Vol. 45, No. 1 (2023) 178-190, DOI: 10.24874/ti.1394.10.22.02

Tribology in Industry

www.tribology.rs

RESEARCH

Lubrication of Radial Loaded Tapered Roller Bearings Under High Rotational Speeds

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Keywords:

Tapered roller bearings
Grease lubrication
Oil-air lubrication
Spindle bearings
Speed capability
Radial loads
Temperature behavior
Tool spindle

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Received: 30 October 2022
Revised: 1 December 2022
Accepted: 9 February 2023

A B S T R A C T

To improve the manufacturing accuracy in the machining process, the stiffness of the machine tool spindle must be increased. One approach is to replace the conventionally used spindle bearings by tapered roller bearings to achieve a higher stiffness and load carrying capacity. However, the speed capability of this bearing type is limited, though can be enhanced by sufficient lubrication, especially in the critically loaded rib-roller-contact. The objective of this paper is to validate the lubricant influence on the speed capability of radially loaded tapered roller bearings. In various speed step runs, the operating behavior of the bearings under oil-air and grease lubrication was evaluated based on the outer ring temperature and the vibration level in the test rig. With oil-air lubrication, a speed parameter up to 900,000 mm/min could be achieved, whereas with grease lubrication, a speed parameter of 540,000 mm/min could be reached. In comparison, the setup with spindle bearings can achieve the highest investigated speed parameter of 900,000 mm/min with both grease and oil-air lubrication. For grease lubricated ball bearings, significantly higher temperature fluctuations are present with variation of the radial load. In the case of oil-air lubrication, the steady-state temperature improved with increasing oil viscosity.

1. INTRODUCTION

Increasing the performance of machine tool spindles is an important research and development challenge. In the aerospace sector, for instance, high-strength steels and titanium alloys are milled from a solid piece of metal, such that the spindle is permanently exposed to high loads in the machining process [1]. Moreover, increasing speeds lead to reduced machining accuracy due to imbalances of the spindle and tools. Hence, high requirements for the machining precision are demanded. A value commonly used in tool spindles for the speed

capability, which allows a comparison between bearings of different sizes, is the speed parameter nxD_m. This value results from the product of rotational speed n and bearing pitch diameter D_m. A new high precision tapered roller bearing with ceramic rollers is presented in [2], which can reach a speed parameter of $1.25 \cdot 10^6$ mm/min under axial load. Furthermore, a high-speed tapered roller bearing with an axial moveable and hydraulically preloaded rib was developed to achieve a comparable high speed limit [3,4]. Another precision hybrid tapered roller bearing with a direct lubricated rib-roller contact on the outer ring was successfully tested for a speed parameter up to $1 \cdot 10^6$ mm/min in [5]. A further development of the bearing by mathematical optimization of the rib geometries, however, did not result in an improvement of the speed limits [6]. Furthermore, it has been shown that tapered roller bearings can handle an axial load of 53.4 kN and a maximum radial load of 26.7 kN for speed parameters of $2.45 \cdot 10^6$ mm/min under stable operational conditions [7]. However, the bearings with a pitch diameter of 163.5 mm were lubricated with a, for machine tools, comparatively high oil amount of $15.1 \cdot 10^6$ mm3/h. Nevertheless, with a sufficiently high lubrication quantity the speed capability of the bearing type could be increased. Since there is no comparable work, this paper represents a first approach lubricating radially loaded tapered roller bearings with oil-air lubrication and grease lubrication under rotational speeds that are common for machine tool spindles.

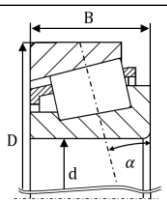
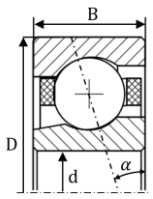
2. EXPERIMENTAL METHODS

In the following section, the bearings, test rig and conditions for the test series are described.

2.1 Bearings

The characteristics of the investigated bearings are given in Table 1. Both bearings are steel bearings, consisting of 100Cr6 rolling bearing steel. The tapered roller bearing has a sheet metal cage, whereas the spindle bearing is designed with a resin cage. The rollers of the tapered roller bearing have a logarithmic profile. As a reference to the tapered roller bearing, a spindle bearing of the same size was tested, which is widely used in tool spindles.

Table 1. Characteristics of the test bearings [8, 9].

	Tapered roller bearing (TRB)	Spindle bearing (SB)
Cross section:		
Type:	32014-X	B7014
Outer diameter D [mm]:	110	110
Inner diameter d [mm]:	70	70
Width B [mm]:	25	20
Contact angle α [°]:	14.7	25
Speed limit n_{oil} (oil) [rpm]:	5,600	16,000
Speed limit n_{grease} (grease) [rpm]:	4,500	12,000
Basic static load rating C_0 [kN]:	153	43.5
Basic dynamic load rating C [kN]:	125	46
Precision class:	P5	P4

2.2 Test rig

The test rig used for the presented study is depicted in Fig. 1.

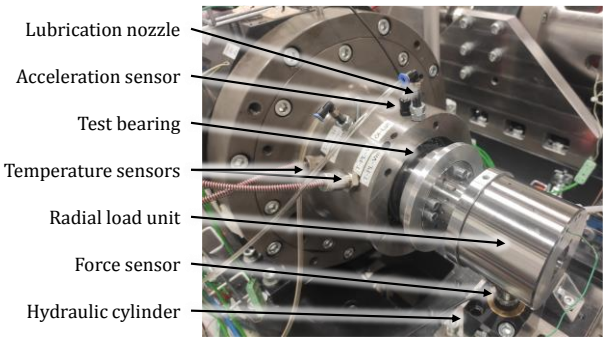


Fig. 1. Test rig (front view).

Due to its modular design, this test rig concept can be easily adapted in order to mount different bearing types and sizes by making minor adjustments to the housing and the shaft [10]. For oil-air lubrication, the lubricant is injected in the front and back side of the bearing. During a test, the vibration level is measured with a piezoelectric acceleration sensor with a sampling rate of 51.2 kHz and evaluated in the form of effective values of the acceleration a_{RMS} and the vibration velocity v_{RMS}. A resistance thermometer detects the bearing temperature of the outer ring. If the limit value of the bearing outer ring temperature of 80 C is exceeded, the

measuring software automatically switches off the test rig. The radial load on the bearing is applied through a mounted radial load unit, which is connected to the shaft. The radial load is variable, and applied by a hydraulic cylinder. For control purposes the radial load is measured with a strain gauge based force sensor.

The cross-section in Fig. 2 of the test bench illustrates the force flux. Three hydraulic cylinders apply the preload force though the sliding seat. The axially applied force is controlled by the feedback of the force sensors. Therefore, the bearing arrangement is a hydraulically controlled fixed-floating O-arrangement, which keeps the preload in the bearing at a constant level and compensates thermal expansions [11]. A force of maximum 5 and 6 kN can be applied in axial and radial direction respectively. Besides this, the configurations with TRBs and the SBs are shown with the injection directions, positions and diameters. Furthermore, a 15 kW synchronous motor drives that shaft via a bellows coupling up to a rotational speed of 30,000 rpm. Besides the hydraulically applied force, the speed is also controlled during the test.

