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OPTIMIZATION OF GEARBOX EFFICIENCY

Abstract

Although mechanical gearboxes used as torque and speed converters have already very high efficiency it is not only a task in automotive applications to further decrease gearbox power losses but also in many industrial applications.

Different methods are discussed for power loss reduction in a gearbox. 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 of the components. This in turn influences the cooling properties in the gear and bearing meshes. Bearing systems can be optimized 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 average of 50 % is technically feasible.

The challenge is 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.

INTRODUCTION

Future energy shortages have not only to be fought 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 overall optimization although absolute efficiency in gearboxes and rear axles is already high, (Xu, 2007). However, 1 kilowatt savings in the gearbox mean 4 kilowatts savings in fuel energy.

Looking at wind turbines as a growing market for alternative energy production a modern equipment of the 5 megawatt class consist of 8 or more gear meshes and more than 12 bearing meshes. A reduction of the overall losses by 50 % 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.

BASIC CONSIDERATIONS

Power loss in a gearbox consists of gear, bearing, seal and auxiliary losses (Fig. 1). Gear and bearing losses can be separated in 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, 2007). Load losses depend on transmitted load, coefficient of friction and sliding velocity in the contact areas of the components.

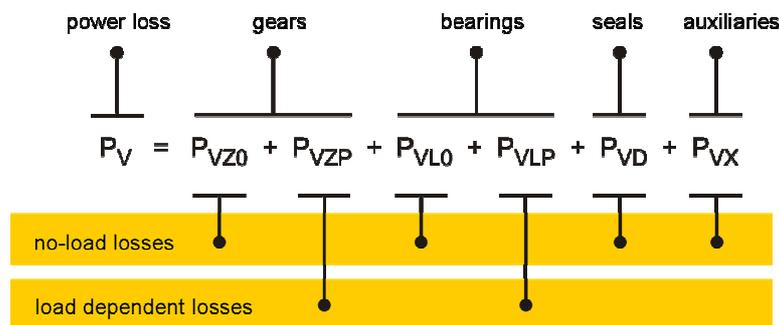


Fig. 1: Composition of transmission power loss

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 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 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. Lowest no load losses of radial bearings are expected for cylindrical roller bearings. The also 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.

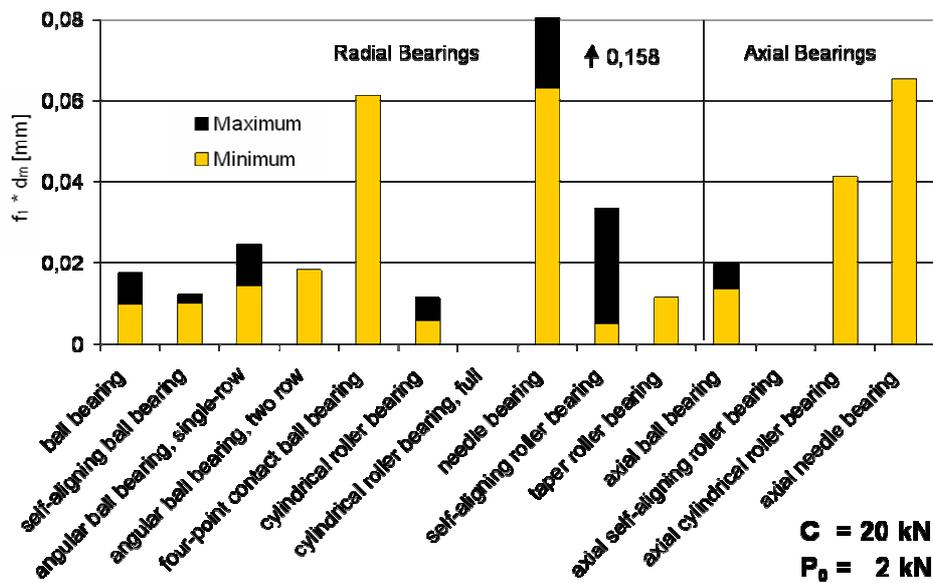


Fig. 2: Influence of bearing type on load losses (according Wimmer, 2003)

Load dependent bearing losses depend also on bearing type and size, load and sliding conditions in the bearing and on lubricant type (Wimmer, 2003).

Fig. 2 shows load dependent losses of bearings with same load capacity $C = 20 \text{ kN}$ 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 same load capacity.

A comparison of the bearings losses for the 6th 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 (Fig. 3) was calculated according SKF, 2004.

For medium load and medium speed conditions at low gear oil temperatures of 40°C , relevant for the new European drive cycle NED C, a reduction of the bearing losses of more than 50% 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% for the alternative design (Fig. 4).

