Industrial Lubrication and Tribology

Influence factors on gearbox power loss

Klaus Michaelis Bernd-Robert Höhn Michael Hinterstoißer

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Klaus Michaelis, Bernd-Robert Höhn and Michael Hinterstoißer
Gear Research Centre FZG, Technische Universitaet München, Garching, Germany

Abstract

Purpose – Besides other approaches, fuel savings in automotive applications and energy savings, in general, also require high-efficiency gearboxes. Different approaches are shown regarding how to further improve gearbox efficiency. This paper aims to address these issues.

Design/methodology/approach – The paper takes the following approach: theoretical and experimental investigations of bearing arrangements and gear design as well as lubricant type and lubricant supply to the components lead to efficiency optimisation.

Findings – No-load losses can be reduced, especially at low temperatures and part-load conditions when using low-viscosity oils with a high viscosity index and low oil immersion depth or low spray oil supply of the components. Bearing systems can be optimised when using more efficient systems than cross-loading arrangements with high preload. Low-loss gears can contribute substantially to load-dependent power loss reduction in the gear mesh. Low-friction oils are available for further reduction of gear and bearing mesh losses. All in all, a reduction of the gearbox losses in an average of 50 per cent is technically feasible.

Originality/value – Results from different projects of the authors and from the literature are combined to quantitatively evaluate the potential of power loss reduction in gearboxes.

Keywords Gearing, Power measurement

Paper type Research paper

Introduction

Future energy shortages have to be fought not only with exploitation of new renewable energy resources but also with reduction of energy consumption in all technical fields.

For automotive applications, optimisation attempts are made in all operating areas and for all components of vehicles to achieve minimum fuel consumption. Weight reduction and thermal management are possible approaches, as well as hybrid systems and mechanical and software features for high-efficient engines. Power loss reduction at the end of the power train has a large impact on the overall optimisation, although absolute efficiency in gearboxes and rear axles is already high (Xu *et al.*, 2007). However, 1 kW savings in the gearbox means 4 kW savings in fuel energy.

Looking at wind turbines as a growing market for alternative energy production, a modern equipment of the 5 mW class consists of eight or more gear meshes and more than 12 bearing meshes. A reduction in the overall losses by 50 per cent would save some 200 kW power losses per wind turbine unit.

The challenges are, therefore, a substantial power loss reduction with only minor impact on load-carrying capacity, component size and weight and noise generation. Adequate compromises have to be proposed.

Basic considerations

Power loss in a gearbox consists of gear, bearing, seal and auxiliary losses (Figure 1). Gear and bearing losses can be separated into no-load losses, which occur even without power transmission, and load-dependent losses in the contact of the power transmitting components. Besides operating conditions and internal housing design, no-load losses are mainly related to lubricant viscosity and density, as well as immersion depth of the components of a sump lubricated gearbox (Changenet and Velex, 2007). Load losses depend on transmitted load, coefficient of friction and sliding velocity in the contact areas of the components.

For nominal power transmission, the load losses of the gear mesh are typically dominant. For part load and high speed, high no-load losses dominate the total losses. For an optimisation of the whole operating range of a gearbox, load losses and no-load losses have to be addressed. In the following sections, the major contributors to gearbox power losses, namely, bearings and gears are considered.

Bearing power loss

No-load bearing losses depend on bearing type and size, bearing arrangement, lubricant viscosity and supply. Figure 2 shows a comparison of the no-load losses of different bearing types for same load capacity C = 20 kN (Wimmer *et al.*, 2003). Lowest no-load losses of radial bearings are expected for cylindrical roller bearings. Also, the low values of taper roller bearings are valid for unloaded bearing arrangements; however, for the typical cross-loading bearing arrangement, axial preloading is required. This requirement of preload in a cross-locating bearing arrangement with taper roller bearings increases the no-load losses substantially.

Load-dependent bearing losses depend also on bearing type and size, load and sliding conditions in the bearing and on the lubricant type (Wimmer *et al.*, 2003). Figure 3 shows load-dependent losses of bearings with same load capacity C = 20 kN

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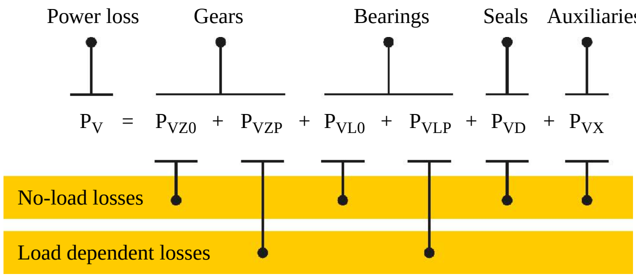
Industrial Lubrication and Tribology
63/1 (2011) 46–55
 [ISSN 0036-8792]
[DOI 10.1108/00368791111101830]

This paper was presented during the 3rd International Conference on Integrity, Reliability and Failure – IRF'2009, within the Symposium on Integrity, Reliability and Failure of GEARS and TRANSMISSIONS, held at Faculdade de Engenharia da Universidade do Porto, Portugal, July 20-24, 2009.

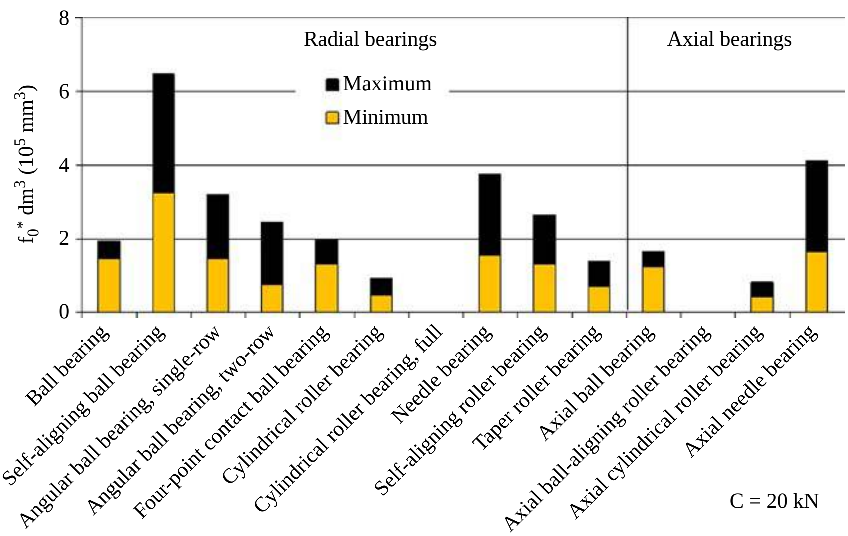
Figure 1 Composition of transmission power loss



and same utilisation ratio $P_0/C = 0.1$. Again, cylindrical roller bearings show the lowest power loss of radial bearings. Taper roller bearings for same load capacity have also low-load power loss due to small diameters for the same load capacity.