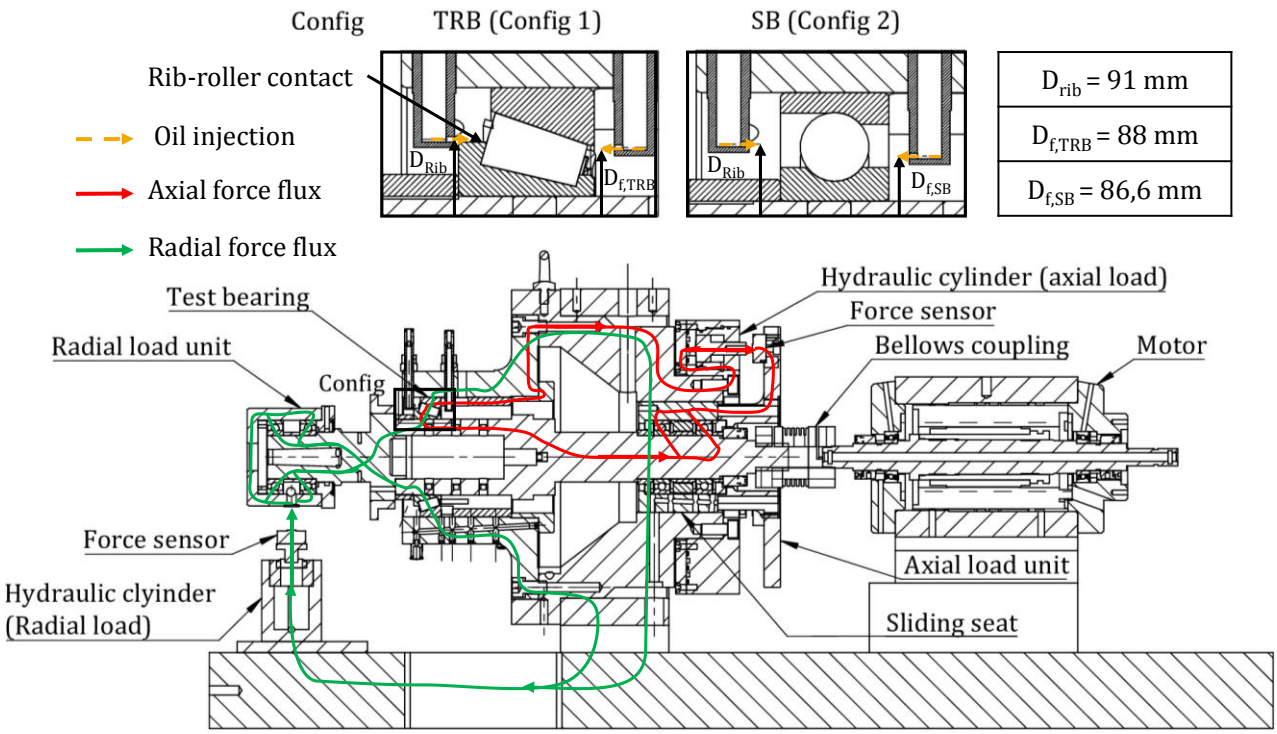


Fig. 2. Cross section of test rig, force flux, the bearing configurations and the lubrication injection diameters.

2.3 Test conditions and lubricants

The test program with rotational speed step run and the axial force are shown in Fig. 3. Each speed level is maintained for 4 hours such that a steady-state bearing temperature is achieved. Rotational speed steps up to a maximum of 10,000 rpm are approached in steps of 1,000 rpm.

To establish test conditions that are similar to those in machine tool spindles the axially pre-loaded test bearing is subjected to a variable radial forces, as shown in Fig. 4. The spindle is loaded by a radial force of 3.5 kN for three minutes, and is unloaded in 40 cycles per speed step.

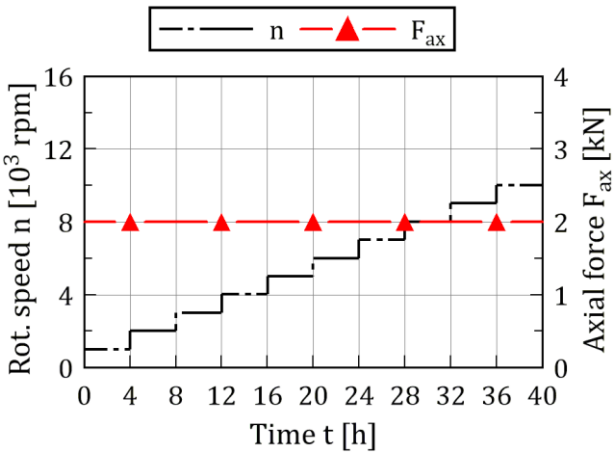


Fig. 3. Test program (rot. speed and axial force).

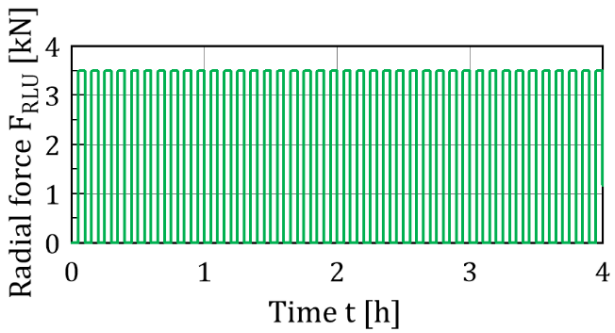
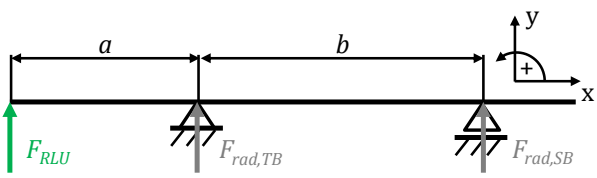


Fig. 4. Radial force applied at the radial load unit for one rotational speed step.

Due to the radial force application at the radial load unit, the test bearing is subjected to a bending moment and a higher radial bearing force compared to the applied force at the load unit. The simplified shaft with the test bearing and the support bearings, which are combined as one bearing, is depicted in Fig. 5.



F_{RLU}:	Radial force at the radial load unit (TCP)
$F_{rad,TB}$:	Radial force at the test bearing
$F_{rad,SB}$:	Radial force at the support bearings
a:	Distance between radial load unit and test bearing (a=125.2 mm)
b:	Distance between test bearing and support bearing (b=196.3 mm)

Fig. 5. Static bearing force at the test bearing.

With an equilibrium of forces and moments for the static case (see equation (1) and (2)), the radial bearing forces can be determined.

$$\sum F_y = F_{RLU} + F_{rad,TB} + F_{rad,SB} = 0 \quad (1)$$

$$\sum M_{SB} = -F_{RLU} \cdot (a+b) - F_{rad,TB} \cdot b = 0 \quad (2)$$

Solving equation (2) to the radial force of the test bearing results in a force higher by a factor of 1.6 compared to the force introduced at the load unit, given in equation (3).

$$F_{rad,TB} = -\frac{a+b}{b} \cdot F_{RLU} = -1.6 \cdot F_{RLU} \quad (3)$$

With a maximum radial force of 3.5 kN on the load unit, a radial force of 5.6 kN acts on the test bearing.

