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Bearings for Aircraft Gas Turbine Engines (part 1)

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1. Introduction

Growth in aircraft manufacturing has been very strong in recent years and forecasts suggest that this will continue into the millennium. Programs include the development of long-range wide-body aircraft such as the Airbus Industrie's A340-500/600 model (to start service in 2000), the Airbus A3XX with 550 and 650 seats, and Boeing's ultra-long-range 777-200/300X derivative. The production of regional aircraft such as the Boeing 717 and Airbus A319M5 is also increasing.

The propulsion system of these aircraft typically has two or three rotating shafts in order to meet the aerodynamics requirements of the engine. The bearings that support these shafts are subject to very demanding operating conditions such as high axial loads.

For application in advanced aircraft gas turbine engines with a wide-chord fan, the main-shaft bearing is expected to have higher load capacity and higher-speed performance due to the need for a higher thrust-to-weight ratio. Moreover, higher reliability of the main-shaft bearing is required (e.g., frequency of failure: less than

0.001 per 1000 hours operation) to meet demand for increased engine reliability, particularly as ETOPS (Extended-Range Twin-Engine Operations) certification becomes more necessary for increasing the efficiency of aircraft operation.

This report outlines the design and operating requirements of bearings for advanced aircraft gas turbine engines, and discusses technical solutions developed to meet these requirements.

2. Bearing Design Requirements

Corresponding to the above requirements for aircraft gas turbine engines, bearings are increasingly required to survive with little or no oil under severe operating conditions such as high speed and high load. As higher bearing rotational speeds result in higher hoop stress within the rotating ring (inner ring and/or outer ring of a bearing), the bearing failure mode tends to gravitate toward ring fracture rather than rolling contact fatigue. As countermeasures, material with high fracture toughness such as M50NiL has been used and materials with similar

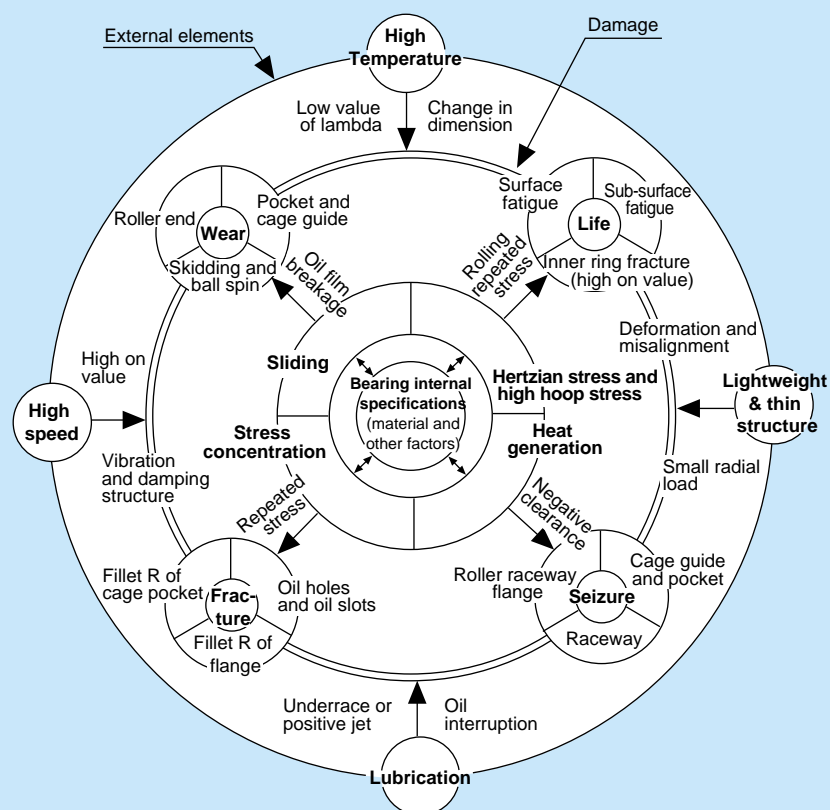
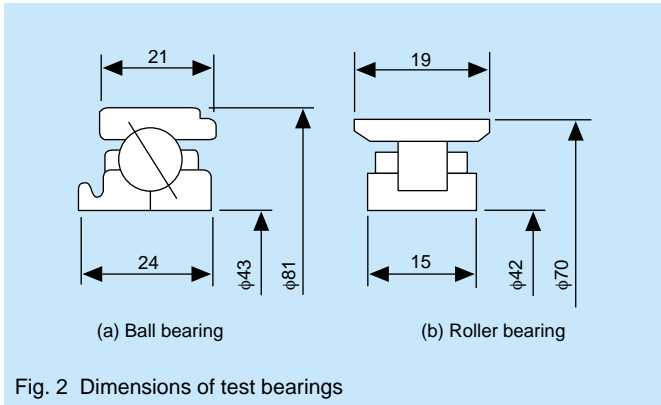


Fig. 1 Basic design concepts and influencing factors



fracture toughness but better corrosion resistance like PYROWEAR675 and CSS-42L are being developed. On the other hand, weight reduction and the load cycle of engine operation means that bearings sometimes operate at high speed and low load resulting in damage or failure due to skidding of the rolling elements. Hence the prevention of failure due to skidding has become an important issue.

Key performance parameters in the design of rolling bearings for aircraft gas turbine engines are:

- Resistance to seizure
- Resistance to wear
- Longer life
- No fracturing
- Component integration

The relation between a bearing's design and external elements, which influence the above bearing performance parameters, is shown in Fig. 1.

From Fig. 1, some important bearing parameters to consider are:

- Stress concentration
- Heat generation
- Excessive slip
- Hertzian stress and hoop stress

Therefore, in order to create the best bearing design for a given position in the engine—including appropriate materials, cage design and bearing internal design—a

detailed examination of all the bearing operating conditions is required.

3. Bearing Operating Requirements

3.1 Improved resistance to seizure

In some engine applications, due to sudden flight maneuvers for example, oil supply to the bearing may be interrupted for between 15 and 30 seconds. When this occurs, the bearings are not only required to function properly but to resume normal operation without any damage once the oil supply is re-established. Ball bearings are more susceptible to damage during oil-supply interruptions than roller bearings due to the excessive heat generated by the spinning motion of the balls. Seizure due to oil interruption may occur in one of two ways:

- Adverse $\Delta T \rightarrow$ Negative bearing clearance \rightarrow Excessive Hertzian stress \rightarrow High heat generation \rightarrow Bearing seizure
- Surface damage \rightarrow High heat generation \rightarrow Negative bearing clearance \rightarrow Bearing seizure

For a number of years, NSK-RHP has been developing a surface treatment to minimize the risk of bearing seizure under adverse operating conditions.¹⁾ This new treatment is discussed in the next section.

3.1.1 Surface modification by “AP Treatment”

In the AP (Advanced Phosphate) Treatment, a phosphate composite coating approximately 0.1- μm thick is applied to all normally treated surfaces of the outer ring, inner ring and rolling elements. At present, NSK main-shaft bearings with AP Treatment are performing successfully in a turbo-shaft engine. Through research it was confirmed that the AP Treatment has no adverse effect on rolling contact fatigue and is compatible with the lubricants used in aircraft gas turbine engines. Furthermore, a pin-on-disc test showed that the AP Treatment significantly improved the load-carrying performance by at least 25% (i.e., improved wear resistance), and the fact that the phosphate composite of the AP Treatment was still present after the test, despite

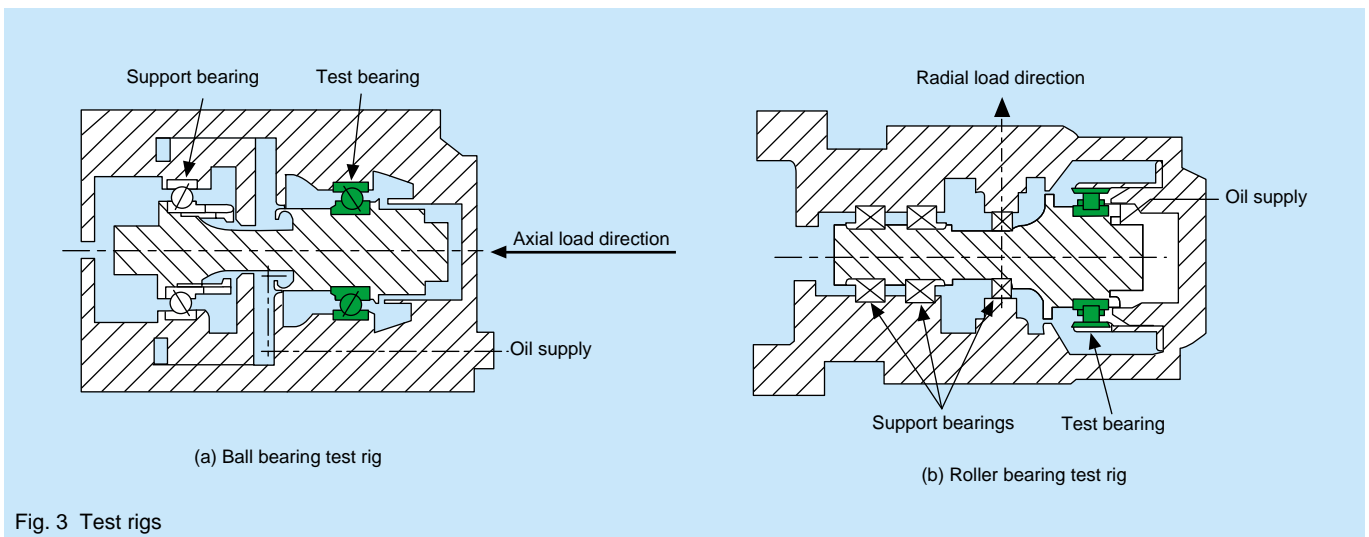


Fig. 3 Test rigs

a change in surface color, demonstrated the endurance of the surface coating.

3.1.2 Oil interruption test

In this test, the benefit of the AP Treatment during oil interruption was shown. Fig. 2 shows the dimensions of the test bearings, one is a split inner ring angular-contact ball bearing and the other is a cylindrical roller bearing. The oil interruption time was six minutes. Table 1 and Fig. 3 show the test conditions and the test rigs respectively. The rolling element orbital speed was measured along with the inner- and outer-ring temperatures.



Photo 1 Tested ball (with AP treatment)

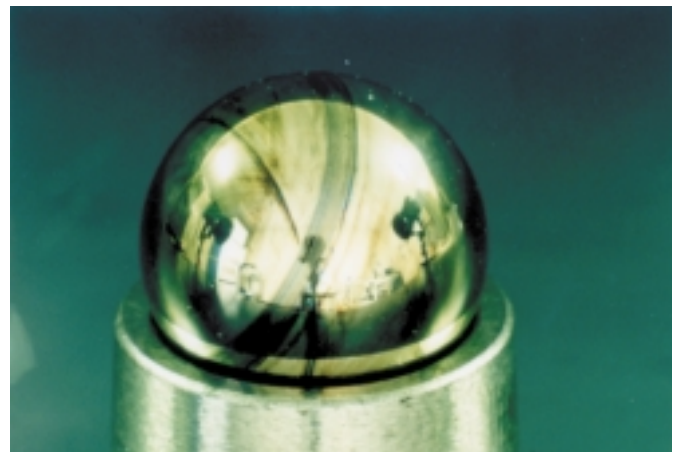
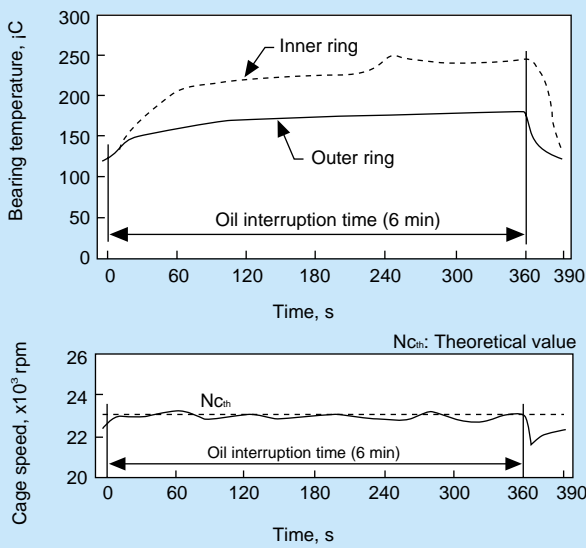
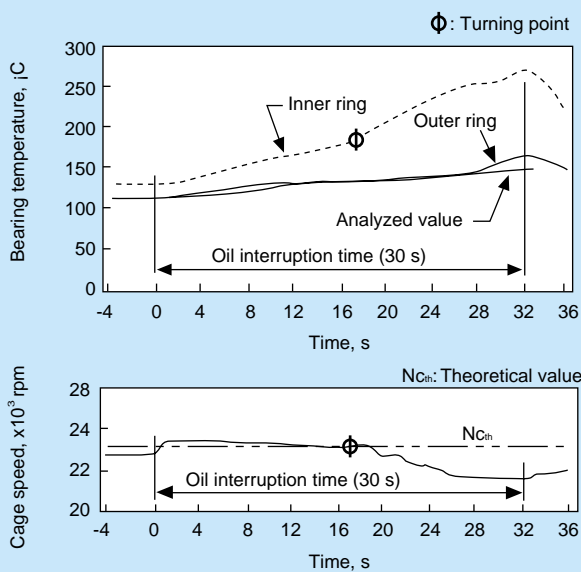


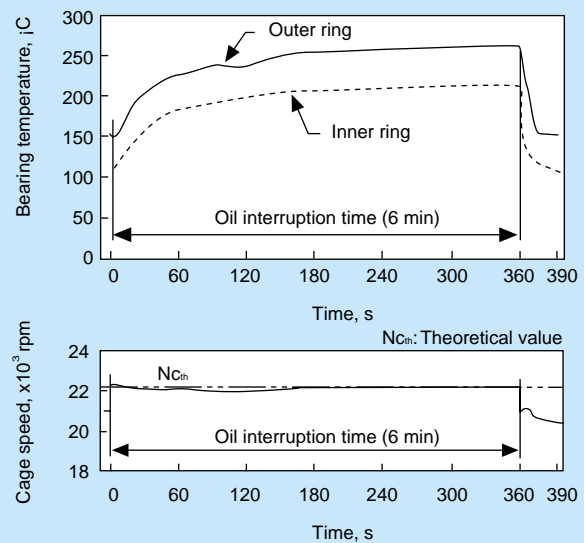
Photo 2 Tested ball (without AP treatment)



(a) Ball bearing (with surface modification)



(b) Ball bearing (without surface modification)



(c) Roller bearing

Fig. 4 Results of oil interruption test



Photo 3 Tested rollers (with AP treatment)

Table 1 Test conditions

(a) Ball bearing		
Shaft speed	(rpm)	52 000
Axial load	(kN)	1.8
Supply oil temperature	(°C)	80
Ambient temperature		Room temperature
Oil flow rate	(l/min)	3 → 0
PV	(GPa · m/s)	4.5
(b) Roller bearing		
Shaft speed	(rpm)	50 000
Radial load	(kN)	1.0
Supply oil temperature	(°C)	80
Ambient temperature	(°C)	200
Oil flow rate	(l/min)	2 → 0

3.1.3 Test results

(1) Ball bearing

Within the six-minute oil interruption period the rate of the temperature rise began to decrease after 40 seconds in the AP-treated bearing, achieving almost steady-state conditions. The orbital speed of the balls showed no abnormality and the motion of the balls was not unusual. The balls could have been re-used after the test, as their surfaces were in good condition with no signs of deterioration. Fig. 4 compares measured data on the AP-treated bearing (a) and the non-modified bearing (b). Photo 1 shows the surface condition after the test of a ball from the AP-treated bearing.

