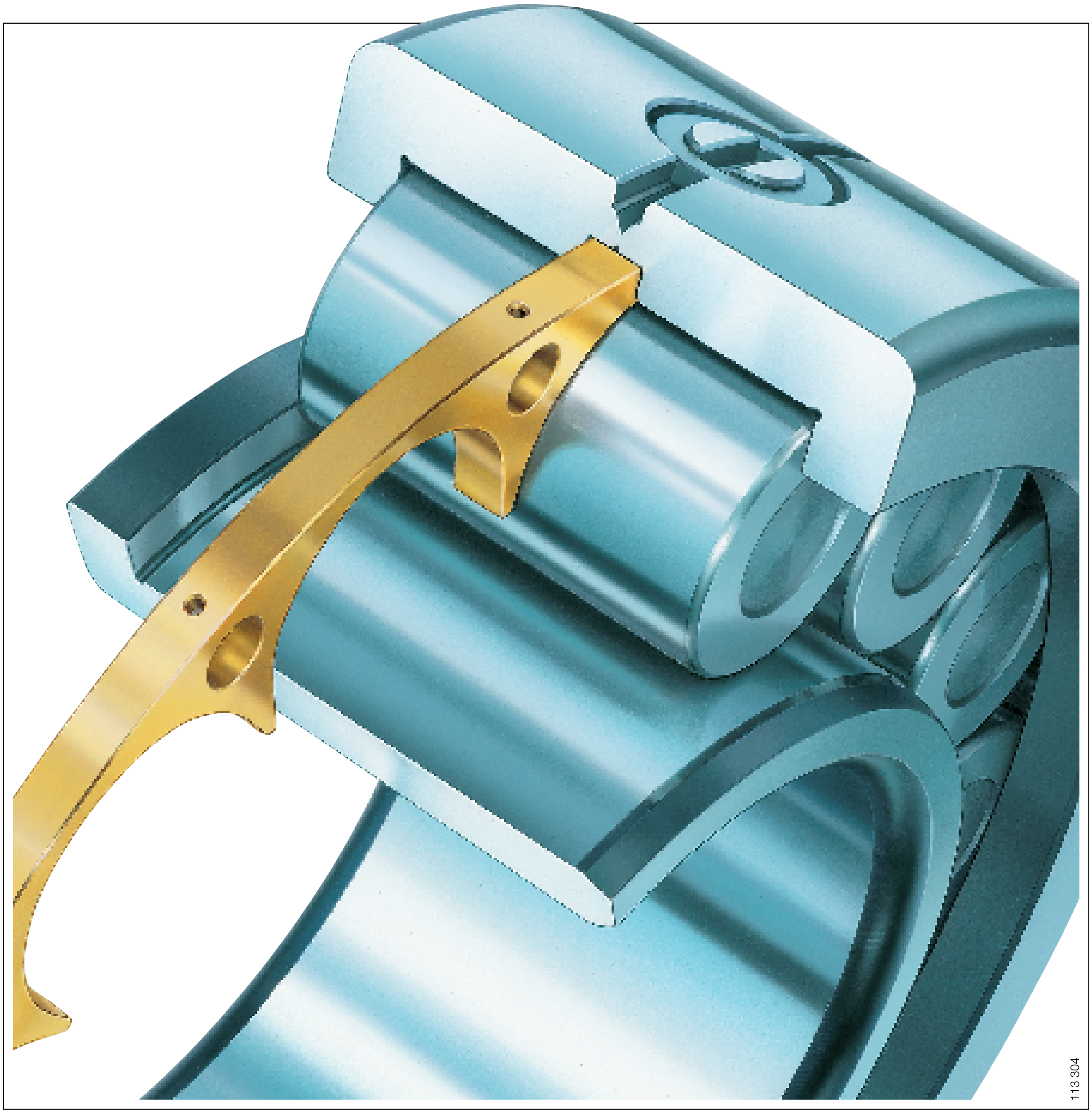

Low friction cylindrical roller bearings

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INA reprint from „antriebstechnik“
April 1993, No. 4
Vereinigte Fachverlage GmbH
Revised April 1995

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The high performance density of many modern machines places enormous demands on the rolling bearings used; these often exceed the capabilities of conventional designs of bearings, necessitating early and frequent replacement.

In such machines bearings with high limiting speeds even under high radial loads are required. In general, convential cylindrical roller bearings cannot meet these requirements.

INA has developed two new types of cylindrical roller bearing – series LSL with a disc cage and series ZSL with plastic spacers between the rollers – which can fulfil the requirements. This allows reliable and economical operation of modern high-performance machines.

The following paper presents these two new developments from INA, which combine the advantages of full complement roller bearings (high load carrying capacity) and cage-guided bearings (high limiting speed).

1 Introduction

Cylindrical roller bearings have proved to be very successful in both mechanical engineering and the automotive industry. They ensure reliable power transmission in both the radial and axial directions. Modern aggregates are becoming increasingly compact due to optimized manufacturing processes, high performance materials and accurate calculations. There is also a corresponding increase in the power density. This bearing is now virtually indispensable for designs with high power transmission in limited spaces.