According to the findings of [7], for the tests, the highest possible oil air feed rate of the lubrication system suitable for long term operation of 960 mm³/h is selected. In case of the TRB the lubricant is injected at rib-roller contact side and the front side with a flow rate of 480 mm³/h. The injection diameters at the front side for the TRB $D_{f,TRB}$, the SB $D_{f,SB}$ and at the rib-roller contact side D_{Rib} and their locations for the bearing are shown for both configurations (i.e., Config1 and Config 2) in Fig. 2.

In the case of grease lubrication, a grease quantity of 6.5 g per bearing is used, which corresponds to about 20% of the bearing free volume with an absolute variance of ±1 %depending on bearing type and grease density. This lubrication quantity is based on the recommendation of [12] for spindle bearings for a speed parameter of 900,000 mm/min.

For the test with grease lubrication, a grease distribution run is performed before the test run, as shown in Fig. 6. After a short run-in period at 500 rpm, the test bearing is ran up to 2,500 rpm five times for 20 s and then decelerated again. The same procedure is then repeated 25 times for the maximum recommended bearing speed of 5,000 rpm. After a short cooling phase, a two-hour constant run is performed such that the grease distribution can be finalized and a steady-state temperature is reached.

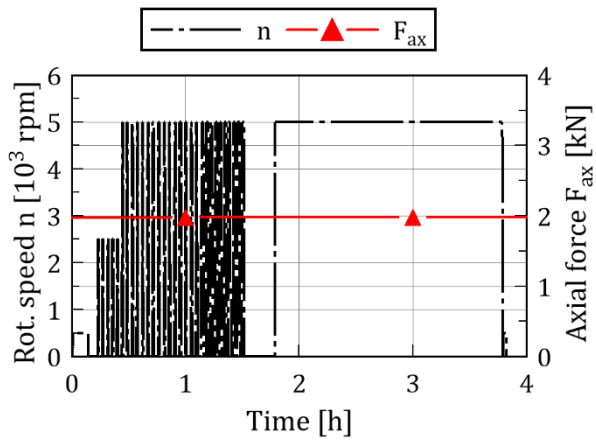


Fig. 6. Grease distribution run.

Both bearing types are subjected to the same test and lubrication conditions to ensure comparability. The tests with TRBs were repeated twice for each lubricant. The SBs were each tested three times with a selected oil and grease as a reference.

An overview of the lubricants is listed in Table 2. In total, three oils and three greases were tested in a step run. Oil 1 and 2 are fine filtered PAO-based oils for machine tool spindle applications, only differing in their viscosity [13]. Oil 3 has a relatively high viscosity of 100 mm^2/s, to investigate the influence of the viscosity. This oil is a sintered bearing oil, which provides a good corrosion protection [14]. Grease A is typically used in machine tool spindles and has a high speed parameter of $2 \cdot 10^6$ mm/min. It is based on a synthetic hydrocarbon and ester oil with a relatively low base oil viscosity of 22 mm^2/s. Furthermore, it is composed of a synthetic hydrocarbon and ester oil and polyurea thickener with good corrosion protection properties [14]. The effect of high pressure and anti-wear additives on wear in the rib-roller contact is validated with the tests with Grease B [14].

The influence of the base oil viscosity is in the focus of the investigations with Grease C. It consists of a synthetic hydrocarbon oil with a high base oil viscosity of 100 mm^2/s, a lithium soap and has good corrosion protection and anti-ageing properties. This grease is mainly used in the electro and traction motor rail sector and has the lowest speed characteristic value [15]. The different properties of the greases should provide a better understanding of the experimental results presented in the following section.

3. RESULTS AND DISCUSSION

The following section analyses the experimental results with TRBs and SBs. The focus is on oil-air lubrication, which tends to give a higher speed capability, though grease lubrication is more common for machine tool spindles.

3.1 Oil-air lubrication

In Fig. 7, the outer ring temperature of the tested TRB is shown for tests with the three oils listed in Table 2. During the tests, the load conditions mentioned in Section 2.2 have been applied. The first number in the legend indicates the tested oil, whereas the second number for the respective test run. For each lubricant three tests were conducted with a new bearing. In all tests with oils, the maximum speed of 10,000 rpm was reached without failure. In contrast, the limiting speed with oil-air lubrication of the bearing type 32014-X is 5,600 rpm. The reached rotational speeds are far above the manufacturers recommended speed limits. In general, all tests led to comparable temperature levels.

Table 2. Investigated lubricants [13–17].

	Oil 1	Oil 2	Oil 3	Grease A	Grease B	Grease C
Name	Klübersynth FB-4 68	Klübersynth FB-4 32	CONSTANT OY 100 K	Klüberspeed BF 72-22	Klübersynth BM 44-42	Isoflex Topas L152
Oil type	Finely filtered speciality lubricant on the basis of a PAO		Synthetic hydrocarbon oil	Synthetic hydrocarbon, ester oil	Synthetic hydrocarbon oil	Synthetic hydrocarbo n oil
Thickener	-		-	Polyurea	Special lithium soap	Lithium soap
Additives	Corrosion protection		Corrosion protection	Corrosion protection	Extreme pressure/ anti wear	Corrosion and ageing protection
Oil separation (DIN 51817 [18])	-		-	≤ 3	-	≤ 2
Density ρ (20 °C) [g/cm³]	0.86	0.85	0.84	0.92	0.88	0.91
Kin. Viscosity of base oil ν (40 °C) [mm²/s]	68	32	100	22	47	100
Speed parameter [10⁶ mm/min]	2.5		-	2	2	0.6
Main application	Tool spindle		Sintered metal plain bearings	Tool spindle	Automotive rolling bearings	Electromo-tor rolling bearings

