Application of Ceramics

TO NU-TYPE CYLINDRICAL ROLLER BEARINGS FOR MACHINE TOOL MAIN SPINDLES

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Management Summary

Ultra-high-speed operation of air-oil lubricated NU-type cylindrical roller bearings has been made possible by using a ceramic inner ring. A maximum speed of up to 35,000 min^{-1} is possible ($d_m n$ value of 3.25 million; inner ring bore of 70 mm). Devising the outer ring rib structure to streamline lubricant drainage resolves the occurrence of high and broad temperature rises around the mid-speed range, which is typical of conventional NU-type cylindrical roller bearings, as well as rapid temperature rises at high shaft speeds. The developed bearing will allow the practical application of NU-type cylindrical roller bearings to machine tools that require high bearing stiffness over a wide range of operation speeds. The cage made of PEEK (polyether-ether ketone) is guided on the air-oil-nozzle outside surfaces, while rollers made of steel can be used even at 35,000 min^{-1} and control the inner ring temperature below 70° C.

Introduction

Bearings used to support main spindles on machine tools must be capable of higher speed and greater rigidity. This is true in that any main spindle that turns together with a tool or work piece mounted onto it is one of the critical machine tool components that directly affects machining efficiency and accuracy of the machine tool; and, the bearings that support the main spindle are the most critical machine elements on the machine tool (Ref. 1). Other mechanical characteristics any main spindle bearing needs to satisfy include higher bearing accuracy, lower vibration and lower noise. Rolling bearings are most often used to support main spindles because they satisfy various requirements, including cost effectiveness and maintainability of balance compared with hydrodynamic (static or dynamic pressure) bearings and magnetic bearings.

Typical rolling-bearing types used to support machine tool main spindles are angular contact ball bearings, cylindrical roller bearings and tapered roller bearings. In particular, cylindrical roller bearings are preferred as non-locating bearings because they boast higher load capacity and greater rigidity in the radial direction, and their inner and outer rings are capable of moving in the axial direction relative to the main spindle. Since requirements appear to be increasing for higher speed with the fixed position preload bearing system (which features greater rigidity) for rolling bearings on machine tool main spindles, capability for much higher speed will be needed for rear-position (that is, free-side) single-row cylindrical roller bearings.

To address this challenge, we attempted to use ceramic inner rings (this topic will be described in detail later) to prevent occurrence of excessive preload that will pose a direct obstacle to achievement of higher main spindle bearing speed. In a previous paper (Ref. 2), we reported our experience in developing the N-type cylindrical roller bearings series having a ceramic inner ring (featuring double-rib inner ring); this cylindrical roller bearing type, lubricated with an air-oil lubrication system, achieved ultra-high-speed bearing operation as fast as $d_m n$ (bearing-pitch-diameter mm × inner-ring-running-speed min^{-1}) value = 3.25×10^6. This speed level is equivalent to that obtained from not-yet- mounted, ultra-high-speed, constant-pressure preloaded angular contact ball bearings (Ref. 3). However, the N-type is uniquely structured in that its ceramic inner ring is tightly fitted with steel spacer rings also serving as ribs: therefore, a simpler-structured ceramic inner ring is needed to simplify formation and mounting of the inner ring.

To address this challenge we have developed NU-type cylindrical roller bearings (featuring double-rib outer ring) that have ceramic—but no spacer—rings, and achieved ultra-high-speed bearing operation as fast as $d_m n$ value = 3.25×10^6 with air-oil lubrication. This article reports the performance of this new engineering development.

Use of ceramic materials in elements of rolling bearings has long been proposed (Ref. 4). In the technical field of machine tools, there have been an increasing number of cases (Ref. 6) where ceramic rolling elements are used in angular contact ball bearings in order to inhibit adverse effects of gyro-moment (Ref. 5) that poses particular problems for machine tool main spindles running at higher speeds. However, there have been a limited number of applications of ceramic materials to cylindrical roller bearings for machine tools. In addition to the information already presented in NTN Technical Review No. 76 (Ref. 2), we provide here additional information in order to demonstrate that by utilizing the benefits of ceramic materials, cylindrical roller bearings can offer high-speed performance comparable to that of constant-pressure, pre-load, angular-contact ball bearings.

Structure and Elemental Technologies for High-Speed Operation

Figure 1 shows a cross-sectional view of NTN's newly developed NU-type cylindrical roller bearing.

