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ORIGINAL ARTICLE

Study on cone roller bearing surface roughness improvement and the effect of surface roughness on tapered roller bearing service life

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Abstract The paper deals with the impact of internal geometry and micro-geometry of functional surfaces of the cone roller bearing on internal resistance—friction of roller bearings. It describes an ideal point of contact at intersection of cone axes and raceway values, as well as the necessary values of micro-geometry, which enable the use of the bearings under the most demanding installations in the automotive industry—in differential gears, or for installation of pinions of differential gears and cog wheels in gearboxes. It describes optimisation (modification) of production of supporting face of the inner ring of the bearing, the purpose of which is to ensure this perfect point of contact, as well as required values of micro-geometry of functional surfaces.

Keywords Bearing · Internal geometry · Function surface · Friction · Bearing life · Rollers · Raceway · Micro-geometry

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1 Introduction

Bearing service life is based on many factors. Depending on the application requirements, the actual service life can greatly vary. Even if bearings are firstly properly operated, they may eventually fail to perform satisfactorily due to some reasons, such as: an increase in noise, vibration, deterioration of grease or fatigue flaking of the rolling surfaces. Many technicians and researchers are involved into the process of the performance improvement. They are focused mainly on special points for improved performance, especially heat-treated material, higher dimensional and running accuracy, optimised surfaces and improved geometry [1–8].

The necessary pre-requisite for the proper operation of single-row tapered roller bearings is compliance with the principles of its internal geometry, i.e. macro-geometry and micro-geometry during production. When speaking about macro-geometry, the principle is obvious from the picture. Axes of the tapered rollers and raceways of the outer and inner ring should intersect at one point, as shown in Fig. 1. Ideal point of contact between supporting face of the inner ring and the large face of the cone is the centre of the effective width of the supporting face and centre of the annular area of the tapered roller face [9, 10]. When speaking about the micro-geometry, we have in mind surface roughness, waviness and roundness of functional areas. The quality of the bearings with tapered rollers is influenced primarily by roughness of the large faces of inner rings and roughness of large faces of rolling elements—tapered rollers. Depending on technology and values of these achieved parameters of the manufactured bearings, we may determine the state-of-the-art of the given manufacturer of bearings. We may consequently assess the stability of such production and quality of bearings.

