Efficiency and temperature of spray lubricated superfinished spur gears

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Abstract

Gearboxes are one of the most power dense systems used today, and in certain instances their limiting factor is the ability to evacuate heat from the gear contact. This work analyses the efficiency (i.e. heat generation) and tooth temperature in the three lubricating conditions dip, into mesh spray and out of mesh spray for superfinished gears which are then compared to ground gears. A back-to-back gear test rig is employed to test maximum contact pressures at the pitch of 0.59 to 0.96 GPa and pitch velocities from 0.5 to 20 m/s at a controlled lubricant temperature of 90 °C. The results show superfinished gears have higher mesh efficiency and lower gear tooth and bulk temperatures, hence lower heat flux compared to ground gears in all lubricating conditions.

Keywords: Spray lubrication, efficiency, gear temperature, superfinish

1. Introduction

To keep gearboxes simultaneously efficient and at low temperatures can be challenging.

Gear related power losses in gearboxes can be separated into load-independent and load-dependent losses, which origins from the gear mesh, the lubricant and windage of the gearbox medium. When gears are dip lubricated, load-independent losses can be reduced by lowering the immersion depth [1]. This effect is also seen when spray lubrication is used and there is no oil sump [2]. Lubricants with lower viscosities also reduced load-independent losses due to easier shear of the lubricant [3]. On the other hand, higher immersion depths of gears help to cool the gears and higher viscosity grades of lubricants separates gear flanks easier.

Load-dependent power losses related to the gear mesh have been studied previously. Xiao et al. suggested lower gear mesh power losses as the surface gets smoother [4]. Petry-Johnson et al. studied the difference between ground and chemically polished gears in dip lubrication and found a higher efficiency for superfinished gears [5]. Previous studies by the authors investigated the effect of two different running-in procedures on ground and superfinished gears

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and found the superfinished gears overall to be more efficient [6]. When testing spray lubrication, Britton et al. found a higher efficiency for superfinished gears compared to ground gears [7].

In ideal conditions the lubricant can cool and separate gear flanks in all running conditions. Höhn and Michaelis investigated gear damages and found lower risk of pitting and scuffing for lower lubricant temperatures [8]. Castro et al. investigated gear scuffing and suggested that there is a critical lubricant film thickness below which scuffing occurs [9]. Townsend and Shimski studied the influence of lubricant viscosity on the fatigue life of gears. They found a positive correlation of the fatigue lives of the tested gears and for calculated thicker lubricating films [10]. Krantz et al. compared fatigue life of ground (R_a 0.38 µm) and superfinished gears (R_a 0.07 µm) and found strong evidence for longer gear life for superfinished gears [11].

Previous work by the authors investigated spray lubricated ground gears regarding efficiency, gear temperature and surface roughness compared to dip lubricated ground gears. The results showed a higher total efficiency as well as higher gear temperatures for spray lubricated gears [12].

Previous work have shown superfinished gears to be more efficient and the importance of keeping lubricant temperatures low for longer gear lives [6, 7]. The temperature of gears need to be further investigated for a possible reduction of lubricant necessary for cooling. This work investigates the efficiency and temperature of spray lubricated superfinished gears compared to spray lubricated ground gears. The following questions are investigated; Is there and to what degree, a temperature difference between spray lubricated superfinished gears compared to ground gears, and to what extent are spray lubricated ground and superfinished gears cooled.

2. Method

2.1. Equipment

Tests were made in an FZG gear test rig with an efficiency setup, Figure 1. In each test, two unused pairs of gears with the same geometry and surface finish were mounted in the test gearbox (1) and the slave gearbox (3). To load the gears, dead weights were put on the load clutch (2). In this rig configuration power losses which occur equal the torque which the motor provides (5). The input torque is measured by the torque and speed sensor (4).

The gears used in all tests were FZG C-Pt gears (16MnCr5) with the inclusion of tip relief, Table 1. To enable temperature measurements in the gear wheel in the test gearbox during testing, two holes were drilled axially 44 and 55 mm from the gear wheel centre where two thermocouples were attached, represented by a green and blue dot respectively in Figure 2 . The shaft on the wheel side of the test gearbox had an extended shaft with a hole drilled where two thermocouples were pulled through to a slip ring.

2.2. Lubrication

A spray lubrication unit from Strama MPS was used to lubricate the gears during spray lubrication testing. The lubricant was injected with a velocity of 0.4 m/s corresponding to 0.025 l/s. A polyalphaolefin lubricant with nominal viscosities of 64.1 cSt at 40 °C and 11.8 cSt at 100 °C and a density of 837 kg/m^3 was used in all tests. The nozzle was positioned above the gear mesh (directly above the pitch point), Figure 2. When dip lubrication was used the oil level reached the centre of the gears.