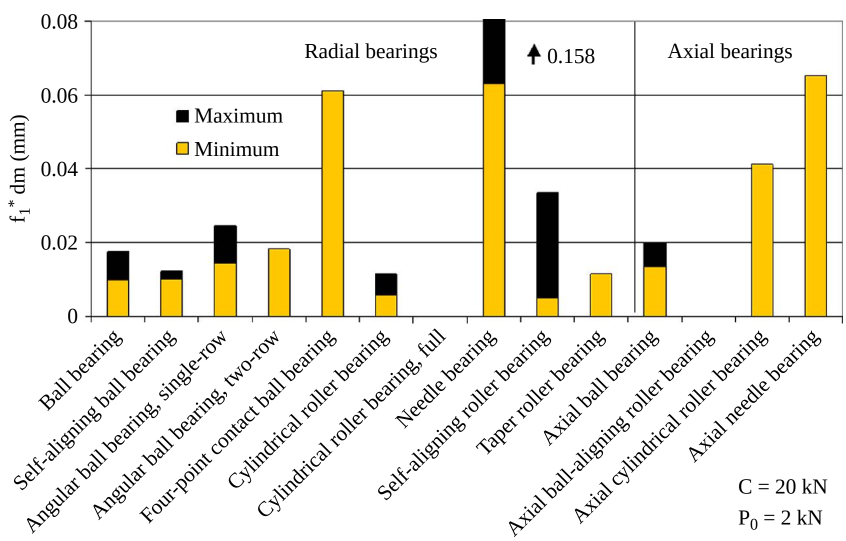
A comparison of the bearing losses for the sixth gear in a manual transmission of a middle class car for original design with preloaded cross-locating taper roller bearing arrangement and alternative design with locating four-point contact ball bearings and non-locating cylindrical roller bearings on the gearbox shafts and cross-locating angular contact ball bearings of the final drive wheel (Figure 4) was calculated according SKF-GRUPPE (Hrsg.) (2004). For medium load and medium speed conditions at low gear oil temperatures of 40°C, relevant for the new European drive cycle (NEDC), a reduction of the bearing losses of more than 50 per cent was found for the alternative design, because of the preload on the cross-locating taper roller bearings. At high gear oil temperatures of 90°C, where the preload is reduced to almost zero, the bearing loss reduction is still around 20 per cent for the alternative design (Figure 5).

Figure 2 Influence of bearing type on no-load losses



Source: Wimmer *et al.* (2003)

Figure 3 Influence of bearing type on load losses



Source: Wimmer *et al.* (2003)

Figure 4 Alternative bearing design in a manual transmission with final drive

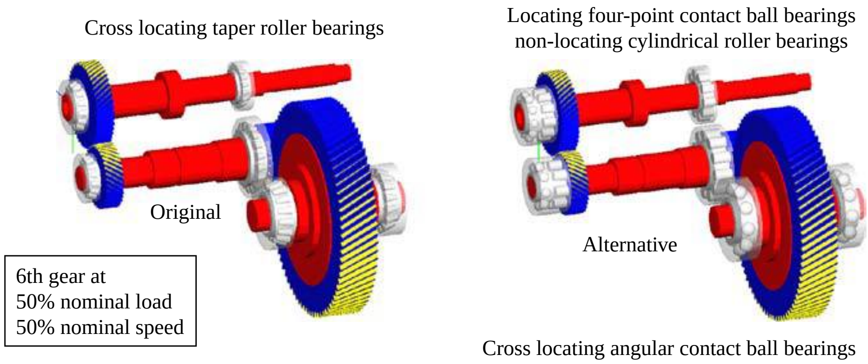
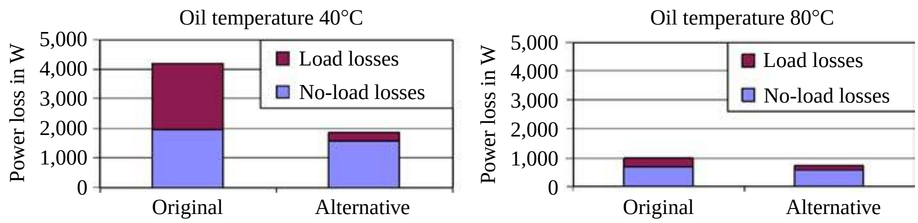