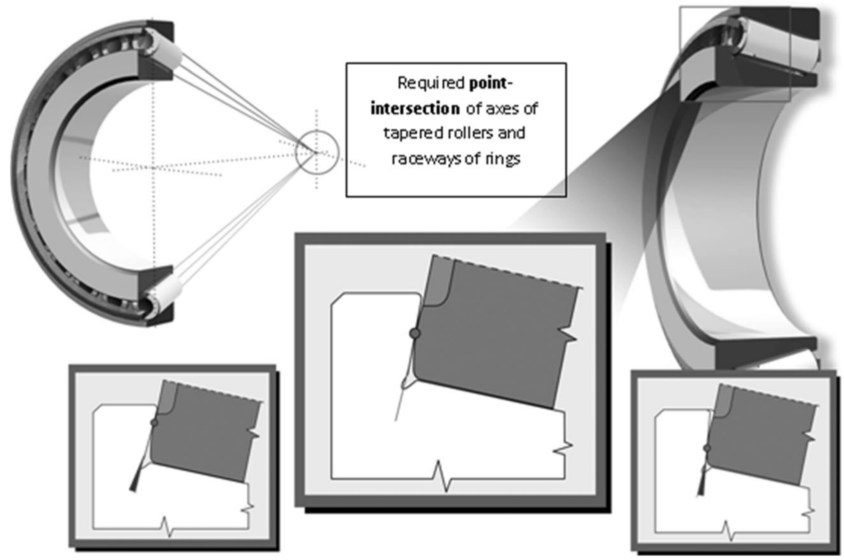

Fig. 1 Principle of internal geometry of bearings with tapered rollers

2 Analysis of the current state

The rolling contact bearing is an element of machinery with a very important role, and it dominates the performance of the machine. Therefore, investigation of bearing geometry becomes essential. Considerable research studies were carried out to examine the geometric configuration, contact geometry, contact stresses, vibratory loads, heat generation rates and bearing failures [11–15]. It is not an easy task to comply with these requirements, especially when we realise that the bearings with tapered rollers consist of four parts and each part has a number of parameters, which must be produced and adhered to within the stipulated tolerances. Since production of bearings is more or less series or mass production (depending on the magnitude of a series in number of pieces), it is possible to use for ensuring the correct (stipulated) design of the bearings or the required parameters of functional surfaces with an advantage preventive method (not the quality measurement performed only after fabrication of the components), namely statistical process control (SPC). It is necessary to choose one of the methods, depending on whether it is possible to use computer support of production (in such case, we choose, e.g. the methods of mean values and of standard deviations x_{priem}—S with number of selected samples=5 pcs), or it is possible to use computer support of production (in this case, we choose the method of medians and the range Xm_{ed}—R with number of selected samples=3 pcs). Magnitude of tolerance for these products is of the order of several microns. The tolerance analysis of the bearings must be performed in such a manner that after the production of individual components of the whole bearings and after their assembly, the pre-requisite for their correct functioning mentioned above is fulfilled. It may seem easy, but this one of the basic engineering components must be manufactured with use of real machining tools and instruments, which also have their production capabilities and limitations [16–20]. After their production at each manufacturing operation, it is necessary to measure the individual produced parameters with the use of appropriate measurement instruments, which also have some uncertainty of their measurements. Environment temperature and temperature of process media also influence the precision of the manufactured components and of the whole bearing. We have established by experiments, observation and measurement that change of temperature by one centigrade causes change of the length dimension (bearing ring diameter) by 1 μm. When we realise that, for example total tolerance of the inner hole of the inner bearing ring is 0.008 mm, we will then, during variation of the ambient temperature within the interval of 8 °C (which can realistically occur during one working shift), theoretically use this whole tolerance even without direct influence of the system machine—tool—workpiece. That is why it is necessary to ensure a constant temperature during machining of components. Operation and control of all these processes is ensured by people, who also influence the manufacturing process. Nowadays, the automotive industry requires from its suppliers the so called 'zero defect' supplies [21]. If formerly it was sufficient that production process was in the area of $+3\sigma$ (σ—standard deviation), today this is no longer sufficient and no defects are required, i.e. that the manufacturing process must be in the area of $+6\sigma$. This means that if the customer's receiving inspection (or inspection on the assembly line, if receiving inspection is not made) finds a single defective product, the whole supply of components is sent back to the supplier, who must usually also pay any additional costs incurred. It is dangerous and it may even lead to a liquidation of such supplier, if the components had already been assembled into some units or even into the whole cars, and consequently the affected cars must be called to the service centres. The supplier of components, who is unable to exculpate himself, pays all the costs associated with the replacement of defective components (some insurance products already exist nowadays in our country, which insure against such situations and eliminate the losses incurred). When this internal geometry is not adhered to, it has an impact on the reliability and service life of bearings and subsequently on the operation and service life of the entire bearing. The automotive industry recognises in principle 4 basic types of supports, where the bearings with taper rollers are used:

– Support of wheel
– Support of the gear shafts
– Support of cog wheels in the gear (differential gear)
– Support of pinion in the gear (differential gear)

Individual supports were presented in the order of their complexity. The most difficult support is the support of pinion in the differential gear. The support is made with a pre-stress. The pre-stress may be understood as another additional axial force in bearings, which ensures the necessary rigidity of the support. It is the force, which is in the support in addition to the force, which the device must transmit [9, 10]. The bearings, which do not have an optimum point of contact in the internal geometry of the bearing, cannot be used for this support. Generally speaking, such bearings are not suitable for other supports, either, but this insufficiency affects the most the differential gear. Friction between individual components, and namely between the supporting face of the inner ring and faces of tapered rollers gets excessively increased due to bad geometry, which in turn causes an increase in temperature. If this increased temperature is not removed from the environment, i.e. if it is not cooled, for example by oil or by suitably shaped ribbed case, a premature termination of service life of the bearings, and thus of the entire device, takes place. The bearings in such device have, after its disassembly, blue colour. It is an unmistakable sign of an overheating (burning) of the bearing.

