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Bearing Power Losses with Water-Containing Gear Fluids

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Abstract: Lubricants have a large influence on gearbox power losses. Recent investigations at a gear efficiency test rig have shown the high potential of water-containing gear fluids in drastically reducing load-dependent gear losses and temperatures. In this study, the bearing power losses with water-containing gear fluids were evaluated at a specific bearing power loss test rig explicitly and compared with mineral and polyalphaolefine oils. For all investigated lubricants, a Stribeck curve behavior of the load-dependent losses is observed. The water-containing gear fluids demonstrate lower no-load bearing losses and higher load-dependent bearing losses at higher rotational speeds. The comparison of measured bearing losses with typical calculation procedures showe partially large differences. The results underline the importance of having detailed knowledge of bearing losses when evaluating gear losses in gearboxes.

Keywords: EHL; friction; efficiency; bearings; water-containing fluids

1. Introduction

Bearings in gearboxes ensure guidance of shafts and bearing of axial and radial forces. Bearing power losses can be divided into no-load and load-dependent losses. Hinterstoißer et al. [1,2] and Jurkschat et al. [3] used a bearing power loss test rig to analyze the influence of operating conditions on roller bearing losses under dip lubrication. Their results show a degressive increase of no-load losses with increasing rotational speed. In case of load-dependent losses, a Stribeck curve behavior of the measured load-dependent losses was observed with increasing rotational speed. Aul et al. [4] investigated, based on a bearing power loss test rig and multibody simulations, the influence of operating conditions and bearing size on the frictional behavior of roller bearings. Their findings show increasing bearing losses with increasing load and bearing size. Talbot et al. [5] examined the power losses of a planetary gear set with different needle bearings and found increasing losses with increasing load and rotational speed. They also show the potential of double-row needle bearings in reducing the measured power losses by up to 23% compared to single-row needle bearings.

Several authors investigated the influence of base oil and viscosity on bearing losses. Hinterstoißer et al. [1,2] show that the measured load-dependent losses of a mineral, polyalphaolefine, and polyether oil evaluated at the same operating viscosity are comparable. The reduction of the viscosity from approximately 10 to 5 mm^2/s resulted in a decrease of the measured total losses by up to 40% at high rotational speeds. Balan et al. [6] and Koryciak [7] confirmed these results. Thereby, besides the lower frictional losses, a lower viscosity also simplifies the motion of the rolling elements and the displacement of the lubricant.

Hinterstoißer et al. [2] also investigated the influence of the immersion depth on bearing losses. Their results show negligible influence on the measured load-dependent losses. However, the no-load losses decreased, especially at a higher rotational speed, resulting in a decrease of no-load losses by up to 20%. Koryciak [7] confirmed these results and show decreasing bearing losses along with a reduction of the immersion depth by up to 50%. Aul et al. [8] conducted experimental investigations under minimum quantity lubrication and found decreased bearing losses of up to almost 40% compared to dip lubrication.

Jurkschat et al. [9] and Koryciak [7] compared measured bearing losses with calculated ones according to SKF [10] and show partially large deviations. According to Jurkschat et al. [9], the calculation methods can be improved if the influence of the base oil, additives, and the heat balance of the bearings is taken into account more precisely. Wang [11] developed a calculation model based on local contact conditions in bearings. This calculation model was implemented in the program LAGER2HP (Wang et al. [12]). Schleich [13] also developed a calculation model based on the load distribution in bearings to calculate bearing power losses.

Recent investigations by the authors have shown that water-containing gear fluids can strongly reduce friction in elastohydrodynamically lubricated (EHL) contacts. Experimental investigations conducted by Yilmaz et al. [14] at a twin-disk test rig show measured coefficients of friction smaller than 0.01 for a wide range of operating conditions. This is commonly referred to as superlubricity (Hirano et al. [15]). In addition, the film thickness measured at an EHL tribometer in [14] demonstrated a good lubricant film formation of water-containing gear fluids. A pressure–viscosity coefficient of approximately 6 1/GPa was derived. In [16], the loss and thermal behavior of these water-containing gear fluids was investigated at a gear efficiency test rig. Mean gear coefficients of friction smaller than 0.01 and considerably lower gear bulk and steady-state excess temperatures were found. The results show a great potential of water-containing gear fluids to reduce friction and improve gearbox efficiency compared to conventional gear oils. This is accompanied by good calorific properties. Challenges with water-containing gear fluids are to avoid vaporization of water and incompatibilities with materials.

The mean gear coefficients of friction identified by the authors in [16] were derived with the load-dependent bearing losses presented in this study. Therefore, the aim of this study is to provide a detailed insight into these results obtained from a specific bearing power loss test rig and into the evaluation of the loss behavior of roller bearings with water-containing gear fluids. Some of the findings of this study were presented during a technical session at the 60th German Tribology Conference held in Göttingen, Germany, in 2019 (Yilmaz et al. [17]).

