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Flank Load Carrying Capacity and Power Loss Reduction by Minimized Lubrication

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

The lubrication of gears has two major functions: Reducing friction and wear as well as dissipating heat. The power losses, especially the no-load losses, decrease with decreasing immersion depth using dip lubrication. The load-dependent gear power losses are nearly unaffected by minimized lubrication. However, the gear bulk temperatures rise dramatically by using minimized lubrication due to a lack of heat dissipation.

With minimized lubrication the scuffing load carrying capacity decreased by up to more than 60% compared to rich lubrication conditions. The dominating influence of the bulk temperature is therefore very clear. Starved lubrication leads to more frequent metal-to-metal contact and the generation of high local flash temperatures must be considered. An additional factor for the scuffing load carrying capacity calculation in case of minimized lubrication conditions is proposed.

Concerning pitting damage test runs showed that by lowering the oil level the load cycles without pitting damage decreased by approximately 50% up to 75% for minimized lubrication compared to the results with rich lubrication conditions. The allowable contact stress is clearly reduced (up to 30%) by minimized lubrication. A reduced oil film thickness as a consequence of increased bulk temperatures results in more frequent metal-to-metal contacts causing a higher surface shear stress. In combination with a decreased material strength due to a possible tempering effect at high bulk temperatures the failure risk of pitting damage is clearly increased. The common pitting load carrying capacity calculation algorithms according to DIN/ISO are only valid for moderate oil temperatures and rich lubrication conditions. For increased thermal conditions, the reduction of the pitting endurance level at increased gear bulk temperatures can be approximated with the method of Knauer (FZG TU München, 1988). An advanced calculation algorithm for pitting load carrying capacity calculation at high gear bulk temperatures (valid for high oil temperatures as well as for minimized lubrication) is proposed.

The micropitting risk was increased by low oil levels, especially at high loads and during the endurance test. The micropitting damage is caused by poor lubrication conditions which are characterized by a too low relative oil film thickness due to high bulk temperatures. Again, the actual bulk temperatures are of major significance for calculation of the micropitting load carrying capacity.

The wear rate of the gears is almost unaffected by the oil level. Only a slight increase of wear could be observed with minimized lubrication. This increase can be explained by the higher bulk temperature of the gears running under minimized lubrication conditions. The investigations showed that there exists a natural limitation for lowering the oil quantity in transmissions without detrimental influence on the load carrying capacity. Knowing these limitations enables the user to determine the possible potential benefits of reduced oil lubrication. The correct prediction of the actual gear bulk temperatures is of major importance in this context. A method for the estimation of the gear bulk temperature at reduced immersion depth respectively poor lubrication conditions is proposed.

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Introduction

Gears are machine components which determine the capability and reliability of many technical products. Continuous demand for higher efficiency and reliability, increased load carrying capacity and endurance life, smaller size, lower noise and vibrations, prolonged service intervals, low environmental impact and low costs will remain the main driving forces in the development of gear drives in the future. The fatigue of contacting surfaces in gears is often the life-limiting factor in transmissions which should, ideally, operate reliably for twenty years or more.

Low oil levels and thus reduced oil quantities are sometimes used for the reduction of no-load losses in dip lubricated automotive and industrial transmissions. Specifically, a higher efficiency can be obtained by reducing no-load power losses such as squeezing, splashing and ventilation losses. These losses can be reduced by lowering the oil volume, namely the oil level in dip lubricated transmissions and the oil flow rate with oil injection lubrication. In these cases the oil amount required for lubrication may be sufficient, but there may be a lack of cooling oil. This leads to high gear bulk temperatures resulting in thin separating films with higher friction and wear on the mating surfaces and therefore an increased risk of gear failures such as scuffing, pitting, micropitting and low speed wear.

Therefore the objective of this study was to investigate the limits concerning possible reduction

of lubricant quantity in gears which could be tolerated without detrimental effects on their load carrying capacity.

Test apparatus

Test gears

Table 1 shows the main geometrical values of the two gear types used in this investigation.