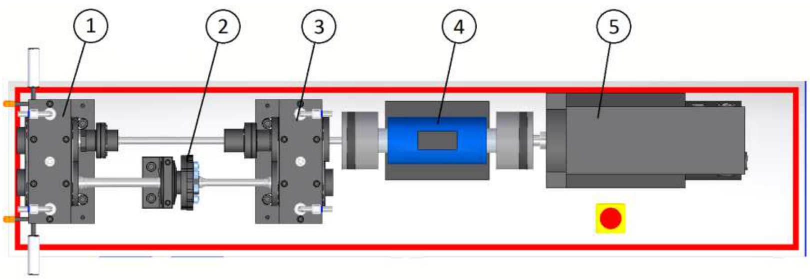

Figure 1: Top view of the FZG gear test rig, showing the test gearbox (1), load clutch (2), the slave gearbox (3), the torque and speed sensor (4) and the motor (5).

Table 1: FZG C-Pt geometrical parameters.

Parameter		Standard	Modified	Unit
Centre distance	a	91.5		mm
Face width	b	14		mm
Pitch diameter	d_{w1}	73.2		mm
	d_{w2}	109.8		mm
Tip diameter	d_{a1}	82.46		mm
	d_{a2}	118.36		mm
Module	m_n	4.5		mm
Number of teeth	z_1	16		
	z_2	24		
Addendum modification factor	x_1	0.18		
	x_2	0.17		
Pressure angle	α	20		°
Working pressure angle	α_w	22.44		°
Helix angle	β	0		°
Lead crowning	C_{b1}	0		μm
	C_{b2}	0		μm
Tip relief	C_{a1}	0	20	μm
	C_{a2}	0	20	μm
Starting diameter for tip relief	d_{g1}	0	80.3	mm
	d_{g2}	0	115.9	mm

2.3. Manufacturing method

The gear specimens in this work were first ground, and later superfinished. Superfinishing of gears in this work refers to a chemical mechanical process in which the gears are subject to in order to decrease their surface roughness. The gears superfinished came from the same batch as the ground gears. The R_a decreased from 0.3 µm to 0.1 µm. The superfinish process did not change the gear's macro-geometrical and micro-geometrical dimensions.

2.4. Experiment setup and surface measurements

Two unused pairs of gears with the same surface finish were used in each test. The gears were initially run-in at a maximum contact pressure of 1.66 GPa at the pitch for four hours, with a pitch velocity of 0.5 m/s and with a lubricant temperature controlled at 90 °C.

An efficiency test followed the running-in process. At maximum contact pressures of 0.59, 0.80 and 0.96 GPa (nominal torques of 35.5, 60.8 and 94.1 Nm on the pinion) at the pitch, the gears were tested at eight different pitch velocities for five minutes each. The tested velocities were 0.5, 1, 2, 3.2, 8.3, 10, 15 and 20 m/s. Tests were performed both using dip and spray lubrication. In both lubrication types the lubricant temperature was controlled at 90 °C. The gears were run in both normal (lubrication into mesh) and reverse (lubrication out of mesh) direction, Figure 2. In order to estimate the gear mesh efficiency, tests without any load applied (0 GPa) were performed to measure load-independent losses at each velocity. This was performed in dip lubrication and in both spray lubrication conditions. In all tests, input torque from the motor and temperatures from gear wheel thermocouples were sampled at one Hertz. All conditions were tested three times.

To follow the eventual surface roughness change of a gear wheel flank in a test, surface roughness profiles were measured in situ. The surface profiles were measured with a Form Talysurf Series 50 mm Intra 2 by Taylor Hobson. To position the Talysurf Intra in the same position between measurements, its holder was fixed with two pins drilled into the top of the test gearbox and then tightened with the same screws as for the lid. A spirit level was placed on two specific gear teeth of the wheel to position the gear wheel in the correct angular position. The stylus were equipped with a 2 µm tip and a positioning stage enabling traceable measurements. Six profiles were taken 100 µm apart. The profiles were 7 mm long and in the analysis, a cut-off length of 0.8 mm was used. Profile measurements were taken without disassembling the gear test rig. More about the profile measurements can be found in [13].

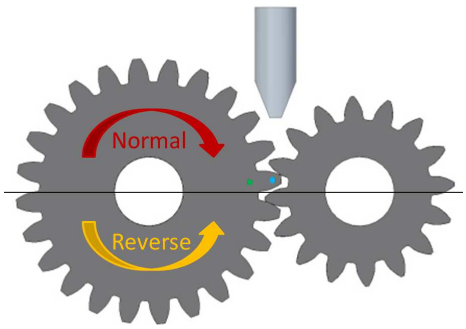

Figure 2: Sketch of the gears and the spray nozzle. In one tooth the wheel has two holes for one thermocouple each (blue and green dot). The red arrow shows when running in normal direction and the green arrow when running in reverse direction.