2. Experimental Setups

This section describes the experimental setup of the bearing power loss test rig considered in the study as well as the operating conditions and lubricants.

2.1. Bearing Power Loss Test Rig

Experiments at an FZG bearing power loss test rig were conducted to investigate the loss behavior of roller bearings with water-containing gear fluids. Figure 1a shows the mechanical layout of the test rig, consisting of a speed-controlled electric engine, a loss torque meter, a radial and axial load unit, and a test housing. Figure 1b shows the assembly of the test bearings onto the shaft in the test housing.

The following description of the bearing power loss test rig is mainly based on the work and formulations of Jurkschat et al. [3,9]. The two outer roller bearings are supported in the housing and the two inner bearings are mounted in a yoke, which can be loaded by the radial load unit. The axial and radial force are applied by screw jacks. The bearings are loaded equally due to the symmetrical assembly of the roller bearings on the shafts. In order to realize an automatic force application and ensure accurate settings, the spindle is driven by a spur-geared motor. A spring assembly allows a wide range of forces without replacing the test rig components. The force is controlled by coupling the spindle drive with the load cell. The total loss torque of the four roller bearings is measured by a loss torque meter shaft. Contactless gap seals with negligible losses are used. Due to the symmetrical structure of the construction, the measured losses are approximately equal for all bearings and can be quartered.

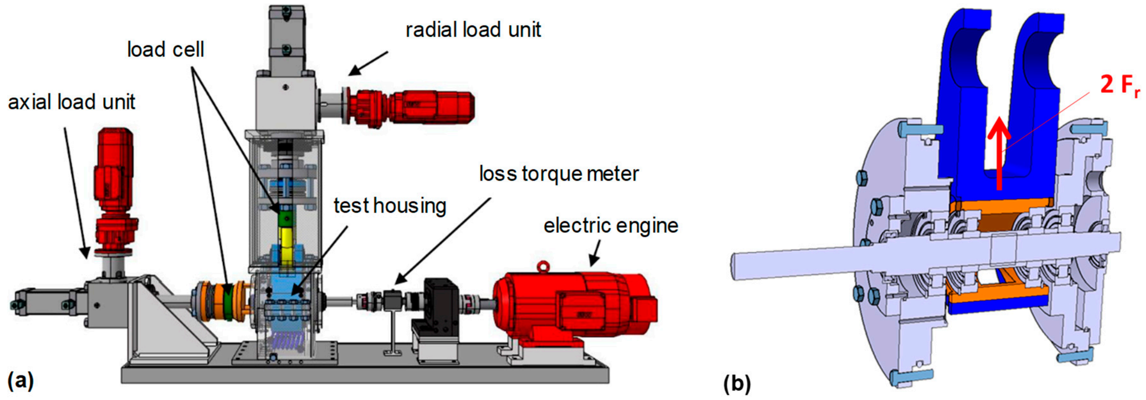

Figure 1. Mechanical layout of the FZG bearing power loss test rig (**a**) and assembly of the test bearings in the test housing (**b**) according to Jurkschat et al. [3,9].

Figure 2 illustrates the considered test bearings. To avoid incompatibilities with the investigated water-containing lubricants, type NU406 and NJ406 cylindrical roller bearings (CEROBEAR GmbH, Herzogenrath, Germany) (d = 30 mm, D = 90 mm, B = 23 mm) made of Si_3N_4 ceramic cylindrical rollers, cronidur© races, and a polyether ether ketone (PEEK) cage were investigated. Two type NJ406 test bearings were mounted as outer bearings, and two type NU406 test bearings were mounted as inner bearings, as shown in Figure 1b. The test bearings were the same as those used in the test gearbox utilized in Yilmaz et al. [16].

Figure 2. Type NU406 and NJ406 test bearings with common front view (**a**) and top view (**b**).

2.2. Operating Conditions and Lubricants

Table 1 shows the considered operating conditions. The experimental procedure included a running-in for t = 30 min at a radial force of F_r = 4.5 kN (Hertzian pressure of p_H = 1564 N/mm^2) and a rotational speed of n = 87 min^{-1}. Corresponding Hertzian pressures between the inner race and the roller were calculated by the bearing producer. Following the running-in, the rotational speed n was increased from 87 to 5218 min^{-1} for no-load and radial loads F_r of 1.4, 2.7, and 4.5 kN. Each rotational speed n was held for t = 5 min in order to allow quasi-stationary conditions. Lubrication regimes from boundary to mixed and fluid film lubrication were covered. The oil sump in the test housing was cooled and heated to control the oil sump temperature of ϑ_{Oil} = 75 °C. All experiments were conducted with an oil filling level 30 mm below the shaft axes.

Table 1. Considered operating conditions at the FZG bearing power loss test rig.