The structure of the NU-type cylindrical roller bearing (Fig. 1) is characterized as follows: the inner ring is made

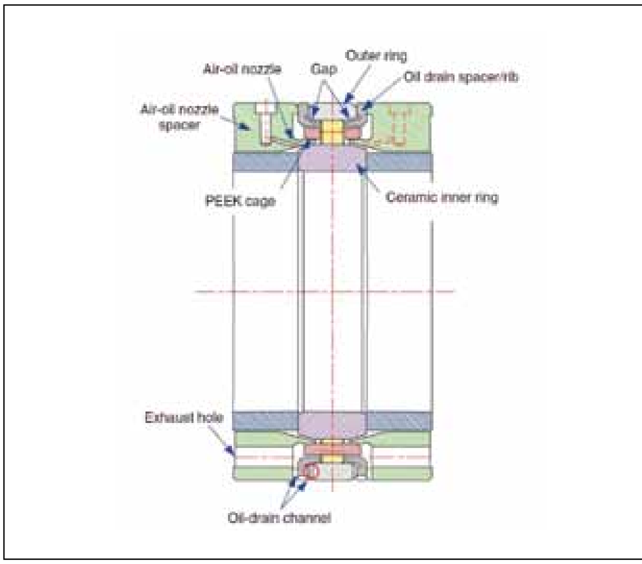

Figure 1—Developed NU-type cylindrical roller bearing.

of silicon nitride (Si_3N_4)—astructural ceramic material; the cage is made of PEEK ; the bore surface of the cage rides on the outer circumferential surface of the air-oil nozzle spacer; the rollers and outer ring are made of common bearing steel (SUJ2); the outer ring is fitted with oil drain spacers that double as ribs; and lubricating oil is drained via the gaps (marked with a red circle) between outer ring and oil-drain spacer/rig.

One characteristic that any machine tool main spindle bearing needs to satisfy is avoidance of excessive preload. On a cylindrical roller bearing, the inner ring and outer ring freely move relative to each other in the axial direction, so no axial preload occurs. On the other hand, a problem can occur in the radial direction in that the inner ring expands—owing to heat build-up and greater centrifugal force resulting, in particular, from high-speed bearing operation—leading to radial over pre-load. Heat build-up then increases between the rollers and raceway surface and the resultant, rapid temperature rise can potentially lead to bearing failure. In machine tool main spindles, jacket cooling is typically provided on the outer surface side of the outer ring—which is a stationary body—in order to prevent heat generation on the main spindle system from adversely affecting the entire machine tool. Temperature on the inner ring side will soon rise due to heat generation on the bearing and built-in motor, and, due as well to a structure that does not readily release heat. Consequently, a steep heat gradient occurs across the inner and outer ring, and pre-load on the bearing at higher speed can be excessively taxing. Therefore, problem-free, high-speed bearings operation is possible through reduction of heat generation inside the bearing and thermal expansion of the bearing.

Based on the abovementioned assumption, the elemental technologies for the elements inside the bearing that allow higher speed operation are now described.

First, the physical properties of ceramic material (silicon nitride) are compared with those of steel for the inner ring

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(Table 1.) A low linear expansion coefficient of the ceramic material—including 30% steel material—effectively limits thermal expansion of the inner ring. Despite the fact that the physical density of this ceramic material is as low as 40% compared to that of the steel material, the modulus of longitudinal elasticity with the ceramic material is 150% as great as the steel material. Yet, at the same time, the difference in Poisson's ratio between these two materials is very small. Consequently, the centrifugal expansion on the inner ring is limited to approximately 30%. More specifically, compared with the steel inner ring, an increase in preload is reduced with the ceramic inner ring and heat generation inside the bearing is more efficiently prevented.

As previously reported (Ref. 2), it is important to note that when using a cage riding system, controlled-temperature lubrication is always supplied to the cage lands and is promptly drained away to prevent the lubricating oil (which has become very hot from shear heat generation) from remaining on the guide surface; this arrangement helps inhibit heat build-up in the bearing. As shown in Figure 1, the lubricating oil ejected from the air-oil nozzle—with compressed air—hits the ramp of the rotating inner ring, rises along the ramp via surface tension and centrifugal force, and lubricates the rollers and raceway surface. At the same time, the compressed, air-propelled lubricating oil passes the cage riding clearance from the inside of the bearing and is drained away. In other words, fresh lubricating oil is always supplied

Table 1-Properties of Si_3N_4 and steel

	Si_3N_4	Steel
Linear expansion coefficient 1/K	3.2×10^{-6}	11×10^{-6}
Density kg/m³	3.2×10^3	7.8×10^3
Modulus of longitudinal elasticity GPa	314	211
Poisson's ratio	0.26	0.3

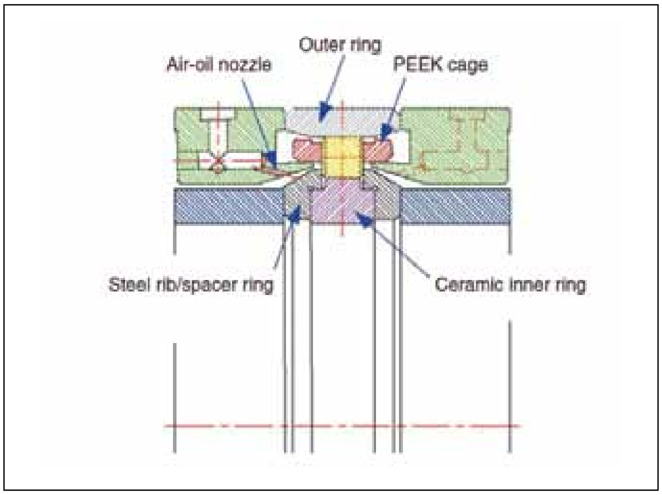

Figure 2—N-type cylindrical roller bearing with ceramic inner ring.