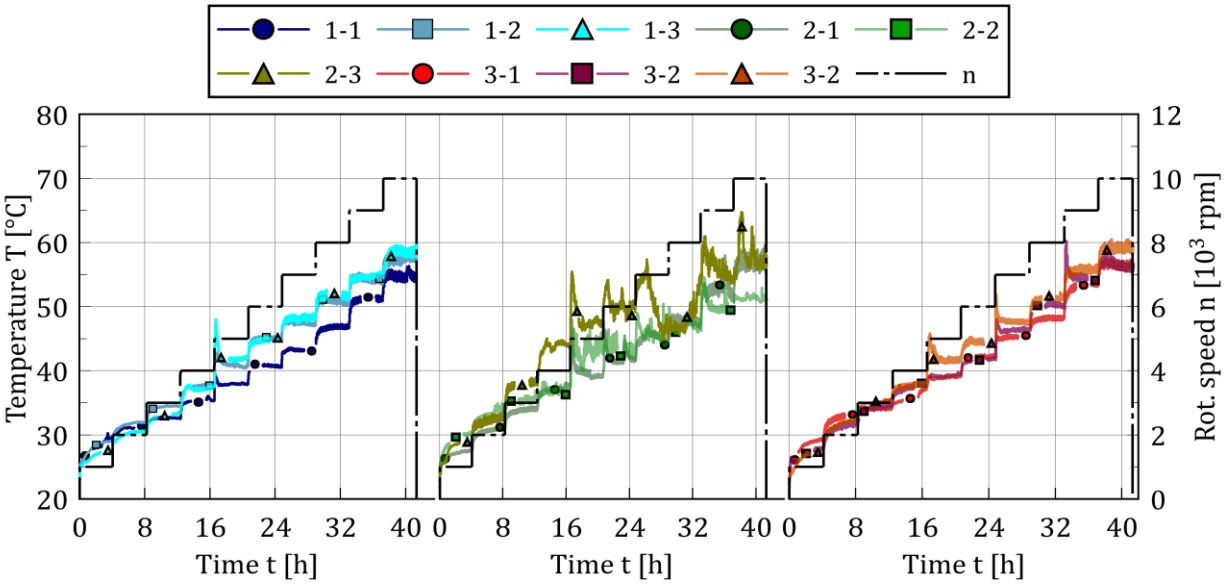


Fig. 7. Temperature behavior of TRBs with different oils.

Nevertheless, the tests with Oil 2, which has the lowest viscosity of 32 mm²/s, shows the greatest fluctuations. Oil 1 and 3 reached constant steady-state temperatures. In almost all tests with TRBs a significant increase in the temperature level could be detected at 5,000 rpm. This effect can be attributed to an excessive amount of lubrication in the bearing resulting in a temporary higher fluid friction, since the tests are performed without an oil suction and the oil is only ejected by the rotating bearing. If a steady-state temperature is reached, the deviations between the measurements per step are below 5 °C. The tests with Oil 3 achieve comparable temperature levels. Even though Oil 3 initially has peak temperatures at the speed levels of 7,000 rpm and 9,000 rpm, though after a while similar steady-state temperature levels occur compared to tests with Oil 1. In contrast, the tests with the low-viscosity Oil 2 exhibit the greatest fluctuations.

One possible reason for this can be the limited lubricating film formation of the low-viscosity oil, especially in the critically stressed rib-roller contact [19]. In [20] the hydrodynamic lubrication film formation in the rib-roller contact of TRBs were investigated. In comparison with different calculation methods for the lubrication film height the approach of [21] showed the best agreement with the experimental data. According to Archard [21] the film height h for the point contact is calculated with equation (4).

$$h = 2.04 \cdot \phi^{0.74} \cdot (\alpha \nu_0 \rho_0 u)^{0.704} \cdot R^{0.407} \cdot \left(\frac{E'}{w'}\right)^{0.074} \quad (4)$$

This semi analytical approach is dependent on the side-leakage factor ϕ, the pressure coefficient of viscosity α, the lubricant density ρ_0, the mean vector sum of the velocities in the rolling contact u, the effective curvature Radius R, the reduced young's modulus E' and the load in the point contact w'. In this respect, an increased kinematic viscosity ν_0 can favor the formation of a lubricant film in the rib-roller contact.

Oil 1 and 3 show the best performance. Hence, Oil 2 is ranked worse due to the greater fluctuations. The wear of the rollers at the rib-roller contact and the raceway-roller contact along with roughness measurements of the oil-air lubricated bearings after the tests are shown in Fig. 8. For one roller, the roughness in the raceway-roller and rib-roller contact was measured at five different positions, which is shown with the mean measurement results in Fig. 8 c). The mean value of the arithmetic average of the profile height R_a and the maximum peak to valley height of the profile R_z with its deviation were evaluated according to the ISO 21920-2 standard [22].

The wear in the rib-roller contact appears to be greater, which roughness values are significantly higher compared to those of the raceway-roller contact (see Fig. 8 c). The stresses in the rib-roller contact lead to a curved elliptical rolling contact at the rib side, which is subjected to high sliding friction components due to the roller kinematics. Consequently cycloidal grooves were formed

on the rib side of the roller (see Fig. 8 a), which result from the sliding and skewing movement of the roller. In general, the wear of the bearing of test Oil 1-1 appeared to be mild on the raceway-roller contact side meaning that its operational life expects the service life. Nevertheless, the roughness values in the rib-roller contact are the highest under oil-air lubrication compared to the other tests (see Fig. 8 c.). The wear of the TRB lubricated with Oil 2, which is shown in Fig. 8 a) and b), appeared different. There is a clear brown discoloration and wear stripes are apparent on the lower side of the rollers in the raceway contact [23]. The macroscopic wear appears to be most severe in the rolling of the bearing from test Oil 2-1 compared to the other bearings run under oil-air lubrication. This also corresponds to the significantly higher temperature fluctuations during the test and indicates a higher stress on the bearing. However, the roughness values of this bearing are slightly below the measured values of the bearing from test Oil 1-1 (see Fig. 8 c).

The discoloration is less severe in case of the bearing lubricated with Oil 3, shown in Fig. 8 b). Furthermore, the limited wear on the rib-roller side indicates a good wear protection of the lubricant. This is also confirmed by the roughness values measured in Fig. 8 c), which are the lowest compared to the other oils tests.