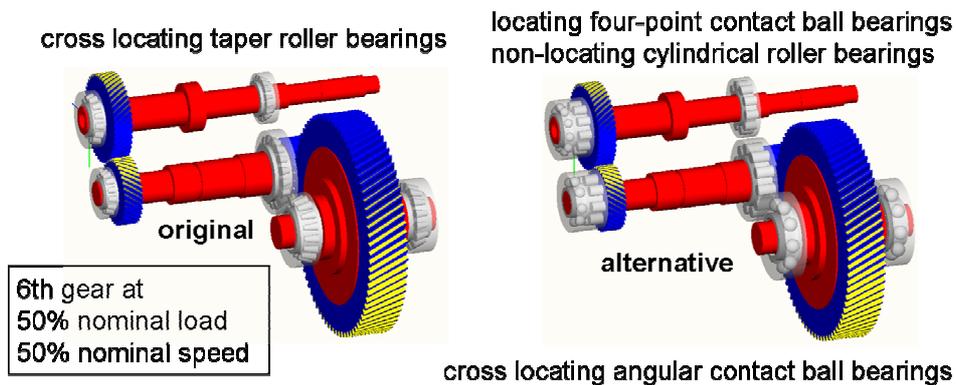


Fig. 3: Alternative bearing design in a manual transmission with final drive

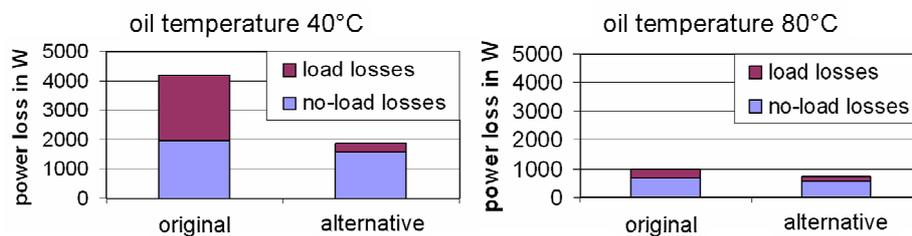


Fig. 4: Influence of design and operating temperature on bearing losses

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. Compared to the reference oil level at shaft centre line, three times module at pinion ($3 \cdot m$ pinion) with pinion and gear immersed in oil, three times module at gear ($3 \cdot m$ gear) as well as one times module at gear ($1 \cdot m$ gear) with only the gear immersed in oil were investigated. The situation in the test gearbox for the different oil levels is shown in Fig. 5. 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. Fig. 6 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.

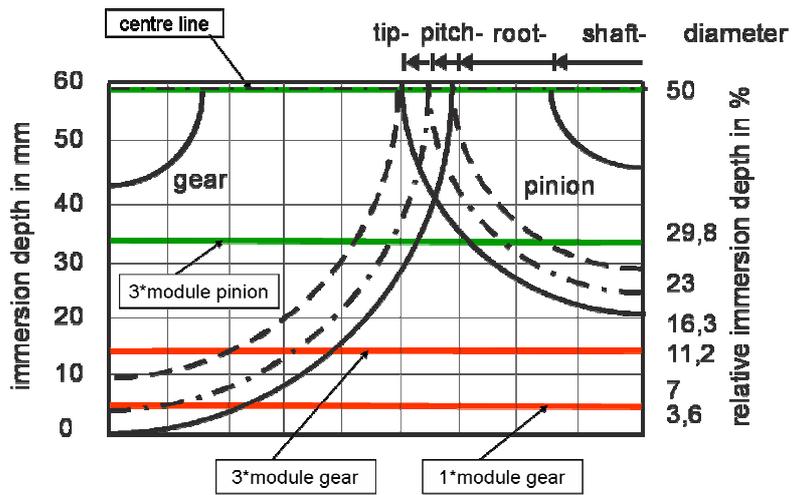


Fig. 5: Immersion depth in test gearbox (according Otto, 2009)