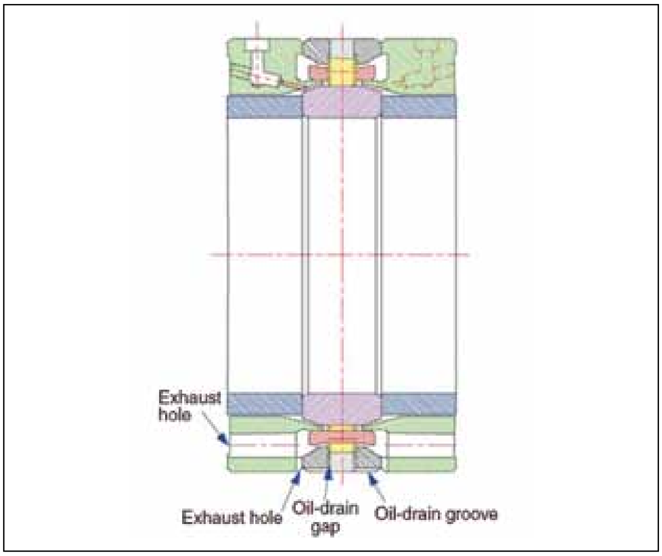

Figure 4—Oil-drain/groove-structure, NU-type cylindrical roller bearing.

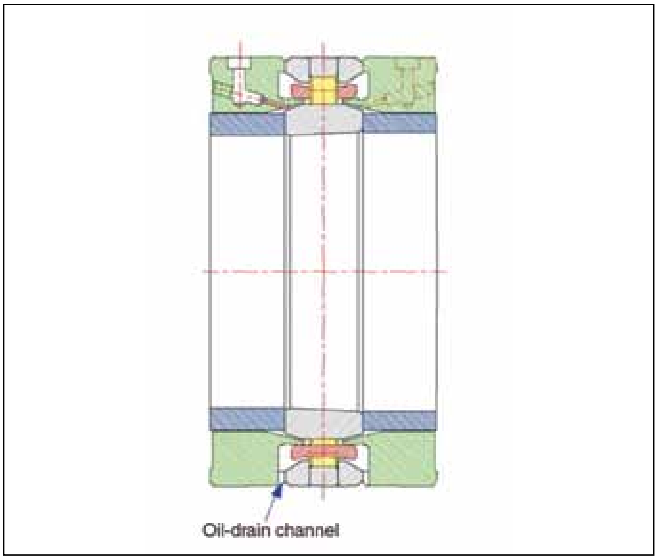

Figure 3—Standard-structure, NU-type cylindrical roller bearing.

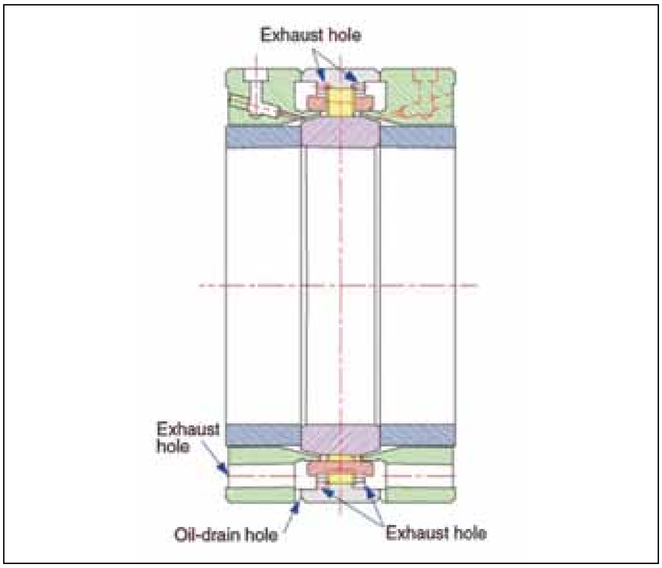

Figure 5—Oil-drain/hole-structure, NU-type cylindrical roller bearing.

to the cage lands and then promptly drained away from the bearing.

Another unique arrangement has been incorporated into the oil-drain structure in the outer ring side. When the bearing is running at a much higher speed, the fresh lubricating oil supplied to the bearing tends to remain in the vicinity of the outer ring bore due to the impact of centrifugal force. An excessive amount of lubricating oil remaining in and around the bore of the outer ring will lead to an increase in stir resistance and, as a result, heat generation inside the bearing. Compared with the N-type bearings, this tendency is more apparent with the NU-type cylindrical roller bearings, whose outer ring includes ribs. Therefore, for trouble-free, high-speed bearing operation, the oil-drain structure on the outer ring side requires a special solution.

For the purpose of comparison, the N-type cylindrical roller bearing (Ref. 2) is illustrated in Figure 2. The structure of this bearing is characterized by a ceramic inner ring fitted with spacer rings on both ends. The ceramic inner ring is interference-fit onto the shaft and the ring spacers slip-fit onto the shaft. In order to replace the steel inner ring on the N-type cylindrical roller bearing with a ceramic version, it is necessary to use a ring with integrated ribs or separate ribs. Compared with the NU-type bearing in Figure 1, either ceramic inner ring variant described above complicates the inner ring design; this "ceramic-based solution" poses a drawback of significantly increased machining costs, as ceramic material machining cost is much higher compared with steel. But to the advantage of N-type bearings, the outer ring does not tend to remain in the bearing; note that the cage riding system and material of the cage in the N-type bearings are essentially identical to those of NTN's newly developed NU-type bearings (Fig. 1).