2.5. Efficiency estimation

The gearbox losses were separated into load-dependent and load-independent losses, equation 1. Load-dependent losses were summed up from those by bearings and gears under loaded conditions and load-independent losses from drag and churning of the lubricant caused by bearings, gears and seals. The measured load-independent losses were subtracted from the total losses to end up with the load-dependent losses.

$$T_{total} = T_{load-dependent} + T_{load-independent} \tag{1}$$

To calculate the gearbox efficiency, equation 2 was used. This was made by dividing the measured torque loss by the nominal torque transferred by the pinion (T_1) and gear ratio (u). The ratio was then multiplied by 0.5 for the efficiency of one gearbox.

$$\eta_{total} = 1 - 0.5\frac{T_{total}}{uT_1} \tag{2}$$

To estimate the gear mesh efficiency, a bearing model by SKF was first tested, but bearing frictional losses, for the gear contact pressures of 0.59 and 0.80 GPa, were overestimated at the higher speeds which yielded a mesh efficiency above 100 %. Instead, a bearing model developed at the Department of Machine Design KTH was used, equation 3. It was developed using a modified version of the gear test rig shown in Figure 1. The values for constants A, B and C are presented in Appendix 1. More information about the bearing model can be found in [14].

$$T_{STA,1,2} = An + \frac{B}{n} + C \tag{3}$$

It is important to note that equation 3 is a function only of rotational speed *(n)*; hence, for each oil, load, temperature, type of bearing and lubrication method a bearing test is needed and the data in this work is only valid for the conditions presented previously.

The load-dependent bearing losses were finally estimated with equation 4, where ω_2 is the ingoing angular speed of the motor. In order to calculate the frictional moment in the gear mesh, the load-dependent bearing losses were subtracted from the load-dependent losses, equation 5. The gear mesh efficiency could then be calculated using equation 6, where u is the gear ratio, T_1 the nominal torque and 0.5 for the mesh efficiency in one gearbox.

$$T_{bearing} = \frac{4T_{STA,1}\omega_2 1.5 + 4T_{STA,2}\omega_2}{\omega_2} \tag{4}$$

$$T_{mesh} = T_{load-dependent} - T_{bearing} \tag{5}$$

$$\eta_{mesh} = 1 - 0.5\frac{T_{mesh}}{uT_1} \tag{6}$$

The difference between measured tooth temperature (ϑ_{Tooth}) and measured bulk temperature (ϑ_{Bulk}) in the gear wheel (ΔWheel), is calculated with equation 7.

$$\Delta Wheel = \vartheta_{Tooth} - \vartheta_{Bulk} \tag{7}$$

3. Lubricant heat absorption estimation

To estimate the amount of heat absorbed by the lubricant in spray lubrication, a second test setup different from the efficiency tests was performed. To estimate the lubricant's heat absorption, the injection lubricant temperature from the nozzle was compared to the lubricant temperature at the outlet, measured with a thermocouple. This was performed for contact pressures of 0 GPa and 0.96 GPa. Pitch velocities of 0.5, 2, 8.3 and 20 m/s were run for two minutes each. After two minutes the gear wheel temperature had reached its median temperature. To reduce influence on the lubricant outlet temperature from the casing, the inside of the test gearbox was insulated internally with PTFE. PTFE plates with a thickness of five mm covered the bottom and the walls and a one mm thick plate was put under the gearbox lid. This test setup was run in both normal and reverse direction for ground and superfinished gears.

Lubricant outlet temperatures (ϑ_0) when run at 0 GPa were subtracted from outlet temperatures ($\vartheta_{0.96}$) when 0.96 GPa was run to achieve the temperature difference between nozzle and outlet ($\Delta\vartheta_{Lubricant}$) , equation 8.

$$\Delta\vartheta_{Lubricant} = \vartheta_{0.96} - \vartheta_0 \tag{8}$$

The amount of heat absorbed by the lubricant was calculated with equation 9. The lubricant had a density (ρ) of 837 kg/m^3, a specific heat capacity (C) of 2180 J/kgK and the amount of lubricant injected ($\dot{V}$) was 0.025 l/s. The injected lubricant temperature was 90 °C.

$$\dot{W} = C\rho\dot{V}\Delta\vartheta_{Lubricant} \tag{9}$$

By using equation 9, one degree Celsius difference will equal a change of 43 W in the oil energy balance.

4. Results

4.1. Gear efficiency and temperature

All three tests from the gear efficiency testing and temperature measurements are presented as the median, the minimum and the maximum value of each test. The results are presented as subplots (1×3), the tested maximum contact pressures increase from left to right, 0.59, 0.80 and 0.96 GPa respectively.