The rate of temperature rise in the non-modified bearing increased continuously over the oil interruption period, particularly after 17 seconds when the temperature began to increase rapidly, indicative of bearing seizure. A ball removed from this bearing after the test is shown in Photo 2.

(2) Roller bearing

As with the ball bearing, initially the temperature rose steadily in the AP-treated roller bearing but after about 50 seconds the rate of the temperature rise began to decrease significantly. After the test, the condition of the rollers also showed no abnormality. Fig. 4(c) shows the test results and Photo 3 the surface condition of some of the

rollers after the test.

The above test results demonstrate how the AP Treatment can significantly increase the durability of a bearing in severe operating conditions such as those encountered during oil interruption. Equally, it is believed that the AP Treatment can offer benefits under marginal lubrication conditions.

The ceramic/steel hybrid bearing with Si₃N₄ rolling elements has also demonstrated excellent seizure resistance.²⁾ However, until ceramic hybrid bearings become more widely accepted and improvements in quality assurance (i.e., non-destructive test methods) and shock-toughness are made, the AP Treatment represents a unique and cost-effective solution for extreme operating conditions such as those encountered during oil interruption or marginal lubrication. Other areas of application may include conditions in which PV values are high (P: contact pressure, V: sliding speed within contact).

3.2 Resistance to wear

Slip is an inherent feature of rolling bearings and at high speeds can result in a significant increase in wear and heat generation. The major types of slip are:

- ① Rolling elements and raceway
 - Micro-slip under pure rolling: differential sliding and sliding due to spinning and gyroscopic motions
 - Macro-slide: cage speed slip
- ② Sliding between rolling elements and cage pockets
- ③ Sliding at cage location lands

3.2.1 Skidding

Skidding is a term used to describe extreme sliding between the rolling elements and raceway. At high speeds, skidding causes wear of the rolling contact surfaces and results in surface damage that is called smearing. Skidding failure can be avoided, even under high sliding conditions, if the lubricant film is sufficient to separate the sliding surfaces. However, changing lubricant and improving surface roughness may not be sufficient to prevent skidding in some cases, so other countermeasures have to be considered.

Skidding is related to the balance between the drive force and drag force. Therefore, in order to minimize the risk of skidding, the drive force needs to be increased and/or the drag force reduced. Methods for reducing skidding are summarized below.

- Increasing drive force:
 - a) Load Preload ----- Multi-lobe raceway, Negative clearance
 - External load--- Un-balance, Offset
 - Reduced no. of rolling elements
- b) High-traction oil
- c) Raceway rib ----- N-type (inner-ring rib)
- d) Cage location land ----- Inner-ring riding
- Reducing drag force:
 - a) Reduction of oil flow
 - b) Extraction of oil ----- N-type (inner-ring rib)
 - c) Reduced no. of rolling elements

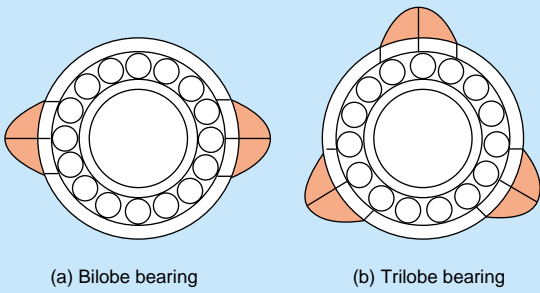
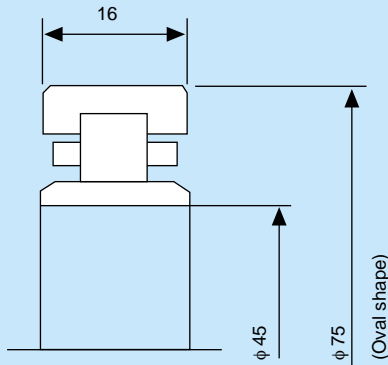
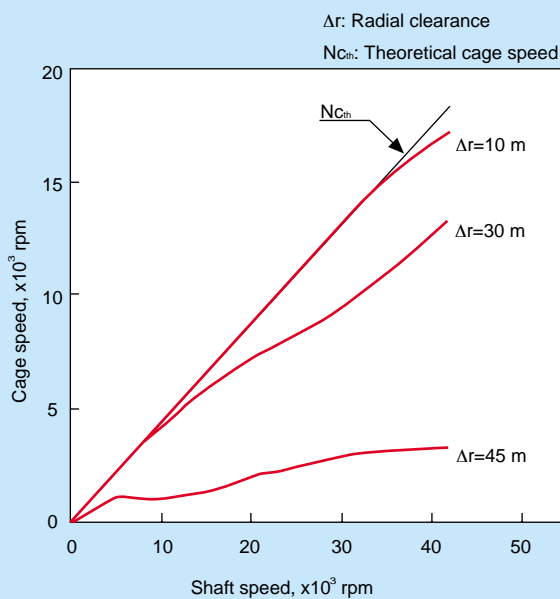


Fig. 5 Multi-lobe bearing rings to prevent skidding

Shaft speed	(rpm)	0~42,000
Radial load	(kN)	50.0
Supply oil temperature	(°C)	100
Ambient temperature		Room temperature
Oil flow rate	(l/min)	2.1



(a) Test conditions and test bearing dimensions



(b) Test results - round shape

- Other: Material, Misalignment, Oil-cleanliness, Rapid acceleration/deceleration

Probably one of the most common ways to increase the drive force is to apply radial preload by designing multi-lobe rings like the ones shown in Fig. 5. However, the housing stiffness, bearing fits and radial internal clearance need to be considered thoroughly in the design of a multi-lobe bearing.

Test results showing the influence of bearing radial internal clearance on roller orbital slip in a normal and bilobe bearing are shown in Fig. 6. It is evident from Fig. 6 that the bilobe bearing is less sensitive to changes in radial internal clearance and therefore also less sensitive to changes in bearing temperature.

3.2.2 Roller end wear

(1) Characteristics of wear

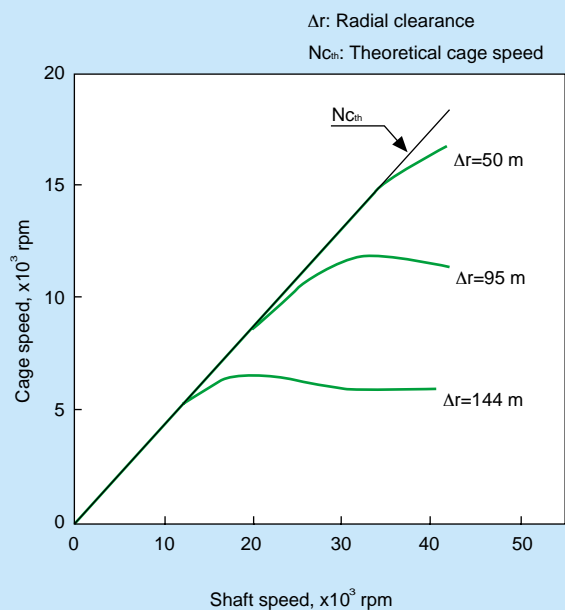
As the rollers in cylindrical roller bearings will always tend to skew during operation, sliding between roller end faces and ribs is unavoidable. As forming an oil film between the roller end face and rib is difficult due to the geometry involved, these parts can be seriously worn during high-speed operation. Roller end wear is progressive by nature and, once started, is unlikely to stop. In the worst case, rollers can turn crosswise resulting in failure of the cage. In high-speed operation of cylindrical roller bearings, two wear processes and corresponding wear patterns are generally recognized:

- Eccentric wear (see Fig. 7a)

This wear is due to the unstable movement of the roller (cyclic change of skewing angle). The wear pattern of the opposite end face is 180° out of phase.

- Concentric wear (see Fig. 7b)

Though the roller rotates while maintaining a certain skewing angle, the wear occurs due to a high PV value at



(c) Test results - bilobe shape

Fig. 6 Effect of bilobe design on preventing skidding

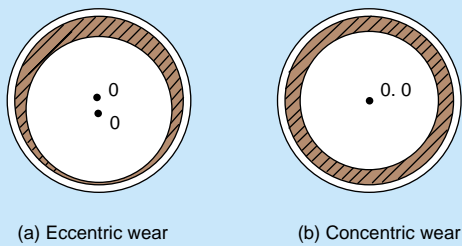


Fig. 7 Wear patterns of roller end surface

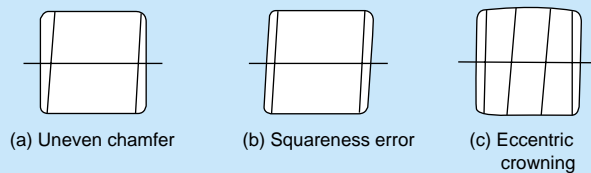


Fig. 8 Examples of poor roller geometry

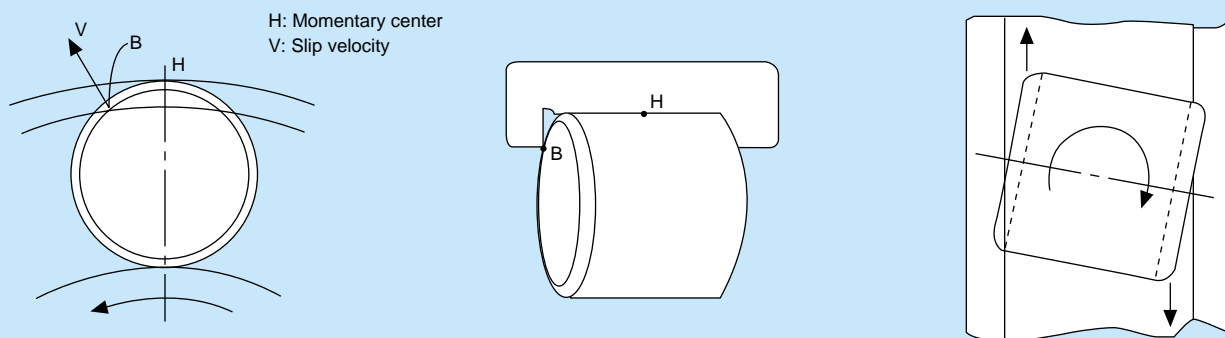


Fig. 9 Conditions of skewed roller

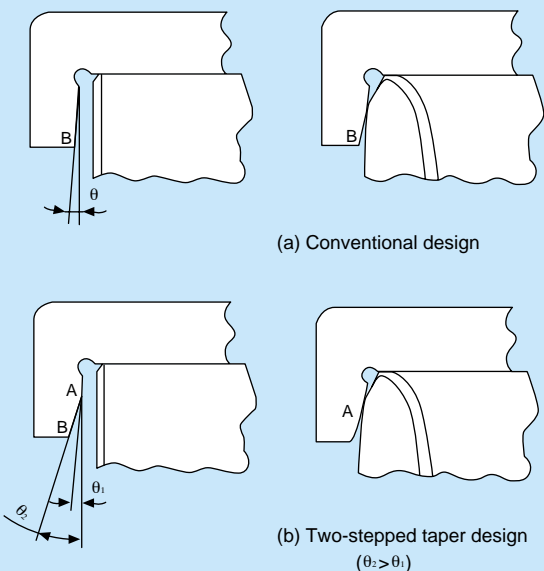


Fig. 10 NSK's two-stepped tapered rib

the contact area between the roller end face and rib surface.

(2) Countermeasures

• Eccentric wear

This type of wear is due to the rollers not being symmetrical. Uneven chamfering, squareness errors and

eccentric crowning are examples of poor roller geometry that can lead to eccentric wear (see Fig. 8). The countermeasure for eccentric wear is therefore high geometrical accuracy of the rollers.

• Concentric wear

Even with symmetrical rollers, skewing and hence concentric wear will occur, but can be minimized by:

- Minimizing the PV value at the contact point (Point B in Fig. 9) of the roller and rib in the skewing condition.
- Ensuring that there is sufficient lubricant at the contact point to avoid direct metal-to-metal contact.³⁾

In order to minimize the risk of this type of wear, NSK has designed the two-stepped tapered rib shown in Fig. 10.

4. Integrated Bearings

In order to minimize the number of parts and reduce cost and weight, bearing designs are often required to have integrated components such as flanges, squirrel cages, gears and stub-shafts. As a result, bearing designs become much more complex and the analysis of bearing structural behavior (i.e., stress concentration and structural resonance), material choice and production engineering methods (i.e., welding technology and riveting) must be carried out in greater detail.