Figure 5 Influence of design and operating temperature on bearing losses



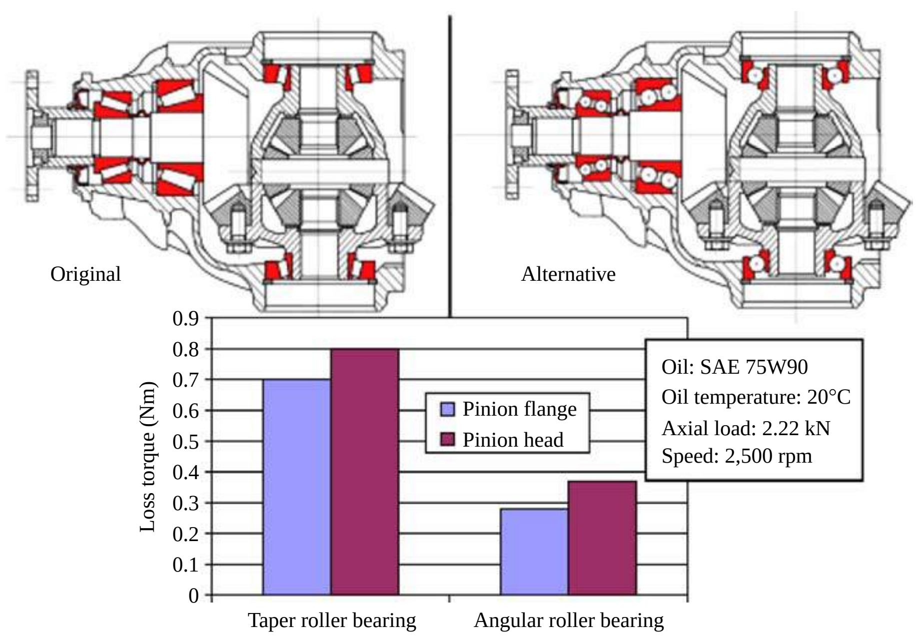
von Petery (2004) reports power loss measurements of the original bearing design of a BMW axle with cross-locating taper roller bearing arrangement and an alternative design with cross-locating double- and single-row angular ball bearing arrangement (Figure 6). For medium load and speed and low temperatures, relevant in the NEDC, the bearing loss reduction for the alternative design was over 50 per cent.

Gear power loss

No-load gear losses

Besides operating conditions, no-load gear losses mainly depend on immersion depth in sump lubricated gearboxes as well as on lubricant viscosity. Otto (2009) investigated systematically the influence of oil immersion depth in a sump lubricated test gearbox.

Figure 6 Influence of bearing type in the BMW rear axle

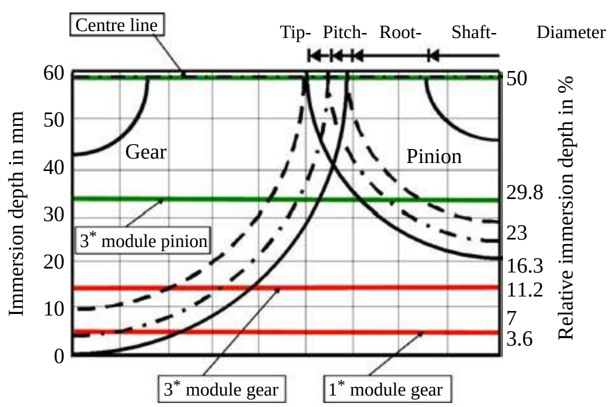


Source: von Petery (2004)

Compared to the reference oil level at shaft centre line, three times module at pinion (3 *m pinion) with pinion and gear immersed in oil, three times module at gear (3 *m gear) as well as one time module at gear (1 *m gear) with only the gear immersed in oil were investigated. The situation in the test gearbox for the different oil levels is shown in Figure 7. The test gearbox was equipped with transparent front and top covers to visualize the oil churning in the test gearbox at different conditions of oil level, pitch line velocity and sense of rotation. Figure 8 shows the distribution of an ATF ISO VG 32 at room temperature in the test gearbox at medium speed of v = 8.3 m/s and outward rotation. The reduction of churning losses with reduced immersion depth is clearly visible.

No-load loss measurements at pitch line velocities v = 8.3 and v = 20 m/s with a mineral oil ISO VG 100 at oil temperatures of 90°C and 120°C showed a substantial reduction of the gear no-load losses with decreased immersion depth (Figure 9). As expected, the effect is higher at high-speed conditions compared to lower speeds. However, in both speed conditions, the churning losses can be reduced by more than 50 per cent when the immersion depth is reduced from centre line to three times module of the gear.

Figure 7 Immersion depth in test gearbox



Source: Otto (2009)

In contrary to the beneficial effect of churning loss reduction with reduced immersion depth, the detrimental effect of reduced cooling of the gear mesh has to be considered. Figure 10 shows measured pinion bulk temperatures at different immersion depths. For high loads and high speeds, the bulk temperature may even exceed the tempering temperature of the case carburised material. A substantial reduction of the load-carrying capacity has then to be expected.

There are different opinions of the influence of lubricant viscosity on no-load gear losses. Terekhov (1975) reports increasing gear churning losses for increasing gear oil viscosities when using relatively high-viscosity oils (Figure 11). Michaelis and Winter (1994) confirm increasing gear churning losses with increasing lubricant viscosity, independent of the oil type (Figure 12), also for low operating viscosities. Depending on the operating conditions, a change from, for example, ISO VG 150 to VG 100 can reduce the no-load power losses by some 10 per cent. Systematic investigations of Mauz (1987) showed, with increasing viscosity, increasing churning losses for low speeds and decreasing churning losses for high speeds (Figure 13). He explains this phenomenon that less oil volume is in motion at higher viscosities and thus lesser losses are generated.

Load gear losses

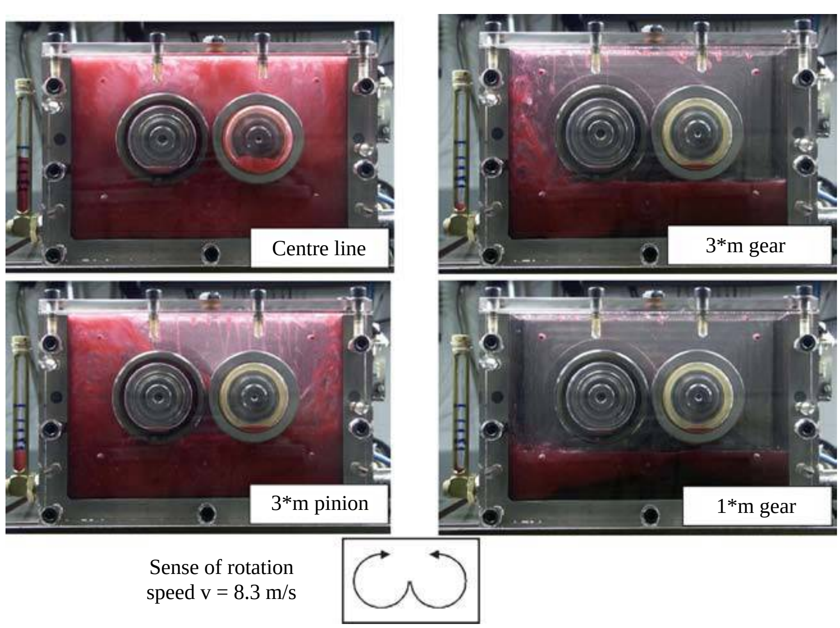
The load gear losses P_{VZP} in the mesh while power is transmitted follow the basic Coulomb law:

$$P_{VZP} = F_R(x) \cdot V_{rel}(x) \tag{1}$$

with:

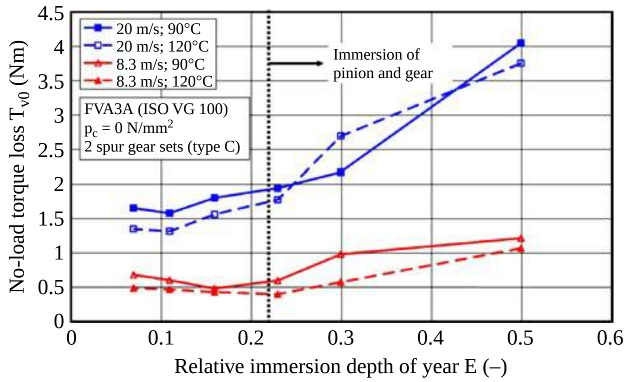
P_{VZP} load gear losses (kW).
F_R friction force (kN).
v_{rel} relative velocity (m/s).

Figure 8 Gear churning as a function of immersion depth



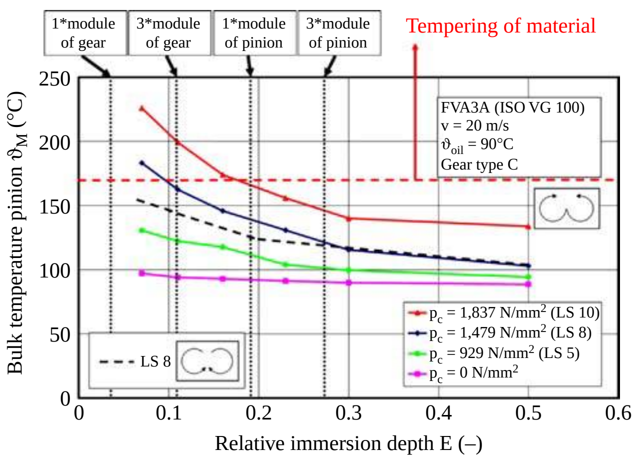
Source: Otto (2009)

Figure 9 Influence of immersion depth on gear churning loss



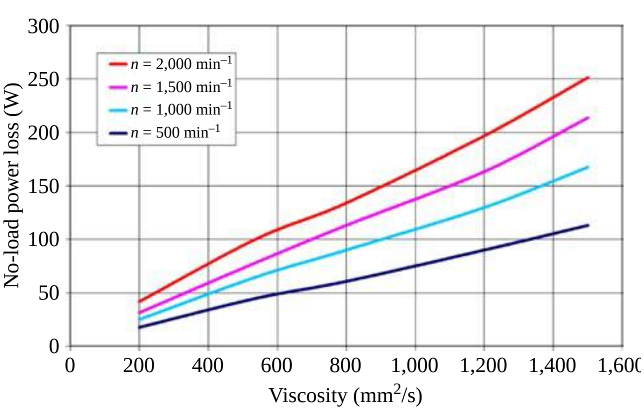
Source: Otto (2009)

Figure 10 Influence of immersion depth on pinion bulk temperature



Source: Otto (2009)

Figure 11 Influence of oil viscosity on gear churning losses



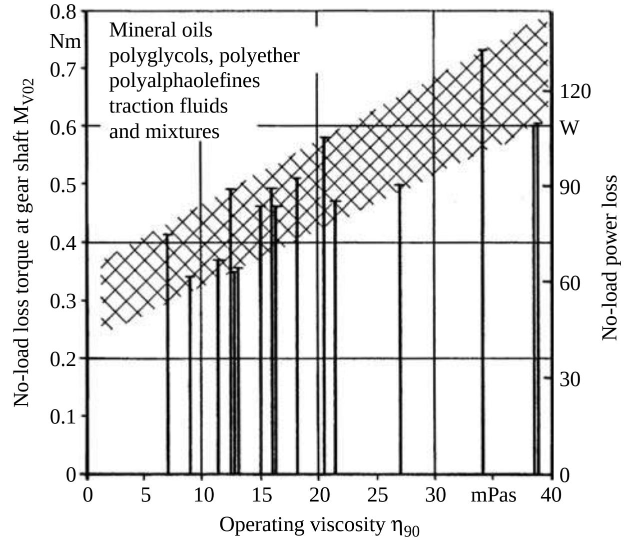
Source: Terekhov (1975)

The local friction force in the gear mesh can be calculated from the local normal force and the local coefficient of friction along the path of contact:

$$P_{VZP} = F_N(x) \cdot \mu(x) \cdot V_g(x) \tag{2}$$

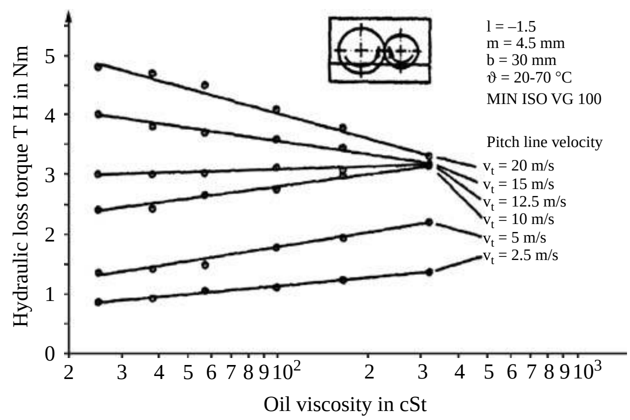
with:

Figure 12 Influence of oil viscosity on gear churning losses



Source: Michaelis (1994)

Figure 13 Influence of oil viscosity on gear churning losses



Source: Mauz (1987)

P_{VZP} load gear losses (kW).
F_N normal force (kN).
μ friction coefficient (-).
v_g sliding velocity (m/s).