Oil-Draining Capability and High-Speed Running Performance of NU-Type Cylindrical Roller Bearings

Encouraged by the oil-draining performance of the NU-type cylindrical roller bearings, we have developed various bearing prototypes. In this section we will describe the result of our investigation into temperature-dependent characteristics of these prototypes being run at various speeds. These prototypes are essentially NU-type cylindrical roller bearings, categorized as: "standard oil-drain structure variant," "oil-drain groove variant" and "oil-drain hole variant." Their structures are schematically illustrated in Figures 3–5.

In the standard oil-drain structure variant (Fig. 3), the inner ring is made of steel (SUJ2) while the rollers are made of ceramic material. The outer ring rib is independent of the outer ring and the PEEK cage is the nozzle outer surface riding type (Figs. 1–2); however, the oil-drain structure on the outer ring side is the standard type.

This oil-drain groove variant (Fig. 4) is unique in that a provided *separate* outer ring rib facilitates the delivery of fresh lubricating oil through the gaps on both ends of rollers as well as the gap between the outer ring and rib toward the outer surface side of outer ring; the heated lubricating oil is drained away from the bearing through the groove formed on the outer surface of the outer ring.

Regarding the oil-drain hole variant (Fig. 5), outer ring ribs on both sides—each with six equally spaced oil-drain holes toward their outer circumference to direct the heated lubricating oil to the outside of the bearing—shift the phases of both outer ring ribs with each other so that the locations of the oil-drain holes on one outer ring rib are not directly opposite the oil-drain holes on the other outer ring rib.

Note that the oil-drain groove variant (Fig. 4) and oil-drain hole variant (Fig. 5) both have inner rings and rollers made of ceramic material; the PEEK cage is riding on the outer circumferential surface of the air-oil nozzle.

Major technical data for these test cylindrical roller bearings and test conditions are summarized in Table 2, while the cross-sectional view of the spindle test rig used throughout our present development work is illustrated in Figure 6. The test results obtained from the NU-type cylindrical roller bearings of Figures 3–5 are illustrated in Figure 7.

From the graphs displayed (Fig. 7) it should be understood that apparent temperature peaks occur at around 10,000 min^{-1} with all designs—i.e., "standard oil-drain structure variant," "oil-drain groove variant" and "oil-drain hole variant."

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Table 2—Test bearings (Figs. 3–5) and conditions associated with Fig. 7

Standard oil-drain structure variant	Cross-sectional plan	Fig. 3
	Size	Φ70 x Φ110 x 20
	Pitch diameter	93 mm
	Inner ring	SUJ2 (tapered hole: 1/12 bore diameter)
	Outer ring	SUJ2
	Rollers	Si_3N_4, Φ7 x 7, 22 pcs.
	Cage	PEEK+CF30%, Nozzle outer surface riding
Oil-drain groove variant	Cross-sectional plan	Fig. 4
	Size	Φ70 x Φ110 x 20
	Pitch diameter	93 mm
	Inner ring	Si_3N (cylindrical bore)
	Outer ring	SUJ2
	Rollers	Si_3N_4, Φ7 x 7, 22 pcs.
	Cage	PEEK+CF30%, Nozzle outer surface riding
Oil-drain hole variant	Cross-sectional plan	Fig. 5
	Size	Φ70 x Φ110 x 20
	Pitch diameter	93 mm
	Inner ring	Si_3N_4 (cylindrical bore)
	Outer ring	SUJ2
	Rollers	Si_3N_4, Φ7 x 7, 22 pcs.
	Cage	PEEK+CF30%, Nozzle outer surface riding
Test conditions	Initial radial clearance	-3 – -4 μm
	Bearing lubrication	Air-oil ISO VG32 Oil is supplied from both sides of bearing. 0.01cm^3/10 min x 2
	Jacket cooling temperature	Room temperature ± 1° C

This is the major reason why NU-type bearings have not yet been used as air-oil-lubricated cylindrical roller bearings for machine tools. Therefore, the challenges for the present development work were to increase the maximum allowable bearing speed and provide a bearing that can maintain its rigidity in a wider speed range without developing heat build-up—all in an economically viable design. Our original objective for prototyping the oil-drain groove and oil-drain hole variants was to improve oil-draining performance at higher bearing speeds. Though the maximum-allowable bearing speed with the oil-drain groove variant reached 35,000 min^{-1}, the temperature peak in the medium-speed range at around 10,000 min^{-1} still persists with either variant. Note that the test for the standard oil-drain structure variant and oil-drain hole was suspended because of sudden temperature rise.