Fig. 11 shows one of the most advanced turbofan engine

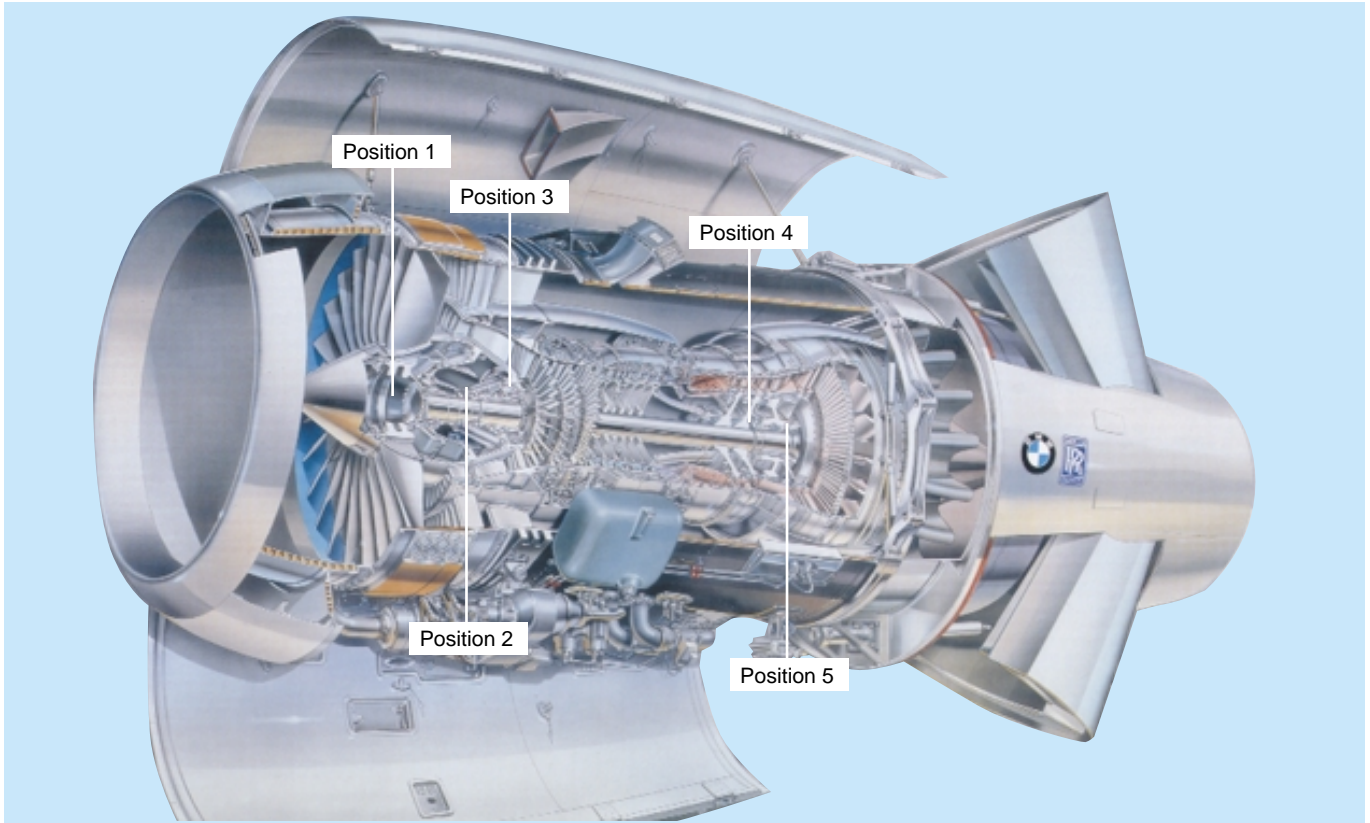
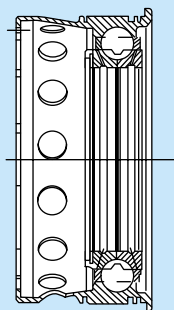
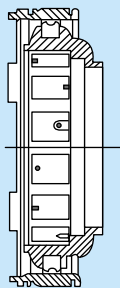


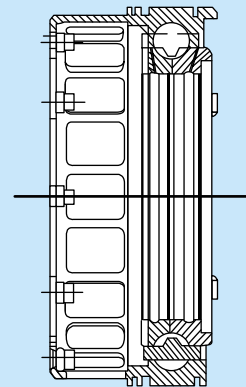
Fig. 11 Advanced turbofan engine (courtesy BMW Rolls-Royce AeroEngines)



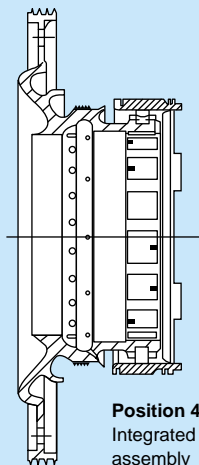
Position 1:
Bearing with structural
outer ring



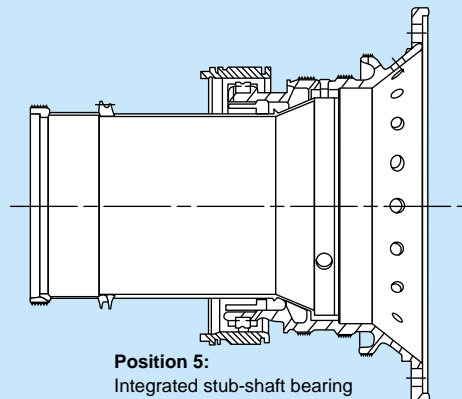
Position 2:
Bearing with structural
inner ring



Position 3:
Bearing with squirrel cage



Position 4:
Integrated stub-shaft bearing
assembly



Position 5:
Integrated stub-shaft bearing
assembly with seal tube

Fig. 12 Integrated bearing designs

assemblies for powering regional aircraft. The engine has five main-shaft bearings including both ball and roller bearings. In the drawings of these bearings in Fig. 12, it can be clearly seen how components adjacent to the bearing have been integrated within a single bearing design.

5. Conclusion

The need by engine manufacturers to increase performance while reducing cost and weight requires rolling bearings to cope with increasingly extreme operating conditions including higher speeds and loads. Rolling element bearings are relatively cheap and easy to use but in order to meet the performance requirements of new engines, development of bearing technology including materials, surface treatments and bearing design must continue.

This article has identified some of the key operating requirements of aircraft gas turbine engine bearings and explained how new developments within NSK-RHP have helped to ensure that rolling bearings continue to provide cost-effective solutions for the aircraft engines of today and tomorrow.

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Development of NSK Linear Guides

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ABSTRACT

Though the fundamental concept of the linear guide is included in a 1932 French patent, it was not until the early 1970s that linear guides were commercialized. Progress with the numerical control of machine tools led to higher speed and accuracy of machines that exposed limitations of conventional sliding guides in terms of durability and response capability. As a result, rolling guides, having better high-speed performance and greater compatibility with electronics, began to be used widely. NSK began selling precision linear guides for machine tools in the early 1980s, and has been developing new series in response to market needs ever since.

This report discusses the basic design concept of NSK linear guides, introduces new NSK linear guides for machine tools, and presents technologies developed by NSK in response to the changing needs of the market.

1. Introduction

Rolling linear guide bearings may trace their origin to an ancient means of carrying massive rocks on round logs, but they did not become widely established as mechanical elements until the twentieth century. The basic design of today's rolling linear guides with rails is outlined in a French patent from 1932 (see Fig. 1), but it took until the early 1970s before this type of linear guide was commercialized, as a variation of a ball spline, for use in general industrial machines.

In the modern machine tool industry, when numerical control began to be widely used for higher speed and precision in machine tools, conventional sliding guides often presented problems due to their lack of endurance and response capability to commands from numerical

controllers. It was at this point that rolling guides, having better high-speed performance and greater compatibility with electronics, began to overtake sliding guides as the method of choice for controlling linear motion in machine tools. In the early 1980s, NSK began selling its LY Series of precision linear guides mainly for machine tools. Almost at the same time, the use of rolling guides in machine tools began to increase dramatically. Since then, a wide variety of linear guides has been developed for myriad applications in many fields. Today, linear guides are the most common type of rolling guide, and they are used extensively in the expanding market for products that facilitate machine automation and saving labor.

This report first describes the basic design, construction and characteristics of NSK linear guides, then presents NSK's linear guides for machine tools, and finally discusses technologies developed by NSK in response to the changing needs of the market.

2. Basic Design, Construction and Characteristics of NSK Linear Guides

The wide variety of NSK linear guides described in Tables 1 and 2 enables users to select the optimum product for their specific application and installation requirements. While all of the linear guides have different functions and characteristics depending on how they are

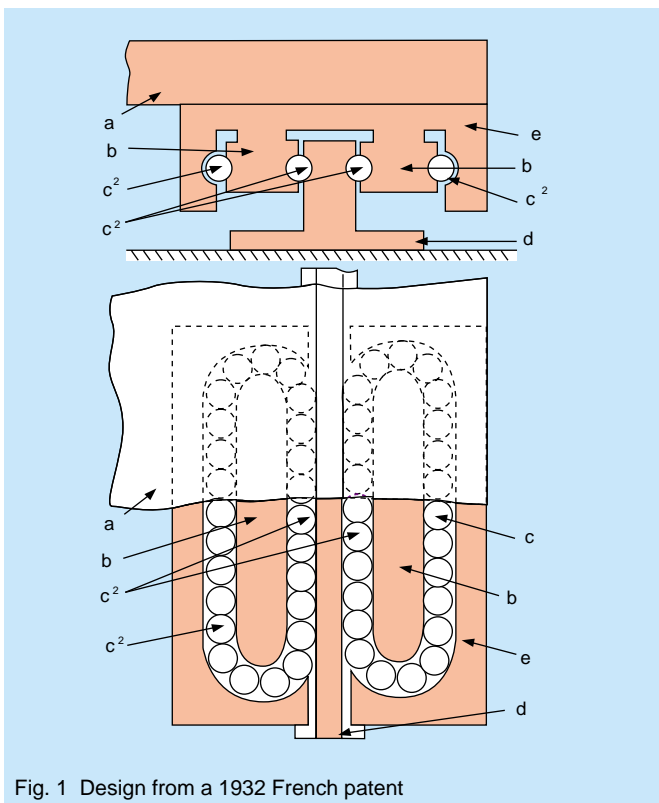


Fig. 1 Design from a 1932 French patent

Table 1 Various series of NSK linear guides

Characteristics	Series	(1)		(2)		(1)(2)		(2)	
		LS	LH	LY	LW	LU	LE	LL	
Direction of load capability	Equal load rating in four directions			○		○	○	○	○
	High load rating in vertical direction	○	○		○				
Rated load	Ultra high load		○	○					
	High load	○	○	○	○				
	Medium load	○				○	○	○	
Stiffness against moment load	Self aligning	○	○						
	High stiffness			○	○		○		
	Intermediate					○		○	
Interchangeability of rails	○	○		○	○	○			
Stainless steel	○			○	○	○			

(1) Wide-rail type (2) Miniature type

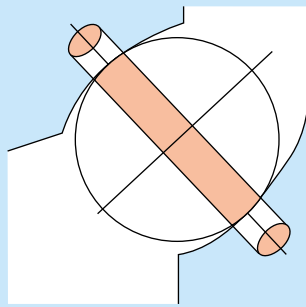


Fig. 2 Circular-arc groove

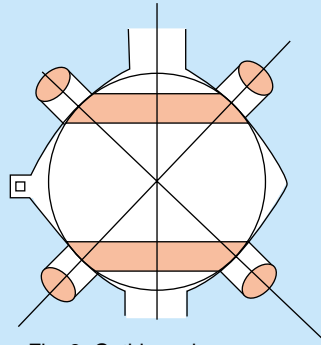


Fig. 3 Gothic-arch groove

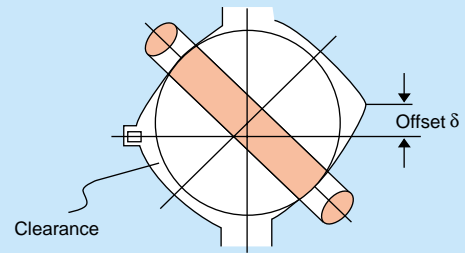
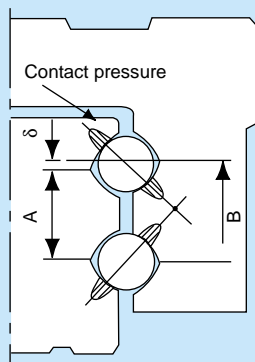
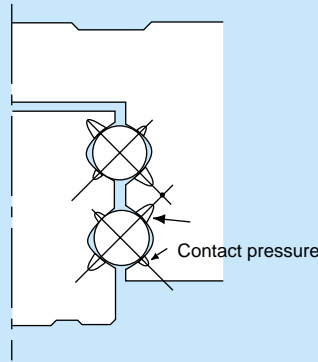


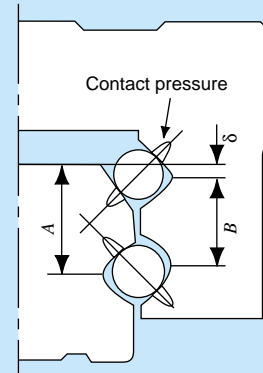
Fig. 4 Offset gothic-arch groove



Light preload



Heavy preload



LH · LS Series

LY Series

Fig. 5 Cross-sectional views of the LY and LH · LS Series

meant to be used, there are some basic elements of their design which they have in common. These include Gothic-arch grooves.

2.1 Offset Gothic-arch grooves and their characteristics

The characteristics of a rolling guide greatly depend on the geometry of its ball grooves. In terms of ball groove geometry, there are two types: circular-arc grooves (comprised of a single circular arc, Fig. 2) and Gothic-arch grooves (a combination of two circular arcs, Fig. 3). In gothic-arch grooves, the ball makes contact with the groove at four points when the rail groove and bearing groove centers are aligned with each other as in Fig. 3. However, when these two groove centers are offset as in Fig. 4, two-point contact results. This type of arrangement is called an offset Gothic-arch groove.