Radial Force F_r in kN	Hertzian Pressure p_H in N/mm^2	Rotational Speed n in min^{-1}	Time t in min	Oil Temperature ϑ_{Oil} in °C
no-load	no-load	87, 131, 348, 522, 1444, 2166, 3479, 5218	5 (for each rotational speed)	75
1.4	1078			
2.7	1320			
4.5	1564			

The considered lubricants were the same as those used in the studies [14,16] of the authors. Therefore, the following description is mainly based on these studies. Three water-containing gear fluids were compared with conventional gear oils. Table 2 shows the corresponding kinematic viscosities and densities. The mineral oil MIN-10 was specified by Laukotka [18] and used as a link to the results of Hinterstoißer [1,2]. The polyalphaolefine oils PAO-09 and PAO-05 have the same base oil type with typical gear oil additives incorporated. Their kinematic viscosity levels are specified at 100 °C and differ strongly. The water-containing fluids PAGW-09, PAGW-05A, and PAGW-05B were based on water-soluble polyalkylenglycols with water concentrations up to 70%. Note that the given kinematic viscosities at 100 °C were approximated. PAGW-09 and PAGW-05A have the same additives incorporated but different kinematic viscosities, similar to PAO-09 and PAO-05. PAGW-05B has the same kinematic viscosity as PAGW-05A but has different additives incorporated. The densities of the water-containing gear fluids were approximately 30% higher than those of conventional gear oils.

Table 2. Properties of the investigated lubricants.

	Conventional Gear Oils			Water-Containing Gear Fluids		
	MIN-10	**PAO-09**	**PAO-05**	**PAGW-09**	**PAGW-05A**	**PAGW-05B**
ν (40 °C) in mm^2/s	94.1	50.2	20.4	45.7	23.6	22.9
ν (60 °C) in mm^2/s	38.4	25.7	12.3	24.7	13.6	12.9
ν (100 °C) in mm^2/s	10.6	9.0	5.0	9.2	5.3	4.8
Viscosity index VI	95	165	185	189	167	135
ρ (15 °C) in kg/m^3	884.5	850.0	840.0	1115.0	1109.0	1097.0

2.3. *Load-Dependent Bearing Losses for Derivation of Mean Gear Coefficients of Friction*

In Yilmaz et al.'s study [16], the loss and thermal behavior of water-containing gear fluids were investigated at the FZG gear efficiency test rig for pitch line velocities v_t of 0.5 to 20.0 m/s, Hertzian pressures at the pitch point p_C from 962 to 1723 N/mm^2, and oil sump temperatures ϑ_{Oil} from 40 to 90 °C. The mean gear coefficients of friction were derived for an oil temperature of ϑ_{Oil} = 60 °C based on the measured load-dependent bearing losses presented in this study. Based on [16], Figure 3 exemplarily shows the measured no-load losses T_{L0}, load-dependent bearing losses T_{LBP}, and the derived load-dependent gear losses T_{LGP} for p_C = 1343 N/mm^2 and ϑ_{Oil} = 60 °C over the pitch line velocity v_t for the conventional gear oils and water-containing gear fluids. Note that the load-dependent bearing losses T_{LBP} comply with the results in Section 3.1. Whereas the load-dependent gear losses T_{LGP} make up the majority of the total losses for the conventional gear oils PAO-09 and PAO-05, T_{LGP} is drastically smaller for the water-containing gear fluids PAGW-09, PAGW-05A, and PAGW-05B. On the other hand, the no-load losses T_{L0} of the water-containing gear fluids are slightly higher compared to the conventional gear oils. The load-dependent bearing losses T_{LBP} of the water-containing gear fluids also show higher values compared to the conventional gear oils. These losses were the basis for deriving the load-dependent gear losses T_{LGP} and the mean coefficients of friction and are the focus of the present study.