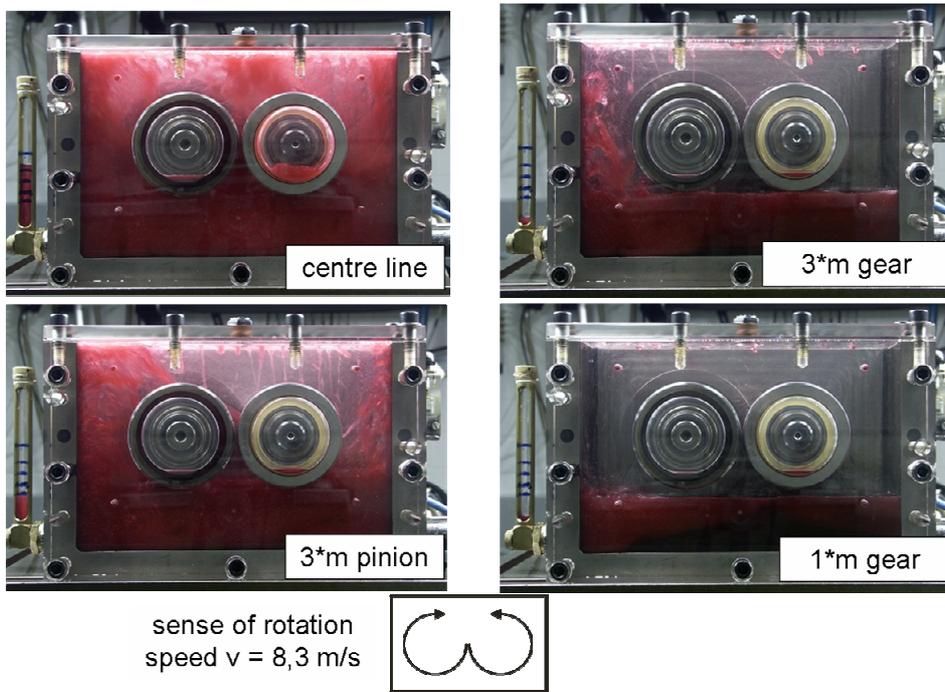


Fig. 6: Gear churning as a function of immersion depth (according Otto, 2009)

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 and 120 °C showed a substantial reduction of the gear no load losses with decreased immersion depth (Fig. 7). 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 % when the immersion depth is reduced from centre line to 3 times module of the gear. In contrary to the beneficial effect of churning loss reduction with reduced immersion depth at the same time the detrimental effect of reduced cooling of the gear mesh has to be considered. Fig. 8 shows measured pinion bulk temperatures at different immersion depth. 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.

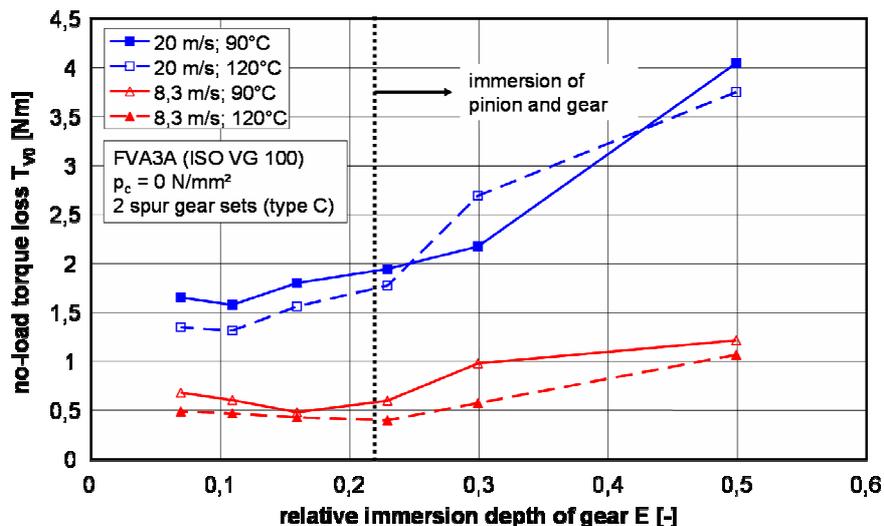


Fig. 7: Influence of immersion depth on gear churning loss (accord. Otto, 2009)

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 (Fig. 9). Michaelis, 1994 confirms increasing gear churning losses with increasing lubricant viscosity independent of the oil type (Fig. 10) also for low operating viscosities. Depending on the operating conditions a change from e.g. ISO VG 150 to VG 100 can reduce the no load power losses by some 10%. Investigations of Mauz, 1987 showed with increasing viscosity increasing churning losses for low speeds and decreasing churning losses for high speeds (Fig. 11). He explains this phenomenon that less oil volume is in motion at higher viscosities and thus less losses are generated.