Naturally, the ball contact geometry in offset Gothic-arch grooves varies with the degree to which the grooves are offset. If the offset is small, the balls will have two-point contact when preload is light and four-point contact when preload is heavy. Four-point contact results in higher rigidity and higher load-carrying capacity than two-point contact. Additionally, the greater differential sliding which occurs with four-point contact results in higher friction and thereby increases the vibration damping capacity of the linear guide. If the offset is large, however,

Table 2 Types of ball slides

Series	Height	Ball slide length	Shape		
			Mounting hole		Flanged
			Square Tapped	Square Drilled	Square Tapped
LS	Low	Standard	LS-AL	LS-FL	LS-EL
		Short	LS-CL	LS-KL	
LH	High	Standard	LH-AN		
		Long	LH-BN		
	Low	Standard		LH-FL	LH-EL
		Long		LH-HL	LH-GL
LY	High	Standard	LY-AN		
		Long	LY-BN		
	Low	Standard	LY-AL	LY-FL	LY-EL
		Long	LY-BL	LY-HL	LY-GL
LW	Low	Standard			LW-EL
LU	Low	Standard	LU-AL(TL) ⁽¹⁾		
		Long	LU-BL(UL) ⁽¹⁾		
LE	Low	Standard	LE-AL(TL) ⁽¹⁾		
		Long	LE-BL(UL) ⁽¹⁾		
		Short	LE-CL(SL) ⁽¹⁾		
LL	Low	Standard	LL-PL		

(1) Large mounting holes on ball slide

the balls will not have four-point contact even when preload is heavy, and friction will remain low.

Fig. 5 shows sectional views of the LY and LH · LS Series of NSK linear guides. The small offset of the grooves in LY-series linear guides makes them suitable for

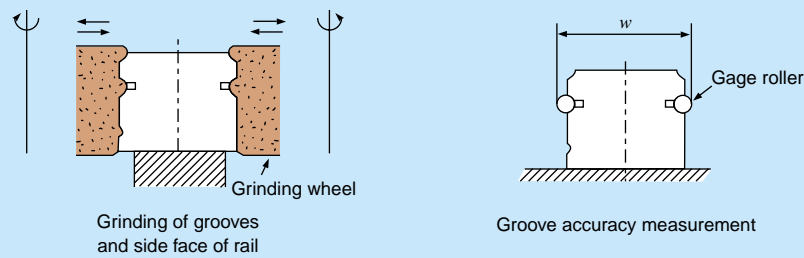


Fig. 6 Grinding and measuring of rail grooves

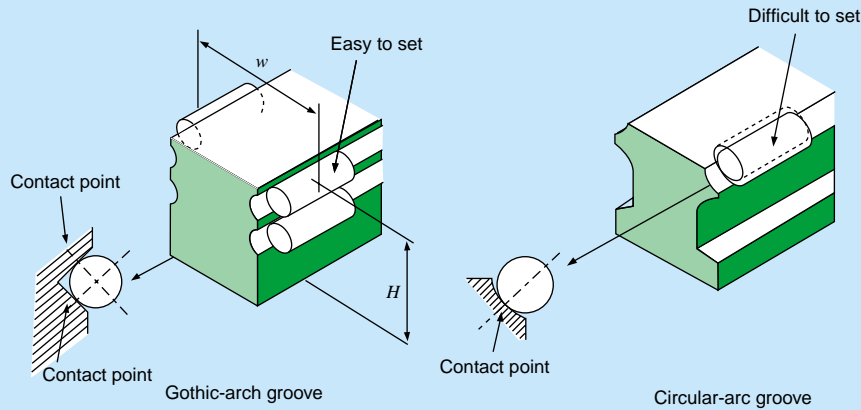


Fig. 7 Groove location measuring technique

cutting machine tools, which require high rigidity and high vibration absorption. In contrast, the large offset of the grooves in the LH and LS Series makes these linear guides suitable for non-cutting machine tools and other industrial machines requiring low friction. Thus, NSK creates linear guides with offset Gothic-arch grooves adapted to the specific requirements of their intended use.

2.2 Basic design and accuracy assurance

The stiffness, life and accuracy of a linear guide largely depend on whether its grooves are as precisely formed and located as they are designed to be. Therefore, accurate measurement and control of the geometric and positional accuracy of the grooves is essential to producing high-quality linear guides. In view of this, NSK linear guides have Gothic-arch grooves on their rail sides.

Fig. 6 depicts the methods for grinding rail grooves and measuring the groove-to-groove distance, W . Grooves are formed in both sides of a rail simultaneously by two grinding wheels. As a result, the accuracy of the geometry and distance from top to bottom of the groove is ensured by precise control of the dresser that determines the forming accuracy of the grinding wheels. The other important factor for groove accuracy is therefore the accurate measurement and control of the distance, W , between the grooves on opposite sides of the rail.

Fig. 7 illustrates the technique for measuring the location of the grooves. As the master rollers remain in steady two-point contact with the Gothic-arch grooves, easy and accurate measurement is made possible.

As a result of the basic design and accuracy control

methods described above, stable ball contact is ensured, even for multi-point-contact designs like the LY Series, and rails and ball slides can be randomly matched, as in the LH and LS Series.

2.3 Characteristics of NSK linear guides

NSK linear guides are made of extremely pure material while utilizing over eighty years of rolling bearing technology and experience. The transformation from raw material to completed precision product is supported by comprehensive production technology and overseen by NSK's strict quality assurance system every step of the way. As a result, NSK linear guides consistently demonstrate outstanding performance characteristics in both experiments and actual use. The following sections present examples of these characteristics as demonstrated in experiments.

2.3.1 Basic dynamic load rating and life

The method for calculating the rated load of rolling bearings was established by the ISO in Directive 281. ISO 281 is based on the Lundberg-Palmgren fatigue theory. Calculations based on this theory for the load-carrying capacity, i.e., the rated dynamic load and life, of rolling bearings are widely supported and standardized by the ISO, as well as many national standardizing organizations (DIN, ASA, JIS, etc.). However, as no such international standard has been established for linear guides, manufacturers use their own calculation methods. The essential characteristics of rolling element and raceway contact in a linear guide are identical to those of a rolling

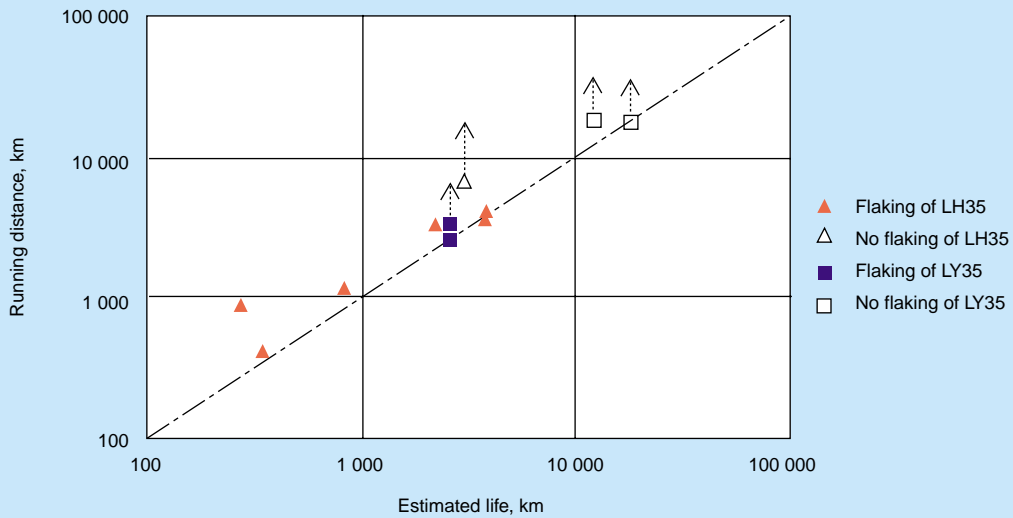


Fig. 8 Comparison of estimated life and test results

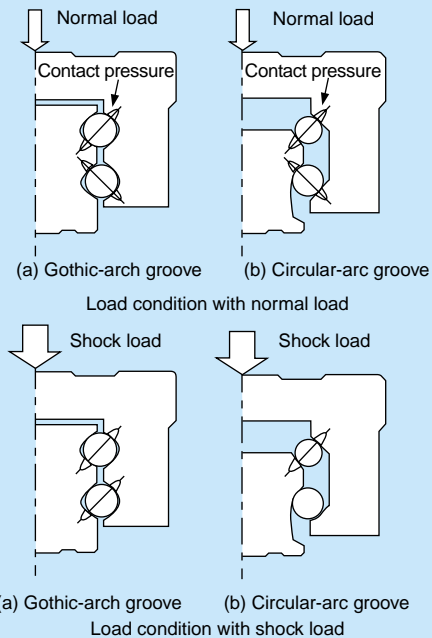


Fig. 9 Load conditions of gothic-arch and circular-arc grooves

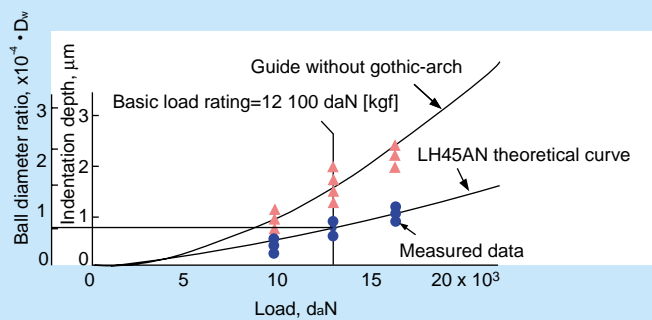


Fig. 10 Load and indentation depth

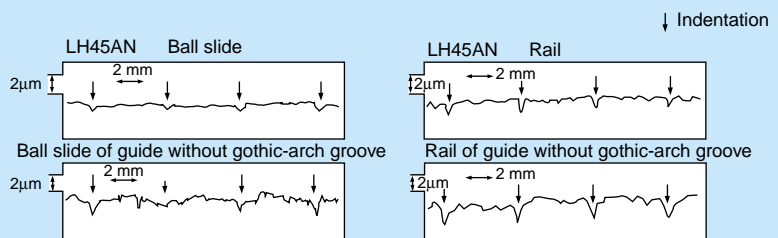


Fig. 11 Indentation under static load rating

bearing, so the theory for rolling bearings is also applicable to linear guides. Based on the Lundberg-Palmgren fatigue theory, NSK developed the following formula for calculating the rated dynamic load of a linear guide²⁾:

$$C = bm \cdot fc \cdot i^{0.7} \cdot Dw^{2.1} \cdot Z^{2.3} \cdot Le^{1/30} \cdot \cos \alpha \text{ -----(1)}$$

- bm*: compensation coefficient
- fc*: material and shape coefficient
- i*: number of rows of balls to bear load
- Dw*: ball diameter
- Z*: number of balls per row to bear load
- Le*: effective length of ball slide groove
- α : angle between contact direction and loading direction

To check the validity of the formula, NSK performed various endurance tests. Fig. 8 compares results of these tests with estimates obtained using the formula. In this graph, the x-axis represents calculated life in terms of running distance while the y-axis represents the distance actually run, both on a logarithmic scale. If the actual life is exactly the same as the calculated life, the test results should fall along the diagonal line. As it turned out, all of the plotted points fell on or above the diagonal line, indicating longer actual life and verifying the validity of the formula.

2.3.2 Shock resistance

A drawback common to rolling elements is insufficient

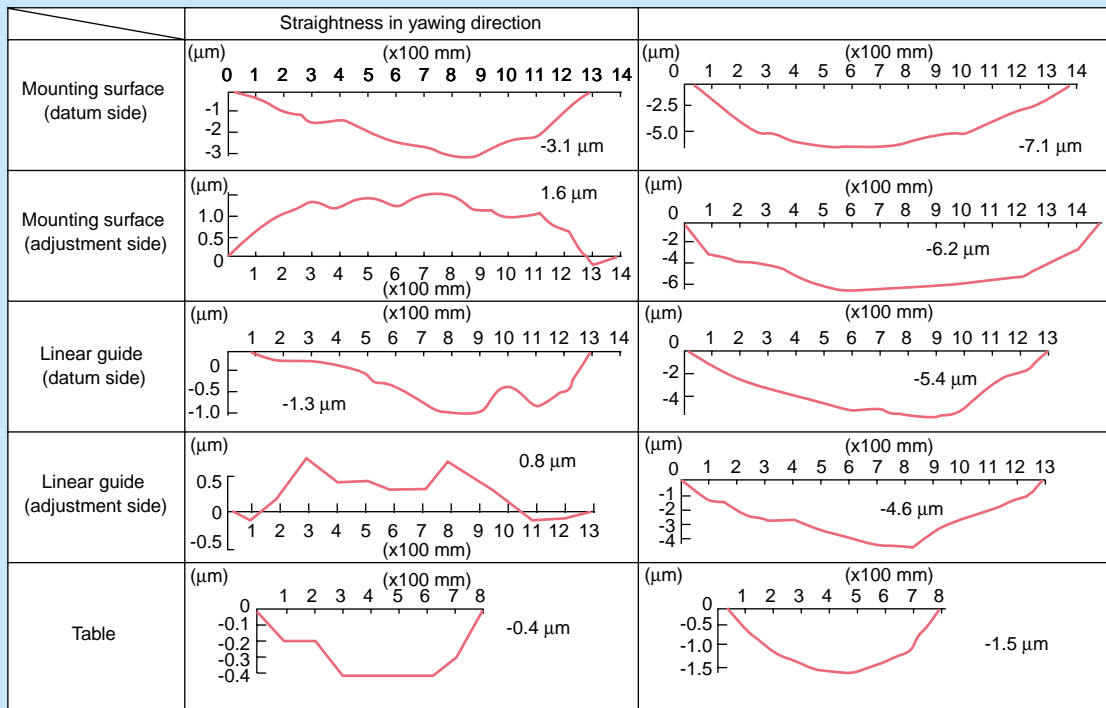


Fig. 12 Straightness of table using linear guide

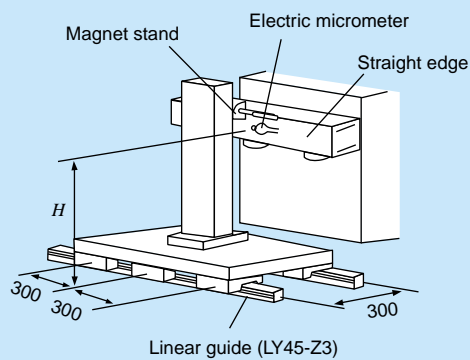


Fig. 13 Measuring equipment of vibration caused by ball passage

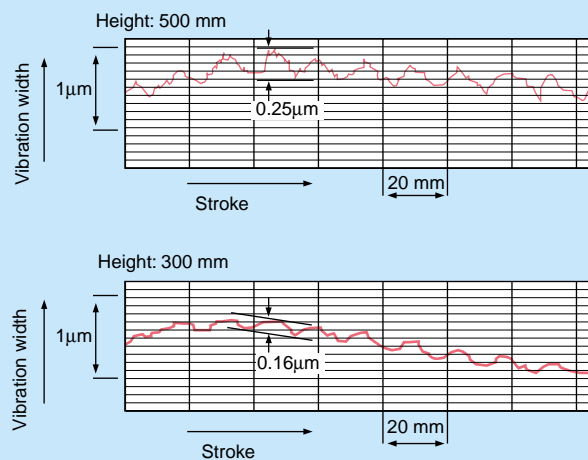


Fig. 14 Vibration caused by ball passage

capacity to accommodate very high loads such as shock loads. When a very high load is applied to a linear guide with Gothic-arch grooves, the load distribution changes so that both rows of balls help support the load, as shown in Fig. 9. Thus Gothic-arch grooves are more capable of accommodating high impact loads.