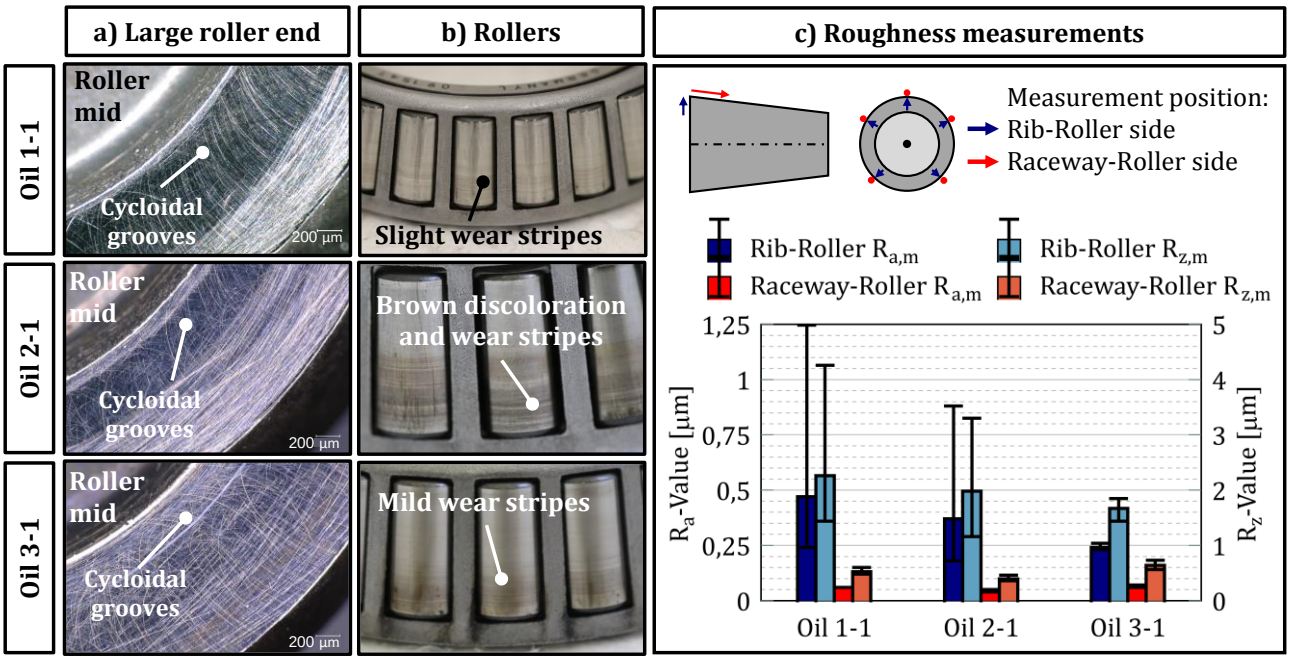


Fig. 8. Microscope views of the rollers (a) & (b) and roughness measurements c) after the tests with oil air lubrication.

Since the test with Oil 3 showed the best temperature and wear behavior, the step run was repeated three times with a SB. The results of the tests with Oil 3 are depicted in Fig. 9 a). In general, all speed steps are reached without failure, even with a single SB setup. The temperature levels are significantly lower, i.e. a steady-state temperature of 43°C at 10,000 rpm. This can be explained by better frictional performance of the rolling contacts (compared to the TRB), which in turn gives a reduced heat generation. Based on the outer ring temperature, the SB shows a comparatively better speed capability. Nevertheless, slight discontinuities in the temperature level between 6,000 and 9,000 rpm can be observed. The corresponding speed steps are illustrated in detail with different colors and the time-resolved root square mean value of the acceleration signal a_{RMS} in Fig. 9 b). Furthermore, the acceleration was evaluated with a sample rate of 51.2 kHz. Due to resonance effects in the test rig, increased accelerations occur, which prevent a transition to a steady state. For the speed steps from 6,000 to 8,000 rpm, there are considerable deviations in the outer ring temperatures, which slowly equalize from 9,000 rpm and reach a comparable temperature level again at 10,000 rpm with low vibration values.

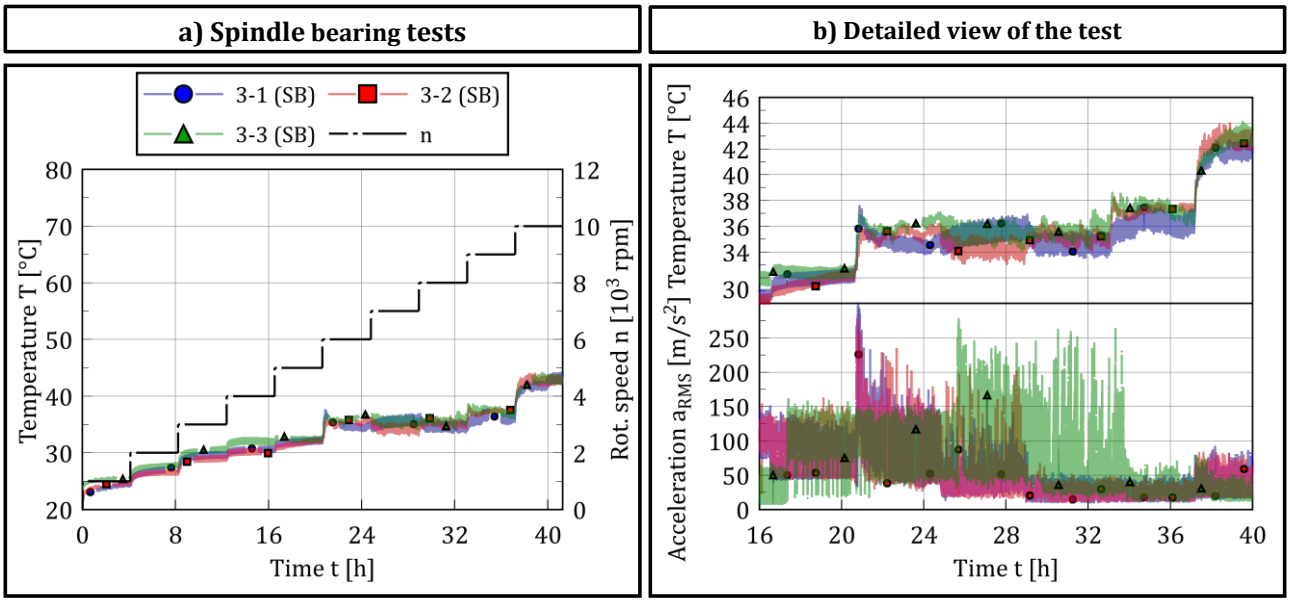


Fig. 9. Spindle bearing test with Oil 3 a) and detailed view of the temperature and acceleration during the test b).

These findings show that a validation of the speed capability only based on the outer ring temperature is not sufficient. Therefore, the acceleration values were also considered in this test series.

The mean effective vibration velocity for each speed step with oil-air lubrication of all test runs is presented as bar charts in Fig. 10. The first number again represents the respective oil, the second number the test run. In this case, the results with the TRBs along with the tests of the SBs are shown. To evaluate the vibration velocities, the standard ISO 10816-3 gives a recommendation regarding acceptable velocities, which lie at around 5 mm/s for the tests performed [24]. In comparison, Oil 2 shows the lowest vibration velocities below 8,000 rpm, but then exceeds the critical vibration value at 9,000 rpm and 10,000 rpm. Above 9,000 rpm the vibration velocity limits for the TRB lubricated with Oil 2 and the SB lubricated with Oil 3 are exceeded. Since SBs are known for their low vibration level compared to other bearings, the measured vibration values are higher than expected and exceed the critical level at 10,000 rpm.