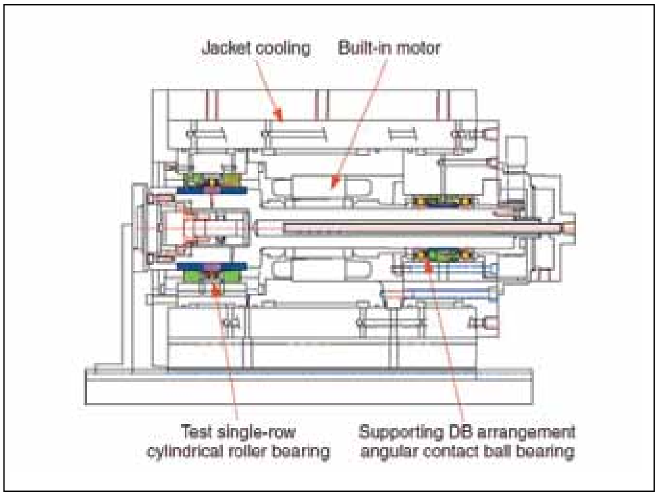

Figure 6—Section view of spindle test rig.

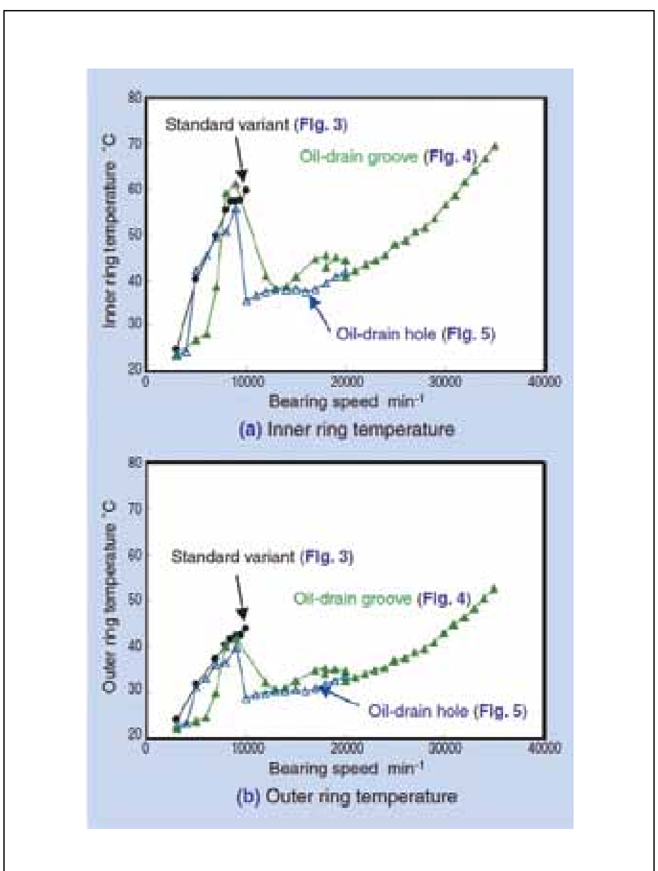

Figure 7—Inner and outer ring temperatures vs. rotational speed.

To find a solution we first assumed the cause for this temperature peak to be poor oil-draining performance and attempted to verify this assumption. Figure 8 includes two sets of data obtained from two cases of bearing operation (Fig. 4); one case corresponds with a scenario where a sufficient amount of air-oil mixture was supplied to the bearing prior to operation; in the other scenario, no air-oil mixture was supplied to the bearing prior to operation.

In the two cases in Figure 8, the test bearings were quickly accelerated to 13,000 min^{-1} while being lubricated with an oil-air flow rate of 0.01 cm^3/10 min × 2. In the scenario in Figure 8-a, air-oil mixture was supplied to the test bearing for 90 minutes prior to the test operation; in the scenario in Figure 8-b, no air-oil mixture was supplied to the test bearing prior to start of the test operation. When comparing the data in scenario "a" with that of scenario "b," the heat rise on the inner ring with scenario "a" is approximately as much as 30° C higher and on the outer ring is approximately as much as 15° C higher. From these findings we suspect the cause of the temperature peak at the medium-speed range is "residual heated lubricating oil" remaining in the bearing. Therefore, efficient oil-draining capability in both high-speed and medium-speed ranges is needed. Also, since such temperature peaks do not occur with the N-type bearings, we feel we should consider oil-draining behavior at the outer ring rib; we believe this approach will lead to improved oil-draining capability on the outer ring side of a bearing running at higher speed.

With these findings we have continued review and prototyping activities, resulting in the NU-type cylindrical roller bearing structure shown in Figure 1; test conditions associated with these activities are summarized in Table 3. Test results from the NU-type bearing and those from the N-type bearing in Figure 2 (Ref. 2) are shown (Fig. 9). Figure 9-a shows the test data from bearing samples with ceramic rollers, and Figure 9-b the test data from bearing samples with steel rollers.