Fig 10 compares the depth of indentations caused by static loads applied from above in an LH45AN linear guide and a comparable linear guide without Gothic-arch grooves. The indentations in the LH45AN guide were less than half as deep as those in the linear guide without Gothic-arch grooves. Fig. 11 shows test data on

indentations under rated static load. The results of both of these tests demonstrate the effectiveness of Gothic-arch grooves.

2.3.3 Straightness of movement

With the primary function of linear guides being to guide linear movement, their accuracy in terms of straightness is a very important part of their overall performance. In fact, general accuracy requirements have been established for linear guides. However, the straightness accuracy of a linear guide can be affected by

the accuracy of the bed on which it is mounted. In an actual machine, interference between ball slides and between rails may often have the effect of averaging straightness errors and thereby reducing individual straightness errors to a lower level than the entire table. Fig. 12 shows actual measurements of the straightness of a table using linear guides. The straightness error of the entire table is almost one-fifth of the mounting surface. Generally, the straightness error of a table using linear guides is reduced to between one-half and one-tenth of the mounting surface straightness error.

Rolling bearings usually have periodic vibration produced by the circulation of their rolling elements. In the case of linear guides, the position of the ball slides is subject to periodic displacements caused by the balls as they enter and exit the loaded zone. These displacements are called ball passage vibrations. In applications where very high accuracy is required or in cases where a point at which accuracy is required is away from the linear guide, the displacements may be enlarged and lead to problems. To avoid this, the groove ends of the bearing have a gentle slope called crowning.

Fig. 14 shows ball passage vibrations measured using the measuring equipment shown in Fig. 13 with heights of 300 and 500 mm, from the linear guide to the point of measurement. The magnitude of the ball passage vibration is almost proportional to the distance from the linear guide (overhang distance). It is smaller when the number of bearings per rail is larger, when the bearing mounting span is wider, and when the preload is lighter. This means that the use of fewer ball slides under light preload can more readily reduce ball passage vibration than the use of a large number of ball slides under high preload.³⁾

3. Linear Guides for Machine Tools

As machine tools have been designed to operate at increasingly higher speeds, the need for high-speed linear guides has grown considerably. In the design and development of linear guides for machine tools, NSK places emphasis on providing them with the following characteristics:

- high rigidity and good balance in all directions with moderate preload
- moderate friction and damping ability
- high accuracy in straightness of movement and other functions
- long life and high reliability
- high impact resistance

3.1 LY Series of NSK linear guides

LY-series linear guides for machine tools have been on the market for 15 years. They have two Gothic-arch grooves on either side of the rail. The center of each of the grooves is offset in such a way that the balls contact the groove at four points under a preset load. This four-groove, four-point contact gives the linear guide high rigidity under a moderate preload. Additionally, setting the contact angle at 45° makes the linear guide well balanced both vertically and horizontally, and provides high rigidity against cutting forces in all directions.

Fig. 15 compares the rigidity of linear guides from different manufacturers as measured by the University of Technology-Aachen on commitment by the German Association of Machine Tools (VDW4). Presenting data on linear guides with balls only, the figure shows that the rigidity of linear guides in the LY Series is higher than the others.

The moderately higher friction force that results from four-point-contact increases the damping capacity of the

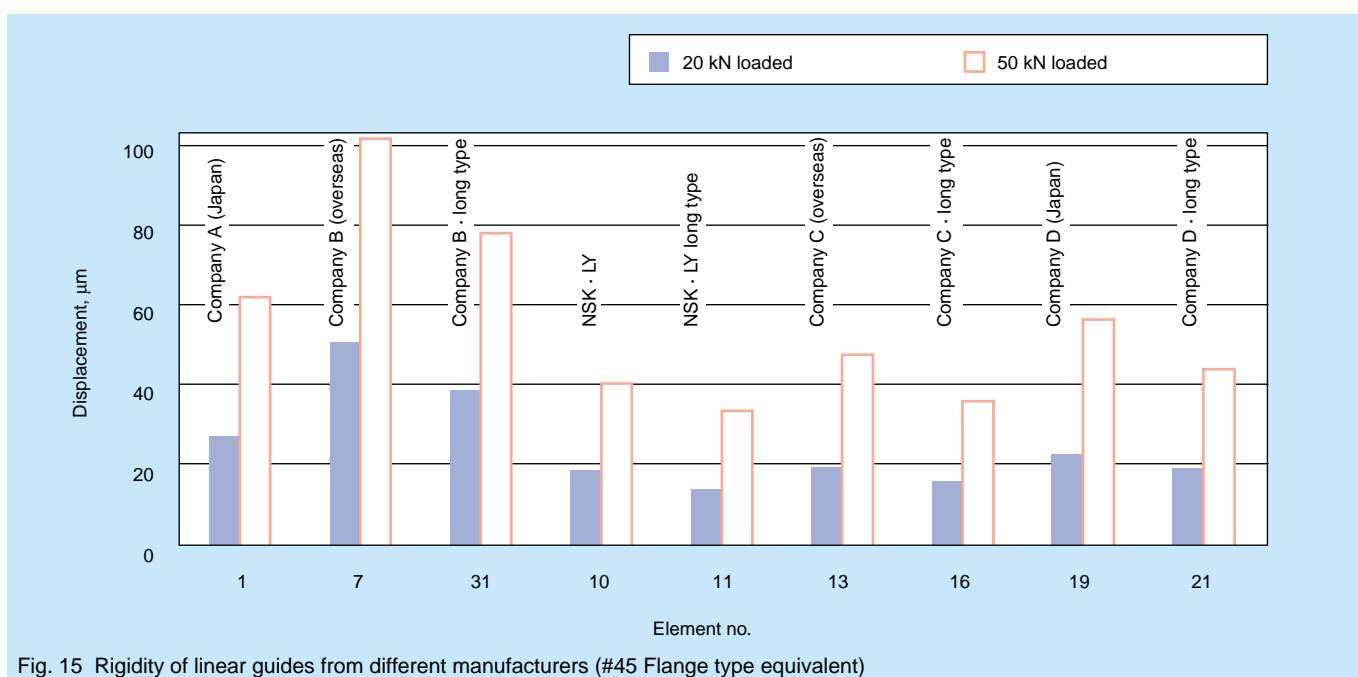


Fig. 15 Rigidity of linear guides from different manufacturers (#45 Flange type equivalent)

LY Series.⁹ While higher than that of two-point contact, this damping capacity is still less than that of sliding guides. Fig. 16 illustrates the device used to measure the damping factors that are compared in Fig. 17.

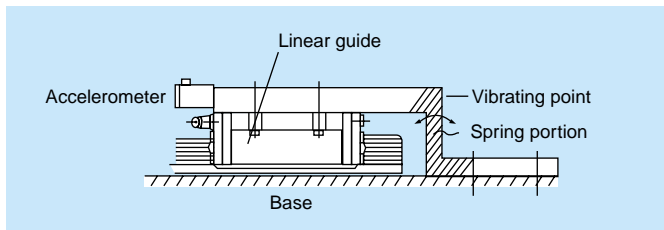


Fig. 16 Damping evaluation device

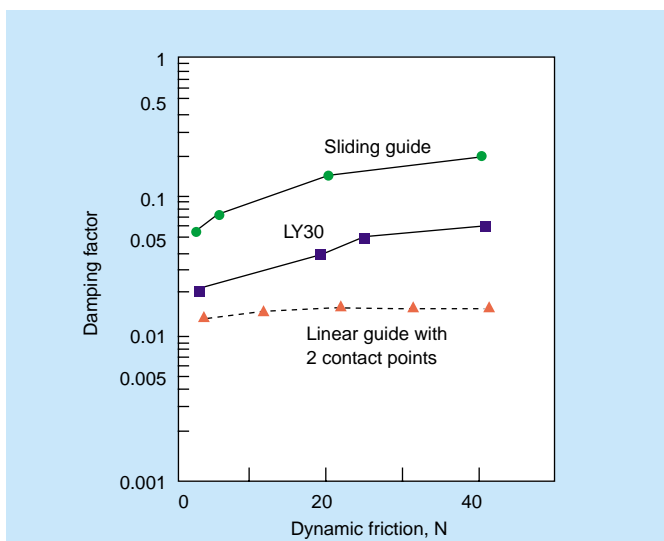


Fig. 17 Comparison of damping factors

3.2 New NSK Linear Guides for machine tools, LA Series

As described above, LY-series linear guides have high rigidity and a high damping factor. Together these features make LY linear guides suitable for machine-tool applications. However, the latest, most-advanced machine tools like machining centers present requirements for even higher speed and accuracy. Some of them have a feed speed of 60 m/min or higher. Higher machine speed necessarily involves higher acceleration and deceleration, imposing greater load on guide systems. Higher machine accuracy, on the other hand, requires greater rigidity of machines and, naturally, the linear guides they contain. In view of these circumstances, NSK developed its LA Series of linear guides for state-of-the-art machine tools. The objectives behind the development of the LA Series were:

- Making both the rigidity and load-carrying capacity 1.4 to 1.5 times higher than in the LY Series through the inclusion of a third groove on each side of the rail
- Reducing the friction force to 50 to 60% of what it is in the LY Series by achieving a better balance between four- and two-point contact

Table 3 Comparison of the characteristics of the LA and LY Series

	LA45 long type P5Z4	LY45 long type P5Z4
Ball diameter D_a (mm)	6.35	6.35
Contact angle	45°	45°
Number of contact points	4	4
Dynamic load rating C (kN)	89	63
Static load rating C_0 (kN)	166	103
Preload F_p (N {kgf})	7 750 {790}	4 410 {450}
Calculated rigidity (N/ μ m {kgf/ μ m})	1 640 {167}	1 100 {112}
Friction rating (N {kgf})	66 to 93 {6.7 to 9.5}	108 to 157 {11 to 16}

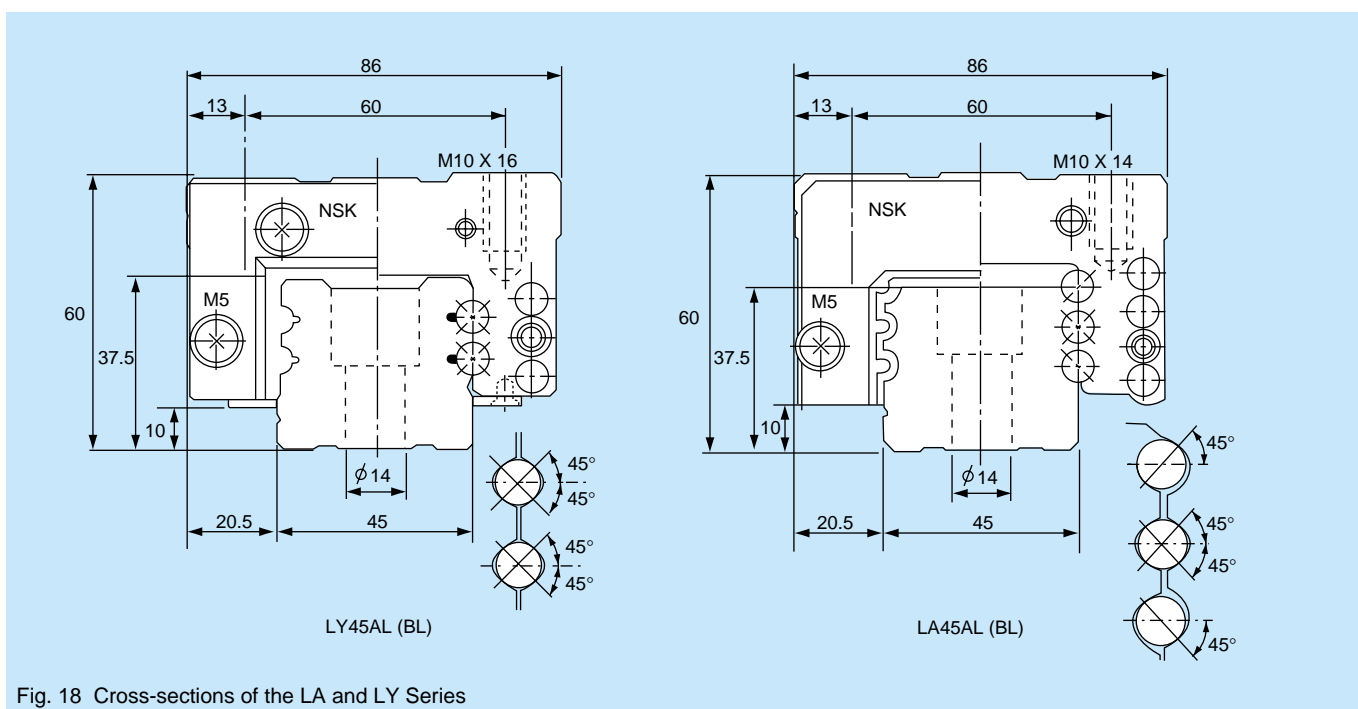


Fig. 18 Cross-sections of the LA and LY Series

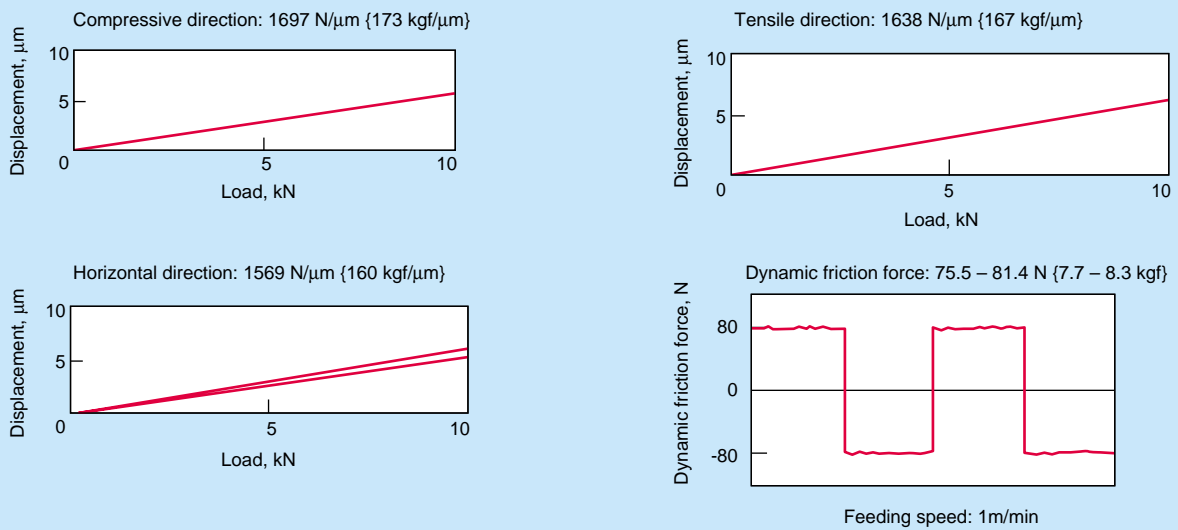


Fig. 19 Rigidity and friction force of an LA45 long-type (heavy pre-load) linear guide