3 Realisation of experiments for achievement of the desired micro-geometry

Low friction in bearings is influenced also by micro-geometry achieved at production, i.e. namely the surface roughness, waviness and roundness of functional surfaces. The most important influence has the supporting face of the inner ring and the large face of tapered rollers and the micro-geometry achieved on them, and then also other parameters, such as micro-geometry of the surface of raceways of the outer and inner ring and of the rolling elements [22]. However, non-observance of the internal geometry has worse consequences for the bearing than non-observance of micro-geometry. If the micro-geometry is not observed, then it is possible to that after certain time of operation individual surfaces will adjust to each other by their mutual interaction, which may eliminate a failure of the bearing. Nevertheless, when the internal geometry of the bearing is not observed, the effective active area serving for running or slip of rollers is lost, and danger of the so-called edge contact becomes real. When this edge contact appears, service life of the bearing is terminated very quickly, and consequently, also the service life of the whole equipment. Necessity of the optimal internal bearing geometry for the bearings used in the most demanding support—in the differential gear, is self evident. In order to reduce the friction between individual components and their functional surfaces, it is necessary to observe the stipulated parameters of micro-geometry, in order to avoid damage of the oil film and to maintain the hydrodynamic lubrication. Experiments and tests of bearings have shown that in practise, it means to observe the roughness *Ra* at the level of 0.04 μm on the supporting face of the inner rings and on the large face of tapered rollers. On standard bearings for other supports a roughness of these surfaces between 0.16 to 0.20 μm, it means even 5 times worse (higher), is sufficient. The roughness was measured on the measuring equipment Talysurf (Fig. 2) [10].

Such roughness cannot be anymore normally achieved by grinding, and the surface must be subjected to super-finishing with use of stones with finer granularity than that of the grinding wheels. Designation (label) of such bearings contains in addition standard marking of the bearing also the symbol ‘CL7C’. This symbol means that these bearings have a guaranteed friction torque without the necessity of any running-in of such bearings. These bearings can be right from the beginning (without any running-in) charged to the maximum load without a risk of their damaging or seizing. Apart from that, we have also the bearings with the symbol ‘CL7A’, which have the same characteristics, but which require at least the minimal short running-in after their installation into the equipment.

The company, in which we carried out our experiments and observations, came with the requirement for a shaft support

Fig. 2 Measurement of roughness on the supporting face of inner ring of the bearing with tapered rollers

gearboxes (differential gears) for automotive and tractor industries (for unnamed renowned brands) with use of the bearings in executions CL7A or CL7C. As already mentioned above, the quality of machining of the supporting face of the inner ring and of the large face of tapered rollers, as well as micro-geometry reached on these surfaces, have the most important impact on such supports. The unnamed company producing bearings was capable of meeting the other parameters required for such bearings. However, the company could not so far perform machining by grinding [8, 23–27] of the supporting face of the inner ring and of the large face of the tapered roller in conditions of the mentioned serial production of bearings in order to achieve the roughness on these functional surfaces of Ra ≤0.04 μm, due to the lack of super-finishing equipment for super-finishing of the supporting faces of inner rings. Such roughness can generally be achieved only by super-finishing machines. Figure 3 shows a view into the working space of the grinder, where we see an inner ring of the tapered roller, and on the right side, we see an abrasive disc grinding the support face. Grinding is performed in supports, and it is sufficiently precise on this type of machine made in Germany.

Some manufacturers of bearings, who want to supply to their clients the bearings in executions CL7A and CL7C, and who are unable to meet manufacture the required qualitative parameters, replace this production by running-in of the bearings. This is, however, very lengthy time consuming and expensive solution requiring testing stations, which can perform such running-in. We did not want to use this not-quite-professional approach that is technically not optimal. On the contrary, we chose the route of planned experiments, and we wanted to achieve the repeatability of production of qualitatively better bearings for demanding applications by modification of technology for manufacture of functional surfaces of the bearing components. At the beginning, we tried to grind the support face of the inner ring with use of grinding wheels with the finest possible granularity that was offered by the manufacturers of grinding wheels. This was, however, insufficient, and no manufacturer of grinding wheels was able to supply us the grinding wheels with the granularity required by us, stating that they did not have the appropriate technology for production of such grinding wheels, and moreover, they were unable to guarantee achievement of the surface roughness required by us. We then, nevertheless, found a manufacturer of grinding wheels, whom we asked to manufacture a grinding wheel from abrasive material used for manufacture of grinding stones for super-finishing. He supplied us with such a grinding wheel with granularity of the grinding stone closest possible to the material of the super-finishing stone, but without any guarantee of achieving the desired roughness. The bonding agent of these discs was also modified by the manufacturer, which influenced the results of the grinding.