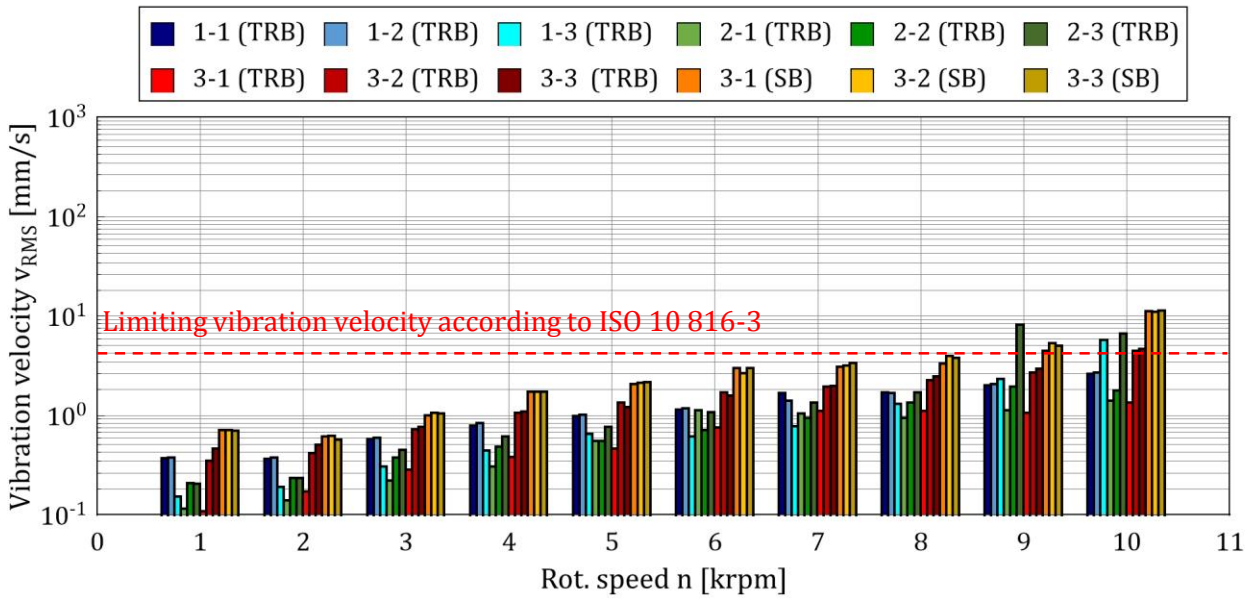


Fig. 10. Mean vibration velocities per speed step for the tests with oil-air lubrication.

The tests with Oil 1 and Oil 3 with the TRBs showed the best performance regarding the vibration velocities, since they do not exceed the critical limit value until 10,000 rpm. The measurement results with oil-air lubrication for radially loaded standard TRBs prove a good speed capability. Since grease lubrication is more widespread in machine tools due to its ease of implementation, the next section focuses on the speed capability of different greases.

3.2 Grease lubrication

The test results for the temperature behavior with grease lubrication are depicted in Fig. 11. With grease lubrication under varying radial load, the bearings failed at the latest at 10,000 rpm due to an exceeding of the outer ring temperature limit of 80 °C. Nevertheless, a suitable grease can increase the performance limits of the bearings. With Grease A, for example, a steady-state temperature could be achieved for rotational speeds up to 5,000 rpm, whereas the recommendation for the TRB under grease lubrication is only 4,500 rpm. The first and third run with Grease A reached 10,000 rpm, but lost a steady-state behavior starting at 6,000 rpm. As an example, the wear pattern of the rollers of the first test with Grease A is shown in Fig. 12 a) and b).This bearing also reached the 10,000 rpm step, but failed after a few hours. The bearing failure and damage analyses catalogue [25] indicates abrasive wear in the rib-roller contact as main cause of failure.

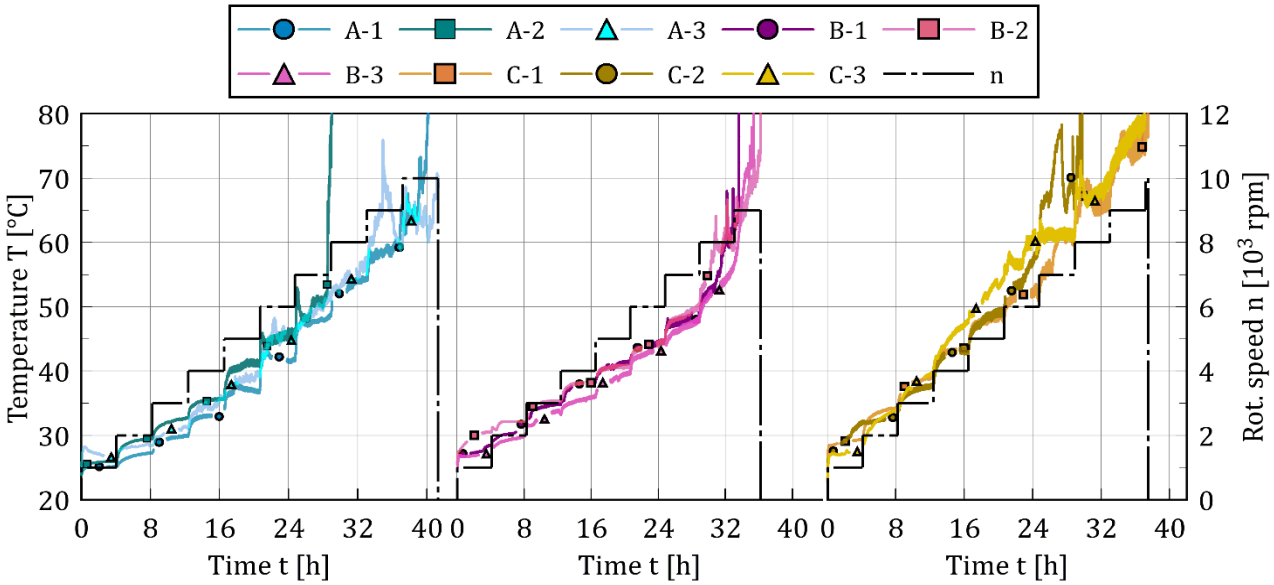


Fig. 11. Temperature behavior of tapered roller bearings with different greases.

In particular, the strong discoloration of the rollers and cage shows that temperatures inside the bearing were significantly higher than the measured outer ring temperature. Apart from the wear images, the roughness measurements on the raceway roller contact also illustrate its heavy wear, which is highest in comparison with the different greases.

Grease B with a high-pressure additive shows a more stable and better reproducible temperature behavior in the step run compared to Grease A and C (see Fig. 12). Furthermore, the wear patterns in the raceway roller contact in Fig. 12 a) and b) indicate the effectiveness of anti-wear additives contact. The comparatively low roughness values in rib-roller contact can be attributed, among other things, to the anti-wear additives of the grease (see Fig. 12 c).

Since the bearing has already failed at 9,000 rpm, the damage pattern is not as severe as for bearing lubricated with Grease A. The speed capability of Grease C is expected lower, due to unstable outer ring temperatures above 5,000 rpm. Overall, this grease can also reach speeds of up to 9,000 rpm, but without a steady-state level (see Fig. 11 C-1 & C-3). Nevertheless, in test run C-3, the steady state temperature is already lost at 4,000 rpm. Grease C has, in comparison to the other greases, the highest heat generation, which is a consequence of the higher base oil viscosity and higher value of oil separation (see Table 2).

