leaflet)
Klüberoil GEM 1-N series	46–680	-15 to 100	mineral oil	High performance gear and multi-purpose oils which meet CLP requirements. FZG test A/8.3/90, scuffing load stage ≥ 12. High micropitting load capability.
Klübersynth GEM 4-N series	32–680	-45 to 140	synthetic hydrocarbon oil	Synthetic high-performance gear oils for spur, bevel and worm gears with excellent low-temperature behaviour. They meet CLP and AGMA requirements. Miscible with mineral oils. Compatible with common sealing materials and paints. FZG test A/8.3/90, scuffing load stage > 12. High micropitting load capability.
Klübersynth EG 4-series	150–1000	-35 to 140	synthetic hydrocarbon oil	Synthetic high-performance gear oils for spur, bevel and worm gears with excellent low-temperature behaviour. They meet CLP and AGMA requirements. Miscible with mineral oils. Compatible with common sealing materials and paints. FZG A10/16.6R/90, scuffing load stage > 12. API GL 4.
Klüberoil 4 UH 1N series	32–1500	-30 to 110	synthetic hydrocarbon oil	Food grade lubricants authorized in accordance with USDA-H1. Especially suitable for the lubrication of spur, bevel and worm gears used in the food processing and pharmaceutical industries. They meet CLP. Compatible with common sealing materials and paints. FZG test A/8.3/90, scuffing load stage > 12.
Klübersynth GH 6-series	32–1000	-45 to 160	polyglycols	High-temperature gear oils with a high scuffing load capacity and good wear protection, especially suitable for the lubrication of worm gears. FZG test A/16.6/140, scuffing load stage > 12. Not miscible with mineral oils. Compatibility tests with sealing materials and paints are required.
SYNTHESO D … EP-series	68–1000	-35 to 100	polyglycols	Synthetic high-performance gear oils for spur and bevel gears operating under extreme loads, also suitable for the lubrication of hypoid gears. Oils of ISO VG 220 and 460 pass the FZG-L-42 test, API GL 5. High micropitting load capability. Not miscible with mineral oils. Compatibility tests with sealing materials and paints are required.
Klüberbio CA 2-460	460	-20 to 110	biodegradable ester	Especially suitable for the lubrication of worm gears which are used in applications where leaking lubricant may be dangerous to the environment. Biodegradability acc. to CEC-L-33-A-94 after 21 days > 70%.
Klübersynth GEM 2-series	220–320	-25 to 110	biodegradable ester	Synthetic high-performance gear oils for spur, bevel and worm gears. They meet CLP requirements. Miscible with mineral oils. Compatible with common sealing materials and paints. FZG test A/8.3/90, scuffing load stage ≥ 12. High micropitting load capability. Biodegradability acc. to CEC-L-33-A-94 after 21 days > 70%.

* see page 78

Product survey

Gear greases for industrial gears

Lubricant / Parameters	Base oil/ thickener	NLGI Grade	Service temperature range* °C, approx. in case of splash lubrication	Colour	Use in industrial gears	Use in small gears	Properties, Application notes (for detailed information, please see product information leaflet)
MICROLUBE GB 0 and 00	mineral oil/ silicate	0 / 000	-10 to 150	reddish brown	x	x	Fluid gear grease for splash lubrication of gear systems subject to increased loads at normal temperatures: FZG test A/2.76/50, load stage > 12.
Klüberplex GE 11-680	mineral oil/ Al complex	0 / 00	-10 to 140	brown	x	x	Noise dampening adhesive grease for spray lubrication of open gears, also suitable for splash lubrication of gearboxes operating at normal temperatures. FZG test A/2.76/50, load stage > 12.
UNIGEAR ST 2 M	mineral oil/ sodium/MoS_2	0 / 1	-10 to 150	dark brown	x		Gear grease for open and closed spur gears of larger dimensions subject to high loads. FZG test A/8.3/90, load stage > 12.
Klübersynth GE 46-1200	polyglycol/ lithium	00	-30 to 120	brown	x	x	Synthetic semi fluid grease for splash lubrication at low and high temperatures. For high loads as well as long-term and lifetime lubrication, up to 8 m/s pheriperal speed. FZG test A/8.3/90, load stage > 12. Not suitable for gears made of plastics or aluminium alloys.
ISOFLEX TOPAS NCA 5051	synth. hydro-carbon oil/ special Ca	0	-30 to 150	yellowish		x	Noise dampening grease for splash lubrication open gears that are not oil tight. Suitable for flank greasing. Especially suitable for gears made of plastic components.
ISOFLEX TOPAS L 32	synth. hydro-carbon oil/ lithium	2	-60 to 130	beige		x	Low temperature grease for tooth flanks with a wide service temperature range. Especially suitable for plastic/plastic and plastic/steel wheels. Ensures low starting torques and smooth operation at low temperatures. Meets specifications of various automobile manufacturers.
ISOFLEX TOPAS NCA 52	synth. hydro-carbon oil/ special Ca	2	-50 to 150	beige		x	Lubricating grease with a wide service temperature range for tooth flanks and lifetime lubrication of small gears subject to increased loads. Especially suitable for plastic/plastic and plastic/steel gear wheels. Low base oil viscosity, thus low starting torques at low temperatures.