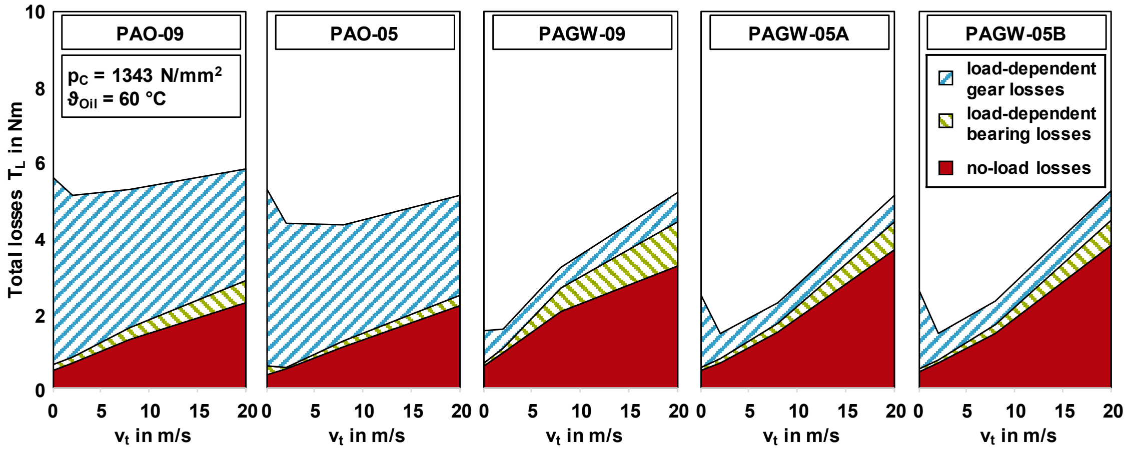

Figure 3. Total losses, no-load losses, load-dependent bearing losses, and load-dependent gear losses for p_C = 1343 N/mm^2 and ϑ_{Oil} = 60 °C over the pitch line velocity v_t based on [16].

Radial forces, rotational speeds, and test times in Section 2.2 were derived from the operating conditions of the gear efficiency tests in [16]. The radial forces in Table 1 were calculated with a RIKOR [19] model of the test gearbox, considering shaft, bearing, and gear stiffness. In order to achieve an oil supply and heat balance of the test bearings mounted at the FZG gear efficiency and bearing power loss test rig that are as comparable as possible, the oil filling level and oil temperature at the bearing power loss test rig were adjusted. Based on the set oil filling level of 30 mm below the shaft axes, half of the bottom cylindrical roller immerges into the oil sump. According to Hinterstoißer [2], this oil filling level approximates the dynamic oil filling level of the bearings during the experiments at the FZG gear efficiency test rig. The oil temperature of ϑ_{Oil} = 75 °C results according to Schleich [13] in an averaged measured bearing bulk temperature of ϑ_M = 60 °C at the bearing power loss test rig. This bearing bulk temperature was measured at the FZG gear efficiency test rig for an adjusted oil sump temperature of ϑ_{Oil} = 60 °C.

3. Results and Discussion

In this section, measured loss torques are presented, discussed, and compared with calculated loss torques. Note that the presented losses consist of the sum of the losses of four bearings. The same bearings were used for each considered lubricant, and all experiments were repeated once. The average values of the two test runs are shown. Error bars in bar charts indicate the result of test runs one and two.

3.1. Bearing Loss Torques

For all lubricants investigated, Figure 4 shows the measured no-load losses T_{LB0} and total losses T_{LB} for radial forces of F_r = {1.4, 2.7, 4.5} kN as well as the derived load-dependent losses T_{LBP} for radial forces of F_r = {2.7, 4.5} kN over the rotational speed.

3.1.1. No-load Losses T_{LB0}

The no-load loss curves (Figure 4a) show an increasing trend for all lubricants, mainly due to increasing drag and churning losses in the dip-lubricated roller bearings. The no-load losses are strongly influenced by the kinematic viscosity and density of the lubricants.

In case of the conventional gear oils MIN-10, PAO-09, and PAO-05, the results are in accordance with the kinematic viscosities shown in Table 2. MIN-10 results in higher no-load losses than PAO-09, which has higher no-load losses than PAO-05. Regarding the water-containing gear fluids PAGW-09, PAGW-05A, and PAGW-05B, higher no-load losses are measured with PAGW-09 compared to PAGW-05A and PAGW-05B, which show almost the same no-load losses due to similar viscosity.