When equation (2) is multiplied with $F_{N\,max}/F_{N\,max}$ * v/v = 1, it reads:

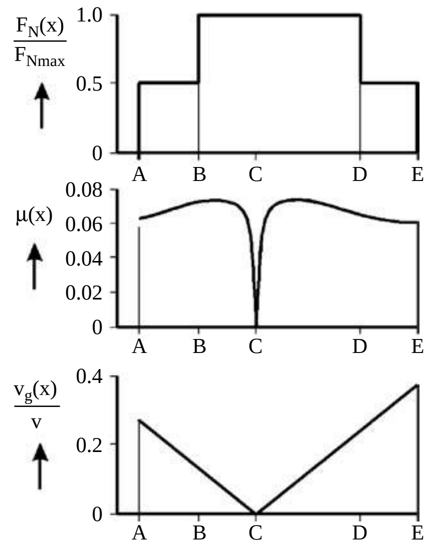
$$P_{VZP} = F_{N\,max} \cdot \frac{F_N(x)}{F_{N\,max}} \cdot \mu(x) \cdot V \cdot \frac{V_g(x)}{V} \tag{3}$$

The distribution of the local relative parameters $F_N(x)/F_{Nmax}$, $\mu(x)$ and $v_g(x)/v$ is shown in Figure 14. With the linear dependence of load and sliding speed and the approximation of a constant friction coefficient along the path of contact, equation (3) can be rewritten and rearranged to:

$$P_{VZP} = F_{tmax} \cdot V \cdot \mu_{mz} \cdot \frac{1}{cos(\alpha_{wt})} \cdot \frac{1}{P_{et}} \cdot \int_A^E \left[\frac{F_N(x)}{F_{Nmax}} \cdot \frac{V_g(x)}{V}\right] dx \tag{4}$$

with:

Figure 14 Load, friction coefficient and sliding speed along path of contact



P_{VZP} load gear losses (kW).
$F_{t\,max}$ tangential force (kN).
v pitch line velocity (m/s).
μ_{mz} mean coefficient of gear friction (-).
α_{wt} working pressure angle (°).
p_{et} transverse pitch (mm).
F_N normal force (kN).
v_g sliding velocity (m/s).

Ohlendorf (1958) introduced a loss factor H_V which only depends on geometrical gear data:

$$H_V = \frac{1}{cos(\alpha_{wt})} \cdot \frac{1}{p_{et}} \cdot \int_A^E \left[\frac{F_N(x)}{F_{N\,max}} \cdot \frac{V_g(x)}{V}\right] dx = \frac{\pi \cdot (u+1)}{z_1 \cdot u \cdot cos(\beta_b)} \cdot \left(1 - \varepsilon_\alpha + \varepsilon_1^2 + \varepsilon_2^2\right) \quad (5)$$

with:

H_V gear loss factor (-).
u gear ratio z_2/z_1 (-).
z_1 number of teeth on the pinion (-).
$ß_b$ helix angle at base cylinder (°).
$\varepsilon\alpha$ profile contact ratio (-).
$\varepsilon_{1,2}$ tip contact ratio, pinion and gear (-).

The load gear losses can then be written as:

$$P_{VZP} = P_a \cdot \mu_{mz} \cdot H_V \quad (6)$$

with:

P_{VZP} load gear losses (kW).
P_a transmitted power (kW).
μ_{mz} mean coefficient of gear friction (-).
H_V gear loss factor (-).

Low mesh losses can be achieved when the gear contact is concentrated around the pitch point with zero sliding (Figure 14) and a low value of the coefficient of gear friction. Low-loss gears with minimum sliding were designed in comparison to FZG standard test gears type C (Table I).

Table I Comparative data of standard and low-loss gears

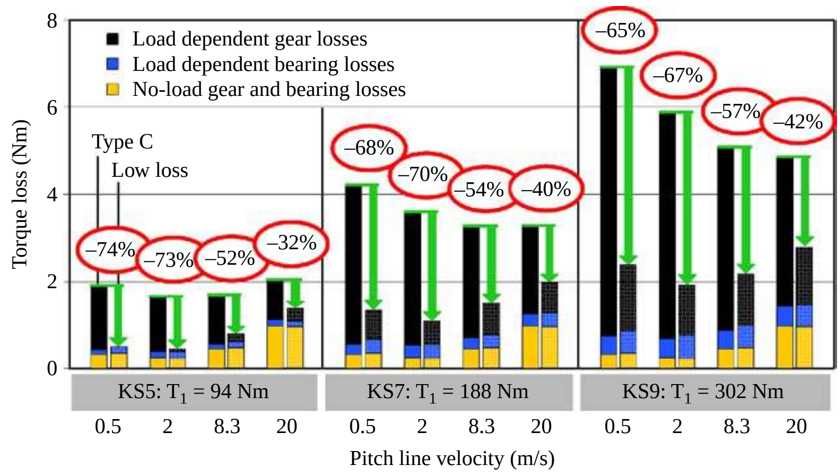
	Symbol	Unit	C-type gear	Low-loss gear
Centre distance	a	mm	91.5	91.5
Normal module	m_n	mm	4.5	1.75
Number of teeth				
Pinion	z_1	–	16	40
Gear	z_2	–	24	60
Pressure angle	α	°	20	40
Helix angle	ß	°	0	15
Face width	b	mm	14	20
Transverse contact ratio	ε_α	–	1.44	0.49
Face contact ratio	ε_β	–	0	0.94
Total contact ratio	ε_γ	–	1.44	1.43
Minimum safety factor pitting	S_H	–	0.81	0.89
Minimum safety factor bending	S_F	–	1.79	1.82
Minimum safety factor scuffing	S_B	–	1.14	11.72

For same nominal load capacity calculated according to DIN 3990, a wider face width is required for low-loss gears compared to the standard gear design. It has to be mentioned that load capacity calculation according to DIN 3990 is no longer valid due to values of pressure angle and profile contact ratio out of the defined parameter field of validity. In an ongoing project, the load-carrying capacity of low-loss gears is investigated and calculation methods will be adjusted.