For all investigations on the flank load carrying capacity under minimized in comparison to rich lubrication conditions standard test gears (Table 1) were used. For the investigations on pitting and wear gear type C-PT ("PT" stands for pitting) was used. For evaluation of the micropitting load carrying capacity gear type C-GF ("GF" stands for Grauflecken – the German word for micropitting) according to [4] was used. The gear type C has a close to practical design with a well balanced sliding speed distribution along the path of contact. For the conducted scuffing tests gear type A according to [3] was used. Gear Type A has an uneven sliding speed distribution along the path of contact in order to increase the scuffing risk especially towards the tip of the pinion. The ground flanks of the test gears had a mean roughness of $R_a = 0.2 \mu\text{m}$ – $0.4 \mu\text{m}$ (gear type C-PT and A) respectively $R_a = 0.4 \mu\text{m}$ – $0.6 \mu\text{m}$ (gear type C-GF). The higher roughness of gear type C-GF increases the risk for the gear failure mode micropitting. The mean surface roughness is the only difference between C-PT and C-GF.

Table 1. Main geometrical data of test gears

Parameter	Symbol	Unit	C	A
Center distance	a	mm	91.5	91.5
Number of teeth pinion	z_1	--	16	16
Number of teeth gear	z_2	--	24	24
Normal module	m_n	mm	4.5	4.5
Normal pressure angle	α_n	°	20	20
Face width	b	mm	14	20
Profile shift factor pinion	x_1	--	0.182	0.864
Profile shift factor gear	x_2	--	0.172	-0.500
Material		--	16MnCr5	20MnCr5

Type of lubrication

For the investigations of power loss and bulk temperature for dip lubrication seven different immersion depths at standstill were chosen (Figure 1).

For the main scuffing, pitting, micropitting and wear tests an immersion depth of one and three times the module of the gear was chosen.

An oil/air lubrication device (Figure 2) was used to lubricate the gears with an oil-air mixture and the bearings with plain oil.

It is a lubrication system in which small measured quantities of oil from the reservoir are introduced into an air/oil mixing device which is connected to a lubricant supply line to the gear set. The air velocity transports the oil along the interior walls of the

lubricant line to the point of application. The continuous air stream to the gear set is only the transporting medium for very small oil quantities – only some milliliters per hour.

Oil/air lubrication is used for tests under severe starved lubrication conditions where the gear mesh is lubricated in order to decrease friction and wear, but no heat dissipation by cooling oil is available due to the very low oil quantities.

Lubricants

For the tests at different immersion depths and corresponding oil quantities the FVA reference oils FVA 3 pure, FVA2A and FVA3A (with a high EP-additive content in order to prevent scuffing during the efficiency and bulk temperature experiments), M220 (a mixture of FVA3A and FVA4A) and the CEC reference oil RL 133, were used (Table 2).

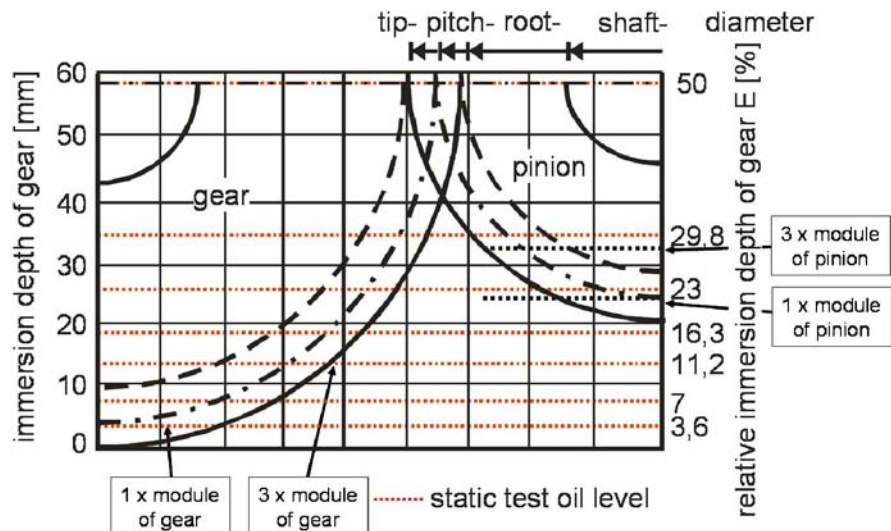
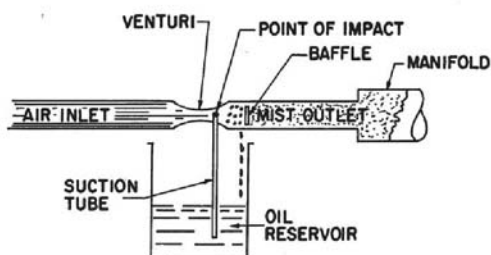