As is shown (Fig. 9-a), the samples of the newly developed NU-type—as well as those of the N-type—do not show a temperature peak in the medium-speed range of around 10,000 min^{-1} and exhibit smooth temperature rise curves up to the targeted maximum running speed of 35,000 min^{-1} ($d_m n$ value = 3.25 × 10^6). Compared with the N-type, the inner ring temperature on the NU-type at 35,000 min^{-1} is 2° C lower. Also, as apparent from the test data of the NU-type samples in the data (Fig. 9-b), there is no temperature peak at around 10,000 min^{-1} and the temperature slowly increases to 35,000 min^{-1}. The inner ring temperature of 35,000 min^{-1} reads 70° C, which is 4° C lower compared with the N-type—a very favorable achievement.

These findings verified that the bearing configuration illustrated in Figure 1 (NU-type)—ceramic inner ring (no spacer rings), cage riding the outside surface of air-oil nozzle, and the outer ring having oil-drain gaps—does not exhibit temperature peak in the medium-speed range and can be run at ultra-high speeds without developing sudden temperature rise. With a variant of this bearing configuration—using steel rollers rather than ceramic—the inner ring temperature at 35,000 min^{-1} is limited to 70° C—the maximum-allowable temperature for commercially acceptable bearing operation. Compared with the N-type (Fig. 2; Ref. 2), the NU-type has the simpler-shaped ceramic inner ring as well as steel rollers, allowing this bearing to be offered at a commercially acceptable price.

It is apparent that use of the ceramic inner ring has helped achieve problem-free, high-speed operation of the bearing. However, use of ceramic material in a bearing leads to additional benefits. When a bearing having a steel inner ring is run at a higher speed, bearing fit to the shaft can loosen, owing to expansion of the inner ring bore due to heat and centrifugal force occurring from bearing operation. To prevent loosening of the bearing relative to the shaft, a rolling bearing of bore diameter 50 to 100 mm, which is often used to support a machine tool main spindle, is interference-fitted over the main spindle with an interference allowance of 30 µm or greater. Consequently, there will be difficulty when mounting the bearing by press-fitting. In contrast, use of a ceramic inner ring—with limited expansion from heat and centrifugal force—will enable the bearing to be interference-fitted over the shaft with an interference allowance of 5 µm or smaller—leading to much easier bearing mounting.

Furthermore, the ceramic material's greater modulus of longitudinal elasticity helps enhance the rigidity of the bearing, which is discussed in the following section.

Improvement in Bearing Rigidity with Ceramic Material

You'll remember that earlier in this article we stated that any cylindrical roller bearing for machine tool main spindle must be capable of not only higher speed, but also greater rigidity. Through calculation we have verified the effect of using ceramic material for improving bearing rigidity.

The internal clearance of a cylindrical roller bearing of bore diameter 70 mm was selected as zero. Combining a steel or ceramic inner ring with steel or ceramic rollers, we prepared various cylindrical bearing samples. To simulate operation of the bearing sample on an actual machine tool, the maximum radial load applied to the bearing has been set to 7 kN.

The relation between radial load and displacement of the bearing center is graphically plotted (Fig. 10). The result of bearing rigidity in the 3-to-7 kN region that features good linearity within the radial load vs. bearing center displacement is shown in Table 4. Improvement in rigidity with the sample using ceramic material only for the inner ring is 7%,

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and with the sample using a ceramic material for the rollers only is 19%. Use of ceramic rollers leads to greater improvement in bearing rigidity since roller to inner ring rigidity and roller to outer ring rigidity are simultaneously improved. The use of ceramic material for the inner ring exclusively results

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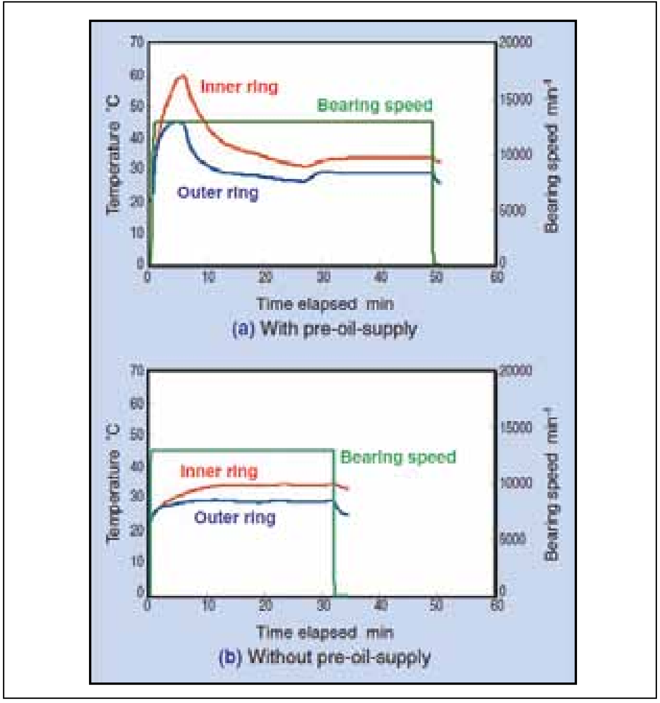

Figure 8—Pre-oil supply and temperature rise.