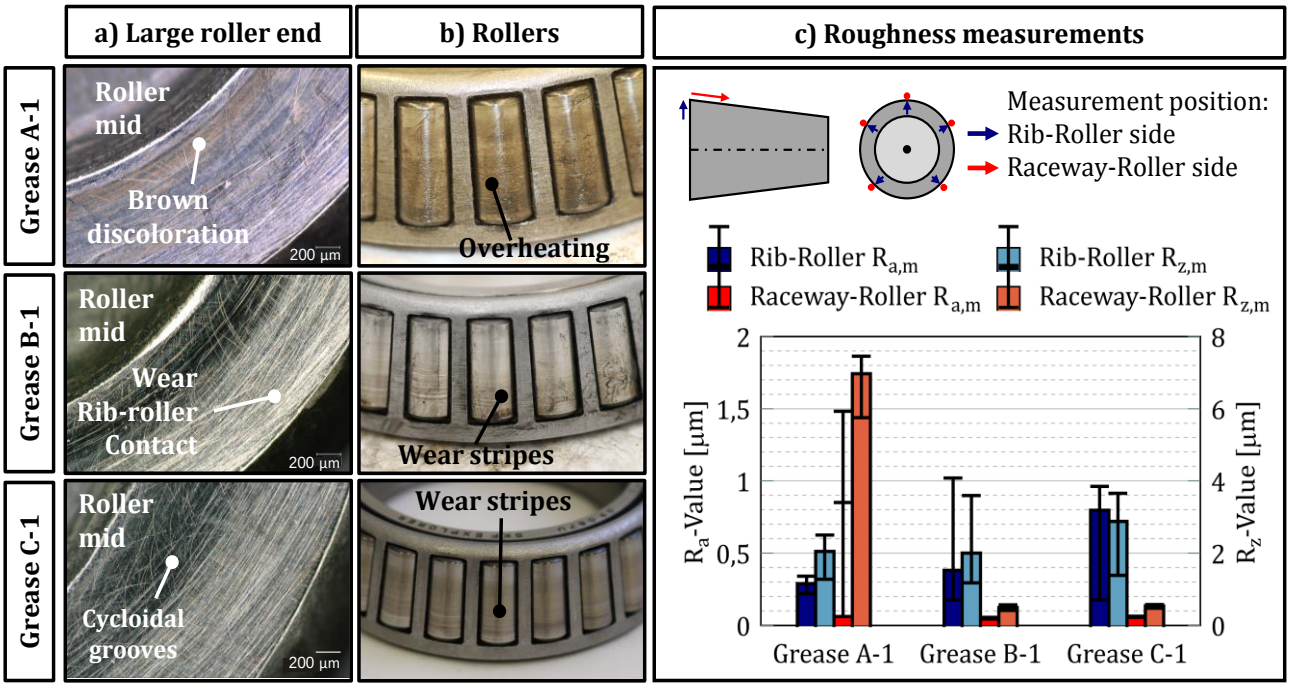


Fig. 12. Microscope views of the rollers (a) & (b) and roughness measurements c) after the tests with grease lubrication.

The wear in rib-roller contact is highest with this grease, which is also apparent from the roughness measurements in addition to the wear pattern shown in Fig. 13 a) and c). Similar to the bearing from test B-1, wear strips have also formed on the lower part of the roller in the bearing from test C-1. However, the results of test run C-2 are remarkable, since a failure due to an increased outer ring temperature occurred with almost no wear on the bearing. Fig. 13 illustrates the wear pattern of the rollers for the bearing from the test Grease C-2.

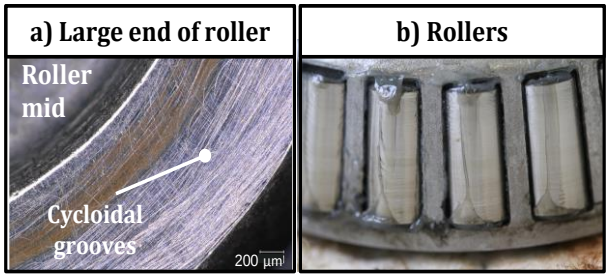


Fig. 13. Wear of rollers (Test C-2).

The results of the grease analysis of a sample of grease C after the second test using infrared spectroscopy is given in Fig. 14. No ageing or major change of the grease could be detected. Due to the low Fe content of 75 ppm, a reusability of the grease is possible.

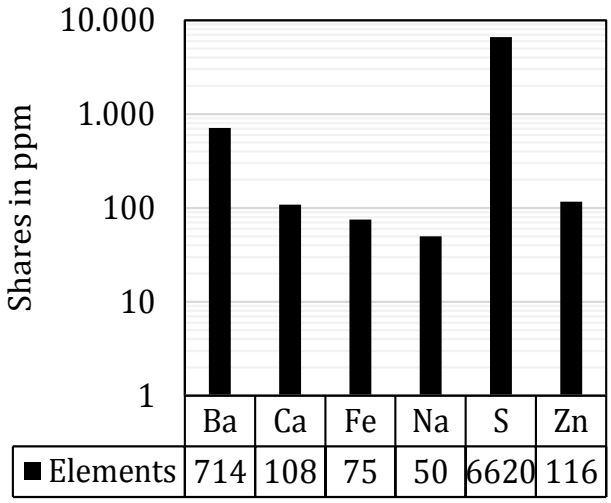


Fig. 14. Element shares of test C-2 after the test.

When comparing all grease-lubricated TRBs, the bearing tests with Grease A shows the best speed capability. Therefore, the temperature behavior of a SB lubricated with Grease A, which is commonly used in machine tool spindles, is investigated (see in Fig. 15). The SB achieves a stable outer ring temperature at a maximum rotational speed of 10,000 rpm. Subsequently, an improved lubricant film formation is established, while bleeding of oil from the grease is increased. This ultimately leads to a reduced load-dependent fluctuation of the heat generation during a speed step.

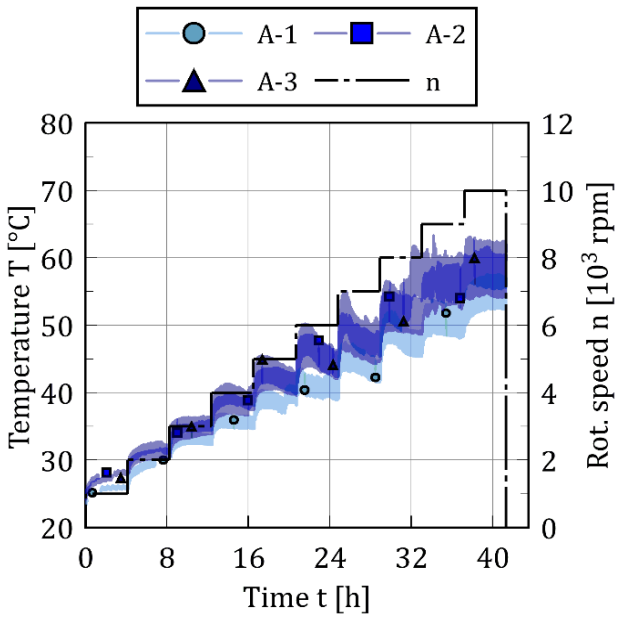


Fig. 15. Temperature behavior of a SB with grease A.