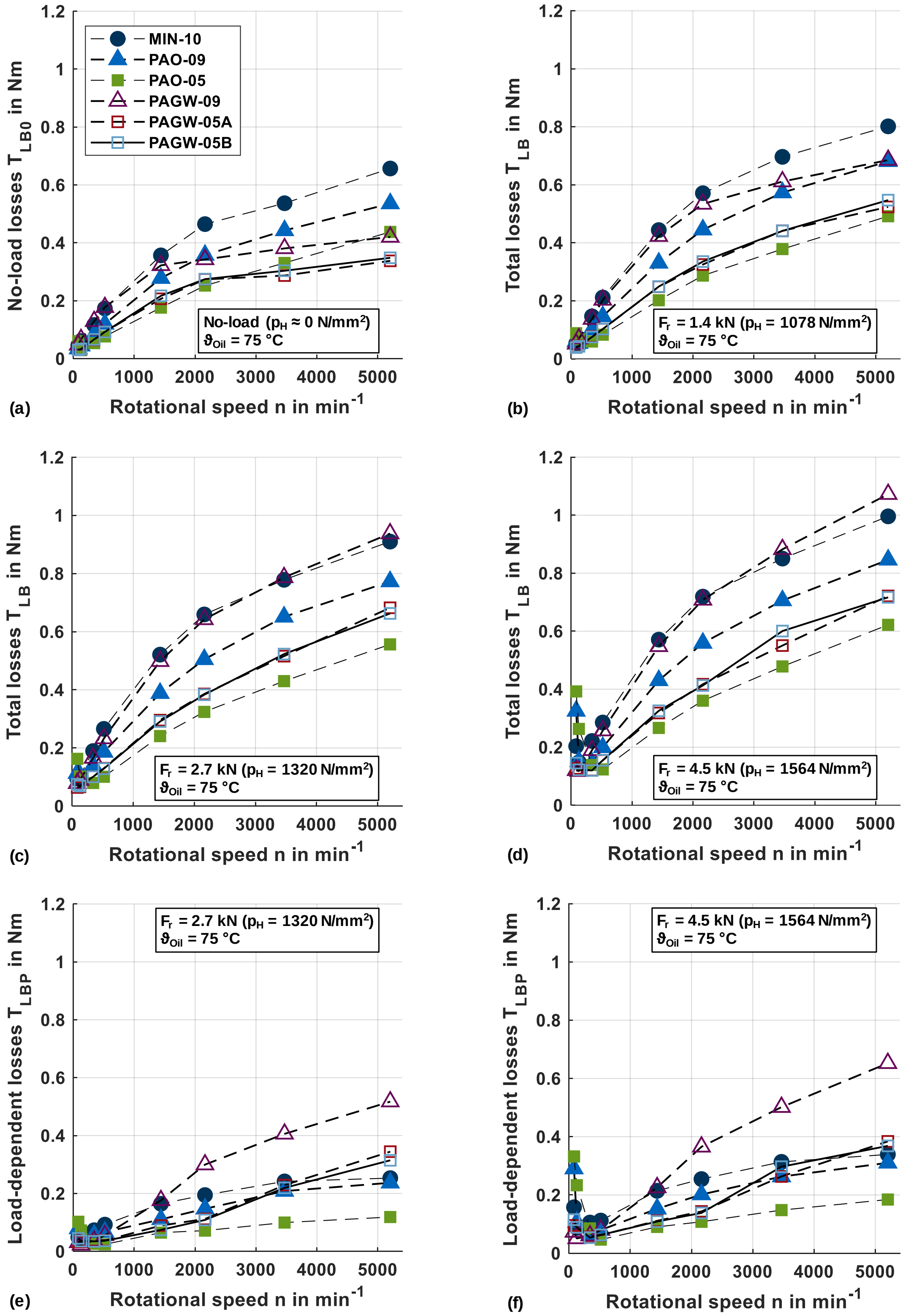

Figure 4. Measured no-load losses (**a**) and total losses for radial forces of F_r = {1.4, 2.7, 4.5} kN (**b**–**d**) as well as derived load-dependent losses for F_r = {2.7, 4.5} kN (**e**,**f**) over the rotational speed for all investigated lubricants (ϑ_{Oil} = 75 °C).

All no-load loss curves show a decrease in the slope between 1444 and 2166 min^{-1}. For rotational speeds of $n \leq 1444 \ldots 2166$ min^{-1}, higher no-load losses are measured with water-containing gear fluids compared to the polyalphaolefine oils with the same kinematic viscosity. This is attributed to the approximately 30% higher density of water-containing gear fluids, resulting in higher dynamic viscosities. For rotational speeds of $n > 1444 \ldots\ 2166\ \mathrm{min}^{-1}$, lower no-load losses are observed with the water-containing gear fluids, as the curvature of their no-load loss curves between 1444 and 2166 min^{-1} is more pronounced. This may be related to a lower sliding friction of these lubricants, resulting in lower acceleration forces on the rolling elements entering the load area (Hinterstoißer [2]). Furthermore, the higher density of the water-containing gear fluids may result in a greater displacement of oil from the bearing due to higher centrifugal forces.

3.1.2. Total Losses T_{LB}

The total loss curves (Figure 4b–d) for radial forces of $F_r = \{1.4, 2.7, 4.5\}$ kN consist of no-load and load-dependent losses. The measured total losses increase with increasing radial force for all investigated lubricants.

In case of the conventional gear oils MIN-10, PAO-09, and PAO-05, higher total losses are measured for MIN-10 compared to PAO-09, which shows higher total losses than PAO-05. Note that, for very low rotational speeds of $n \leq 131\ \mathrm{min}^{-1}$, the relations change to the highest total losses for PAO-05 and the lowest for MIN-10. This becomes more pronounced with increasing radial force.

In case of the water-containing gear fluids PAGW-09, PAGW-05A, and PAGW-05B, higher total losses are measured with PAGW-09 compared to PAGW-05A and PAGW-05B, which show almost the same total losses. In comparison with the polyalphaolefine oils with the same kinematic viscosity, the water-containing gear fluids result in lower total losses for very low rotational speeds of $n \leq 131\ \mathrm{min}^{-1}$ and higher total losses for higher rotational speeds.

3.1.3. Load-Dependent Losses T_{LBP}

The load-dependent loss curves (Figure 4e,f) represent the subtraction between the measured total losses T_{LB} and the measured no-load losses T_{LB0}. For all investigated lubricants, a Stribeck curve behavior of the load-dependent loss curves can be observed. They decrease between $87 \leq n \leq 348$ 1/min and then increase for $n > 348\ \mathrm{min}^{-1}$.

