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Efficiency and temperature of spur gears using spray lubrication compared to dip lubrication

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Abstract

Increased machine efficiency is a requirement in today's world and geared transmissions are no exception. A back-to-back gear test rig was used to compare dip lubrication with spray lubrication regarding gearbox efficiency, mesh efficiency, gear temperature and surface roughness. Gears lubricated at the inlet of the mesh show a lower measured temperature when compared to spray lubrication at the outlet of the mesh. Spray lubrication, when compared to dip lubrication, yields the same efficiency for both rotating directions at the tested speeds of 0.5 to 20 m/s. Spray lubrication shows a significantly higher total gearbox efficiency at higher speeds, higher measured tooth temperature and no measurable change in surface roughness.

Keywords

Spray lubrication, gear efficiency, gear temperature, surface roughness

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Introduction

Reduced load-independent power losses in gear transmissions can lower the fuel consumption in motor vehicles. The sources of load-independent power losses in gearboxes have been studied in the past decades. Anderson studied windage losses in gearboxes and found higher losses as the gear speed was increased.[1] Diab et al. found that load-independent power losses were reduced when gears with smaller pitch diameter and tooth distance were used.[2] Seetharaman et al. reduced load-independent power losses from gears by reducing the gear face width.[3]

Load-independent power losses are also related to the lubricant. Using dip lubrication, Höhn, Michaelis and Otto showed that a decreased immersion depth of gears can reduce load-independent power losses.[4] Terekhov found that a decreased lubricant viscosity yields lower load-independent power losses as well.[5]

To use spray lubrication in which the lubricant is injected at the gears through a nozzle instead of dip lubrication, decreases the load-independent power losses inside the gearbox.[6] When using spray lubrication, the position of spray nozzles will affect the lubrication and cooling of the gears. It is recommended to spray into the mesh inlet for better film formation and at the mesh outlet for better cooling.[7]

In spray lubrication mode, when the lubricant was sprayed into the gear mesh inlet, the best lubrication and cooling was achieved when the injection speed of the lubricant was the same as the pitch line velocity.[8,9]

The position of the spray nozzle affects the life of the gears, in the sense that a shorter distance from the mesh exit has a positive effect on the gear life with less likelihood of scuffing damages and failed gears.[10] However, if a position closer to the mesh exit is best for both lubrication and cooling was not clear. Other authors have found a better load carrying capacity using spray lubrication when the spray nozzle is positioned at the mesh inlet.[11]

The research questions in this paper are the following: is there a difference in efficiency (1), measured tooth temperature (2) and surface roughness (3) when comparing dip lubrication with spray lubrication, when sprayed at the mesh inlet (normal running direction) and at the mesh outlet (reverse running direction).

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Method

Equipment

All tests were performed in an Forschungsstelle fär Zahnrder und Getriebebau (FZG) gear test rig with an efficiency setup, Figure 1. In each test, two unused pairs of gears (16MnCr5) with the same geometry 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 torque losses which occur in the gearboxes equals the torque which is provided by the motor (5). The input torque is measured by the torque and speed sensor (4).

The gears used in all tests had the geometry of FZG C-Pt gears with the inclusion of tip relief (Table 1). To enable temperature measurements in the test gearbox during testing, the tested gear wheels had two holes drilled axially 44 and 55 mm from the gear wheel centre where two thermocouples were inserted (Figure 2). The thermocouple inserted at 44 mm is called bulk and the thermocouple inserted at 54 mm is called tooth. The thermocouples were of type K and with an accuracy of $\pm$ 0.5 °C. The thermocouples were attached with an epoxy glue (LOCTITE 9497). The shaft on the wheel side of the test gearbox had an extended shaft with a hole drilled in the axial direction, where the two thermocouples were pulled through to a slip ring from where the temperature was sampled.

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 25 ml/s. A polyalphaolefin lubricant (PAO) 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), and when gears were dip lubricated, an immersion depth to the centre of the gears was used (Figure 2).