There are two basic designs for this type of bearing:

- Full complement cylindrical roller bearing (figure 1)
- Cylindrical roller bearing with cage (figure 2)

Each of these bearing systems has particular features and is therefore assigned to certain applications accordingly. When selecting the appropriate bearing type, compromises must often be made and the advantages and disadvantages weighed out. The design engineer has to choose between the caged cylindrical roller bearing and the full complement cylin-drical roller bearing for the design of the bearing arrangement. In making this decision, the main criteria for correct selection should be precisely determined. In the case of caged bearings for example, the cage requires about 15 % to 35 % of the space between inner and outer ring. The theoretical 100 % utilization of the load carrying capacity of the rolling elements is thus reduced to approx. 85 % to 65% and the basic rating life decreases accordingly from 58 % to 24 % compared with the full complement roller bearing.

Figure 1 Full complement cylindrical roller bearing SL 1923

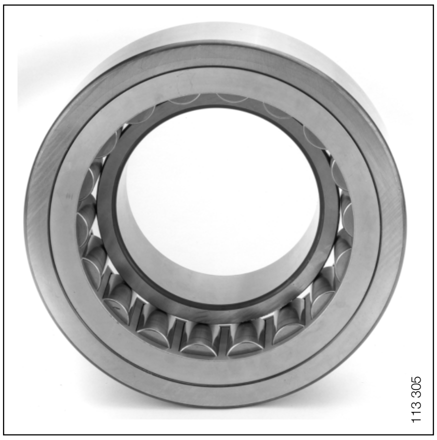

Figure 2 Cylindrical roller bearing with cage NJ 23

In the case of full complement bearings more heat is generated due to the high power density. The limiting speed is therefore reduced to 50 % compared with the caged bearing. The permissible dynamic and static axial loads are directly proportional to the lateral contact faces be-tween the rolling elements and the corresponding ribs on the inner and outer rings. In this case, the full complement bearing is preferable as it has more rolling elements.

Frictional torque, heating, cooling, prevention of smearing, wear, radial dimensions and bearing power density should also be taken into consideration when selecting the appropriate cylindrical roller bearing.

As far as the development of cylindrical roller bearings is concerned, INA's idea is to combine the positive features of both full complement cylindrical roller bearings and cylindrical roller bearings with cage to create a new generation of cylindrical roller bearings.

This new idea has created two highly advanced bearing systems which can be described as follows:

- Cylindrical roller bearing with disc cage (figure 3)
- Cylindrical roller bearing with spacers (figure 4)

Figure 3 Cylindrical roller bearing with disc cage, series LSL 19 23

Figure 4 Cylindrical roller bearing with spacers, series ZSL 19 23

Figure 5 Spacers made of composite material

2 New bearing series LSL and ZSL

2.1 Structure

Cylindrical roller bearings with a solid disc cage guided by ribs on the outer ring and cylindrical roller bearings with spacers are mainly bearings of series NJ. The standard design of these bearings is in the dimensional range 23. ZSL bearings are designed for the bore reference numbers 05 to 24 and LSL for 20 to 60. **They are fully interchangeable with known cylindrical roller bearings of series NJ 23 and NJ 23 VH.** The inner ring, rolling elements and outer ring of series ZSL are identical to those of our established series SL 19 23.

By removing a cylindrical roller from the rolling element unit, space has been created for a spacer to separate the remaining rolling elements from one another (figure 5).

For series LSL, the inner ring and rolling elements are identical to series SL 19 23 and ZSL 19 23. The outer ring is radially split and has an additional groove in the raceway between the ribs; a specially developed brass disc cage is located in the groove (Figure 6).

In conjunction with heat treated retaining rings, the outer ring with the integral disc cage forms a bearing unit.

The rolling element unit in bearings of series LSL 19 23 with this new disc cage also has only one rolling element less than the full complement bearing of series SL 19 23. As a result, extremely high load carrying capacities are obtained compared with cylindrical roller bearings with cage.

LSL and ZSL bearings are also available on request as serial bearings provided economically viable quantities are required. Figure 7 shows the main bearing of a D.C. engine as used in high speed railway applications. The bearing is fitted with this cage.

2.2 Series LSL

This new one-piece cage has been developed as a plane disc with the rolling elements held in pockets. The inside diameter of the cage has been pulled down below the pitch circle line in order to provide retention for the rolling elements, i.e. the inner ring can be mounted separately. The disc cage is located at the exact centre point between the ribs of the outer ring by means of a groove cut in the outer raceway. The pockets of the cage have a special shape (like teeth). They contact the rolling elements exactly in the plane of symmetry. All the tangential forces are therefore directed very safely into the cage. Unlike conventional cages (figure 8) which come into contact with the rolling elements at any point on the surface, the disc cage cannot exert any gyroscopic effect on the rolling elements. In other words, a conventional cage may in some respect force the rolling elements to skew whereas the new disc cage does not interfere with the guidance of the rolling elements through ribs or shoulders. The cage minimizes the distance between two rolling elements which means that there is a greater number of rolling elements in the available space compared with conventional caged bearings.