- Interchangeability with existing linear guides in mounting dimensions
- Incorporating other improvements made possible by recent advances in NSK technologies

Fig. 18 compares cross-sections of the LA and LY Series. While the mounting dimensions and ball diameter are the same, linear guides in the LA Series have three grooves on each side for a total of six. The top and the bottom grooves are single radius grooves with the ratio of groove radius to ball diameter smaller than that of the LY Series. The middle groove is a Gothic-arch groove to allow four-point contact. High groove-accuracy is achieved by employing extremely precise measurement and machining processes. The combination of these highly accurate grooves gives the LA Series high rigidity and high load-carrying capacity with moderate friction force. In addition, as in the LY Series, setting the contact angle at 45° ensures almost equal rigidity and load capacity in all directions. Table 3 compares characteristics of the LA and the LY Series. Fig. 19 shows measurement results on the rigidity and dynamic friction force of a linear guide from the LA Series. As the results indicate, the friction force of the LA Series is about 60% of the LY Series, and the rigidity is 1.4 to 1.5 times higher.

The LA Series incorporates other improvements in addition to those described above. One of these is better sealing performance. The rail's shape has been simplified as much as possible to ensure tighter seal contact and the seal itself has been improved to better prevent the ingress of foreign substances and the outflow of ball slide lubricant. Additionally, linear guides in the LA Series can be equipped with NSK K1 Seals™, which are described briefly below and in more detail in another report in this journal. The improved sealing performance described above in tandem with this option results in better maintenance-free performance of the series. The various types of linear guides available in the LA Series are listed

Table 4 Types of LA-series linear guides

Shape code	AN/BN	AL/BL	EL/GL	FL/HL
Shape	Square		Flanged	
	Higher	Lower		
Model No.	Tapped mounting holes			Drilled mounting holes
LA30	○		○	○
LA35	○	○	○	○
LA45	○	○	○	○
LA55	○	○	○	○
LA65	○		○	○

in Table 4. Two types of preload are available: Z3 (medium preload) and Z4 (heavy preload).

4. Responses to Changing Market Requirements

As mentioned, linear guides are being used in increasingly diverse applications. NSK produces a wide variety of linear guides with features that meet changing market requirements. This section presents some of the special features of NSK linear guides.

4.1 Maintenance-free performance

Effective lubrication and proper maintenance of rolling products are essential in ensuring reliable performance over extended periods. NSK developed "Molded Oil," a unique polyolefin resin compound containing lubricating oil, and used it to create NSK K1 Seals™ for linear guides. Linear guides equipped with K1 Seals are already being supplied to users. Photo 1 shows K1 Seals and a linear guide fitted with them. K1 Seals contain 70% lubricating oil by weight. During operation, lubricating oil gradually

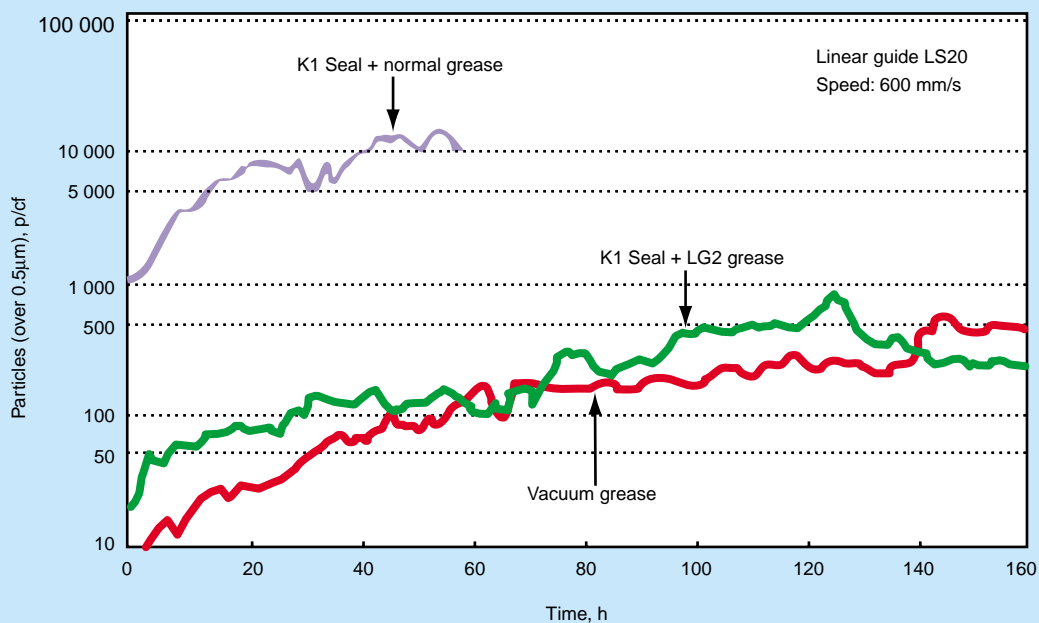


Fig. 20 Comparison of particle emission

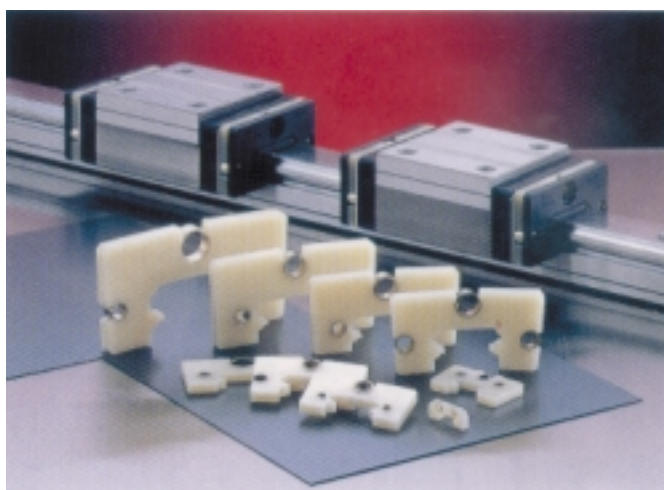


Photo 1 NSK Linear Guide and K1 Seals

seeping from K1 Seals helps keep linear guides lubricated for long periods. K1 Seals have demonstrated excellent lubricating performance in various tests at NSK as well as in actual use on the market.

Initially, K1 Seals were used in the LH and LS Series mainly for conveying equipment. Meeting market requirements for maintenance-free performance, low environmental contamination from lubricant outflow, and availability at a low price, K1 Seals were well received by the market. Today, K1 Seals are employed in LY- and LA-series linear guides for machine tools, and are beginning to be used even with the LU, LE and LW Series.⁶⁾ In addition, an interchangeable ball slides series with K1 Seals is included in the range of NSK standard products.

As requirements for maintenance-free performance are expected to become more demanding in the future, NSK is working to expand applications of K1 Seals.

4.2 Performance in clean environments

Semiconductor and liquid crystal manufacturing require clean environments. For lubrication of linear guides used in clean vacuum environments, fluorine grease has conventionally been used because it emits very few particles into its operating environment. However, fluorine grease lacks durability, and has lower corrosion resistance and higher viscosity resistance at high speeds than general greases. In response, NSK developed LG2, a grease that is free from such drawbacks and yet emits very few particles.⁷⁾ As shown in Fig. 20, a linear guide employing a combination of LG2 grease and K1 Seals emits particles at almost the same rate as one utilizing vacuum grease.

In clean environments and in many other applications as well, linear guides are required to have high corrosion resistance. To meet such requirements, rails and ball slides made of martensite stainless steel are available for all models in the LS, LE and LU Series and some models in the LH Series. Their performance and characteristics such as rated load are equivalent to the standard series.

For applications requiring higher corrosion resistance or reduced reflection of light, linear guides are surface-treated. Considering corrosion resistance, durability and cost, NSK recommends an electrolytic anti-corrosion black chrome coating for such applications. An additional low-temperature fluorinated chrome plating (fluororesin coating) over this black chrome coating can further increase resistance to corrosion and chemicals, and is, for this reason, being used more frequently.

4.3 Reduced size

There has been a marked trend toward smaller information and office automation equipment.

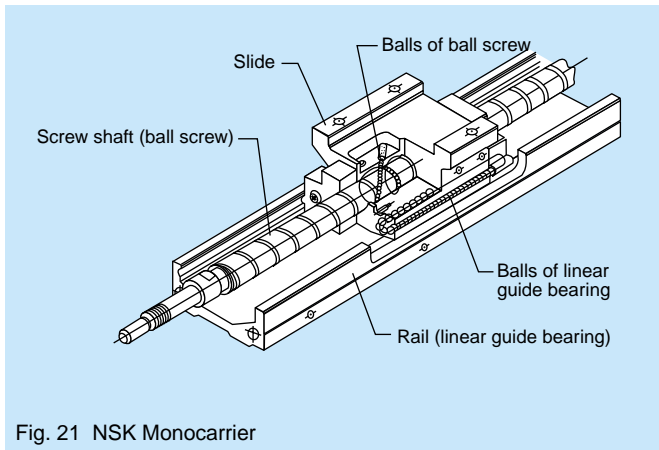


Fig. 21 NSK Monocarrier

Additionally, semiconductor manufacturing equipment needs to be composed of smaller components and parts in consideration of the cost of maintaining clean rooms. To meet such requirements, NSK has been working on further reducing the size of its LU and LE Series of miniature linear guides. The LU05 has a rail 5 mm wide and an overall height of 6 mm, and LE05 has a 10-mm-wide rail and an overall height of 6.5 mm. The wide rail of LE-series linear guides enables them to support high moment loads even when used on a single rail. As a result, they are particularly conducive to saving space.

In addition to reducing the size of individual elements, integrating the component parts of entire units is being actively pursued. The NSK Monocarrier™, shown in Fig. 21, is one such integrated unit that comprises a ball screw and linear guide. Not only is the unit small but the simplicity of its construction reduces design and assembly work.

4.4 Interchangeable linear guide series

Highly precise positioning and machining of Gothic-arch raceway grooves enabled NSK to be the first to develop and market an interchangeable linear guide series. In such a series, rails and ball slides are produced and stored separately, and then matched at random as needed. Interchangeable series contribute to customer satisfaction in two ways:

- Delivery time is reduced by maintaining a standard stock of rails and bearings.
- Adding or replacing ball slides is easier.

Initially, the LH and LS Series were made interchangeable, followed recently by the LU, LE and LW Series in response to market demand.⁹⁾ Making these series interchangeable has increased their overall utility.

5. Conclusion

To meet the demands of the market, NSK has developed linear guides whose fundamental design and characteristics satisfy the requirements of specific applications. Some of them have been presented above. NSK will continue its development of linear guides that meet requirements and offer utility for an expanding array of applications.

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Shiroji Yabe

Rear-wheel Bearings for Fully Floating Axles—Analysis of Load and Optimum-strength Design

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ABSTRACT

The load of rear-wheel bearings for fully floating axles has been determined in the past by simple geometric calculations. In order to achieve an optimum-strength design, we measured bearing load while a vehicle was running and compared it to the calculated one. In our comparison, we found several differences between collected data and calculated values. We determined the causes of the differences and, considering these causes, developed both a simple way to estimate the bearing load and a procedure for optimum bearing design. Finally, to verify the effectiveness of the procedure, we designed a bearing using it and performed various analyses using measurements from actual vehicles.

1. Introduction

With the trend toward increasing the cargo capacity of trucks while reducing the weight and size of their components, wheel bearings are subjected to increasingly severe loads. To develop lightweight, low-cost wheel bearings, a precise understanding of bearing load is necessary. In the past, bearing load was determined using a simple geometric calculation based on calculated input load at the tire. However, for optimum wheel bearing design, reexamining the method for calculating wheel

bearing load has become necessary. The goal of our research was to develop a procedure for achieving optimum wheel bearing designs. To do this, we used rear-wheel bearings in fully floating systems of small trucks operating in extremely severe conditions as a model.

In this report, we first analyze the difference between conventionally calculated load and actual load as measured by NSK's own method¹⁾, then present a new bearing load calculation method and design flow for optimum bearings, and finally, introduce a new, lightweight and compact bearing design that is easier to maintain.