In Figure 3, the total gearbox efficiency of dip and spray lubricated ground and superfinished gears is compared. From a pitch velocity of 8.3 m/s the difference in efficiency between dip and spray lubrication is significant. The biggest difference is seen at 20 m/s at 0.59 GPa where dip lubricated gears yield an efficiency of 94.5 % compared to 97.5 % for the least efficient ground gear. At 0.96 GPa the difference has decreased and is around 1 % at 20 m/s between the two lubrication methods.

Consequently, the superfinished gears are more efficient than the ground gears. A higher efficiency between 0.1 to 0.2 % can be seen for superfinished gears.

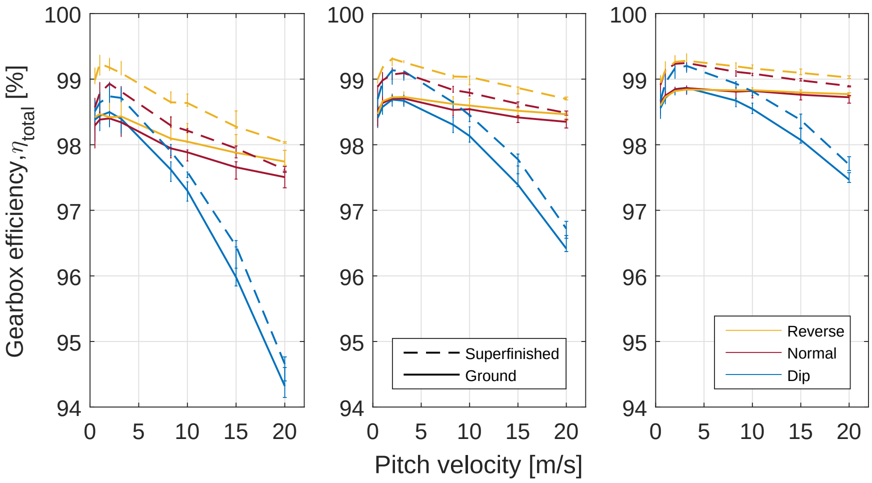

Figure 3: Gearbox total efficiency comparing ground and superfinished gears for dip and spray lubrication, run in normal and reverse direction, at contact pressures of 0.59, 0.80 and 0.96 GPa, presented from left to right. Ground gears are represented as solid lines and superfinished as dashed lines.

Gear mesh efficiency is presented in Figure 4. At 20 m/s, spray lubricated ground gears at 0.59 and 0.80 GPa have a median mesh efficiencies of 99.8 % compared to 99.9 % for superfinished gears. At 0.96 GPa the ground gears have a median mesh efficiency of 99.7 % while the superfinished gears still yield a median efficiency of 99.9 %. For all tested contact pressures, at 20 m/s there is a trend that dip lubricated gears have a slightly lower mesh efficiency than spray lubricated gears.

In Figure 5, the gear wheel tooth temperature is presented. The overall trend for the tooth temperature is that ground gears are warmer than superfinished gears. The difference between ground and superfinished is clearly seen as the

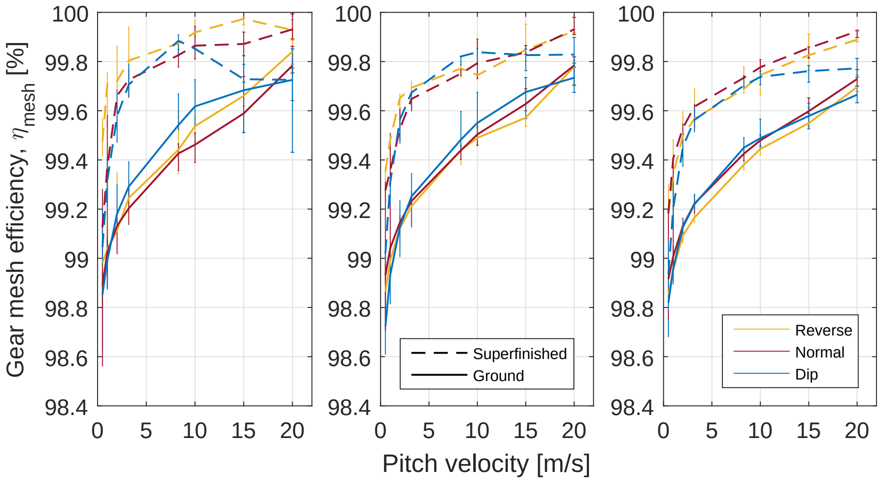

Figure 4: Gear mesh efficiency comparing ground and superfinished gears for dip and spray lubrication, run in normal and reverse direction, at contact pressures of 0.59, 0.80 and 0.96 GPa, presented from left to right. Ground gears are represented as solid lines and superfinished as dashed lines.

contact pressure and speeds are increased. At 0.96 GPa the biggest difference between the medians for ground and superfinished gears in reverse direction is 14 °C and 4 °C in normal direction. In dip lubrication the deviation between ground and superfinished is smaller, around 2 to 3 °C.