Notch stress which can arise with conventional solid cages does not occur any more with the disc cage. As usual the rolling elements are guided between the ribs on the outer ring. As the disc cage, unlike conventional cages, does not interfere with this guidance, the velocity drop of rolling elements outside the load area is very small.The required acceleration moment for the rolling elements is therefore much lower and less frictional torque and heat are generated in the bearing. Due to the lower coefficient of friction and fric-tional torque of these disc cage bearings, much higher limiting speeds are achieved, even with normal accuracy. This theory has been proven by various tests on INA's heavy-duty bearing test rig.

The cage which is guided in the outer ring groove represents an optimal sliding bearing: a sufficient oil film is formed at lower speeds to separate the contact surfaces and provide perfect hydrodynamic running conditions.

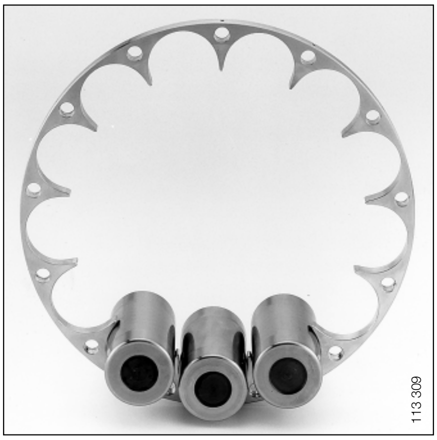

Figure 6 Solid brass disc cage

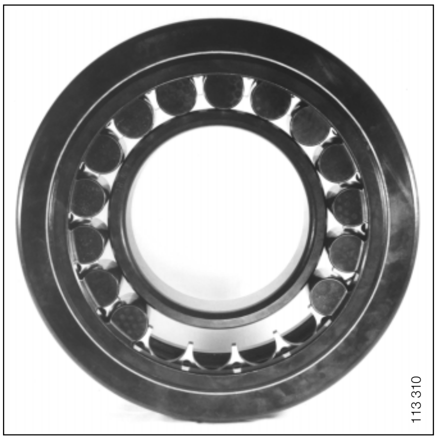

Figure 7 Special bearing with brass disc cage

Figure 8 Conventional solid brass cage

Lubricant exchange occurs by means of radial lubrication holes arranged centrally around the outer edge of the cage which join with axial through holes in the cage.

In rolling bearings where heat dissipation is provided by the lubricant, the free axial space should be as large as possible to ensure a regular and high volume oil flow in the bearing. With full complement bearings, the gap remaining between two rolling elements is relatively small. With conventional caged bearings, this free space may be filled by the cage itself, thus reducing and restricting the oil flow. The open disc cage bearing provides the solution: the lubricant is allowed to flow freely with minumum obstruction. Bearings of series LSL are black-furnished, a further positive feature which provides a level of protection against smearing to a degree not achieved before.

2.3 Series ZSL

These spacers specially developed for this series are designed in such a way that the bearing and the inner ring can be mounted separately.

Due to the special design of the spacer (PES) the point of contact on the rolling element outside diameter is always at the pitch diameter circle of two adjacent rolling elements irrespective of the operating condition. This prevents uncontrolled radial force components which usually occur in other spacer applications. This has a positive effect on the frictional behaviour in the bearing or between the spacers and rolling elements. Furthermore, no additional constraining forces have to be overcome; this can be seen in the low frictional torque of the bearing.

The spacers are axially guided between both ribs on the outer ring. The radial limits in the mounted condition are formed by the raceways. The spacers with radial clearance are pushed outwards, even at low speeds, and lie on the outer ring race-way. The special design of the radial abutment surface facilitates hydro-dynamic separation between the spacer and the raceway as the speed increases. The central recesses at the inner and outer diameter provide optimum bearing lubrication. Stripping the oil film at the raceways can be avoided and the frictional torque independent of the load is reduced due to the free flow of oil. Every effort has been made in the design of the spacers to obtain the best axial lubricant flow.

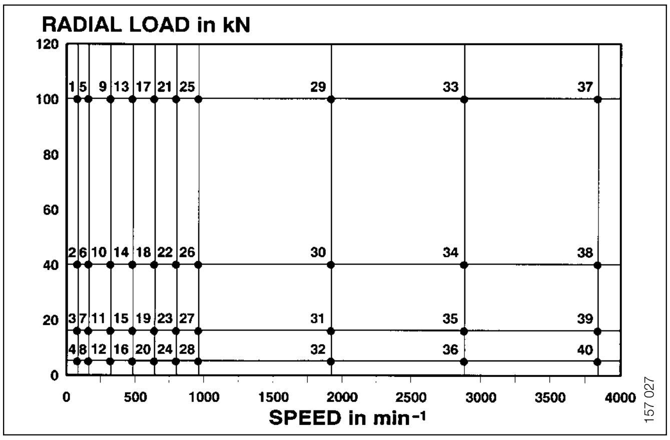

Figure 10 Test matrix

3 Research and tests

3.1 Test rig

Various tests have been carried out on the operating behaviour of the LSL and ZSL bearings as shown in figure 9.

In this case INA has used a test rig with a so-called „flying" test piece arrangement. This test rig has already been described in detail by Giese and Scherb [1], [2].