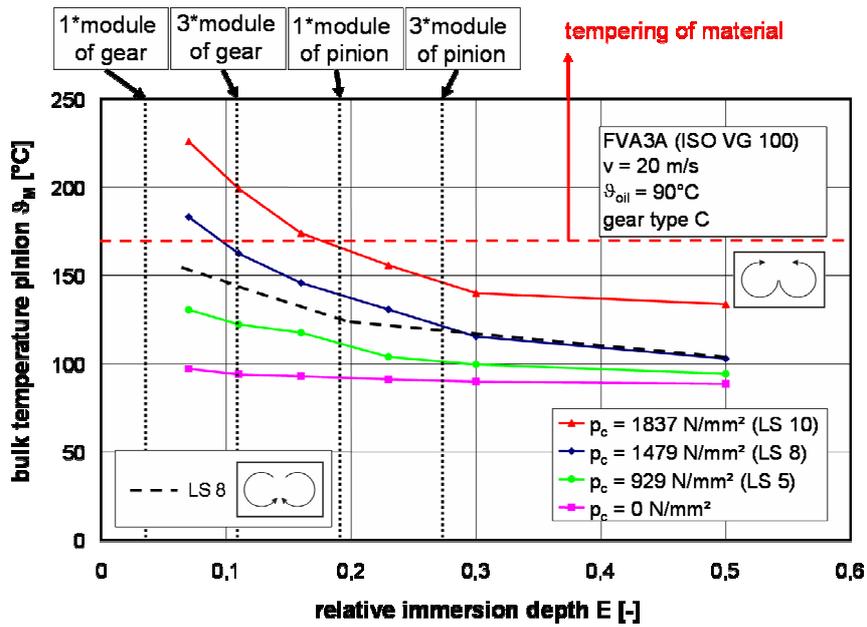


Fig. 8: Influence of immersion depth on pinion bulk temperature (accord. Otto, 2009)

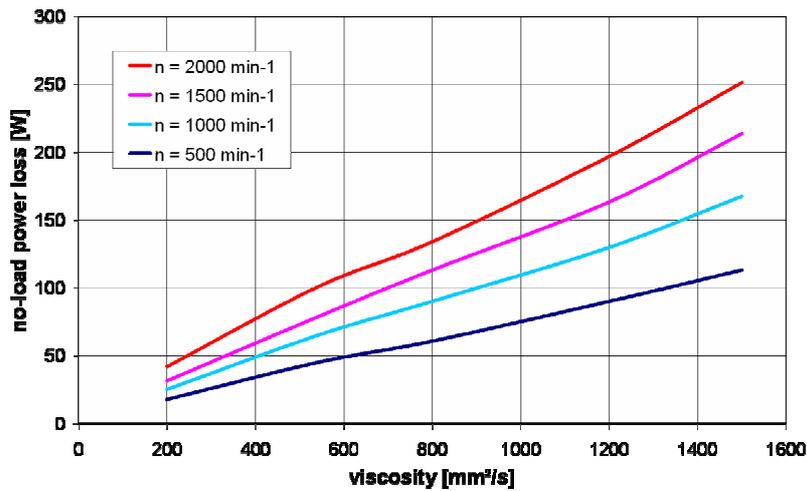


Fig. 9: Influence of oil viscosity on gear churning losses (accord. Terekhov, 1975)

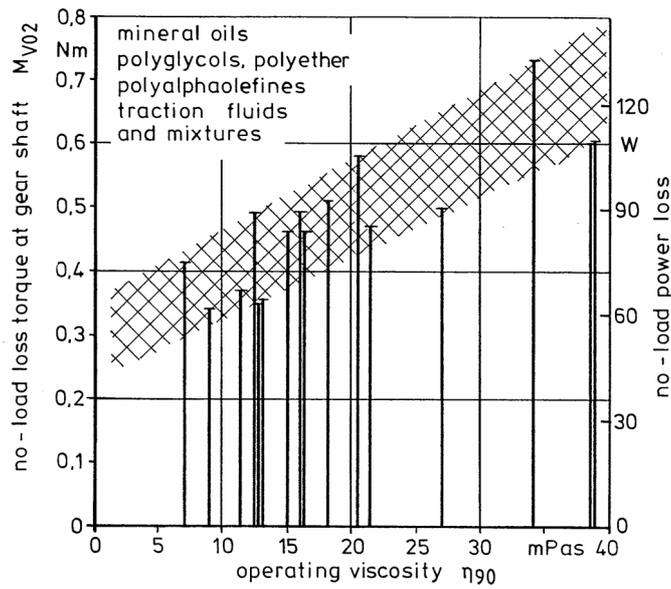


Fig. 10: Influence of oil viscosity on gear churning losses (accord. Michaelis, 1994)

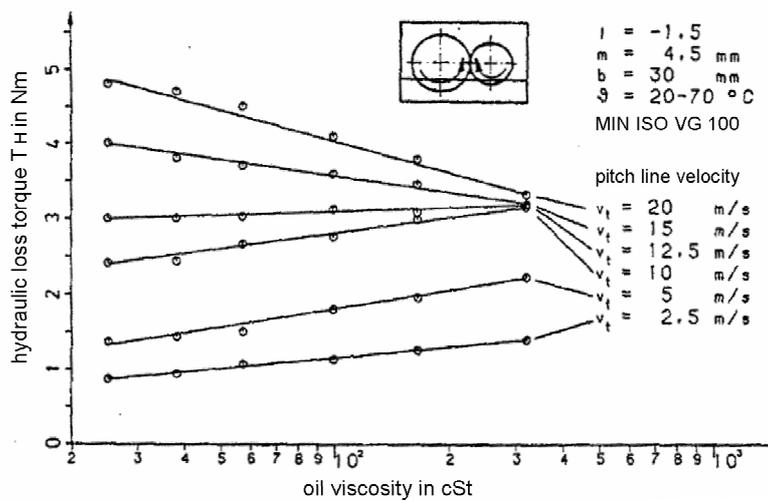


Fig. 11: Influence of oil viscosity on gear churning losses (accord. Mauz, 1987)