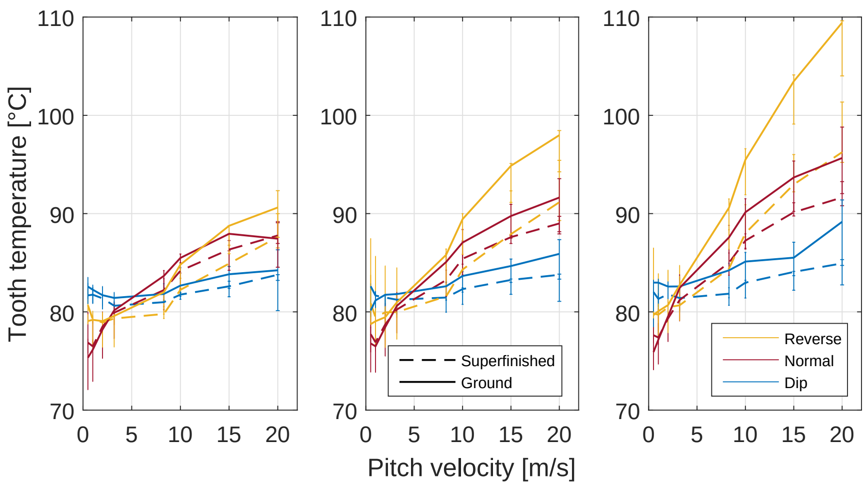

Figure 5: Measured gear tooth temperature comparing ground and superfinished gears for dip and spray lubrication, run in normal and reverse direction, at contact pressures of 0.59, 0.80 and 0.96 GPa, presented from left to right. Ground gears are represented as solid lines and superfinished as dashed lines.

The differences between tooth and bulk temperature measurements are presented in Figure 6. Above 3.2 m/s the medians for spray lubrication start to diverge, this is not as prominent in dip lubrication where the divergence is almost constant. In general the ground gears and gears run in the reverse direction have a higher temperature difference compared to superfinished gears and gears run in normal direction. The median temperature differences for dip lubricated gears are always lower compared to spray lubricated gears.

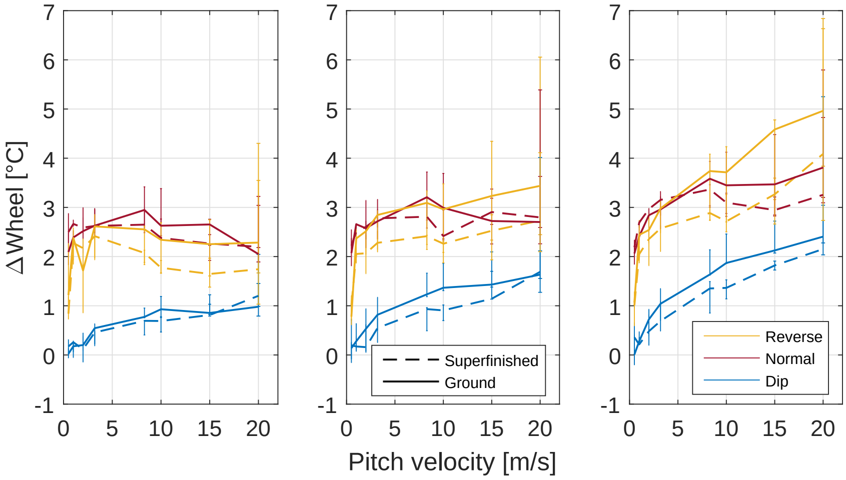

Figure 6: Measured differences between tooth and bulk temperature for ground and superfinished gears for dip and spray lubrication, run in normal and reverse direction at contact pressures of 0.59, 0.80 and 0.96 GPa, presented from left to right. Ground gears are represented as solid lines and superfinished as dashed lines.

Gear surface parameters from surface profile measurements for superfinished gears are presented in Table 2. The initial mean values of R_a 0.10, R_z 0.98 and R_{pk} 0.09 µm did not change neither after running-in nor after efficiency testing.

Table 2: Surface parameters measured in-situ from superfinished gears. Units are presented in µm.