The test rig has been designed to provide easy access to the test piece for the measuring sensor application and for good visual examination by means of stroboscopic video recordings or recordings using high speed cameras. Easy mounting and a clear measurement of the frictional torque in the test bearing should be possible without the interference of any external influences.

The influence of the load direction in the load area with the force of gravity or against it, as well as intermediate positions, together with the influence of a superimposed thrust loading should be able to be investigated. These requirements led to the test rig as shown in figure 9.

The test bearing is loaded via the central bearing unit which exerts a defined test load through servohydraulically controlled pressure cylinders; this test load is constant throughout an extremely wide range of temperatures. In order to determine the limiting speeds of the new bearings it was necessary to replace the two angular contact ball bearings of the central bearing unit with an LSL bearing. It was then possible to set the load and speed conditions without damaging the bearing exerting the load.

The central bearing unit can be rotated through 360° in a supporting ring in order to achieve various load directions. The axial load on the test bearing is applied by means of four servohydraulic controlled pressure cylinders which act on the test bearing through the stationary outer ring of the central bearing unit on the shaft and then through the rib on the inner ring. Variable test load may be simulated by a special actuator. Constant lubricant viscosities are guaranteed for all test conditions due to extremely efficient oil coolers or heaters.

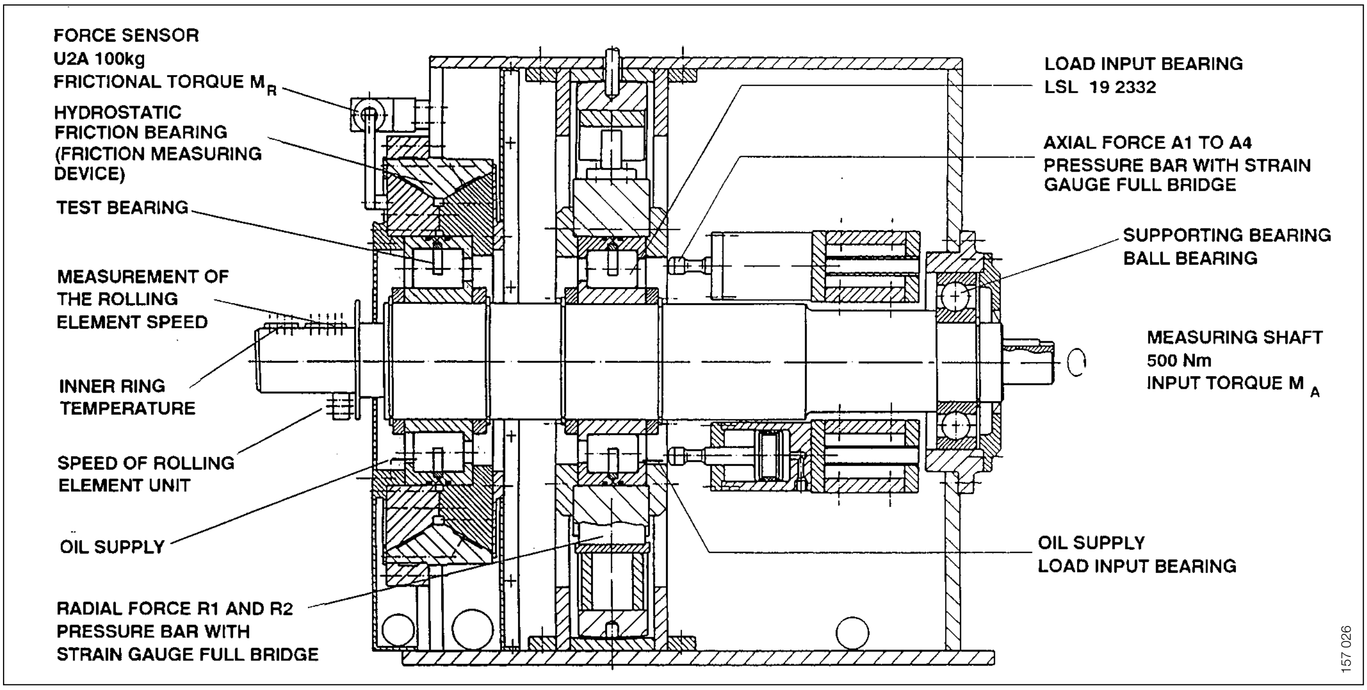

Figure 9 Heavy-duty bearing test rig

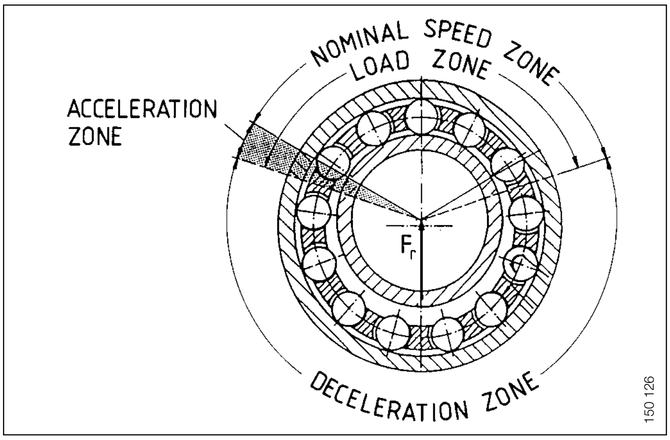

Figure 12 Kinematic areas in a bearing

The variable speed drive motor permits measurements and tests up to a speed of 4000 min^{-1}. This corresponds to a value over 200 % higher than the limiting speed given in the catalogue for a caged bearing of series NJ 23 32 E.