Load Gear Losses

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

$$P_{VZP} = F_R(x) \cdot v_{rel}(x) \quad (1)$$

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

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

$$P_{VZP} = F_N(x) \cdot \mu(x) \cdot v_g(x) \quad (2)$$

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

When Eq. (2) is multiplied with $F_{Nmax}/F_{Nmax} \cdot v/v = 1$ it reads

$$P_{VZP} = F_{Nmax} \cdot \frac{F_N(x)}{F_{Nmax}} \cdot \mu(x) \cdot v \cdot \frac{v_g(x)}{v} \quad (3)$$

The distribution of the local relative parameters $F_N(x)/F_{Nmax}$, $\mu(x)$ and $v_g(x)/v$ is shown in Fig. 12. With the linear dependence of load and sliding speed and the approximation of a constant friction coefficient along the path of contact Eq. (3) can be rewritten and rearranged to

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

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

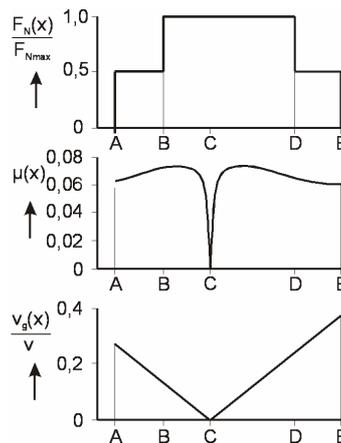


Fig. 12: Load, friction coefficient and sliding speed along path of contact

Ohlendorf, 1958 introduced a loss factor H_V which depends on geometrical gear data

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

with	H_V	-	gear loss factor
	u	-	gear ratio z_2/z_1
	z_1	-	number of teeth pinion
	β_b	°	helix angle at base cylinder
	ε_α	-	profile contact ratio
	$\varepsilon_{1,2}$	-	tip contact ratio, pinion and gear

The load gear losses can then be written

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

with	P_{VZP}	kW	load gear losses
	P_a	kW	transmitted power
	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 (Fig.12) and a low value of the coefficient of gear friction. Lowloss gears with minimum sliding were designed in comparison to FZG standard test gears type C (Table 1). For same nominal load capacity calculated according to DIN 3990 a wider face width is required for lowloss gears compared to 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 lowloss gears is investigated and calculation methods will be adjusted.

	symbol	unit	C-type gear	lowloss 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	β	°	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

Table 1:Comparative data of standard and lowloss gears

C-type gears and lowloss gears were manufactured (Fig.13) and tested with respect to total gearbox power loss savings at different operating conditions. Wimmer, 2005 found for lowloss gears with minimum sliding speeds compared to standard gears a reduction of total gearbox power loss between some 75% at low speed of $v = 0,5$ m/s and some 35% at high speed of $v = 20$ m/s (Fig.14) with a mean potential of some 50% power loss savings.



Fig. 13: Geometry of standard C type gears and lowloss gears

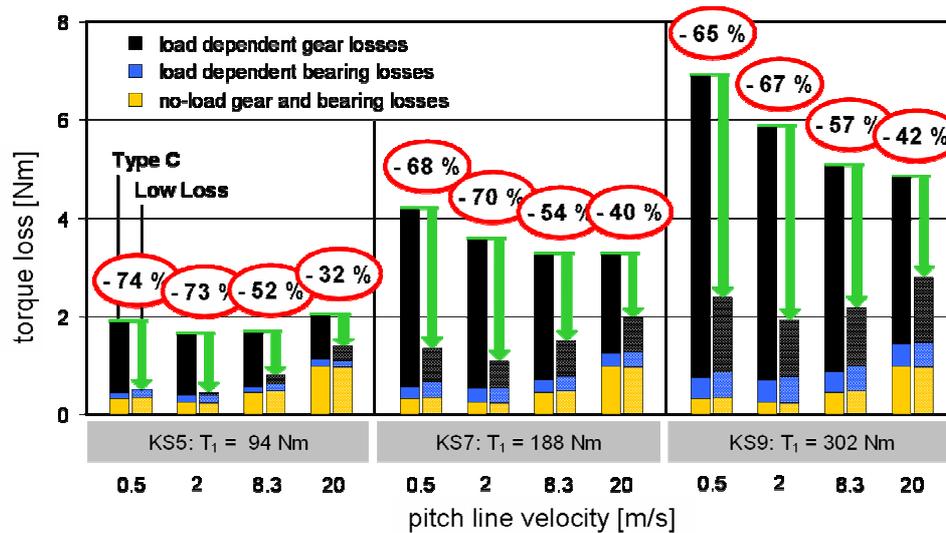


Fig. 14: Power loss of standard gears (C type) compared to lowloss gears (according Wimmer, 2005)