C-type gears and low-loss gears were manufactured (Figure 15) and tested with respect to total gearbox power loss savings at different operating conditions. Wimmer *et al.* (2005) found for low-loss gears with minimum sliding speeds compared to standard gears, a reduction of total gearbox power loss between some 75 per cent at low speed of v = 0.5 m/s and some 35 per cent at high speed of v = 20 m/s (Figure 16) with a mean potential of some 50 per cent power loss savings. Besides the required wider gear face width for adequate load capacity, it has also to be considered that higher bearing forces may occur depending on the designed helix angle and the larger pressure angle compared to standard gears. The influence of low-loss gear design on vibration excitation and noise generation has also to be separately considered. Because of the higher mesh stiffness of low-loss gears, they are also less tolerant to manufacturing tolerances than standard gears. The typical smaller module and higher number of teeth of the low-loss gears compared to standard gears results in a higher mesh frequency which has to be

Figure 15 Geometry of standard C-type gears and low-loss gears

Figure 16 Power loss of standard gears (C type) compared to low-loss gears



Source: Wimmer *et al.* (2005)

considered in the expected vibration excitation. Further research is initiated in this field.

A reduction of the coefficient of friction in the gear mesh is possible in the field of boundary lubrication with thin separating films when using beneficial additive systems. Systematic investigations were made by Wimmer *et al.* (2006) with additives with different sulphur and phosphorus components as well as pure organic and metal-organic systems. In modified FZG-FVA efficiency tests (Doleschel, 2002) at low speeds and high temperatures for thin film conditions, no influence on boundary friction for the different additive systems was found except for a soluble molybdenum-thio-phosphate additive. At high pressure of $p_H = 1{,}720\,N/mm^2$, low speed of $v = 0.5\,m/s$ and high temperature of $\vartheta_{oil} = 120°C$ corresponding to an operating viscosity of the oil of $\nu_{120} = 7.2\,mm^2/s$, the boundary friction coefficient of the molybdenum-thio-phosphate additive was found to be less than 50 per cent of the friction coefficient of standard sulphur-phosphorus additives (Figure 17).

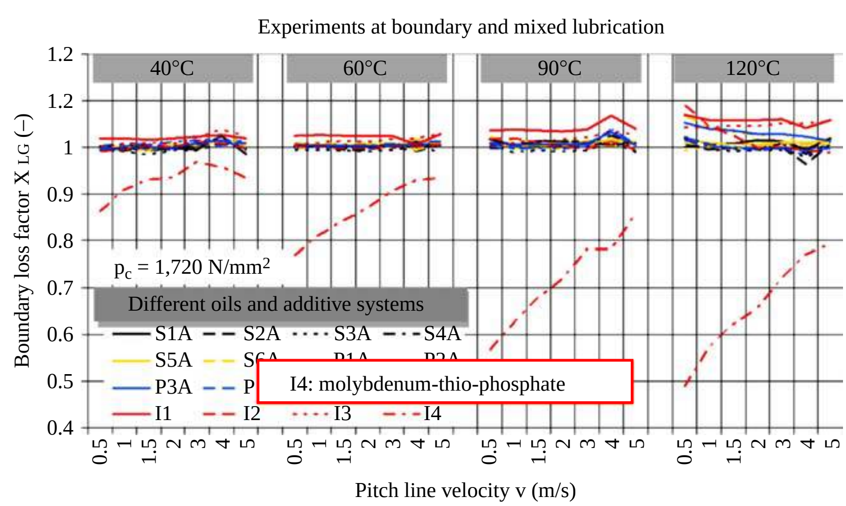
In the operating range of predominantly mixed and elastohydrodynamic (EHD) friction, a large influence of the base oil type on gear mesh friction is found. Doleschel (2003) investigated different base oil types at different viscosity grades including expected high friction of a traction fluid (MYH 68) and a polyether-based low-friction fluid (MYL 68). The range of the measured values for the coefficient of friction in the FZG-FVA efficiency test is shown in Figure 18. In a wide range of operating conditions, friction in a gear mesh can be reduced compared to lubrication with a mineral oil by some 10-20 per cent with a polyalphaolefin plus ester, by some 20-30 per cent with a polyglycol and even by some 50 per cent with a polyether-type base oil compared to a mineral oil. Similar effects are expected for the different base oil types for the load-dependent bearing losses.

Application

Wind turbine gearbox

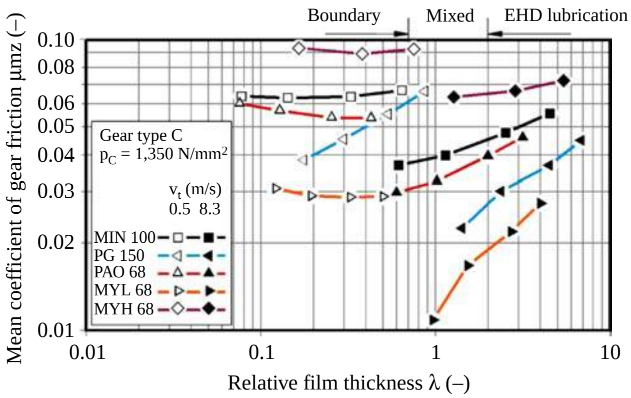
A very simple means of power loss reduction is the use of an efficient lubricant. For a quantitative evaluation of the influence of different lubricants on gearbox power loss, a middle-sized wind turbine gearbox with nominal power capacity of $P = 1.8\,mW$ was investigated. The gearbox with

Figure 17 Influence of additive type on gear mesh loss



Source: Wimmer *et al.* (2006)

Figure 18 Influence of base oil on gear mesh loss



Source: Doleschel (2003)

a planetary low-speed first stage and an intermediate- and-high-speed cylindrical gear stage was modelled in the computer program WTplus (Kurth, 2008) (Figure 19).

The program calculates the expected power loss of gears, bearings and seals for any gearing system. The influence of the lubricant can be introduced into the calculation with the evaluation of the friction coefficient of the lubricant according to Doleschel (2002). From the results of the FZG-FVA efficiency test for the candidate oil at different operating conditions, an empirical equation is derived for the calculation of the mesh friction in gears and bearings.