	Initial	After running-in	After efficiency test		
			Dip	Normal	Reverse
$R_{a,mean}$	0.10	0.10	0.10	0.10	0.10
$R_{a,std}$	0.01	0.01	0.01	0.01	0.01
$R_{z,mean}$	0.98	0.95	0.97	0.96	0.97
$R_{z,std}$	0.31	0.28	0.32	0.33	0.28
$R_{pk,mean}$	0.09	0.10	0.10	0.10	0.10
$R_{pk,std}$	0.03	0.04	0.02	0.02	0.02

4.2. Lubricant heat absorption

In Figure 7 and 8, equation 9 is used to show the estimated heat absorbed by the lubricant in spray lubrication for ground and superfinished gears respectively. To relate the absorbed heat to other losses, gear mesh losses (T_{mesh}) presented in Watts are plotted as well. The median values for absorbed heat are presented to the left. Median, maximum and minimum values for gear mesh losses are presented to the right.

Both for ground and superfinished gears the lubricant absorbs slightly more heat when the gears are run in normal direction. The amount of heat absorbed is larger for ground gears than for superfinished. For the superfinished gears the amount of heat absorbed by the lubricant is larger than the gear mesh power loss at 20 m/s.

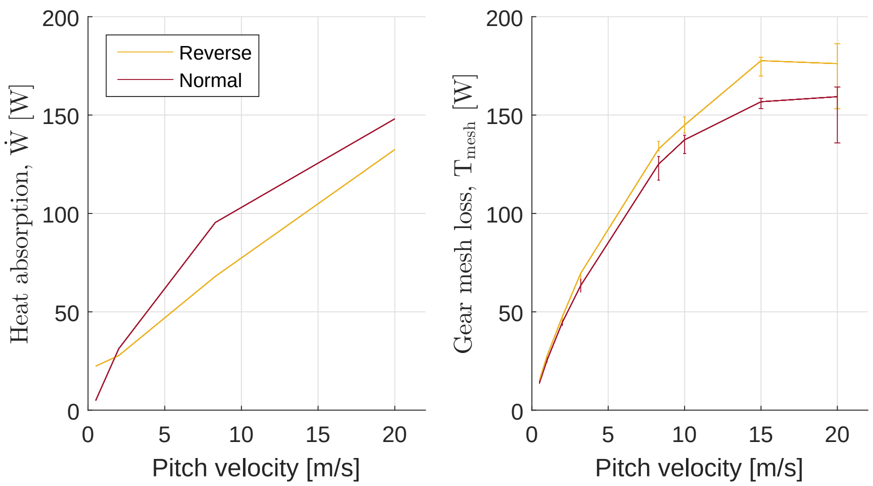

Figure 7: Heat absorbed by the lubricant is plotted on the left and mesh losses on the right for ground gears, at a contact pressure of 0.96 GPa.

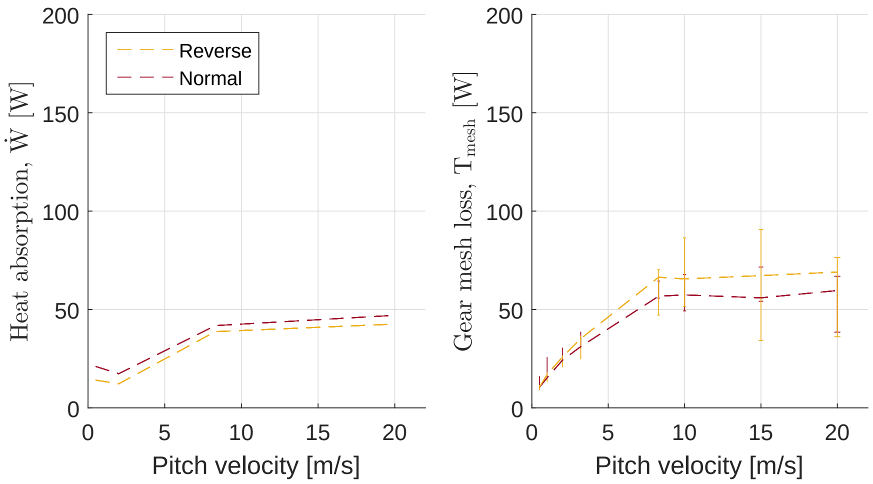

Figure 8: Heat absorbed by the lubricant is plotted on the left and mesh losses on the right for superfinished gears, at a contact pressure of 0.96 GPa.

5. Discussion

For all three tested contact pressures, the gearbox's total efficiency is higher for spray lubricated gears compared to dip lubricated gears, Figure 3. In spray lubrication there is no oil sump as in dip lubrication, and hence drag losses are significantly reduced. This result is in line with other authors who also investigated spray lubricated gears [2].

The higher gearbox efficiency for superfinsihed gears is due to the higher gear mesh efficiency, Figure 4. General higher mesh efficiencies for superfinished gears have also been seen previously when comparing dip lubricated ground gears to superfinished gears at contact pressures between 0.96 to 1.66 GPa [6]. Within the same surface finish, dip and spray lubricated gears have the same or similar mesh efficiency. However, from 10 m/s the mesh efficiency for dip lubrication at 0.59 and 0.80 GPa decreases, while at 0.96 GPa it does not increase as much as spray lubricated gears. Although the mesh efficiencies for the two spray running conditions are the same or similar, a higher load carrying capacity has been seen for gears run in normal direction [15].