3.2 Running tests

A comprehensive series of tests on the friction and temperature behaviour as well as the kinematics of the test bearing have been carried out on the test rig shown in figure 9. The required kinematic revolutions are calculated according to [3].

A matrix of values from previous tests has been compiled starting with the highest load and the lowest speed as shown in figure 10.

Two versions of series 19 2332 bearings were provided for comparison purposes: one as a disc cage design, the other with spacers. A lubrication oil with a viscosity $\nu_{40} = 215\ mm^2 \cdot s^{-1}$, $\nu_{100} = 17{,}5\ mm^2 \cdot s^{-1}$ was used for lubrication. Figure 11 contains an overview of the acquisition and processing of the measured values and of the sensors and monitoring equipment used. All measured values have been monitored and stored by means of a computer so that a subsequent evaluation of the recorded measuring signals can be carried out.

The following measured values were continuously recorded to assess the operating behaviour of these bearings:

- Temperature at the inner and outer ring of the test bearing
- Temperature of oil at inlet and outlet
- Oil flow through the test bearing
- Bearing load
- Bearing frictional torque
- Size and direction of the load area
- Displacement of the shaft
- Structurally–borne noise from the rolling element when entering the load area
- Shaft speed
- Speed of the rolling element mid point
- Cage slip or rolling element slip
- Rotational velocity of an individual rolling element

The most important characteristics for assessing the operating behaviour of large bearings are provided by the frictional behaviour and the sensory analysis of these bearings.

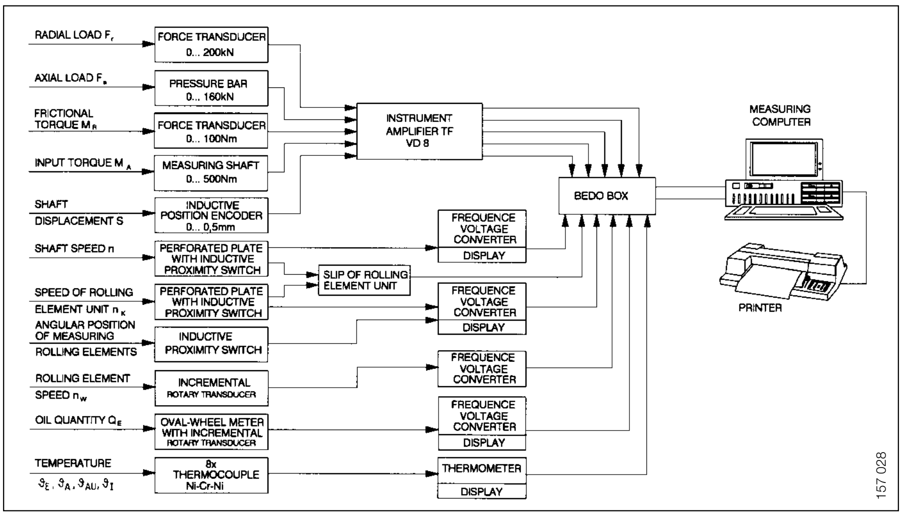

Figure 11 Measuring chain with sensory analysis and monitoring equipment

As described in 3.1, the test bearing is mounted in the hydraulic friction measuring balance. A load sensor is attached via a lever arm which is located on the movable part of the friction balance: the measured reaction force and the calibrated length of the lever arm length provide a scale for the frictional torque of the bearing.

Observations at the test rig have shown that particularly in case of large bearings the rolling elements do not always rotate with their kinematic nominal speed but are subjected alternately to accelerations and decelerations.

Figure 12 shows the various kinematic areas in a bearing.

The rolling element is provided with enough radial load to produce a kinematic gyratory movement around the axis of the rolling element. Only in the area of nominal speed in the deceleration area, the rolling element is slowed down due to the friction and is accelerated again to its kinematic nominal speed in the acceleration area by the frictional forces which occur. This acceleration and deceleration process is repeated periodically with each rotation. Furthermore, cage or rolling element slip can occur with lower external loads, i.e. the cage does not achieve its kinematic nominal speed either.