* see page 78

Lubricant / Parameters	Base oil/ thickener	NLGI Grade	Service temperature range* °C, approx. in case of splash lubrication	Colour	Use in industrial gears	Use in small gears	Properties, Application notes (for detailed information, please see product information leaflet)
Klübersynth GE 14-151	synth. hydro-carbon oil/ Al complex	1	-35 to 140	yellow		x	Lubricating grease for the lubrication of gears bearings, joints in small gears subject to high loads, e.g. power tools. Suitable for applications with a high sliding percentage and peripheral speed of 20 m/s at short term operations.
Klübersynth UH1 14-151	synth. hydro-carbon oil/ Al complex	1	-45 to 120	beige	x		Food grade lubricant approved in acc. to USDA-H1 grease for the food processing and pharmaceutical industries. Excellent protection against wear, good resistance to corrosion, very good low-temperature behaviour.
Klübersynth UH1 14-1600	synth. hydro-carbon oil/ Al complex	00	-45 to 120	yellowish	x		Food grade lubricant authorized in acc. USDA-H1. Semi fluid grease for the food processing and pharmaceutical industries. Excellent protection against wear, good resistance to corrosion, very good low-temperature behaviour. FZG test A/2.76/50 load stage 12.

* see page 78

Product survey

Gear greases for small gears

Parameters / Lubricant	Base oil/ thickener	Con-sistency grade DIN 51 818	Worked penetration DIN ISO 2137 (0.1 mm) ≈	Service temperature range* °C, approx. in case of splash lubrication	Density DIN 51 757 (g/cm^3) at 20 °C ≈	Base oil viscosity DIN 51 562 (mm^2/s), at °C ≈ 40	Base oil viscosity DIN 51 562 (mm^2/s), at °C ≈ 100	Colour	Drop point DIN ISO 2176 (°C) ≈	Speed factor** $n \cdot d_m$ (mm/min)	Apparent viscosity Klüber viscosity range***	Properties Application notes
Mineral oil base lubricating greases												
MICROLUBE GB 00	Mineral oil/ silicate	**00 / 000**	430 to 475	-10 to 150	0.90	700	35	reddish brown	–	–	EL	Fluid gear grease for splash lubrication of spur and bevel gears operating in the normal temperature range and subject to high loads. Good protection against wear and corrosion. Scuffing load stage >12 in the special FZG test A/2.76/50 and a work-related change in weight < 0.2 mg/kWh.
Klüberplex GE 11-680	Mineral oil/ Al complex	**0 / 00**	380 to 420	-10 to 140	0.94	680	35	brown	> 160	–	L	Noise-damping adhesive grease for the lubrication of tooth flanks of open gears of larger dimensions (module > 3 mm) subject to increased loads. Also suitable for splash lubrication. To be applied in the normal temperature range on steel/steel gear wheels. Scuffing load stage >12 in the special FZG test A/2.76/50.

*, ** and *** see page 78

Product survey

Gear greases for small gears

Parameters / Lubricant	Base oil/ thickener	Con-sistency grade DIN 51 818	Worked penetration DIN ISO 2137 (0.1 mm) ≈	Service temperature range* °C, approx.	Density DIN 51 757 (g/cm^3) at 20 °C ≈	Base oil viscosity DIN 51 562 (mm^2/s), at °C ≈ 40	100	Colour	Drop point DIN ISO 2176 (°C) ≈	Speed factor** $n \cdot d_m$ (mm/min)	Apparent viscosity Klüber viscosity range***	Properties Application notes
Lubricating greases with a mineral base oil												
MICROLUBE GB 0	Mineral oil/ silicate	**0**	355 to 385	-25 to 150 in case of splash lubrication	0.90	400	25	reddish brown	> 180	–	L	Gear grease for splash lubrication or tooth flank greasing of small gears subject to high loads. Good protection against wear and corrosion. Scuffing load stage >12 in the special FZG test A/2.76/50 and work-related change in weight < 0.2 mg/kWh.
Lubricating greases with a synthetic base oil												
Klübersynth GE 46-1200	Polyglycol/ lithium	**00**	400 to 430	-30 to 120	0.99	120	20	brown	> 160	–	EL	Fluid gear grease for splash lubrication e.g. gear motors subject to high loads (up to 8 m/s peripheral speed) with gear stages in the form of spur, bevel and worm gears. Not suitable for gear wheels made of synthetic materials or alluminum alloys. Can be applied on miniature gears. Scuffing load stage >12 in the FZG test A/8.3/90, DIN 51 354, Pt 2. *, ** and *** see page 78