2. Process of Research and Development

Table 1 shows the process of research and development.

To review the optimum design, the actual bearing load during operation of a 4-ton truck was measured and compared with the conventionally calculated bearing load. The causes of the differences between actual and calculated loads were determined. Verification with a 2-ton truck was carried out and then a method for calculating the bearing load more accurately was developed. An optimum design flow was proposed. Then, problems with existing bearings for trucks were resolved and, focusing on reducing size and weight, bearings were

Table 1 Research and development process

I. Review of the optimum bearing design	Comparison of actual load measurement and conventional calculated value for a 4-ton truck
	Determination of factors affecting bearing load
	Verification with 2-ton truck and prediction of input load
	Proposal of optimum bearing design flow
II. Verification of the optimum bearing design	Following the optimum design flow, determination of bearing loads
	Study of existing problems of bearings for trucks
	Resolution of problems and improvement of design
	Prediction of the effect of optimum design flow and confirmation through experiments

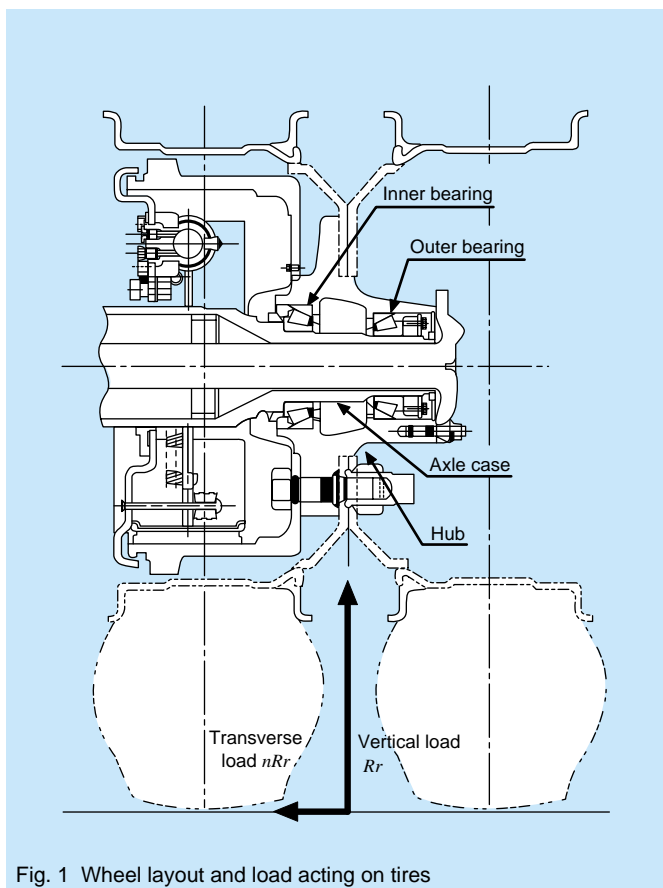


Fig. 1 Wheel layout and load acting on tires

produced and their performance verified for a model of a 4-ton truck.

3. Measurement Method of Actual Bearing Load

3.1 Actual bearing load measurement method

The layout and applied load of wheels for general small truck rear wheels are shown in Fig. 1. The load applied to a wheel bearing is distributed as shown in Fig. 2. By measuring the load of rolling elements, $Qe(j)$, at respective rolling element positions, the bearing radial load (Fr) and axial load (Fa) can be obtained by equations (1) and (2).

$$Fr = \sum_{j=1}^Z (\dot{Q}e(j) \cdot \cos \alpha \cdot \cos \phi(j)) \tag{1}$$

$$Fa = \sum_{j=1}^Z (\dot{Q}e(j) \cdot \sin \alpha) \tag{2}$$

where Z : Number of rolling elements, α : Contact angle.

The measurement of rolling element loads, $Qe(j)$, is obtained by converting the strain caused by the passage of rolling elements. Strain gauges are located in the bottom of a groove in the outer ring as shown in Fig. 3. The strain gauges are calibrated by rotating the outer ring while a known load is applied.

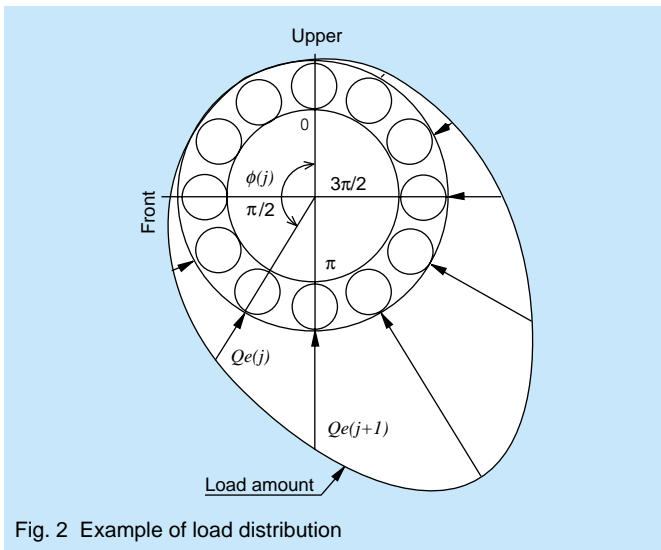


Fig. 2 Example of load distribution

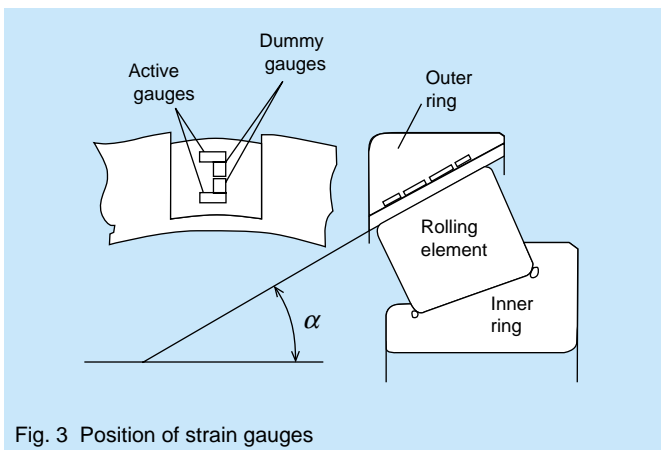


Fig. 3 Position of strain gauges

3.2 Past bearing load calculation method

• Vertical tire load, lateral tire load:

Fig. 4 shows the loads acting on the rear wheels when a vehicle is cornering about a radius, R , on a road inclined laterally. Vertical loads, Rro and Rri , and lateral loads, $nRro$ and $nRri$, acting on the tires, are given by the equilibrium equations (3) and (4) below.

$$Rr = \frac{1}{2} (Wr \cdot \cos \phi + FG \cdot \sin \phi) \pm \frac{h}{tr} (FG \cdot \cos \phi - Wr \cdot \sin \phi) \tag{3}$$

where the positive sign is for Rro and the negative sign is for Rri .

• Bearing load:

$$nRr = Rr \cdot \frac{FG}{Wr} \cdot \cos \phi \tag{4}$$

Fig. 5 shows the loads applied to the bearing. Bearing radial loads are calculated using Equations (5) and (6).

$$Fr_{IN} = \frac{Rro \cdot SM}{SL} \pm \frac{r \cdot nRro}{SL} \tag{5}$$

$$Fr_{OUT} = \frac{Rro \cdot SN}{SL} \mp \frac{r \cdot nRro}{SL} \tag{6}$$

where the positive signs are for the outside cornering wheel and the negative signs are for the inside cornering wheel.

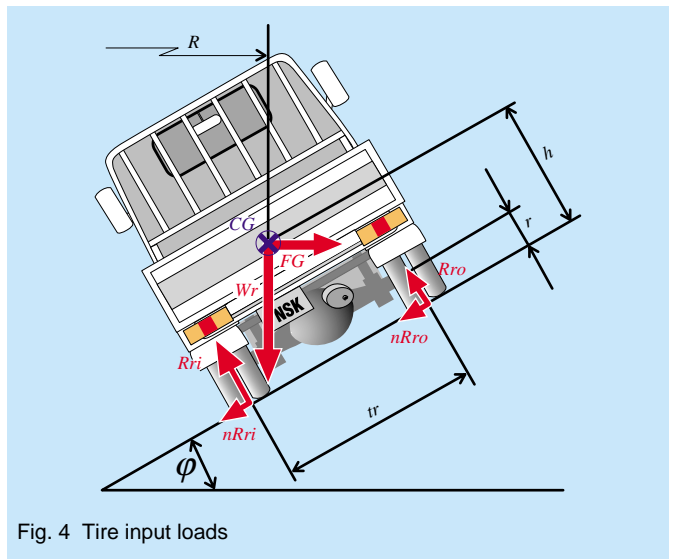


Fig. 4 Tire input loads

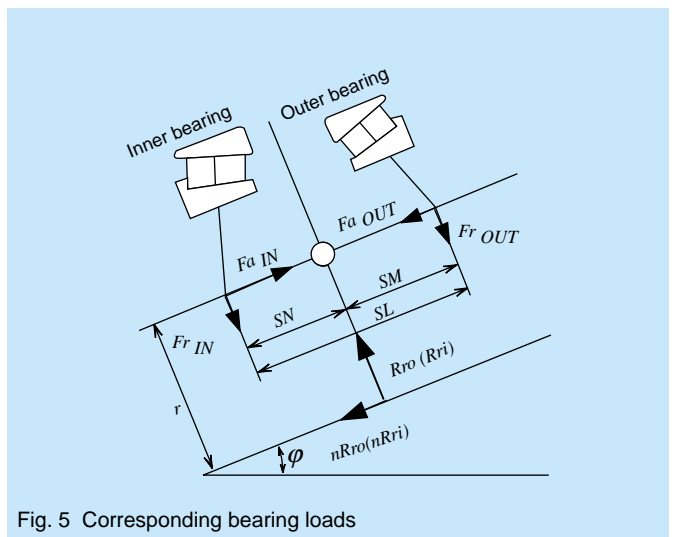


Fig. 5 Corresponding bearing loads

4. Comparison of Measured Load and Calculated Load

4.1 Running conditions during measurements

The running conditions during measurements were:

- (1) Straight: 5, 50, 70, 90 km/h
- (2) 20R cornering: 5, 10, 20, 30 km/h (0.01G ~ 0.35G)

4.2 Comparison between measured and calculated results

Past calculation and measurement results on a 4-ton and 2-ton truck when running on a straight road at 90 km/h and cornering at 0.35G are shown in Fig. 6. Three major differences exist between the calculated and measured values:

- The measured value for vertical tire load during turning is 5 - 10% higher than the calculated value.
- Distribution of radial load during turning is offset depending on the inner bearing.
- The measured axial load during straight travel is much higher than the calculated value.

These differences substantially reduce the accuracy of calculated bearing life.

4.3 Comparison with the simulation calculation

Vertical tire load is 5 - 10% higher than the calculated values when cornering. The major cause of this is the displacement of the vehicle's center of gravity during cornering. Fig. 7 and Fig. 8 show comparisons between the results of two-dimensional sprung-mass simulation calculations and measurements. These calculations can take into account the displacement of the center of gravity that occurs as the vehicle turns. Fig. 7 shows that the simulation calculation and measurement results on displacement of the center of gravity agree well. Fig. 8 shows that the simulation calculations are closer to the measured values than the conventionally calculated ones concerning the vertical load of the tires. The data presented in these figures confirms the validity of the simulation calculation method.

4.4 Calculation of lateral tire load and tire load center when vehicle is running

Measured and conventionally calculated lateral tire load and tire load center values during cornering acceleration

Table 2 Tire cornering force and tire load center

Measured result and calculated result for 4-ton truck		Actual bearing load	Conventional geometric calculation	Calculated by tire characteristics
Lateral load acting on tires (kN)	Outer two wheels during cornering	8.0	11.1	—
	Inner two wheels during cornering	-3.6	-4.4	—
Tire load center (running)		20.0 mm inner side	0 Assumed as a center position	20.8 mm inner side
Measured result and calculated result for 2-ton truck		Actual bearing load	Conventional geometric calculation	Calculated by tire characteristics
Lateral load acting on tires (kN)	Outer two wheels during cornering	4.5	6.2	—
	Inner two wheels during cornering	-2.2	-2.5	—
Tire load center (running)		10.7 mm inner side	0 Assumed as a center position	12.9 mm inner side

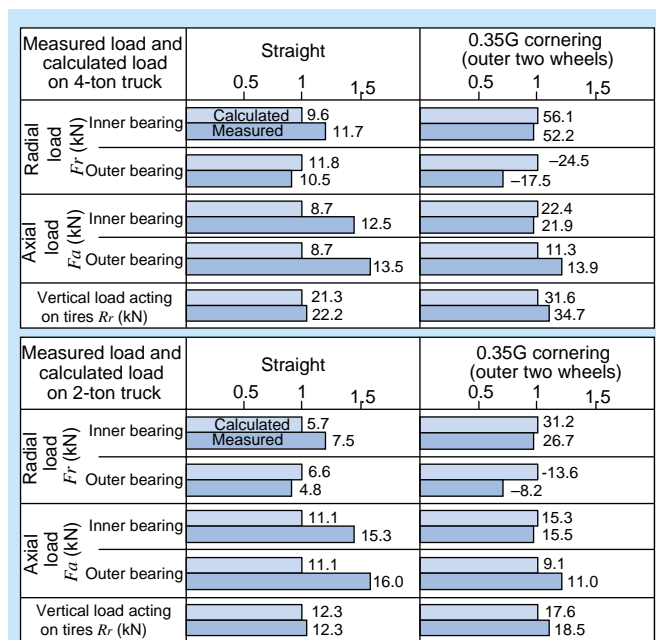


Fig. 6 Measured load and calculated load

of 0.35G are compared in Table 2. Tire load center values obtained using the lateral spring coefficients of the tested tires are also shown. The tires used were 7.50R16-12PR for the 4-ton truck and 195-85R16 for the 2-ton truck. Measured values of lateral tire load are slightly lower than those conventionally calculated. This is assumed to be caused by tire slippage. As for the tire load center position, the values obtained using the lateral spring coefficients of the tires are close enough to the measured values to verify the reliability of this calculation method.