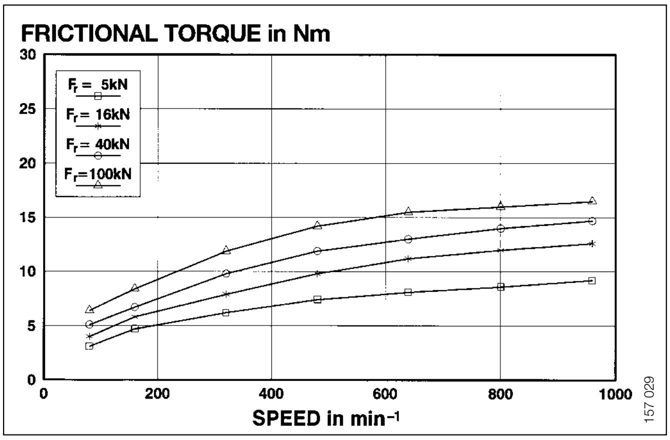

Figure 13 LSL 19 2332 frictional torque curves

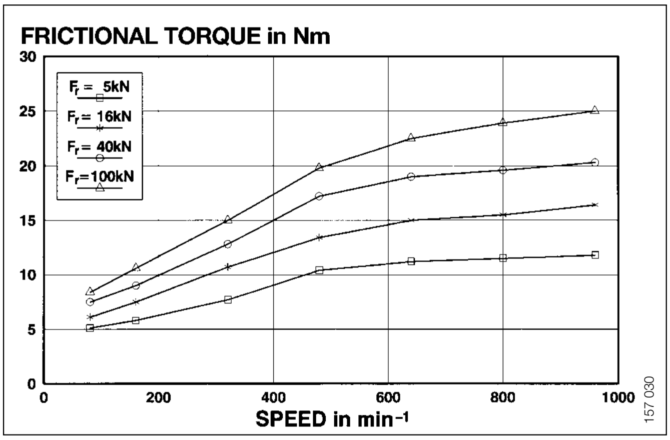

Figure 14 ZSL 19 2332 frictional torque curves

Determination of the self-rotation of the measuring rolling element is by means of an incremental shaft encoder which has a very low mass moment of inertia as well as a very low internal friction. It is connected coaxially to a rolling element via a universal joint. The incremental shaft encoder is mounted on a disc which is driven by another rolling element remote from the measuring rolling element and which rotates at the speed of the cage. This rotating disc is provided with holes and supplies a measure for the cage speed via inductive proximity switch scanning. The signal from the incremental shaft encoder was supplied by an F-U transducer; the transformed stress signal provides a measure for the current speed of the rolling element. Another hole on the rotating disc situated at the angular position of the measuring rolling element allows to locate the coning angle of the measuring rolling element to be created via a stationary inductive proximity switch.

3.3 Test results

As described in section 3.2 the operating behaviour of large bearings is essentially assessed from the frictional behaviour and bearing kinematics. The following deals in greater detail with the frictional behaviour of those new bearings.

3.3.1 Frictional behaviour

Figures 13, 14 and 16 show the charac-teristic frictional torque curves dependent on speed and load.

The graphs in figures 13, 14 and 16 show a significant frictional torque advantage for the disc caged bearing LSL 19 2332. Figure 15 clearly shows the frictional torque difference at operating point nr. 25 (F_r = 100 kN; n = 960 min^{-1}).

The progression over a period of time of the dynamic frictional torque allows con-clusions to be drawn regarding the ope-rating behaviour of the bearing. Figure 17 shows the comparison between the dynamic frictional torques at operating point nr. 25 (F_r = 100 kN; n = 960 min^{-1}).

The slight frictional torque fluctuations of the disc cage at a very low absolute frictional torque are to be taken into consideration.

These low fluctuations in the frictional torque lead to the assumption that there are also differences in the kinematics.

3.3.2 Kinematics

As already described in 3.2 the rollers are periodically accelerated and decelerated.

Figure 18 shows the speed characteris-tics of the measuring rolling element dependent on its relative position for the var-ious bearing types LSL (disc cage), ZSL (spacers), NJ (solid brass cage) and SL (full complement).