The friction coefficient μ_M in a gear mesh consists of a portion of solid body friction μ_F and a portion of fluid film friction μ_{EHD}:

$$\mu_M = (1 - \zeta) \cdot \mu_F + \zeta \cdot \mu_{EHD} \tag{7}$$

with:

μ_M mixed friction coefficient (-).
μ_F solid friction coefficient (-).
μ_{EHD} fluid friction coefficient (-).
ζ portion of fluid friction (-).

The portion ζ of fluid and solid friction depends on the relative film thickness λ in the contact (Figure 20).

Figure 19 Model of wind turbine gear system in WTplus

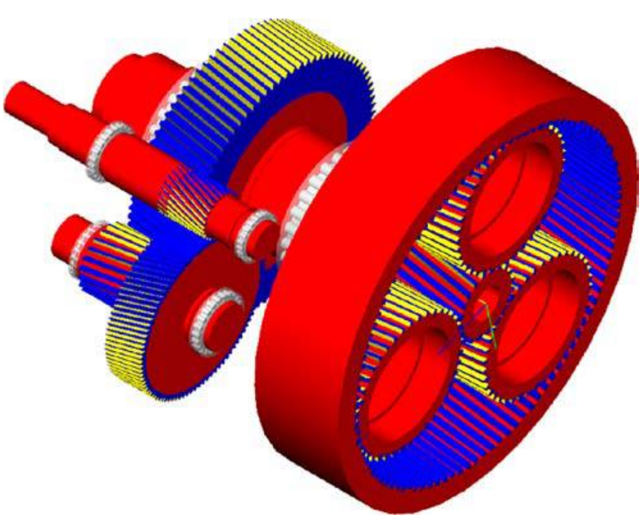
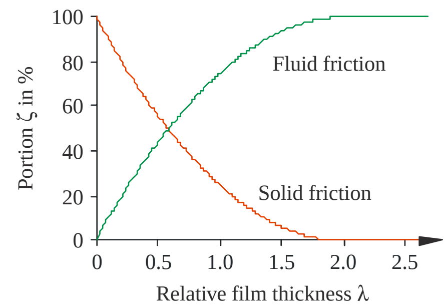


Figure 20 Fluid and solid friction in an EHD contact



Source: Doleschel (2003)

The solid friction coefficient and the fluid friction coefficient can be calculated according to equations (8) and (9) with the parameters for the lubricant from the FZG-FVA efficiency test:

$$\mu_F = \mu_{F,R} \cdot \left(\frac{p_H}{p_R}\right)^{\alpha F} \cdot \left(\frac{V_{\sum}}{V_{R,F}}\right)^{\beta F} \tag{8}$$

with:

μ_F solid friction coefficient (-).
$\mu_{F,R}$ solid friction coefficient, reference value from test (-).
p_H contact pressure (N/mm^2).
p_R reference value of contact pressure, 1,000 N/mm^2 (-).
v_Σ sum velocity (m/s).
$v_{R,F}$ reference value of speed for solid friction, 0.2 m/s (m/s).
α_F pressure exponent for solid friction from test (-).
β_F speed exponent for solid friction from test (-).

$$\mu_{EHD} = \mu_{EHD,R} \cdot \left(\frac{p_H}{p_R}\right)^{\alpha EHD} \cdot \left(\frac{V_{\sum}}{V_{R,EHD}}\right)^{\beta EHD} \cdot \left(\frac{\eta_{oil}}{\eta_R}\right)^{\gamma EHD} \tag{9}$$

with:

μ_{EHD} fluid friction coefficient (-).
$\mu_{EHD,R}$ fluid friction coefficient, reference value from test (-).
p_H contact pressure (N/mm^2).
p_R reference value of contact pressure, 1,000 N/mm^2 (N/mm^2).
v_Σ sum velocity (m/s).
$v_{R,EHD}$ reference value of speed for fluid friction $v_{R,F} = 8.3$ m/s (m/s).
α_{EHD} pressure exponent for fluid friction from test (-).
β_{EHD} speed exponent for fluid friction from test (-).
γ_{EHD} viscosity exponent for fluid friction from test (-).

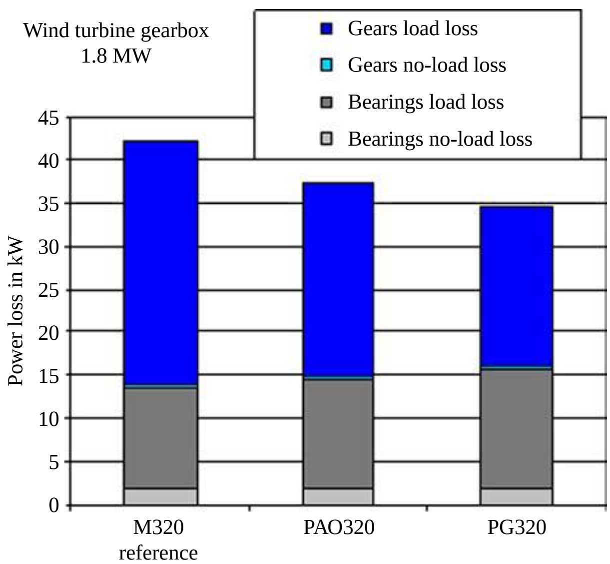
For three different lubricants from the market place typical for wind turbine application, the FZG-FVA efficiency test was performed. The relevant lubricant data can be taken from Table II.

All the lubricants had the same viscosity grade, ISO VG 320, with different base oil types: a mineral oil MIN320, a polyalphaolefin PAO320 and a polyglycol PG320 with typical additive packages for the application. The results of the comparative calculation for nominal power transmission are shown in Figure 21. When changing from a mineral oil to a polyalphaolefin, a reduction of power losses of some 10 per cent are possible; with a polyglycol, even a 20 per cent reduction of power loss is feasible.