Table 3—Test bearings (Figs. 1–2) and conditions associated with Fig. 9

Newly developed NU type	Cross-sectional plan	Fig. 1
	Size	Φ70 x Φ110 x 20
	Pitch diameter	93 mm
	Inner ring	Si_3N_4 (w/o steel spacer rings)
	Outer ring	SUJ2
	Rollers	Steel or $SiaN_4$, dia. Φ7 x 7, 22 pcs.
	Cage	PEEK+CF30%, Nozzle outer surface riding
	Fit between shaft and inner ring	5 µm, interference-fit
N-type with ceramic inner ring	Cross-sectional plan	Fig. 2
	Size	Φ70 x Φ110 x 20
	Pitch diameter	93 mm
	Inner ring	Si_3N_4 (w/ steel spacer rings)
	Outer ring	SUJ2
	Rollers	Steel or $SiaN_4$, dia. Φ7 x 7, 22 pcs.
	Cage	PEEK+CF30%, Nozzle outer surface riding
	Fit between shaft and inner ring	2 µm, interference-fit
Test conditions	Initial radial clearance	0 – 3 µm
	Bearing lubrication	Air-oil ISO VG32 Oil is supplied from both sides of bearing.
	NU type	0.01cm³/10 min x 2 0.01cm³/6min x 2 (steel rollers)
	N type	0.01cm³/10min x 2 (Si_3N_4 rollers) 0.01cm³/5min x 2 (steel rollers)
	Jacket cooling temperature	Room temperature ± 1° C

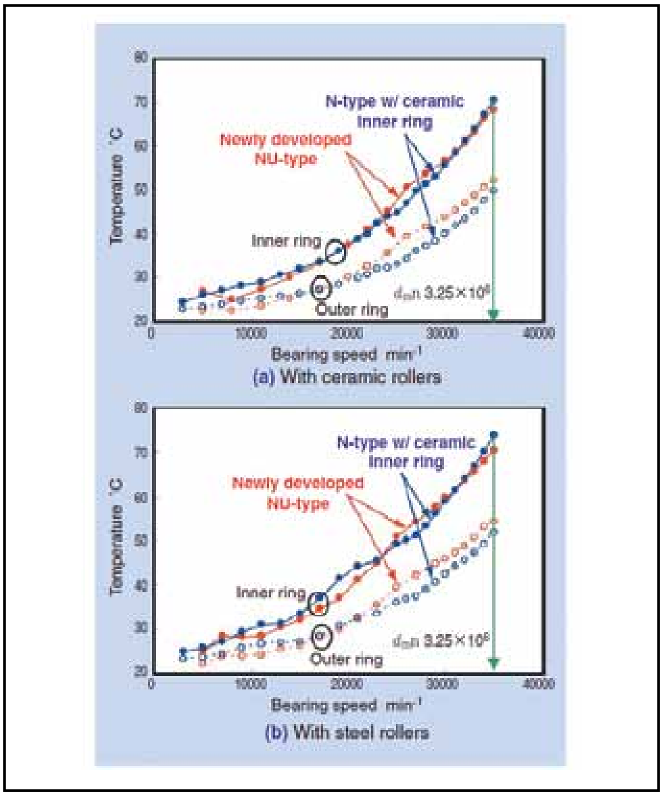

Figure 9—Inner and outer ring temperatures vs. rotational speed.

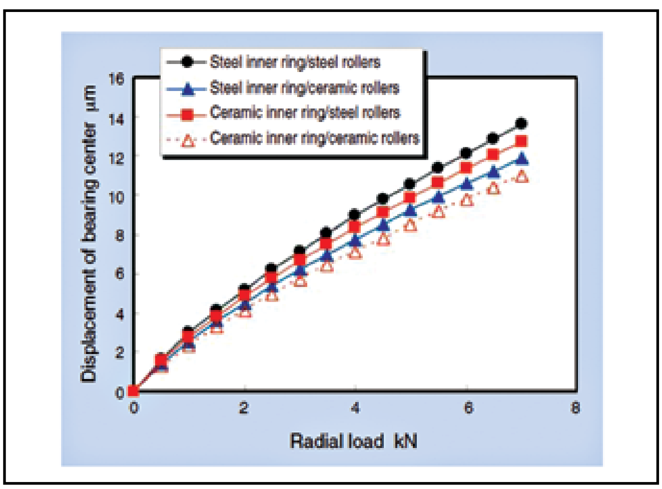

Figure 10—Radial load vs. calculated bearing deflection.

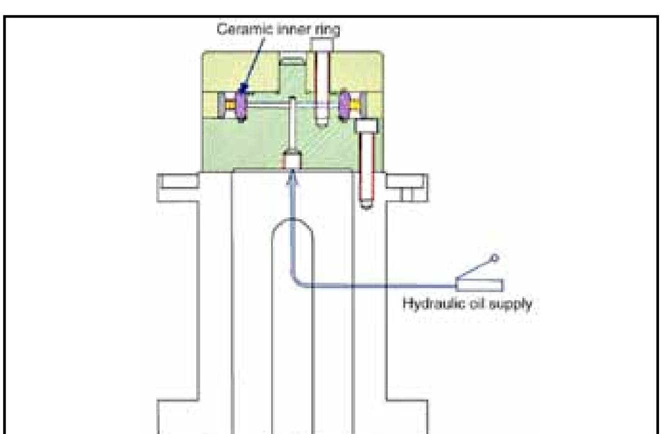

Figure 11—Hydraulic-loading test rig.

in relatively small improvement in bearing rigidity; however, this arrangement greatly contributes to improvement in high-speed performance of the bearing.