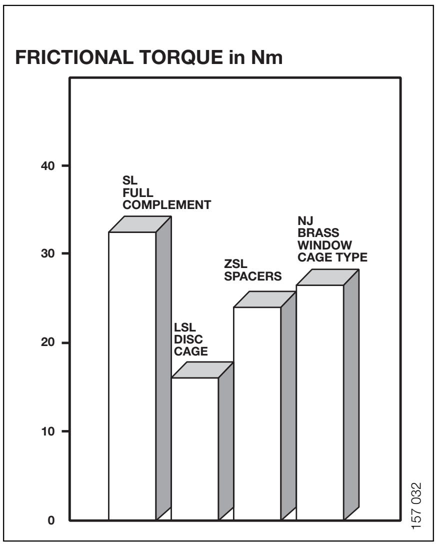

Figure 15 Frictional torque comparison of various bearings

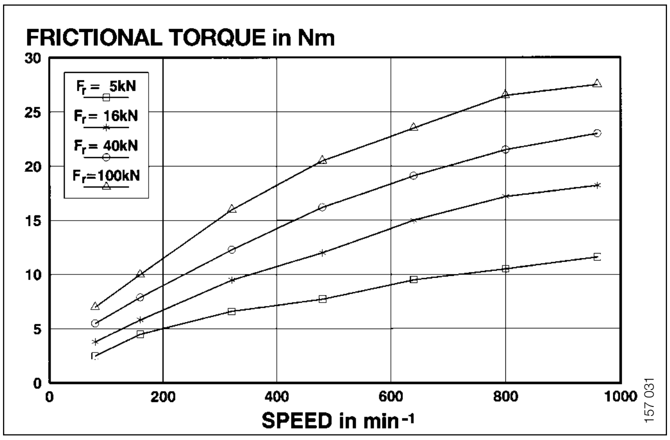

Figure 16 NJ 23 32 E frictional torque curves

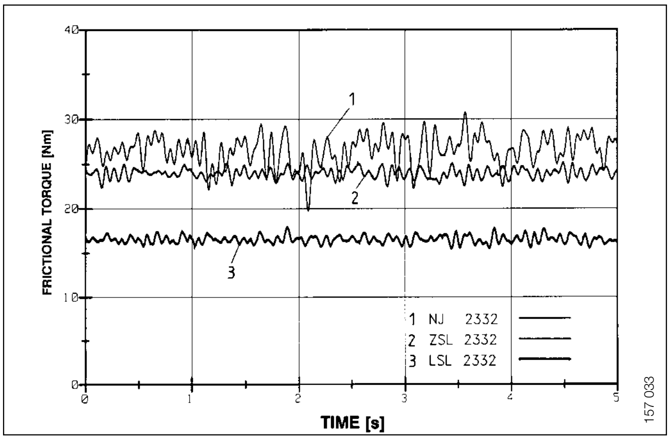

Figure 17 Dynamic frictional torque curves

Under extreme conditions the rolling element slip can reach 100% which means that the rolling element is decelerated to a standstill. The maximum rolling element slip is influenced, on the one hand, by the internal frictional conditions of the bearing at the roller-rib, roller-roller or roller-cage contact points and on the other hand, by lubricant friction. This influence was eliminated in the tests by using a lubricant with exactly the same viscosity and the same oil feed as was used for the test bearing. As a consequence, the internal frictional conditions and the cage design remain the only variables which have an influence on the maximum slip of the rolling element.

Figure 18 shows that very low rolling element slip values are achieved due to the design of the disc cage. The bearing with spacers also has low values for the rolling element slip compared to the convention-al cage (figure 18).

When the rolling elements enter the load area they are accelerated within a few milliseconds from a starting speed to their kinematic nominal speed. The energy which occurs during this acceleration must be transferred to the lubricant film. The tests have established that this abrupt acceleration process is a critical operational condition for smearing and is known as the so called smearing acceleration.

Figure 19 shows the comparison between the maximum accelerations at the test point matrix.

The low frictional torques and the very low values for the rolling element slip lead to very low rolling element accelerations and to a very favourable thermal behaviour in these bearings. An additional criterion for low frictional torques and optimum kinematics is the oil flow and therefore the heat dissipation. This has also been taken into consideration in the design of the disc cage and the spacers which provide an optimum solution due to the shape and the design.

As a result, the limiting speeds of these new bearings can be increased.