Figure 1. Oil levels at standstill for testing for dip lubrication

Oil/air lubrication:



spray nozzle

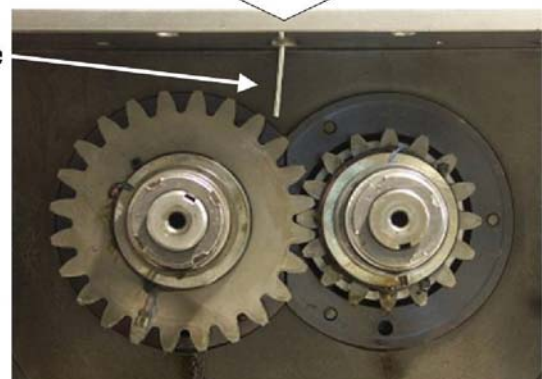


Figure 2. Oil/air lubrication device

Test rig

For all durability tests (pitting, micropitting and wear) and the scuffing tests, a standard FZG back-to-back test rig [5] (Figure 3) with a pitch line velocity of 8.3 m/s was used. For the wear tests at 0.05 m/s a speed reducer was installed between the motor and drive gear. For the test program at a pitch line velocity of 30 m/s one of the FZG test rigs was equipped with a variable speed three-phase asynchronous motor and a speed increaser.

The main elements of the test rig are the two connected gear boxes and an engine behind the

drive gear. The test gear box has an easily removable front and top plate in order to enable a quick change of the test gears. The two spur gear units (test and drive gear set) are connected by two parallel shafts.

In order to put a specific load on the gear flanks one of the shafts can be divided into two separate parts connected with the load clutch. With the load lever and different weights a defined torsion can be generated. Table 3 shows for the different load stages LS in the FZG scuffing, pitting and micropitting test the corresponding pinion torque T_1 and the Hertzian stress at the pitch point p_C .

Table 2. Properties of used lubricants

Name		RL 133	FVA2 + 4% A99	FVA 3 + 6.5% A99	FVA 3 pure	M220 + 4% A99
Abbreviation		RL133	FVA2A	FVA3A	FVA3	M220A
ISO VG		100	32	100	100	200
Kinematic viscosity at 40°C (mm ² /s)	ν_{40}	101.4	29.8	95	950	210
Kinematic viscosity at 100°C (mm ² /s)	ν_{100}	13.45	5.2	10.7	10.7	19
Density (at 15°C) (kg/m ³)	ρ_{15}	892	871	872	872	895

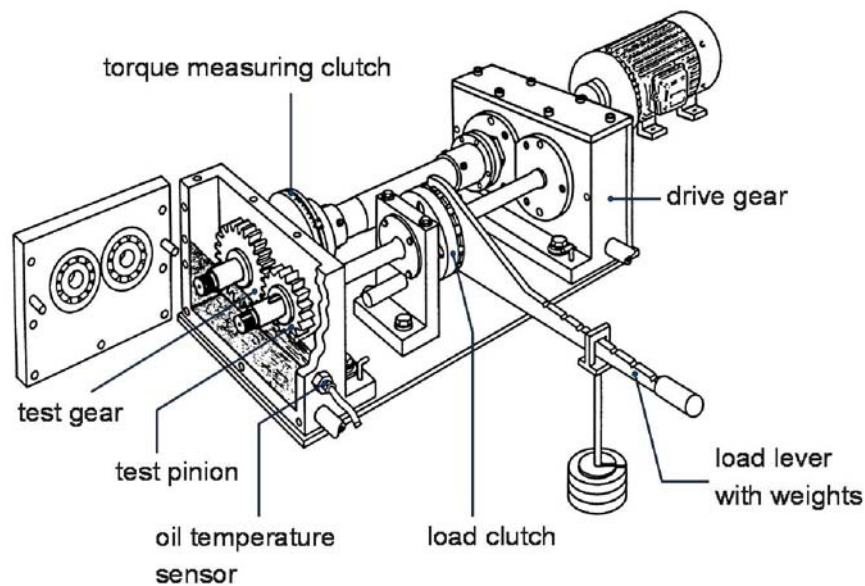


Figure 3. FZG back-to-back test rig