Verification of Mechanical Strength of Ceramic Inner Ring

When ceramic material is used for a bearing inner ring, it is necessary to prove that the inner ring has sufficient mechanical strength against hoop stress occurring from both heat generation and centrifugal expansion. In the axi-symmetric deformation mode, no shear stress occurs; therefore, the major stresses involved are circumferential hoop stress, axial stress and radial stress. Whereas on a thin-walled cylinder subjected to internal pressure and centrifugal force, the stress with the greatest impact is hoop stress.

Figure 11 illustrates the hydraulic loading test rig we have used to test the mechanical strength of the ceramic inner ring. The test piece used is the same inner ring used in the NU-type cylindrical roller bearing (Fig. 1). High-pressure hydraulic oil supplied from an outside hydraulic pump is uniformly distributed within the bore of the inner ring, and either an inner ring alone or an inner ring in a bearing assembly (complete with rollers and outer ring) can be tested. The test result is illustrated in Figure 12, with the hydraulic pressure applied to the bore surface of the inner ring on the X-axis, and the corresponding hoop stress occurring on the bore surface of the inner ring on the Y-axis. The inner ring has developed fracture at a hoop stress of 500 MPa when tested alone, and at a hoop stress of 640 MPa when tested in the bearing assembly. These hoop stress values are approximately three and four times greater than maximum commercially allowable hoop stress (160 MPa) for inner ring in typical cylindrical roller bearings for machine tool main spindles. When assembled with the rest of the bearing, a compressive stress is applied to the inner ring in a direction that helps the compressive stress overcome the hoop stress; therefore the inner ring in this configuration can withstand greater internal pressure than the inner ring alone can withstand.

Thus, we have proven the functionality and mechanical strength of our newly developed NU-type cylindrical roller bearing (Fig. 1). Typical photographic views of this bearing type are given in Figure 13.

Conclusion

To enhance high-speed capability of its NU-type cylindrical roller bearing for machine tool main spindle, NTN has introduced the following elemental technologies:

- Ceramic (silicon nitride) inner ring
- Cage riding on the outer surface of the air-oil nozzle
- Oil drain structure with separate outer ring rib

Ceramic materials boast a low linear expansion coefficient, low density and high modulus of longitudinal elasticity. Thanks to these features, the ceramic inner ring can resist over-preload that can result from expansion of the inner ring while the bearing is running at a greater speed. This improvement helps mitigate heat build-up within the bearing, which is the biggest obstacle to problem-free, high-speed bearings operation.

Incidentally, stagnant lubricating oil within a bearing can lead to higher bearing temperatures due to shear-heat-generation of the lubricating oil. To address this problem we improved the guide surface on the cage to promote draining of oil from its slide-way; at the same time, we introduced a draining structure independent of the outer ring rib in order to promote draining of oil from an area around the outer ring rib.

By adoption of the abovementioned elemental technologies, NTN has successfully developed an improved variant of NU–type cylindrical roller bearings that boasts an ultra-high-speed range that equates to $d_m n$ value = 3.25×10^6 (bore diameter 70 mm; bearing speed 35,000 min^{-1}). At the same time, we analyzed the mechanical strength of the ceramic inner ring and have determined that this inner ring has mechanical strength sufficient for commercial use of the new NU-type bearing.

Higher functionality and improved reliability of bearings directly contribute to better performance of machine tools and pose not-yet-solved challenges for bearing manufacturers. NTN will remain committed to further sophistication of its bearing technologies.

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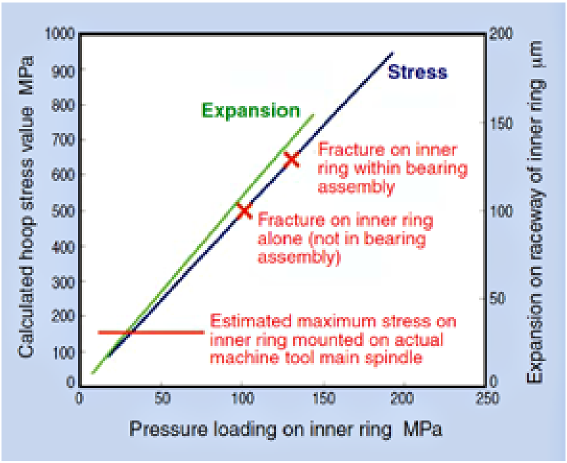

Figure 12—Inner loading pressure vs. hoop stress and inner ring expansion.

Figure 13—Developed, NU-type cylindrical roller bearing.

Table 4—Bearing stiffness improvement due to ceramic elements

Inner ring/rollers	Rigidity N/m	Increase %
Steel/steel	6.23×10^8	0
Si_3N_4/steel	6.64×10^8	+7
Steel/Si_3N_4	7.43×10^8	+19
Si_3N_4/Si_3N_4	8.09×10^8	+30