Test setup and surface measurements

Two unused pairs of ground gears were used in each test. An initial running-in process was made at a contact pressure of 1.66 GPa at the pitch for four hours with a pitch velocity of 0.5 m/s and 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, corresponding to nominal torque of 35.5, 60.8 and 94.1 N m 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 (87, 174, 348, 550,

Table 1. FZG C-Pt geometrical parameters with the inclusion of tip relief.

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

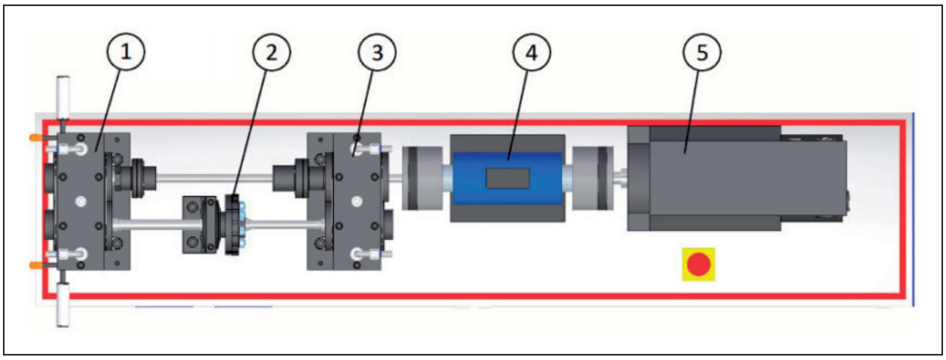

Figure 1. Top view of the FZG gear test rig showing the test gearbox (1), load clutch (2), the slave gearbox (3), the torque and speed sensor (4) and the motor (5).

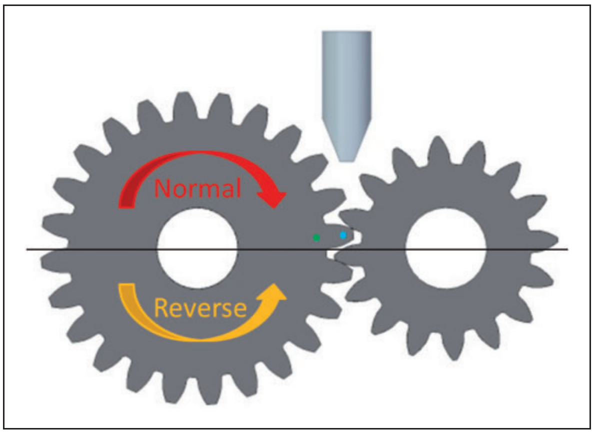

Figure 2. The black line shows the immersion depth of the gears during dip lubrication. The red and the yellow arrows show the running direction during spray lubrication. The blue dot and the green dot show the placement of the thermocouples, tooth and bulk respectively. The spray nozzle is placed directly above the gear mesh.

1444, 1740, 2609, 3479 r/min of the gear wheel). In dip lubrication, an immersed pipe in the lubricant in which 10 °C water passes through cooled the lubricant. Inside the test gearbox, the temperature was measured by a PT100 thermal resistance. In spray lubrication, the lubricant temperature was measured directly before injection. In both cases the lubricant temperature was controlled at $90 \pm 3\,^{\circ}C$. This procedure was run from lowest to highest load and running all eight velocities at each load. In spray lubrication, the gears were run in both normal and reverse direction (Figure 2). In order to estimate the gear mesh efficiency, tests without any load applied (0 GPa) were made first to measure load-independent power losses at each velocity in dip lubrication and for both rotational directions using spray lubrication. In all tests, the power input from the motor, the lubricant temperature and the gear wheel bulk and tooth temperature were sampled at one Hertz. A total of three identical tests were performed at each condition.

The measurement uncertainty taking into account only the uncertainties in the measurement device, (not the possible spread in the bearing friction) is ± 0.075 % ± 0.043 % and ± 0.028 % for 35.5, 60.8 and 94.1 N m respectively.