Regarding the conventional gear oils MIN-10, PAO-09, and PAO-05, the load-dependent loss curves are in good agreement with their viscosities shown in Table 2. For very low rotational speeds of $n \leq 131\ \mathrm{min}^{-1}$, the highest load-dependent losses are derived for PAO-05, followed by PAO-09 and MIN-10. For rotational speeds of $n > 131\ \mathrm{min}^{-1}$, the opposite is observed. These findings can be traced back to solid and fluid friction portions and to the increase of fluid friction with increasing viscosity. The results correlate well with the results of Hinterstoißer [2].

For all water-containing gear fluids PAGW-09, PAGW-05A, and PAGW-05B, similar load-dependent losses are measured for very low rotational speeds. For rotational speeds of $n \geq 348\ \mathrm{min}^{-1}$, the highest load-dependent losses are measured with PAGW-09. The load-dependent losses of the lower viscos PAGW-05A and PAGW-05B are smaller and very similar. In comparison with the polyalphaolefine oils with the same kinematic viscosity, the load-dependent losses are smaller at very low rotational speeds and clearly higher for higher rotational speeds.

Figures 5 and 6 show the no-load losses T_{LB0} and the load-dependent losses T_{LBP} in bar charts in order to provide a more complete overview of the influence of operating conditions and lubricants.

Figure 5 shows the measured no-load and derived load-dependent losses for rotational speeds $n = \{87, 2166, 5218\}\ \mathrm{min}^{-1}$ at $F_r = 4.5$ kN and $\vartheta_{Oil} = 75$ °C. It shows that for conventional gear oils, the load-dependent losses are dominant for very low rotational speeds and the no-load losses are dominant for higher rotational speeds. In contrast, the no-load losses of water-containing gear fluids are lower for higher rotational speeds, whereas their load-dependent losses are lower at very low rotational speeds and higher at higher rotational speeds.

Figure 6 shows the measured no-load and derived load-dependent losses for radial forces of $F_r = \{1.4, 2.7, 4.5\}$ kN at $n = 2166\ \mathrm{min}^{-1}$ and $\vartheta_{Oil} = 75$ °C. For the considered rotational speed of $n = 2166$

min^{-1}, the no-load losses are dominant. For all lubricants, the load-dependent losses increase with increasing radial force. The load-dependent losses of water-containing gear fluids are higher compared to polyalphaolefine oils with the same kinematic viscosity.

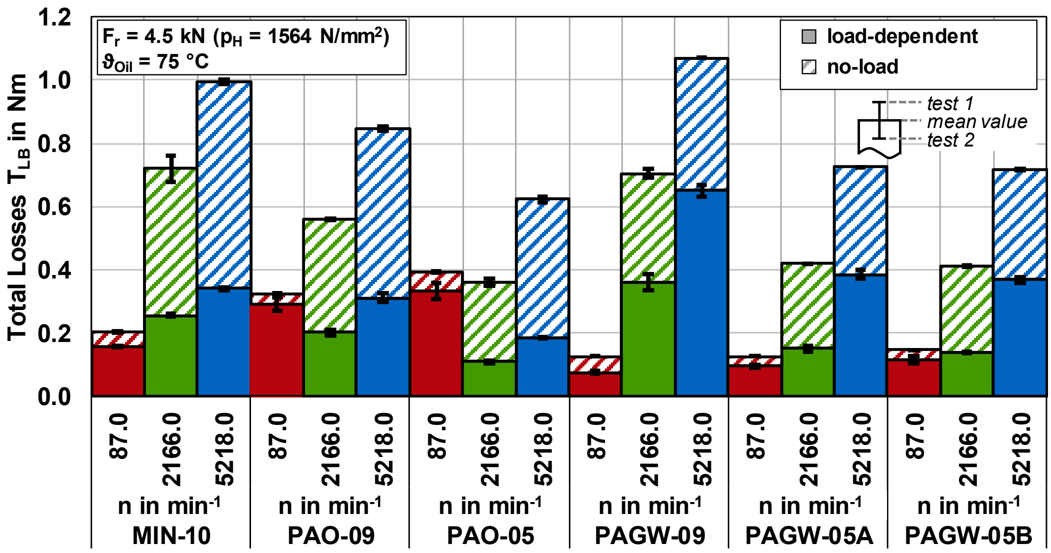

Figure 5. Influence of rotational speed n = {87, 2166, 5218} min^{-1} on the measured no-load and derived load-dependent losses at F_r = 4.5 kN and ϑ_{Oil} = 75 °C for all investigated lubricants.