Fig. 3 View into the working space of the machine

4 Comparison of the original and optimised technology

Original grinding stone for standard bearings: 200×8×51 89A 120K 4V 112 TYROLIT [10]. Newly manufactured grinding stone for optimised bearings for demanding installations: 200×8×51 3CB3 150 K9 VEZ 1 HERMES. We tried for a longer time this grinding wheel with this granularity to find out what technological conditions must be met in order to achieve the desired roughness. As a result of the planned experiments, we have achieved the desired result, i.e. we have achieved the roughness on these functional surfaces (roughness of the large face of the inner ring and of large faces of tapered rollers) of Ra ≤0.04 μm. The used machine: Grinder SWäAGL125 PC 610, made in Germany. Technological conditions of the grinder are as follows:

- Cutting speed: up to 50 m/s
- Revolutions: 4500 rpm
- Revolutions of the working spindle: 1000 rpm
- Allowance for grinding of standard bearings: 0.2–0.35 mm
- Allowance for grinding of optimised bearings:
- Roughing operations 0.2–0.35 mm
- Finish grinding 0.03–0.05 mm

Requirements to the machine qualitative parameters after grinding (inner ring with a detail of the support face—see Fig. 4 [10]):

- Roughness of the support face for standard bearings: Ra=0.20 μm
- Roughness of the support face for optimised bearings: Ra=0.04 μm
- Axial wobbling of the support face in respect to the basic face of the standard bearing: 0.005 mm
- Axial wobbling of the support face in respect to the basic face of the optimised bearing: 0.002 mm

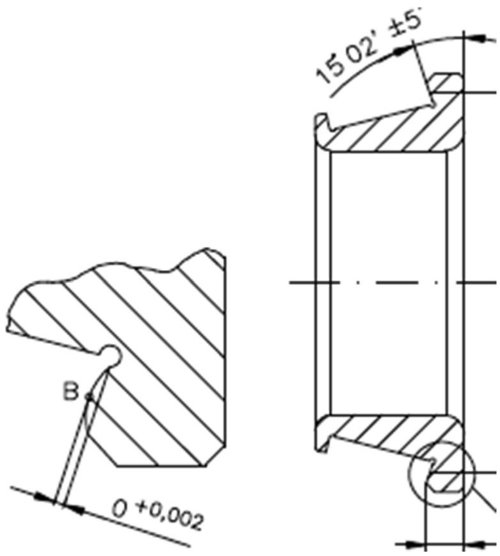

Fig. 4 Inner ring with a detail view of the support face

- Tolerance of the angle of the support face of the standard bearing: 20 angular minutes
- Tolerance of the angle of the support face of the optimised bearing: 10 angular minutes
- Surface of the support face must not be reheated (i.e. decarburised)
- Profile (shape) of the support face must be convex 0.002 mm

5 Results and discussion: use of findings in practise

We were able to manufacture the supporting faces of rings with the required surface roughness on the grinders with the grinding wheels with granularity and material close to those of the stones for super-finishing. It was a great success, because it was not necessary to achieve the qualitative parameters of the support face by lengthy, expensive and inefficient running-in of bearings, as it was usually applied by bearing manufacturers in order to achieve these parameters. This requires moreover testing stations for realisation of this running-in of bearings.

The bearings achieved hydrodynamic lubrication (since we achieved the roughness of $Ra \leq 0.04$ mm) and in the testing laboratory; it was possible to load these bearings to full maximal axial force exceeding 6 t, which the bearing had to withstand. When we imagine for an illustration that a one-passenger vehicle of the category of small cars weighs approximately 1 t, we might have loaded axially on such tested bearing 32309BARJ2Q with diameter of the inner ring bore of 45 mm, six vehicles and the bearing should rotate smoothly at its operating speed.

Explanation of the marking 32309BARJ2Q:

- BA—greater contact angle alpha of 17°
- R—flange on the outer ring required by the customer for support in the gear box
- J2—reduced overlap of the cage in comparison with the standard bearings
- Q—improved quality reached by the above optimisation of production

5.1 Test of basic load dynamic

To complete the picture, we mention that the tests are carried out in the testing laboratories in accordance with the specified methodology on 20 bearings, which are mounted on the shafts in testing stations. The test terminates when five bearings are gradually excluded due to wear, i.e. until pitting is reached. The tests determine after how many revolutions the test was terminated (the requirement is 1 million of revolutions), and what kind of damage was found in the bearings excluded from the testing. When no bearing is excluded during testing, the test runs until reaching of 1 million of revolutions, which means that the test meets the basic dynamic load rating expressed as 100 %. If we want to determine what basic dynamic load can be achieved by the tested bearing even after reaching 100 %, the test continues. This, however, sometimes takes a very long time, and therefore such tests are terminated after 9 months, because the tests are expensive. The bearings achieved at these tests are more than 100 % of the basic dynamic bearing capacity. This means that the optimisation was from the viewpoint of design proposed and realised correctly and successfully.