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 was equipped with a 2 μm tip and a positioning stage enabling traceable measurements. Six profiles were taken 100 μm apart. Each profile measurement was 7 mm long. In the analysis, a cut-off length of 0.8 mm and a Gaussian filter was used. Profile measurements were taken without disassembling the gear test rig. More about the profile measurements can be found in Sosa et al.[12]

Efficiency estimation

The total gearbox losses were separated into load-dependent and load-independent losses, equation (1). Load-dependent losses were summed up from those originating from 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

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

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

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

To estimate the gear mesh efficiency, a bearing model was used. First a bearing model from SKF was applied, as done previously by the authors.[13,14] However, the bearing frictional losses for the tested gear contact pressures of 0.59 and 0.80 GPa were overestimated at the higher speeds which yielded a mesh efficiency above 100%; these loads are also below the SKF model's validity. Instead, a bearing model developed at the Department of Machine Design KTH was used, equation (3), where n is the r/min of the bearing and subscript 1 and 2 are for bearings on the pinion and the gear shaft respectively. It was developed using a modified version of the gear test rig shown in Figure 1. The bearing testing was performed by replacing the gear wheel in the slave gearbox with two NJ 406 cylindrical roller bearings. Loads between 527 to 4509 N were tested, for example loads 527, 908 and 1405 N which corresponds to the gear torques of 35.5, 60.8 and 94.1 N m on the pinion used in this work. The tested rotational speeds were the same as for the gears, 87 to 3479 r/min. In the bearing testing the same PAO lubricant was used. The immersion depths were to centre of the bearings, same as for dip lubricated gears, and 29 mm up on the bearings (half bearing rollers) to mimic a low oil supply for spray lubricated gears. The lower oil level of was determined as the lowest acceptable by the authors. The values for constants A, B and C are presented in Appendix 2. More information about the bearing model can be found in Tu.[15] The gear test rig has eight NJ 406 cylindrical roller bearings

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

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

Load-dependent bearing losses were in the end 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 with equation (6)

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

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

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

Results

Results for efficiency and temperature measurements are presented as the median together with the minimum and maximum value. In the presented figures, the contact pressures increases from left to right. The total efficiency of one gearbox for all three lubrication conditions is shown in Figure 3. For all tested contact pressures, the gearbox efficiency for dip and spray lubrication diverge from 3.2 m/s where spray lubrication is more efficient, and as the contact pressure increases a higher efficiency is yielded. The median for spray lubrication in reverse direction is slightly higher compared to normal direction.

The gear mesh efficiencies are presented in Figure 4. For the tested contact pressures and speeds, the gear mesh efficiencies vary from 98.60% at 0.5 m/s to 99.80% at 20 m/s. The efficiencies overlap each other except at 20 m/s where a trend of a lower efficiency for dip lubrication can be seen when compared to spray lubrication.

In Figure 5, the temperature measured from the thermocouples in the gear wheel teeth are presented. The measured gear wheel tooth temperature starts around 80 °C for all tested contact pressures and lubrication conditions. As the speed increases, the measured temperature is consequently higher for spray lubrication in reverse direction. The lowest temperature is measured for dip lubrication and the highest tooth temperature when using dip lubrication reached 90 °C at 20 m/s, same as the controlled lubricant temperature.

Table 2 shows surface parameters from testing dip lubrication, spray normal direction and spray reverse direction. After the running-in procedure, the surfaces do not change under the used test conditions.

Discussion

In Figure 3, the comparison between dip lubrication (half the gears immersed) and the two spray methods (normal and reverse direction at a volumetric flow of 25 ml/s) described in the method section, in terms of overall total efficiency is shown, with both spray methods proving to be more efficient from a pitch velocity of 3.2 m/s. This is due to the reduction of load-independent losses by the removal of the oil