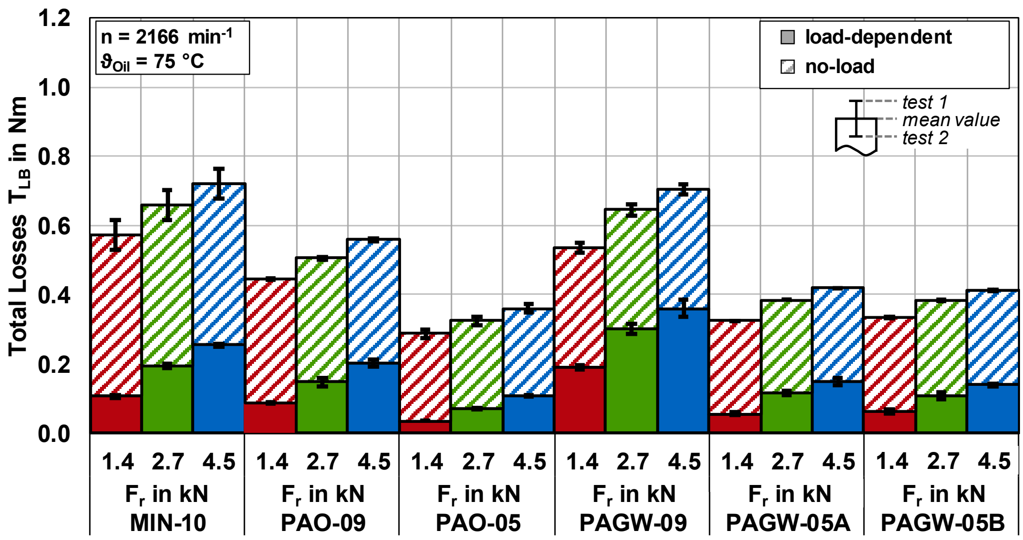

Figure 6. Influence of radial force F_r = {1.4, 2.7, 4.5} kN on the measured no-load and derived load-dependent losses at n = 2166 min^{-1} and ϑ_{Oil} = 75 °C for all investigated lubricants.

3.2. Comparison between Measured and Calculated Bearing Loss Torques

Figure 7 compares the measured bearing losses for F_r = 4.5 kN, n = {87, 131, 348, 522, 1444, 2166, 3479, 5218} min^{-1} and ϑ_{Oil} = 75 °C with calculated bearing losses for the conventional gear oils (Figure 7a) and water-containing gear fluids (Figure 7b). The calculations are based on BEARINX [20] and SKF 2004 [10]. The calculations show a tendency towards higher bearing losses with SKF 2004 and lower bearing losses with BEARINX compared to measurements. The bearing losses calculated with SKF 2004 are closer to the measured bearing losses.

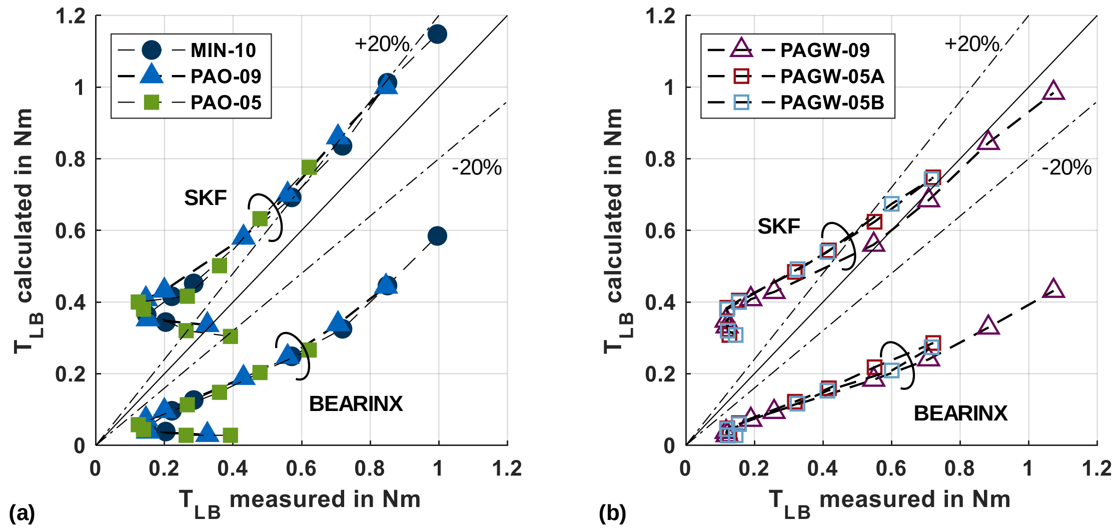

Figure 7. Comparison between calculated and measured total bearing losses T_{LB} at F_r = 4.5 kN, ϑ_{Oil} = 75 °C and n = {87, 131, 348, 522, 1444, 2166, 3479, 5218} min^{-1} for conventional gear oils (**a**) and water-containing gear fluids (**b**).