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 lowloss gear design on vibration excitation and noise generation has also to be separately considered. Because of the higher mesh stiffness of lowloss gears they are also less tolerant to manufacturing tolerances than standard gears. The typical smaller module and higher number of teeth of the lowloss gears compared to standard gears results in a higher mesh frequency which has to be 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, 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 $p_H = 1720 \text{ N/mm}^2$, low speed of $v = 0,5 \text{ m/s}$ and high temperature of $\vartheta_{oil} = 120^\circ\text{C}$ corresponding to an operating viscosity of the oil of $\nu_{120} = 7,2 \text{ mm}^2/\text{s}$ the boundary friction coefficient of the molybdenum-thio-phosphate additive was found to be less than 50% of the friction coefficient of standard sulphur-phosphorus additives (Fig.).

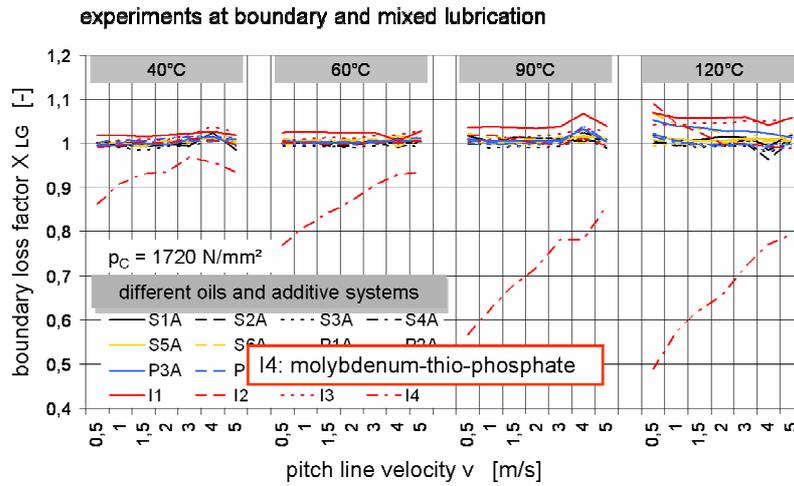


Fig. 15: Influence of additive type on gear mesh loss (accord. Wimmer, 2006)

In the operating range of predominantly mixed and 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).

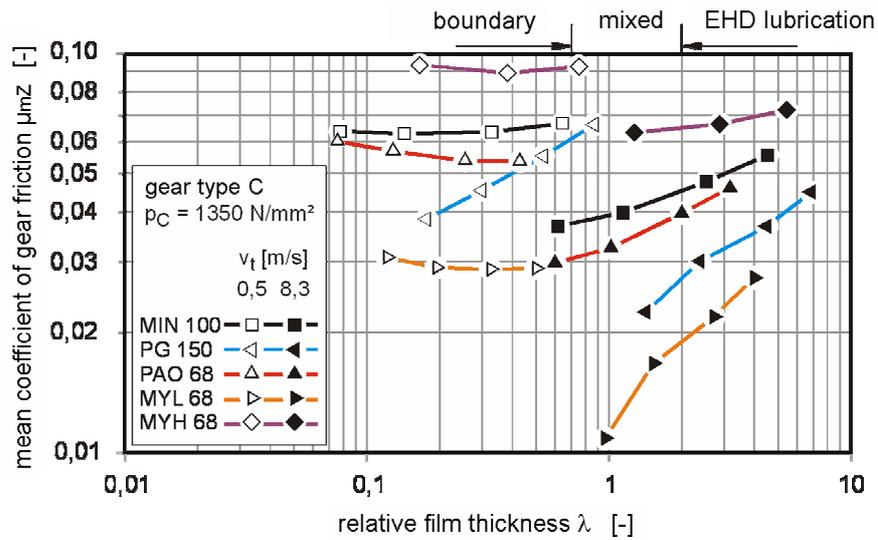


Fig. 16: Influence of base oil on gear mesh loss (according Doleschel, 2003)

The range of the measured values for the coefficient of friction in the FZG-FVA efficiency test is shown in Fig.16. 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 to 20% with a polyalphaolefin plus ester, by some 20 to 30% with a polyglycol and even by some 50% 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 a planetary low speed first stage and an intermediate and high speed cylindrical gear stage was modelled in the computer program WTplus (Kurth, 2008) (Fig. 17).

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.