The mean vibration velocities of the tests with grease lubrication are shown in Fig. 16. In the first test with a TRB lubricated with Grease A, the vibration velocity values are comparatively high and exceed the ISO limit values already when reaching 2,000 rpm. However, the vibration values in the repeat tests are significantly lower, such that the first test represents an outlier. Considering this, the tests with Grease B show the highest vibration levels, whereas the TRB lubricated with Grease C reaches the lowest vibration speeds. This can be, for example, attributed to the dampening characteristics of the thickener. In contrast, with Grease C, the first failures already occurred at 9,000 rpm. Possible influencing factors for this reduced vibration behavior of Grease C may be damping effects of the thickener on one hand and the comparatively high base oil viscosity on the other hand. Again, the SB shows the best speed capability and stays below the critical vibration level of 5 mm/s according to ISO 10816-3 for all speed steps. In comparison the speed capability of Grease C with a steady-state behavior up to 5,000 rpm is only slightly worse compared to Grease B with a steady state temperature up to 6,000 rpm. In contrast, Grease C shows a significant vibration decrease, which is why it is considered as the best grease regarding the reduction of the vibration behavior of TRBs within this study.

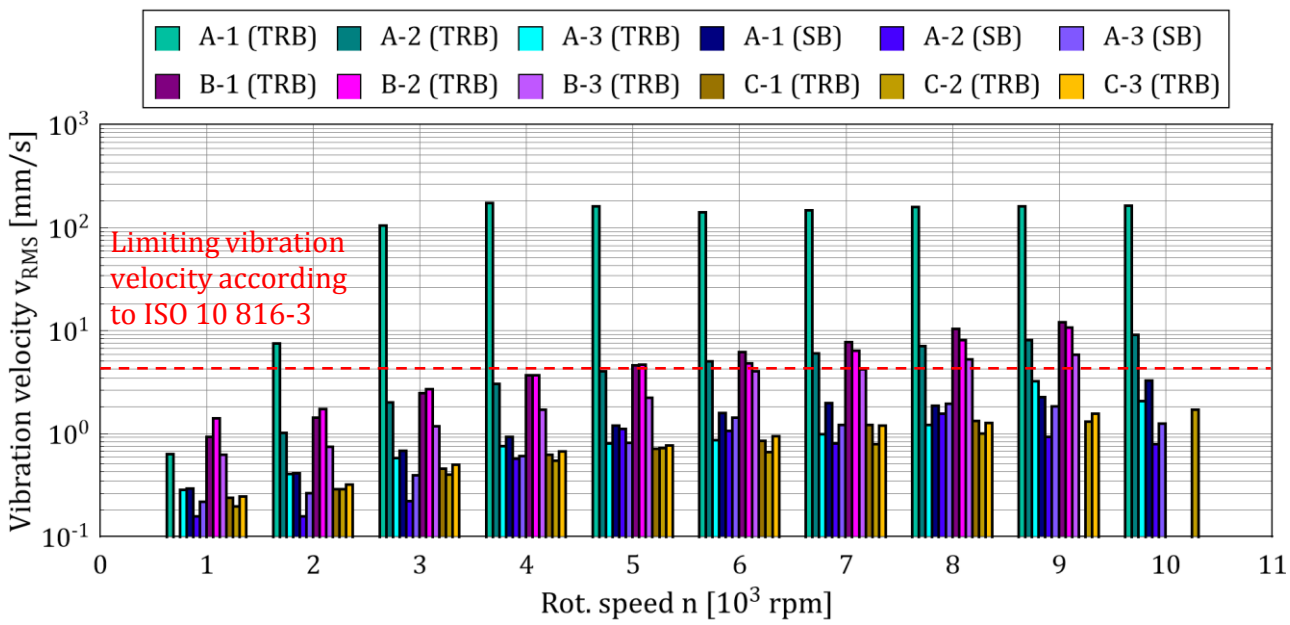


Fig. 16. Mean vibration velocities of each speed step for the greased TRBs in the tests

4. CONCLUSION

This paper presents a study of TRBs (type 32014-X) and SBs (type 7014) lubricated with different oils and greases for speed parameters up to 900,000 mm/min under varying radial loads. The speed capability of the bearings was validated using the bearing outer ring temperature, test rig vibrations. Apart from this wear patterns as well as the surface roughness of the bearings were analysed.

The roughness measurements on the bearings after the test indicate that the rib-roller contact tends to wear more under radial load compared to the raceway roller contact. An improved surface finish in the rib-roller contact, for example, could minimise the wear. The test results showed that TRBs could run up to 10,000 rpm under varying radial loads up to 3.5 kN by using oil-air lubrication. Nevertheless, the vibration velocities increased to a critical level at speeds above 9,000 rpm. The oil with viscosity class ISO VG 100 achieved the most stable steady-state temperatures in the step run compared to the lower viscosity oils. One possible reason for this behavior is a better lubricant film formation of the higher viscosity oil in the critically loaded rib-roller contact. A single SB with steel balls delivered similar results with lower steady-state temperatures and lower vibration levels. In general, the grease lubricated TRBs reached a steady-state temperature until 6,000 rpm corresponding to a speed parameter of 540,000 mm/min. The speed variation of this lubrication type is more limited compared to oil-air lubrication, which is (among other reasons) due to the poorer heat dissipation. With grease lifetime lubrication the cooling effect of the compressed air is omitted and the lubricant remains in the bearing. However, by using Grease C the vibrations are significantly reduced compared to the other tested greases. In contrast, Grease B with high pressure and anti-wear additives indicated a more stable temperature behavior reducing the wear in the critical stressed rib-roller contact. The SB lubricated with Grease A reached the highest speed step of 10,000 rpm without any problems in all tests performed. However, the outer ring temperature shows significant fluctuations compared to the ones of the tests of the oil-air lubricated SB. In general, the studies shows that the speed limits of TRBs can be improved by suitable lubrication. Even with standard bearings, speed limits close to common spindle speeds in machine tools can be achieved. A further developed high-speed TRB combined with an optimized lubrication strategy could be an alternative for SBs in machine tool spindles with a benefit regarding the load carrying capacity and stiffness. Due to the relatively high lubrication quantities for spindles and environmental aspects, further investigations could also focus on the question to what extent the lubrication quantities can be reduced without increasing the steady temperature levels or reducing the speed limits. The influence of lubricants, especially of the individual additives, on the operation of high-speed rolling bearings is difficult or in some cases impossible to calculate. Based on the exact knowledge of the lubricant properties, the influences can be evaluated using the tests via a black box approach. This paper provides a broad and detailed described experimental basis for the investigation of the speed capability of TRB regarding the lubricant influence.

Acknowledgement

The Laboratory for Machine Tools and Production Engineering (WZL) of RWTH Aachen University would like to thank the participants of the work group Spindle Bearing Systems for the technical support, the discussion of the results and the permission to publish the findings. This research is additionally supported by the German Research Foundation (DFG) within the research unit BR 2905/117-1 “Increasing the speed capability of tapered roller bearings for main spindles” (Steigerung der Drehzahleignung von Kegelrollenlagern für den Einsatz in Hauptspindeln”). Furthermore, special thanks go to Klüber Lubrication München GmbH & Co. KG for providing the lubricants and their analyses. In addition, we would also like to thank Mr. Thomas Engelmann of SKF GmbH and Mr. Ewald Förtschbeck of GMN Paul Müller Industrie GmbH & Co. KG for providing test bearings. Further thanks go to Mr. Jürgen Haselwander of the Chiron Group for and Dr. Mark Franken of Klüber Lubrication München GmbH & Co. KG for their expert comments.

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