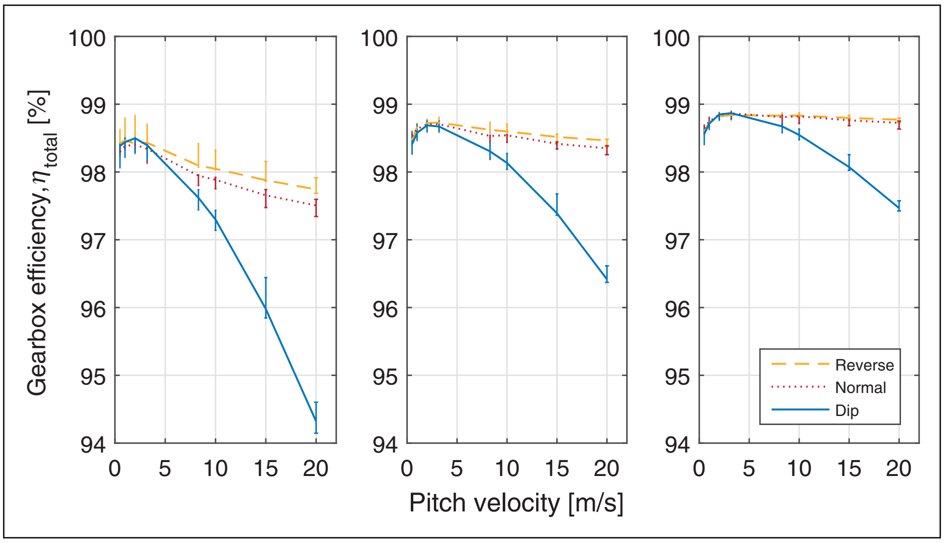

Figure 3. Total gearbox efficiency at contact pressures of 0.59, 0.80 and 0.96 GPa. From 3.2 m/s the gearbox efficiency is significantly lower when dip lubrication is used.

sump which creates churning and windage losses in both the gears and the bearings, which has also been shown by other authors.[3,6] The total efficiency of dip lubrication is in line with previous work.[13]

Overall, dip and spray lubrication have the same mesh efficiency from 0.5 to 20 m/s, Figure 4. However, while at a pitch velocity of 20 m/s, dip lubrication hints at a slightly lower mesh efficiency. The increasing trend of gear mesh efficiency from lowest to highest speed for all loads, is due to increasing film thickness between the gear teeth. Transitions in lubrication regimes has previously been covered in the literature.[16] Utilising the same test equipment as shown in Figure 1, film thickness formation between gear teeth has previously been investigated by the authors, and by other researchers as well.[14,17] The gear mesh efficiency is related to the lubricant temperature i.e. the viscosity, and if one compares the controlled lubricant temperature (90 °C) to the maximum measured tooth temperature (110 °C), Figure 5, the change in viscosity is roughly 4 cSt for the lubricant used in this work. This decrease in viscosity is considered to be negligible here. That spray lubrication can be more efficient at higher velocities can be explained with the fact that the lubricant is sprayed and directed into the contact while at dip lubrication the lubricant is thrown off the gear teeth due to the centrifugal force.[18] However, more tests at higher pitch velocities should be made to clarify whether there is a difference in mesh efficiency at 20 m/s. At lower velocities the