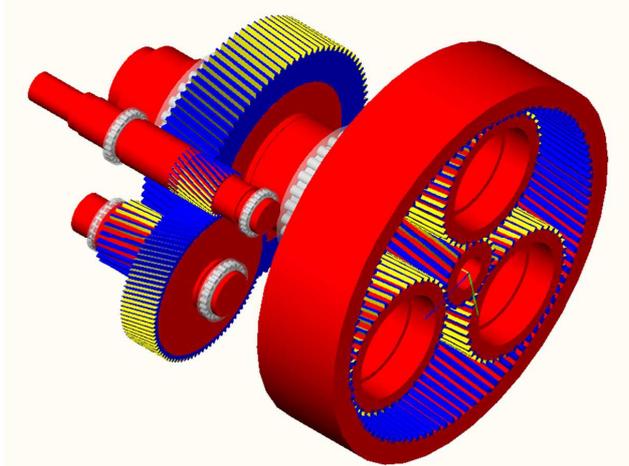


Fig. 17: Model of wind turbine gear system in WTplus

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} :

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

with	μ_M	-	mixed friction coefficient
	μ_F	-	solid friction coefficient
	μ_{EHD}	-	fluid friction coefficient
	ζ	-	portion of fluid friction

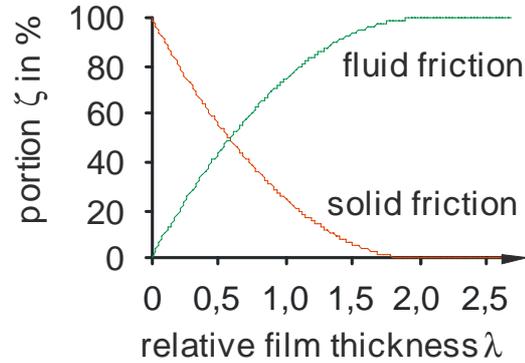


Fig. 18: Fluid and solid friction in an EHD contact (accord. Doleschel, 2003)

The portion ζ of fluid and solid friction depends on the relative film thickness λ in the contact (Fig. 18).

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

$$\mu_F = \mu_{F,R} \cdot \left(\frac{p_H}{p_R} \right)^{\alpha_F} \cdot \left(\frac{v_\Sigma}{v_{R,F}} \right)^{\beta_F} \quad (8)$$

with	μ_F	-	solid friction coefficient
	$\mu_{F,R}$	-	solid friction coefficient, reference value from test
	p_H	N/mm ²	contact pressure
	p_R	N/mm ²	reference value of contact pressure $p_R = 1000$ N/mm ²
	v_Σ	m/s	sum velocity
	$v_{R,F}$	m/s	reference value of speed for solid friction $v_{R,F} = 0,2$ m/s
	α_F	-	pressure exponent for solid friction from test
	β_F	-	speed exponent for solid friction from test

$$\mu_{EHD} = \mu_{EHD,R} \cdot \left(\frac{\rho_H}{\rho_R} \right)^{\alpha_{EHD}} \cdot \left(\frac{v_{\Sigma}}{v_{R,EHD}} \right)^{\beta_{EHD}} \cdot \left(\frac{\eta_{oil}}{\eta_R} \right)^{\gamma_{EHD}} \quad (9)$$

with	μ_{EHD}	-	fluid friction coefficient
	$\mu_{EHD,R}$	-	fluid friction coefficient, reference value from test
	ρ_H	N/mm ²	contact pressure
	ρ_R	N/mm ²	reference value of contact pressure $\rho_R = 1000$ N/mm ²
	v_{Σ}	m/s	sum velocity
	$v_{R,EHD}$	m/s	reference value of speed for fluid friction $v_{R,F} = 8,3$ 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 2. The lubricants had all 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 Fig. 19. When changing from a mineral oil to a polyalphaolefin a reduction of power losses of some 10 % are possible, with a polyglycol even a 20 % reduction of power loss is feasible.

	symbol	unit	M320	PAO 320	PAG 320
type			mineral	poly-alpha-olefin	poly-glycol
viscosity	v_{40}	mm ² /s	327	310	340
	v_{100}	mm ² /s	24,4	37,0	60
viscosity index	VI	-	97	169	247
density	ρ_{15}	kg/dm ³	898	902	1050
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

Table 2: Lubricant data

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.