5.2 Test of moment of friction

Another test, which was carried out on the optimised bearings consisted of measurement of the moment of friction. The testing stand was adapted for simultaneous testing of four bearings. The bearings were rechecked before this test in order to exclude impacts caused by failure to observe the technological process on the outcome of tests of the moment of friction. This was a rapid test of bearings, which on the basis of the internal friction of all rotating parts assessed the quality of bearing more or less from the perspective of technology. The tapered roller bearings were loaded in the test stand by the maximum force, which the bearing should have withstand according to calculations, and it was monitored what is the moment of friction (internal resistance) of the given bearing. It was possible to realise the test either by gradual loading of bearings from the minimum to the maximum force or by immediate loading of the bearing by the maximum force it should withstand. Testing by gradual loading is used rather for standard bearings, to which we enable by this mode their initial running-in. Testing by the maximum load right from the start is applied to the bearings, which should be able at teal operation to withstand an immediate maximal load right from the start of operation. These are particularly the bearings designated for the most demanding installations in the differential gears CL7C and CL7A where the bearings are mounted with a pre-stress. The measured values of the moment of friction of the standard bearing are presented in Table 1, and the measured values of the moment of friction of the optimised bearing are given in Table 2. Graphical representation of evolution of the moment of friction of the standard bearing is given in Fig. 5, and the measured values of the moment of friction of the optimised bearing are given in Fig. 6.

At the beginning, we tried to perform the measurement of the moment of friction of the standard bearing before its optimisation by immediate loading of the bearing by the maximum force F_a=63,000 N, although we knew that the bearing did not have potential to withstand it without the optimisation. Our assumptions were confirmed, the bearings stopped after couple of seconds of running. We therefore continue the planned experiment. We first performed running-in of the bearings for 24 h under the load force F_a=9000 N. The loading was then performed gradually in accordance with Table 1. Every 10 min, we increased the load and read the moment of friction:

– Measurement no. 1—we increased the load to F_a=18, 000 N, and after 20 min. we measured the moment of friction of 3.1 Nm
– Measurement no. 2—we increased the load to F_a=27, 000 N, and after 30 min, we measured the moment of friction of 3.4 Nm
– Measurement no. 3—we increased the load to F_a=36, 000 N, and after 40 min, we measured the moment of friction of 3.6 Nm
– Measurement no. 4—we increased the load to F_a=45, 000 N, and after 50 min, we measured the moment of friction of 4 Nm
– Measurement no. 5—we increased the load to F_a=54, 000 N, and after 60 min, we measured the moment of friction of 5.1 Nm
– Measurement no. 6—could not be made since the bearings at the load of 54,000 N seized up immediately after the attempt to increase the load

Table 1 Measured values of the moment of friction of the standard bearing

Run time [min]	20	30	40	50	60
Moment of friction [N/m]	3.1	3.4	3.6	4.0	5.1

Table 2 Measured values of the moment of friction of the optimised bearing

Run time [min]	10	20	30	40	50	60
Moment of friction [N/m]	4.8	4.5	4.6	4.5	4.5	4.4

Graphical representation of evolution of the moment of friction of the standard bearing in Fig. 5 clearly shows how the curve of the moment of friction steeply deteriorated between the 50th and 60th minute of running, so it was possible to expect the breakdown.

We started the measurement of the moment of friction by 1 min of running at the minimal load force F_a=2300 N. This mode was supposed to ensure only fitting of all bearing components in the testing stand. After 1 min, we measured the moment of friction of 1.3 Nm, and we then immediately subjected the bearings by load of the maximal force F_a=63, 000 N, which the bearings should have been able to withstand according to calculations. Increase to the maximal load was realised within a short time of 45 s. During testing, no undesirable increase in temperature occurred; its maximal value was 65 °C. Noise also did not increase, and no vibrations were detected. The planned experiment then ran gradually in accordance with Table 2; we read the moment of friction every 10 min under the maximal load of F_a=63,000 N:

– Measurement no. 1—after 10 min, we measured the moment of friction of 4.8 Nm
– Measurement no. 2—after 20 min, we measured the moment of friction of 4.5 Nm
– Measurement no. 3—after 30 min, we measured the moment of friction of 4.6 Nm
– Measurement no. 4—after 40 min, we measured the moment of friction of 4.5 Nm
– Measurement no. 5—after 50 min, we measured the moment of friction of 4.5 Nm
– Measurement no. 6—after 60 min, we measured the moment of friction of 4.4 Nm

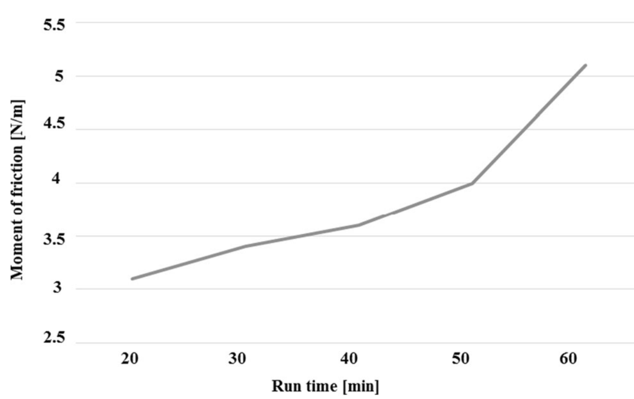

Fig. 5 Graphical representation of evolution of the moment of friction of the standard bearing

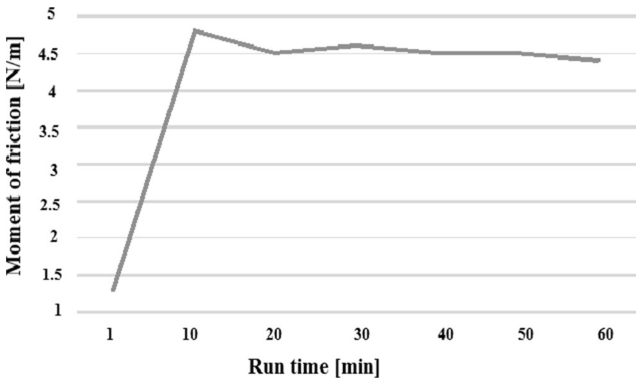

Fig. 6 Graphical representation of evolution of the moment of friction of the optimised bearing

Graphical representation of evolution of the moment of friction of the standard bearing in Fig. 6 clearly shows how the curve of the moment of friction after the maximum load by the force F_a=63,000 N achieved the level of the moment of friction of 4.8 Nm, which is normal; then for 1 h, the moment of friction was constant, or it had a slightly decreasing trend. Evolution of the moment of friction exactly corresponded to that of the bearings CL7C and CL7A, designated for the most demanding installations [20–22], particularly in automotive industry.

6 Conclusion

When we inspect the diagrams in Figs. 5 and 6, and if we compare the graphical evolution of the moment of friction of standard bearing and that of the optimised bearing, we see immediately the quality of the bearing before and after its optimisation. The bearings were before an optimisation first running-in for 24 h under the minimal load of F_a=9000 N; after this long-lasting running-in, they were gradually loaded by a higher load in regular 10 min intervals. The bearings subjected even to such sparing testing did not achieve the maximal load that they were supposed to withstand. The curve of the moment of friction increased after each consecutive load, and after 50 min, it started to increase very steeply. The temperature, as an accompanying phenomenon, also started to increase. After the next load, after 50 min, the noise increased, vibrations appeared and the bearings stopped to roll, as they got seized.

The bearings were after the optimisation after a short 1-min run (in order to ensure fitting of the bearing components) at the minimal load force F_a=2300 N immediately from the start loaded by the maximal force F_a=63,000 N, which the bearings were supposed to withstand according to calculations. We set this drastic mode, because the bearings from the most demanding installations (differential gear, wheel-drive assembly) work at their exploitation with pre-stress, and they are in practise subjected to such loads. The curve of the moment of friction reached after the maximal load its maximum of 4.8 Nm, but it did not increase anymore; it remained constant or even had slightly decreasing trend. When the bearings behaved like that at the maximal load, we can say with certitude that they are suitable for the most demanding installations, where such loads naturally do not last permanently and continuously, since they change.

These achieved results are of great importance for the manufacturer and find practical application, because we were able to implement them also into other bearings of the given dimensional and geometric series. We have verified by laboratory testing the possibility of flat extension of application of the achieved knowledge.

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