For the tested parameters, the gear mesh efficiencies are almost 100 % for both manufacturing methods, and hence the gear mesh losses are small. However, when considering the reduction in gear mesh losses for superfinished gears compared to ground gears, it is significant. For spray lubrication at 10 m/s, the reduction in mesh losses is roughly 80 %, 60 %, and 60 % at each respectively contact pressure. For dip lubrication this effect is around 60 % lower mesh losses for all three tested contact pressures.

At 0.59 GPa the spread in efficiency is relatively large. At this contact pressure the signal to noise ratio is small, but as the contact pressure is increased, the spread in mesh efficiency decreases. Low loaded bearings such as in this case may not behave as expected. For instance, the rollers may not rotate as they should. In the section for bearing losses, the SKF handbook states that the bearing equations should not be used if the load is below 2 % of the critical load of the bearing [16]. In the performed tests the bearing load is 0.87 %, 1.50 % and 2.32 % of the critical load defined in the SKF handbook. However, the bearing model used in this work, equation 3, is developed for the NJ 406 cylindrical roller bearings specifically. They were tested in the same test rig as for the tested gears in this work, under the same loads and speeds used, with the same lubricant at a controlled temperature of 90 °C [14].

There is no or a small difference in mesh efficiency for the two spray running conditions. Nevertheless, the measured tooth temperatures are clearly affected by the direction of the lubricant spray (with respect to the rotational direction) and what manufacturing method is used, Figure 5, and hence the heat flux through the gears, Figure 6. At the highest contact pressure and speed the superfinished gears have a 20 % lower median temperature differece in reverse direction, and in normal direction this effect is around 15 % lower compared to ground gears.

Gear scuffing damages can origin from an increased lubricant temperature or a worse lubricant film formation, and may cause tempering of the gear [8, 9]. With lower measured gear temperatures, superfinished gears show a higher potential of reducing the risk of scuffing. The risk of pitting damages is also significantly reduced for superfinished gears compared to ground gears [11]. Under the tested conditions, to achieve the best cooling dip lubrication is to be preferred.

The amount of heat absorbed by the lubricant is slightly larger for gears run in normal direction, Figure 7 and 8. This corresponds to the lower tooth temperatures when run in normal direction, Figure 5. However, the absorbed heat by the lubricant is significantly lower than the increment in measured tooth temperature at higher speeds and contact pressures. Lubricant injection speeds which are equal to the gears, can possibly cool the gears better [17, 18]. In this work, the spread of the measured heat absorption by the lubricant is left out due to the difficulties of measuring temperatures in this test setup. Even though the spread is left out, heat absorbed by the lubricant is always lower than gear mesh losses, as expected.

The gear's roughness parameters did not change under these testing conditions, Table 2. Previous work by the authors, solely in dip lubrication, shows no change in surface roughness either [6]. The test procedure was the same except that higher contact pressures were tested, 0.96 to 1.66 GPa.

6. Conclusions

Dip and spray lubricated superfinished gears were tested at contact pressures between 0.59 to 0.96 GPa for pitch velocities between of 0.5 to 20 m/s. Spray lubricated gears were lubricated before ingoing mesh (normal direction) and at outgoing mesh (reverse direction). The tested superfinished gears were compared to previously tested ground gears run under the same test conditions.

The following conclusions can be made from this work:

- From a pitch velocity of 8.3 m/s, a higher gearbox efficiency is seen for spray lubricated gears compared to dip lubricated gears.

- Median mesh efficiencies for superfinished gears are higher at all test conditions compared to ground gears.

- Superfinished gears have lower median tooth and bulk temperatures and hence lower heat flux compared to ground gears.
- No change in surface roughness for the tested superfinished gears under these test conditions.
- The amount of heat absorbed by the lubricant is higher for gears lubricated before the ingoing mesh (normal direction) compared to after the outgoing mesh (reverse direction).

7. Conflict of interest

None declared.

Acknowledgments

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References

[1] Bernd-Robert Höhn, Klaus Michaelis, and Hans-Philipp Otto. Influence of immersion depth of dip lubricated gears on power loss, bulk temperature and scuffing load carrying capacity. *International Journal of Mechanics and Materials in Design*, 4(2):145–156, 2008.

[2] Bernd-Robert Höhn, Klaus Michaelis, and Hans-Philipp Otto. Minimised gear lubrication by a minimum oil/air flow rate. In *ASME 2007 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*, pages 717–725. American Society of Mechanical Engineers, 2007.