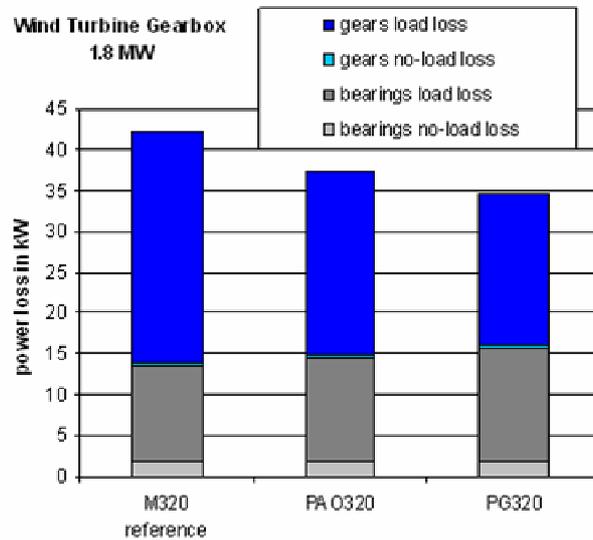


Fig. 19: Calculated power loss with different lubricant types for a wind turbine gearbox

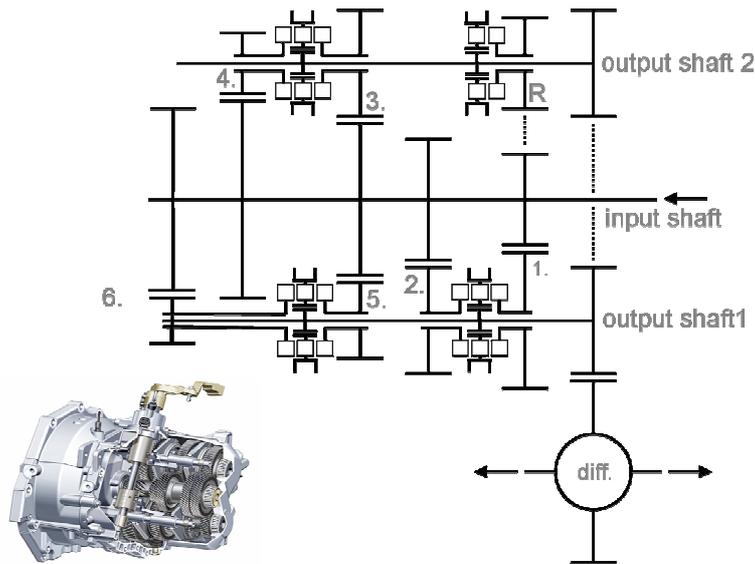


Fig. 20: Manual transmission (according Kurth, 2009)

Automotive Gearbox

Possible power loss reduction in an automotive 6 gear manual transmission (Fig.) was investigated by Kurth, 2009. The conventional gear design was replaced with a lowloss gear design. Due 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 lowloss gears. Thus it was possible by changing to lowloss gears not only to reduce load losses in the gear mesh but also to reduce 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 lowloss gear design with reduced oil level were performed. Fig. 21 shows the possible loss savings for the 2nd gear as a function of speed and load. For a wide range of operating conditions total losses can be reduced by more than 40%. Even for the New European Drive Cycle NEDC with a large part load share and thus a high portion of no-load losses a loss reduction of some 35% is possible.

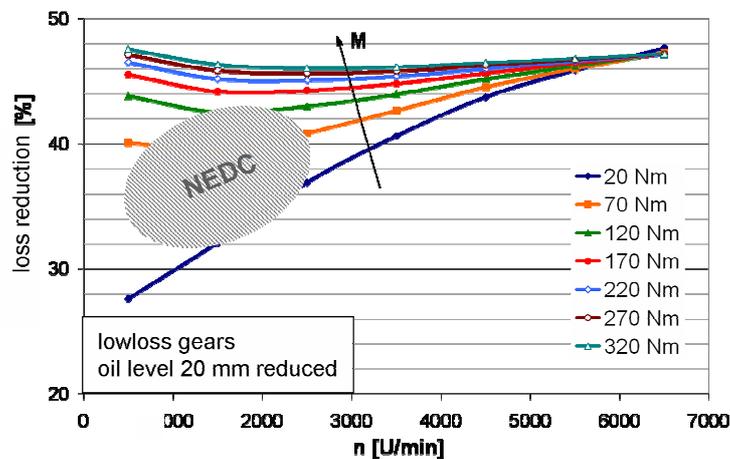


Fig. 21: Loss reduction with lowloss gears and reduced oil level, 2nd gear (acc. Kurth, 2009)

CONCLUSIONS

Dependent on the application and the operating regimes a power loss reduction potential in a gearbox of some 50% was proven to be possible. In some applications only the simple change to a highly efficient lubricant can save some 20% power loss. For maximum efficiency optimization alternative solutions have to be found for gear and bearing design as well as 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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UDK	ključne riječi	key words
621.833.018	učinkovitost zupčaničkih prijenosnika	gear transmission efficiency
621.833-11	zupčanički prijenosnik, projektiranje i konstrukcijska obilježja	transmission gear, design and construction characteristics
621.43.018.2	učinak motora, koeficijent mehaničkog učinka	engine mechanical efficiency
629.113	automobil	automobile
621.548.4	vjetroturbina	wind turbine
.004.183	gledište uštede energije	energy savings viewpoint

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