3.3. Derivation of Mean Gear Coefficients of Friction

In Yilmaz et al.'s study [16], the mean gear coefficients of friction were derived based on the measured load-dependent bearing losses T_{LBP} shown in Sections 2.2 and 2.3. To underline the importance of having detailed knowledge of load-dependent bearing losses T_{LBP} when evaluating load-dependent gear losses T_{LGP}, Figure 8 shows a comparison of derived mean gear coefficients of friction μ_{mz} based on measured and calculated load-dependent bearing losses T_{LBP}. The evaluation refers to pitch line velocities of v_t = {0.5, 2.0, 8.3, 20} m/s, a Hertzian pressure at the pitch point of p_C = 1723 N/mm^2, and an oil sump temperature of ϑ_{Oil} = 60 °C [16]. For the conventional gear oils (Figure 8a), the results show a good correlation of μ_{mz} if the load-dependent bearing losses T_{LBP} are calculated according to BEARINX [20]. The usage of SKF 2004 [10] underestimates μ_{mz}. In case of water-containing gear fluids (Figure 8b), the differences are partially larger compared to conventional gear oils with both calculation procedures.

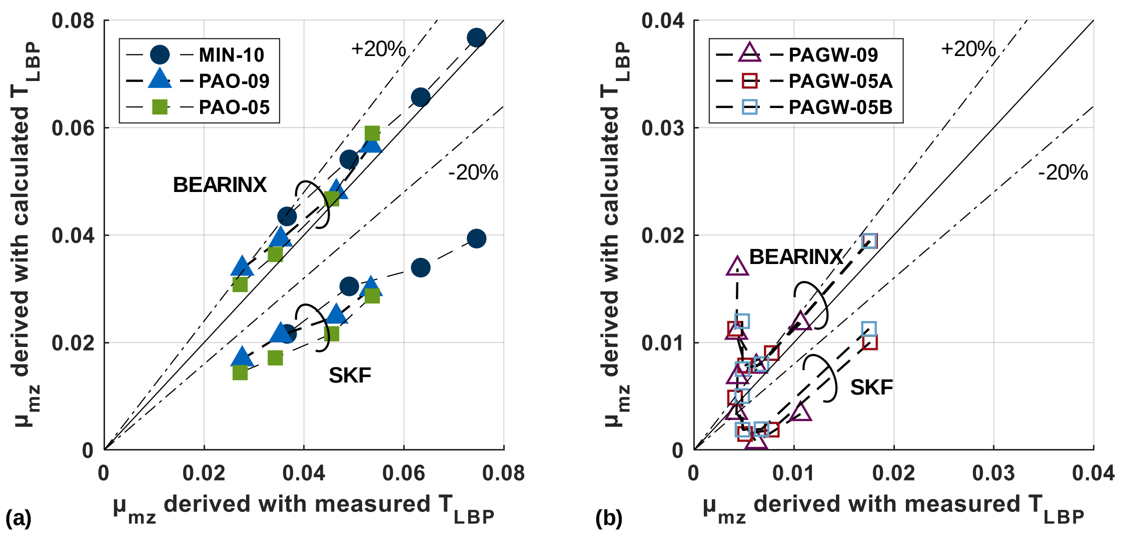

Figure 8. Comparison of derived mean gear coefficients μ_{mz} with measured and calculated load-dependent bearing losses T_{LBP} at p_C = 1723 N/mm^2, ϑ_{Oil} = 60 °C and v_t = {0.5, 2.0, 8.3, 20} m/s for conventional gear oils (**a**) and water-containing gear fluids (**b**).

4. Conclusion

In this study, the loss behavior of roller bearings with water-containing gear fluids was investigated and compared with conventional gear oils. For water-containing gear fluids, lower no-load losses and higher load-dependent losses were observed at higher rotational speeds. For all investigated lubricants, a Stribeck curve behavior of the load-dependent losses was determined. The comparison of the measured bearing losses with typical calculation procedures showed comparatively large differences. The comparison of derived mean gear coefficients of friction based on measured and calculated load-dependent bearing losses underline the importance of having detailed knowledge of bearing losses in order to derive mean gear coefficients of friction precisely.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

B	Bearing width	mm
d	Bearing inner diameter	mm
D	Bearing outer diameter	mm
F_r	Radial force	kN
n	Rotational speed	min^{-1}
p_C	Hertzian pressure at the pitch point	N/mm^2
p_H	Hertzian pressure	N/mm^2
t	Time	s
T	Torque	Nm
v_t	Pitch line velocity	m/s
Greek symbols		
ϑ_M	Bulk temperature	°C
ϑ_{oil}	Oil temperature	°C
ν	Oil kinematic viscosity	mm^2/s
μ_{mz}	Mean gear coefficient of friction	-
ρ	Oil density	kg/m^3
Indices		
0	No-load	
B	Bearing	
G	Gear	
L	Loss	
P	Load-dependent	

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