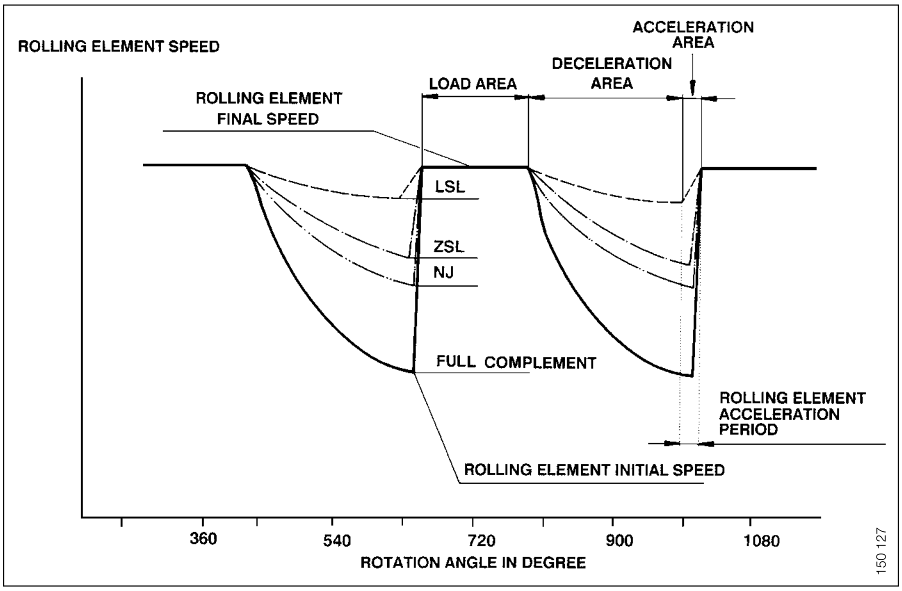

Figure 18 Speed curves of the measuring rolling elements

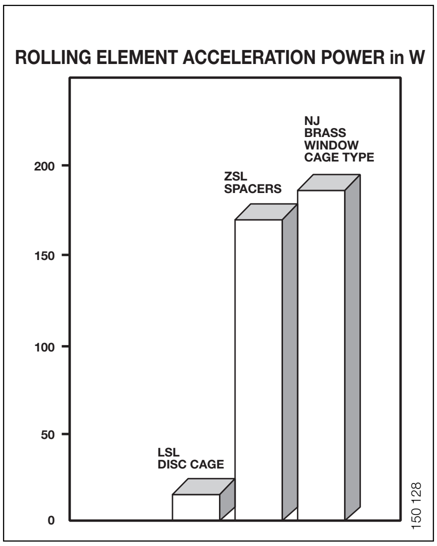

Figure 19 Comparison of the maximum rolling element accelerations

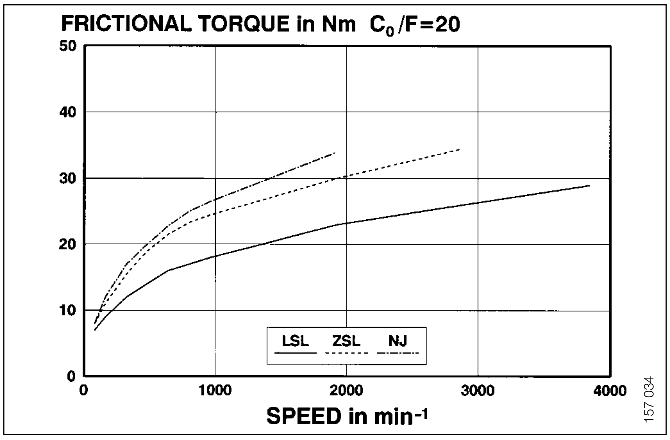

Figure 20 Frictional torque curves – Determining the limiting speed

3.3.3 Determining the limiting speeds

The limiting speeds of the new bearing systems have been established according to the test conditions for determining the limiting speed specified in step [4]. Figure 20 shows the measured frictional torques at $C_0/P = 20$ (the speeds were increased incrementally).

Figure 21 shows the comparison between the limiting speeds for bearings of series NJ 23 32.

Figure 22 shows the speed parameters for the established limiting speeds with respect to the average diameter of these bearings.

The speed parameter $n \cdot d_M$ for the full complement cylindrical roller bearing of 200 000 mm/min could be increased five-fold, which means an increase in the speed parameter of over 200 % compared with the brass window type cage. This speed increase can be achieved with bearings with normal accuracy (PN).

Bearing type	LSL 19 2332	ZSL 19 2332	NJ 2332 E	SL 19 2332
Cage type	Disc cage brass	Spacers composite material	Window type cage brass	Full complement
Rel. dyn. load rating (%)	121	121	100	129
Rel. stat. load rating (%)	110	110	100	120
Rel. life (%)	190	190	100	232
Rel. friction value (%)	75	95	100	150
Rel. speed factor (%)	210	147	100	42

Table 1 Comparison of the main sizes for bearing selection

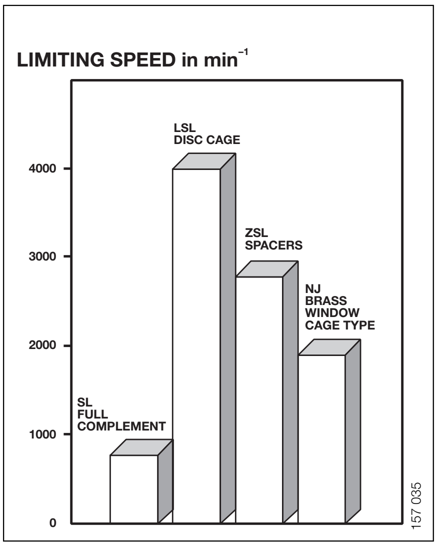

Figure 21 Limiting speed – Bearing series NJ 23 32

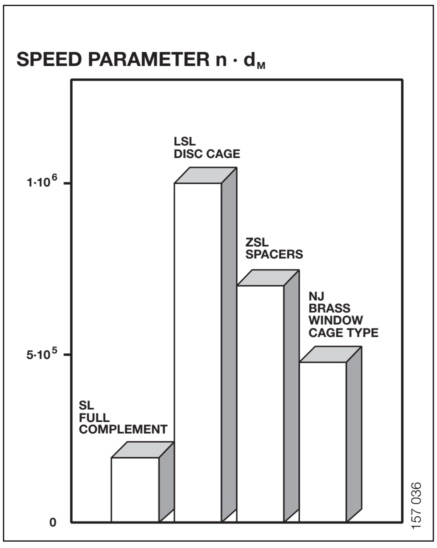

Figure 22 Speed parameter – Bearing series NJ 23 32

4 Summary

This article shows that the new cylindrical roller bearings developed by INA exceed by far the application potential of conventional radial cylindrical roller bearings.

In designing the bearing with spacers and the disc cage bearing INA has succeeded in combining the advantages of the extremely high load ratings of full complement bearings with the advantages of a caged bearing. Furthermore, the frictional torque and the rolling element slip can be minimized due to the shape of the disc cage and the spacers. The result is a very favourable dynamic frictional behaviour with minimal frictional torque fluctuations.

Table 1 shows a summary of the comparison of the different dimensions used for the evaluation of main sizes and bearing selections as shown for the bearing series 23 32.

The advantages of the cylindrical roller bearing with disc cage and of the cylindrical roller bearing with spacers can be summarized as follows:

- high static and dynamic load carrying capacity
- high limiting speed
- low frictional torque over the whole speed range
- quiet running due to optimum rolling element kinematics
- high protection against smearing
- high permissible axial load carrying capacity
- thermal stability due to optimum heat dissipation

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Sach-Nr. 002-540-010/RZL GB-D 08991 · Printed in Germany