Table II Lubricant data

	Symbol	Unit	M320	PAO 320	PAG 320
Type			Mineral	Polyalphaolefin	Polyglycol
Viscosity	ν_{40}	mm^2/s	327	310	340
	ν_{100}	mm^2/s	24.4	37.0	60
Viscosity index	VI	–	97	169	247
Density	ρ_{15}	kg/dm^3	898	902	1,050
Reference solid friction	$\mu_{F,R}$	–	0.047	0.060	0.048
Solid friction exponents	α_F	–	0.62	0.74	1.55
	β_F	–	−0.12	−0.27	−1.60
Reference fluid friction	$\mu_{EHD,R}$	–	0.033	0.022	0.016
Fluid friction exponents	α_F	–	0.19	0.59	−0.11
	β_F	–	−0.05	−0.07	0.01
	γ_F	–	0.19	0.21	0.40

Figure 21 Calculated power loss with different lubricant types for a wind turbine gearbox

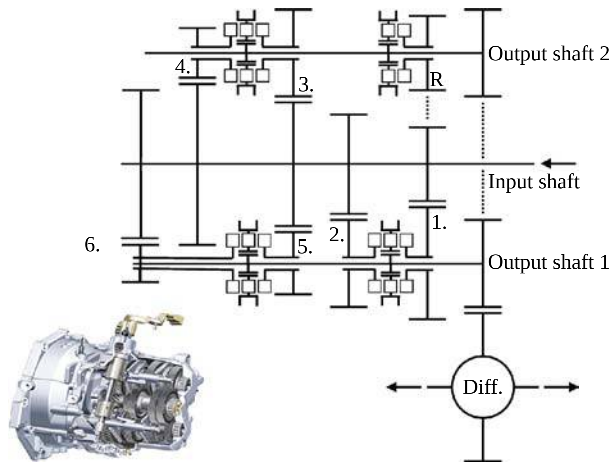


There is an even higher potential of efficiency gain when different viscosity grades are used according to the different viscosity-temperature behaviour of these oils. For further efficiency improvements, the expected film thickness values have to be analyzed taking viscosity and pressure viscosity at the expected gear temperature for the different lubricants into account.

Automotive gearbox

Possible power loss reduction in an automotive six-gear manual transmission (Figure 22) was investigated by Kurth *et al.* (2009). The conventional gear design was replaced with a low-loss gear design. Owing to the lower losses in the gear mesh, the cooling oil requirements are reduced. Therefore, it was possible to reduce the oil level in the gearbox by 20 mm for same calculated gear bulk temperatures of conventional and low-loss gears. Thus, it was possible, by changing to low-loss gears, to reduce not only

Figure 22 Manual transmission



Source: Kurth *et al.* (2009)

load losses in the gear mesh but also no-load losses by reduced oil level.

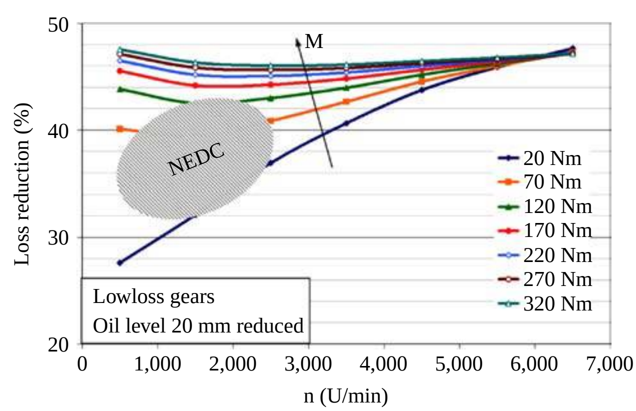
The gearbox was modelled in WTplus (Kurth, 2008) and comparative power loss calculations for the conventional gear design and the low-loss gear design with reduced oil level were performed. Figure 23 shows the possible loss savings for the second gear as a function of speed and load. For a wide range of operating conditions, total losses can be reduced by more than 40 per cent. Even for the NEDC with a large part-load share and thus a high portion of no-load losses, a loss reduction of some 35 per cent is possible.

A direct comparison of the total losses of the gearbox in 2nd gear at 2,500 rpm is shown in Figure 24. It is not only the large gain in the gear load losses that is obvious but also the substantial reduction of the gear no-load losses.

Conclusions

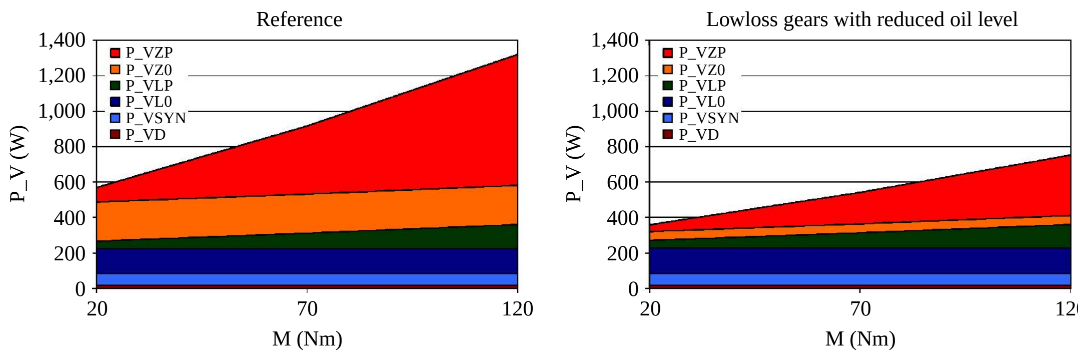
Depending on the application and the operating regimes, a power loss reduction potential in a gearbox of some 50 per cent was proven to be possible. In some applications, only the simple change to a highly efficient lubricant can save some 20 per cent power loss. For maximum efficiency, optimisation alternative solutions have to be found for gear and bearing design as well as

Figure 23 Loss reduction with low-loss gears and reduced oil level, second gear



Source: Kurth *et al.* (2009)

Figure 24 Comparison of losses for the 2nd gear at 2,500 rpm



Source: Kurth *et al*. (2009)

lubricant type, viscosity and supply to the components. The challenges of these new approaches are adequate compromises between power loss reduction on the one hand and load-carrying capacity and noise properties on the other hand.

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Further reading

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Corresponding author

Klaus Michaelis can be contacted at: michaelis@fzg.mw.tum.de

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