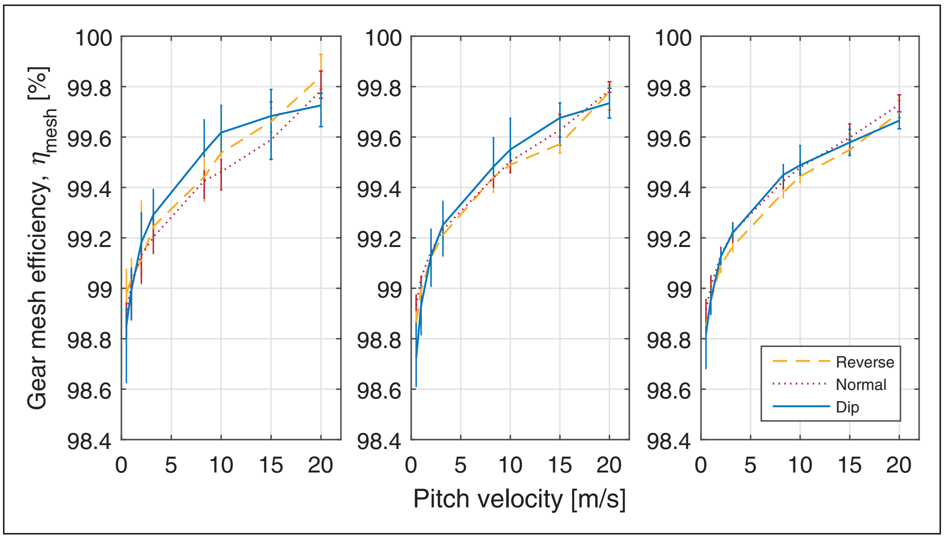

Figure 4. Gear mesh efficiency at a contact pressure of 0.59, 0.80 and 0.96 GPa. At 20 m/s spray lubrication has a trend of starting to be more efficient than dip lubrication.

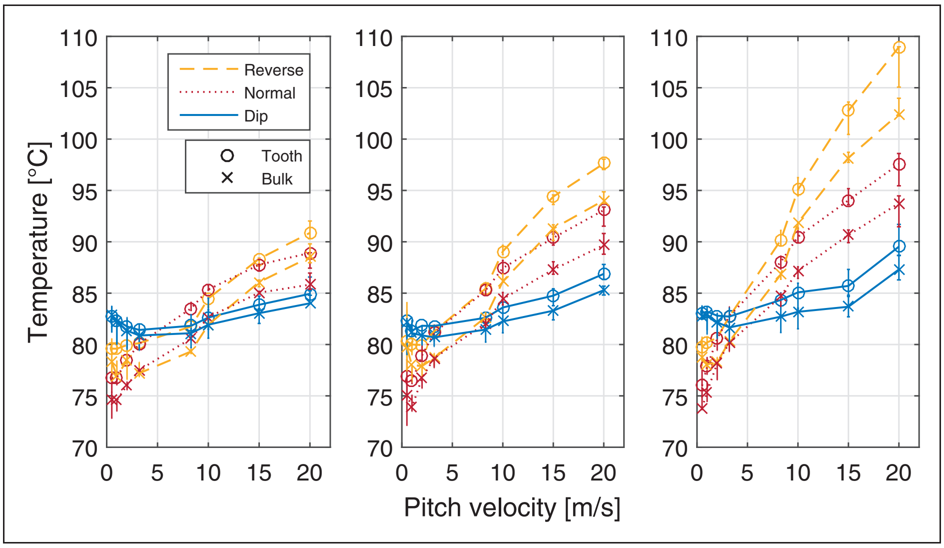

Figure 5. Measured gear wheel temperatures at contact pressures of 0.59, 0.80 and 0.96 GPa during testing. The highest tooth temperature was reached when running in reverse direction.

Table 2. Mean and standard deviation of surface parameters. Units are presented in μm.

	Initial	After running-in	After efficiency test Dip	Normal	Reverse
$R_{a,mean}$	0.45	0.39	0.39	0.39	0.38
$R_{a,std}$	0.01	0.01	0.02	0.01	0.01
$R_{z,mean}$	3.20	2.66	2.66	2.64	2.75
$R_{z,std}$	0.12	0.08	0.13	0.13	0.10
$R_{pk,mean}$	0.43	0.26	0.26	0.26	0.28
$R_{pk,std}$	0.06	0.06	0.06	0.05	0.05

frictional losses in the gear mesh can be the same even when the immersion depth of the gears is reduced, which has been shown previously.[19] This points to that the contact condition in this case is essentially the same during dip and spray lubrication, with the same lubricant temperature, film thickness, contact pressure and surface roughness.