[3] AS Terekhov. Hydraulic losses in gearboxes with oil immersion. *Russian Engineering Journal*, 55(5):7–11, 1975.

[4] Li Xiao, B-G Rosen, Naser Amini, and Per H Nilsson. A study on the effect of surface topography on rough friction in roller contact. *Wear*, 254(11):1162–1169, 2003.

[5] Travis T Petry-Johnson, A Kahraman, NE Anderson, and DR Chase. An experimental investigation of spur gear efficiency. *Journal of Mechanical Design*, 130(6):062601, 2008.

[6] Martin Andersson, Mario Sosa, and Ulf Olofsson. The effect of running-in on the efficiency of superfinished gears. *Tribology International*, 93:71–77, 2016.

[7] RD Britton, CD Elcoate, MP Alanou, HP Evans, and RW Snidle. Effect of surface finish on gear tooth friction. *Journal of tribology*, 122(1):354–360, 2000.

[8] B-R Höhn and K Michaelis. Influence of oil temperature on gear failures. *Tribology International*, 37(2):103–109, 2004.

[9] J Castro and J Seabra. Scuffing and lubricant film breakdown in fzg gears part i. analytical and experimental approach. *Wear*, 215(1):104–113, 1998.

[10] Dennis P Townsend and John Shimski. Evaluation of the ehl film thickness and extreme pressure additives on gear surface fatigue life. 1994.

[11] TL Krantz, MP Alanou, HP Evans, and RW Snidle. Surface fatigue lives of case-carburized gears with an improved surface finish. *Journal of tribology*, 123(4):709–716, 2001.

[12] Martin Andersson, Mario Sosa, and Ulf Olofsson. Efficiency and temperature of spur gears using spray lubrication compared to dip lubrication. *Submitted to Journal of Engineering Tribology*, 2016.

[13] Mario Sosa. *Running-in of gears from a surface transformation and efficiency point of view.* Licientiate thesis, Royal Institute of Technology (KTH), Department of Machine Design, 2015.

[14] Mighui Tu. Validation and modeling of power losses of nj406 cylindrical roller bearings. Master's thesis, Royal Institute of Technology (KTH), Department of Machine Design, 2016.

[15] VN Borsoff. On the mechanism of gear lubrication. *Journal of Basic Engineering*, 80:79–93, 1959.

[16] SKF. *SKF Handbook*. SKF, 2012.

[17] LS Akin and DP Townsend. Into mesh lubrication of spur gears with arbitrary offset oil jet. part 1: For jet velocity less than or equal to gear velocity. *Journal of Mechanisms, Transmissions, and Automation in Design*, 105(4):713–718, 1983.

[18] LS Akin and DP Townsend. Into mesh lubrication of spur gears with arbitrary offset oil jet. part 2: For jet velocities equal to or greater than gear velocity. *Journal of Mechanisms, Transmissions, and Automation in Design*, 105(4):719–724, 1983.

Appendix 1

The constants used in equation 3 are presented in Table 3 and Table 4.

Table 3: Constants for A, B and C in spray.

	0.59 GPa	0.80 GPa	0.96 GPa
A	1.590e-05	1.957e-05	2.225e-05
B	1.821	3.303	3.354
C	-0.005123	-0.006502	-0.004118

Table 4: Constants for A, B and C in dip.

	0.59 GPa	0.80 GPa	0.96 GPa
A	1.438e-05	2.185e-05	2.666e-05
B	0.4556	1.262	3.403
C	-0.001312	-0.005676	-0.009950

Nomenclature

$\Delta\vartheta_{Lubricant}$ Change in lubricant temperature from nozzle to outlet [°C]

$\Delta Wheel$ Measured difference between tooth and bulk temperature in the gear wheel [°C]

$\dot{V}$ Lubricant volume injection [l/s]

$\dot{W}$ Amount of heat absorbed by the lubricant [W/s]

η_{mesh} Gear contact efficiency with respect to transmitted power [-]

η_{total} Gearbox total efficiency with respect to transmitted power [-]

ω_2 Ingoing angular velocity [rad/s]

ρ Lubricant density [kg/m^3]

ϑ_{Bulk} Measured bulk temperature in the gear wheel [°C]

ϑ_{Tooth} Measured tooth temperature in the gear wheel [°C]

T_1 Pinion inside power loop torque [Nm]

$T_{bearing}$ Torque loss from bearings [Nm]

$T_{load-dependent}$ Load-dependent torque loss [Nm]

$T_{load-independent}$ Load-independent torque loss [Nm]

T_{mesh} Equivalent gear contact torque loss [Nm]

$T_{STA,1,2}$ Torque loss from a bearing [Nm]

T_{total} Measured torque loss [Nm]

u Gear ratio [-]

C Lubricant specific heat [J/kgK]

n Rotational speed [rev/s]