Product survey

Gear greases for small gears

Parameters / Lubricant	Base oil/ thickener	Consistency grade DIN 51 818	Worked penetration DIN ISO 2137 (0.1 mm) ≈	Service temperature range* °C, approx. in case of splash lubrication	Density DIN 51 757 (g/cm^3) at 20 °C ≈	Base oil viscosity DIN 51 562 (mm^2/s), at °C ≈ 40	100	Colour	Drop point DIN ISO 2176 (°C) ≈	Speed factor** $n \cdot d_m$ (mm/min)	Apparent viscosity Klüber viscosity range***	Properties Application notes
Lubricating greases with a synthetic base oil												
ISOFLEX TOPAS NCA 5051	Synthetic hydro-carbon oil/ special Ca	**0 / 00**	385 to 415	-50 to 140	0.85	30	6	beige	> 180	1 000 000	EL	Noise-damping grease for splash lubrication of gears that are not oiltight. Suitable for flank greasing. Especially suitable for gears with plastic components. Excellent low-temperature behavioer; low starting torques possible.
Klübersynth G 34-130	Synthetic hydro-carbon oil/ mineral oil/ special Ca	**0**	355 to 385	-30 to 150 in case of splash lubrication	0.87	130	15.5	beige, brownish	> 180	–	EL	Special grease for small gears, e.g. in power hand tools and equipment for craftsmen. Suitable for splash lubrication (up to 5 m/s peripheral speed) and flank greasing (up to 4 m/s). Wide service temperature range. Especially suitable for plastic/plastic and plastic/ steel gears.

*, ** and *** see page 78

Parameters / Lubricant	Base oil/ thickener	Con-sistency grade DIN 51 818	Worked penetration DIN ISO 2137 (0.1 mm) ≈	Service temperature range* °C, approx.	Density DIN 51 757 (g/cm^3) at 20 °C ≈	Base oil viscosity DIN 51 562 (mm^2/s), at °C ≈ 40	Base oil viscosity 100	Colour	Drop point DIN ISO 2176 (°C) ≈	Speed factor** $n \cdot d_m$ (mm/min)	Apparent viscosity Klüber viscosity range****	Properties Application notes
Lubricating greases with a synthetic base oil												
ISOFLEX TOPAS L 32	Synthetic hydro-carbon oil / Li	**2**	265 to 295	-60 to 130	0.88	17.5	3.8	beige	> 185	1 000 000	L	Low-temperature grease for tooth flanks with a wide service temperature range. Especially suitable for plastic/plastic and plastic/steel wheels. Ensures low starting torques and smooth operation at low temperatures. Meets specifications of various automobile manufacturers.
ISOFLEX TOPAS NCA 52	Synthetic hydro-carbon oil / special Ca	**2**	265 to 295	-50 to 150	0.89	30	5.5	beige	> 220	1 000 000	L / M	Lubricating grease with a wide service temperature range for tooth flanks and lifetime lubrication of small gears subject to increased loads. Especially suitable for plastic/plastic and plastic/steel gear wheels. Low base oil viscosity, thus low starting torques at low temperatures.
Klübersynth GE 14-151 Klübersynth UH1 14-151	Ester oil / polyurea	**2 / 3**	250 to 280	-40 to 180	0.96	70	9.4	beige	> 250	600 000	L	High-temperature lubricating grease with a wide service temperature range for flank greasing. Ensures low driving torques at low ambient temperatures (smooth operation). *, ** and *** see page 78

Notes

* Service temperatures are guide values which depend on the lubricant's composition, the intended use and the application method. Lubricants change their consistency, apparent dynamic viscosity or viscosity depending on the mechano-dynamical loads, time, pressure and temperature. These changes in product characteristics may affect the function of a component.

** Speed factors are guide values which depend on the type and size of the rolling bearing type and the local operating conditions, which is why they have to be confirmed in tests carried out by the user in each individual case.

*** Klüber viscosity grades:
EL = extra light lubricating grease;
L = light lubricating grease; M = medium lubricating grease; S = heavy lubricating grease; ES = extra heavy lubricating grease

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Klüber products are continually improved. Therefore, Klüber Lubrication reserves the right to change all the technical data in this technical brochure at any time without notice.