To investigate this further, a transparent gearbox front was used instead of the usual cast iron one and the lubricant distribution was visually inspected. The first observation was that the inlet lubricant clearly covers both gear teeth. When running normal direction the lubricant and the gears are moving in the same direction and the lubricant can pass through the contact, while in reverse the lubricant and gears are colliding. In reverse direction from a pitch velocity of 8.3 m/s, the gear velocity was so high that the lubricant was flung up towards the lid by the gears. Even though the active flanks of the gears are more exposed to the lubricant, the gears are not cooled as well as in normal direction, Figure 5. When running in normal direction, the inlet lubricant is directly hitting the opposite flank, which is better for cooling anyway. For a better cooling at the outlet of the mesh, the nozzle should be directed at the active flank. From this brief investigation it is clear that the lubricant should be injected at a higher velocity for a better cooling of the gears.

Another possible way to reduce the tooth temperature is to conduct away the generated heat quicker or simply generate less frictional heat. This can perhaps be done by a different steel composition or by possibly using smoother gear flanks and is something that should further be investigated in spray lubrication.

The surface parameters, shown in Table 2, show a change in surface topography from initial state to after running-in but no changes due to efficiency testing. This further indicates that the contact condition when spray lubrication was used is the same as when dip lubrication was used. Tests performed with the same conditions except a higher contact pressure (1.66 GPa) in previous work using dip lubrication, the surface parameters slightly changed in efficiency testing.[13] With the positive effect of reducing the load-independent power losses, it should be investigated further whether the surface can change under the conditions when spray lubrication is used. For example, the measured tooth temperature differences between dip and spray lubrication in Figure 5 could influence in the long run, and hence longer test times should be implemented in the future. It should be noted that the reason for choosing a maximum pressure of 0.96 GPa for efficiency testing was the author's caution in avoiding possible scuffing damages at higher loads.

This work is in line with results presented by Greiner.[20] However, no comparison between spray and dip lubrication was presented, higher pitch velocities were studied (20 to 60 m/s) and no surface measurements were performed in Greiner's work. When comparing spray nozzle positions at the inlet and outlet of the mesh for different oil flows at a pitch velocity of 60 m/s, the biggest difference in mesh efficiency was 0.07%. Similar results are presented here for pitch velocities between 0.5 to 20 m/s, Figure 4. There is no significant difference in mesh efficiency between normal and reverse direction. Another similarity is the temperature rise in the gear wheels when running reverse compared to normal direction, where a higher temperature is measured for all tested injection velocities.

Conclusions

The conclusions below are based on comparing dip lubricated ground gears to spray lubricated ground gears. The spray lubrication of 25 ml/s was either made at the gear mesh inlet (normal) or at the gear mesh outlet (reverse). Tests were performed at maximum contact pressures of 0.59, 0.80 and 0.96 GPa at the pitch, with a controlled lubricant temperature of 90 °C and at pitch velocities between 0.5 to 20 m/s.

- Gearbox total efficiency is higher when spray lubrication is used from pitch velocities of 3.2 m/s compared to dip lubrication.
- A higher measured tooth temperature was present in spray lubrication in reverse direction compared to normal direction, with dip lubrication having the lowest measured tooth temperature.
- Mesh efficiency was indistinct between spray and dip from 0.5 to 15 m/s, with a possible difference at 20 m/s.
- No measurable change in surface roughness using spray lubrication under these test conditions.

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Declaration of Conflicting Interests

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Appendix 1

Notation

n	Rotations per minute of the bearing (r/min)
T_1	Pinion inside power loop torque (N m)
$T_{\text{bearing,STA}}$	Torque loss from bearings (N m)
$T_{\text{load-dependent}}$	Load dependent torque loss (N m)
$T_{\text{load-independent}}$	Load independent torque loss (N m)
T_{mesh}	Equivalent gear contact torque loss (N m)
T_{STA}	Torque loss from one bearing (N m)
T_{total}	Measured torque loss (N m)
u	Contact ratio (-)
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)

Appendix 2

The constants used in equation (3) are presented in Tables 3 and 4.

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

	0.59 GPa	0.80 GPa	0.96 GPa
A	1.590×10^{-5}	1.957×10^{-5}	2.225×10^{-5}
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	1.438×10^{-5}	2.185×10^{-5}	2.666×10^{-5}
B	0.4556	1.262	3.403
C	−0.001312	−0.005676	−0.009950