4.5 Generation of bearing internal axial load due to temperature distribution

When a temperature difference occurs between the inside and outside of a bearing, an internal axial load, ΔFa , develops in the following way:

1. Heat is generated and dissipated from inside the bearing
2. A difference in temperature between the inside and outside of the bearing occurs
3. In both the fixed (inner) ring and rotating (outer) ring
 - Thermal expansion of the bearing and its periphery occurs in the axial direction
 - Thermal expansion of the bearing raceway surfaces occurs in the radial direction

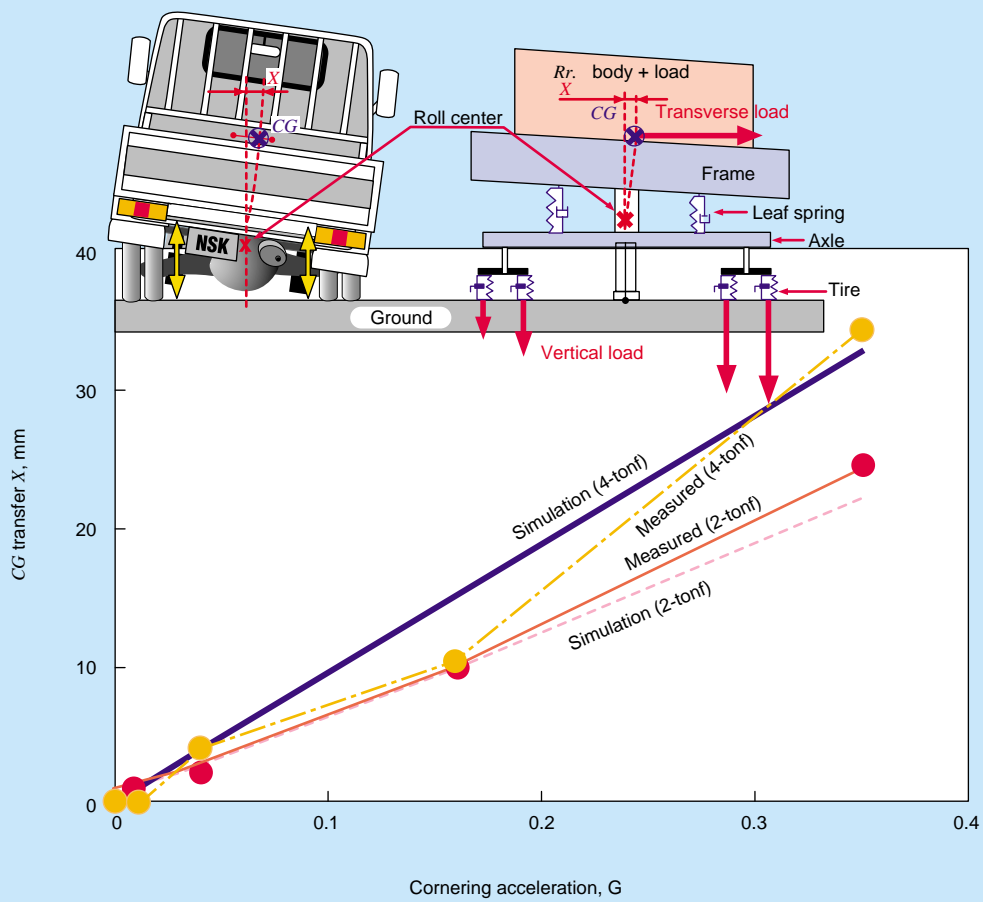


Fig. 7 CG transfer during cornering

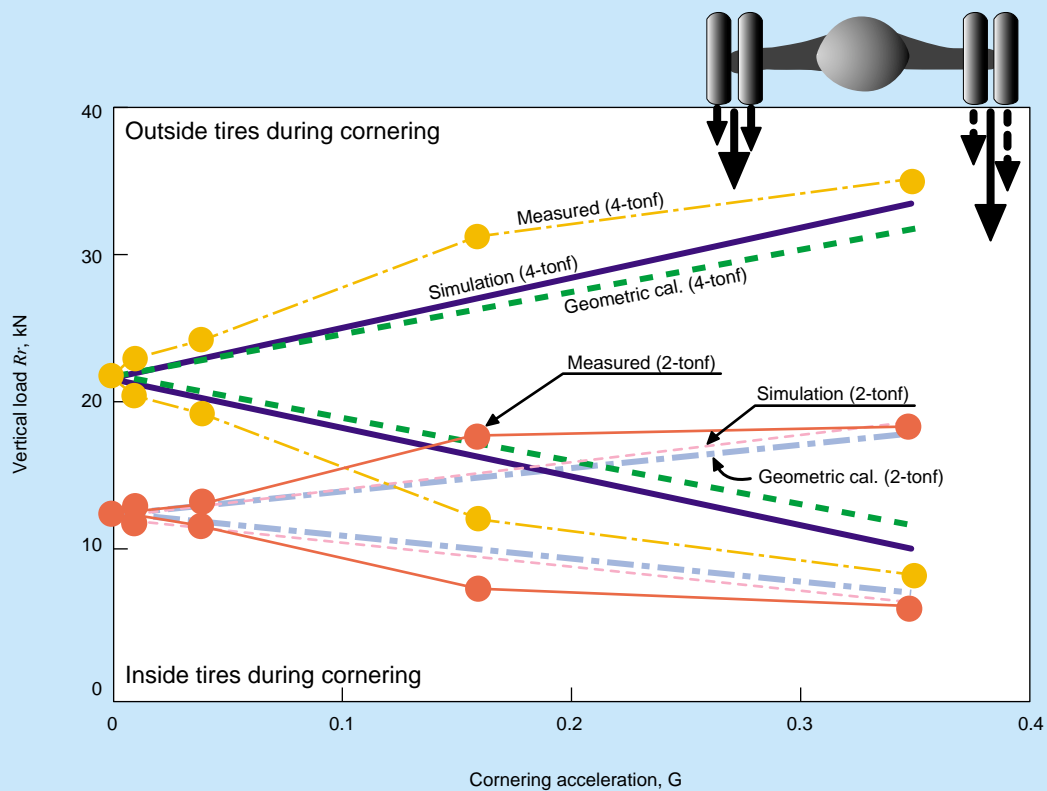


Fig. 8 Vertical load during cornering

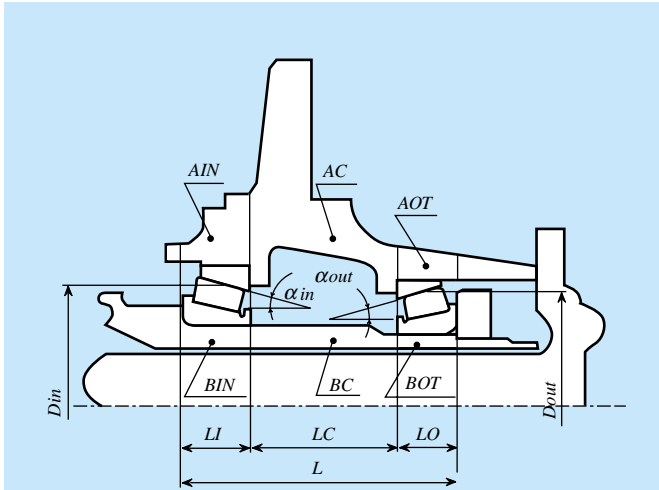


Fig. 9 Sketch of axle hub

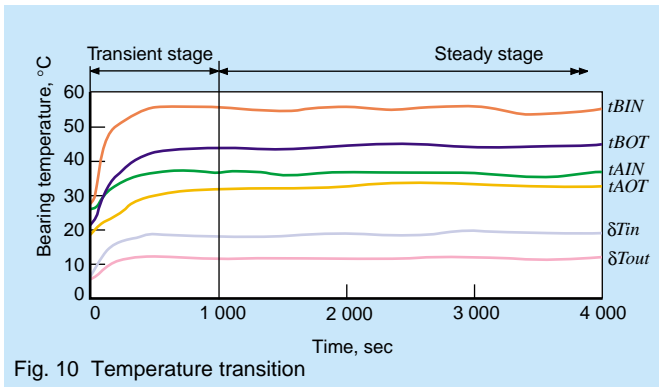


Fig. 10 Temperature transition

4. A difference in axial displacement develops between the fixed ring and rotating ring
5. Axial load, ΔFa , is generated

ΔFa is generated due to differences in axial displacement caused by radial and axial thermal expansion in the bearing. In the simplified method of calculating ΔFa shown below, the axle hub is divided into three parts in the axial direction, each with its own respective temperature (see Fig. 9).

- Axial displacement due to thermal axial expansion, $\Delta\delta a$

$$\Delta\delta a = \frac{(t_{BIN}-t_{AIN}) \cdot \xi_{Brg} \cdot \frac{D_{in}}{2}}{\tan \alpha_{in}} + \frac{(t_{BOT}-t_{AOT}) \cdot \xi_{Brg} \cdot \frac{D_{out}}{2}}{\tan \alpha_{out}} \quad (7)$$

- Axial displacement caused by thermal expansion in the axial direction, ΔL

$$\Delta L = (t_{BIN}-t_{AIN}) \cdot \xi_{Brg} \cdot LI + (t_{BOT}-t_{AOT}) \cdot \xi_{Brg} \cdot LO + (t_{AC} \cdot \xi_{Axle} - t_{BC} \cdot \xi_{Hub}) \cdot LC \quad (8)$$

- Combined axial displacement caused by thermal expansion of the bearing, Δa

$$\Delta a = \Delta\delta a - \Delta L \quad (9)$$

Where ξ_{Brg} , ξ_{Hub} , ξ_{Axle} : Linear thermal expansion

Measured load and calculated load on 4-ton truck	50 km/h straight			90 km/h straight		
	0.5	1	1.5	0.5	1	1.5
Bearing axial load F_r (kN)	ΔFa calculated					
	Measured					
L.H.	8.7	13.6	4.8	8.7	13.6	6.0
R.H.	13.7	13.7	5.0	8.7	15.0	6.2
Measured load and calculated load on 2-ton truck						
Bearing axial load F_r (kN)	50 km/h straight			90 km/h straight		
	0.5	1	1.5	0.5	1	1.5
L.H.	13.7	17.9	4.0	13.7	19.4	5.5
R.H.	11.1	14.5	3.2	11.1	15.7	4.5

Fig. 11 Increase in axial load due to temperature difference between inner and outer rings

For example, in t_{AIN} , A indicates that the temperature is for portion A , and IN indicates the inner bearing side. OT indicates outer bearing temperatures. Consequently, if the relation between displacement and axial load is known beforehand, ΔFa can be obtained.

Fig. 10 shows the temperature transition of various points in the bearing arrangement of a 4-ton truck when it ran at a speed of 90 km/h (outside air temperature 5°C). The temperature of the various points passes through the transient stage quickly to equilibrium conditions. When considering endurance life, since the bearing reaches equilibrium temperature quickly and remains there, it is sufficient to consider the temperature difference only at equilibrium. Fig. 11 shows the calculated value of ΔFa caused by the temperature difference between the bearing inner and outer rings. The fact that ΔFa was not considered in past geometrical calculations accounts for these calculations being different from measured axial load values. The result obtained by adding ΔFa to the geometrical calculation matches the measured axial load well.

5. Consideration of Predicted Input Load and Bearing Life

5.1 Estimation of input load

To more accurately estimate bearing life, we must precisely calculate the radial load (F_r) and axial load (F_a) to be applied to the bearing. To do this, it is necessary to consider the following items:

1. Vertical tire load - take into account the displacement of the center of gravity when the vehicle is cornering
2. Lateral tire load - take into account tire slippage when the vehicle is cornering
3. Changes in the position of the center of the applied load caused by lateral distortion of the tire
4. Bearing internal axial load caused by temperature differences in the bearing and surrounding parts

5.2 Method of estimation for the above items

1. Vertical tire load

We developed a two-dimensional sprung-mass

simulation model using commercially available kinematic analysis software in order to consider the following:

- Roll about roll center
- Roll caused by the difference of lateral strain in the right and left leaf springs
- Roll caused by the difference of lateral strain in the right and left tires

As for the model, constraining conditions, and method used to apply load, refer to Fig. 7.

2. Lateral tire load

The calculation method used for lateral tire load is the same as the past calculation method, i.e.,

$$\text{Lateral load} = \text{Cornering } G \times \text{Vertical load.}$$

In this study, we considered the gripping function of the tire and sought the cornering force as a function of the tire slipping angle. We compared measured values and past calculation values. As a correlation was not clear, we left this problem for future study.

3. Center of applied load position

As the value obtained from the tire lateral spring coefficient is nearly equal to the measured value we

$$\text{Center of applied load position} = \frac{\text{Lateral strain amount of tire}}{\text{Tire lateral spring coefficient}} - \frac{\text{Lateral load}}{\text{Tire lateral spring coefficient}}$$

deemed it possible to calculate using:

4. Bearing internal axial load caused by temperature distribution in the bearing

The bearing internal axial load, ΔFa , can be calculated from the temperature difference between the bearing inner

and outer rings and that between the axle case and hub, δT . It became clear that the measured δT in this study (of 2-ton and 4-ton trucks) correlates with bearing PV values,²⁾ as shown in Fig. 12. The difference in the PV values - δT gradient of the three items in the figure is caused by a difference in the state of heat dissipation.

As to the structure around the bearing, the material characteristics (linear thermal expansion coefficient and coefficient of thermal conductivity) are the same as the specimens in this study. When the PV value is calculated, δT of other parts can also be calculated. Here, the P value indicates Hertzian surface pressure, and the V value indicates peripheral speed of the roller end face in contact with the bearing inner ring rib.

For the above case and hub, as there is no contact point, the PV values cannot be calculated but are derived based on the PV values of the inner bearing since a correlation with the bearing temperature difference was observed. Fig. 12 is based on PV values obtained purely by calculation.

δT of all parts \rightarrow axial displacement amount, $\Delta a \rightarrow \Delta Fa$ can be calculated

5.3 Calculation of bearing radial load- Fr and axial load- Fa

We calculated the bearing load using the tire vertical load (Rr), lateral load (nRr), and load application center position, as defined by SM and SN .

(1) Bearing radial load, Fr : calculated as described in section 3.2

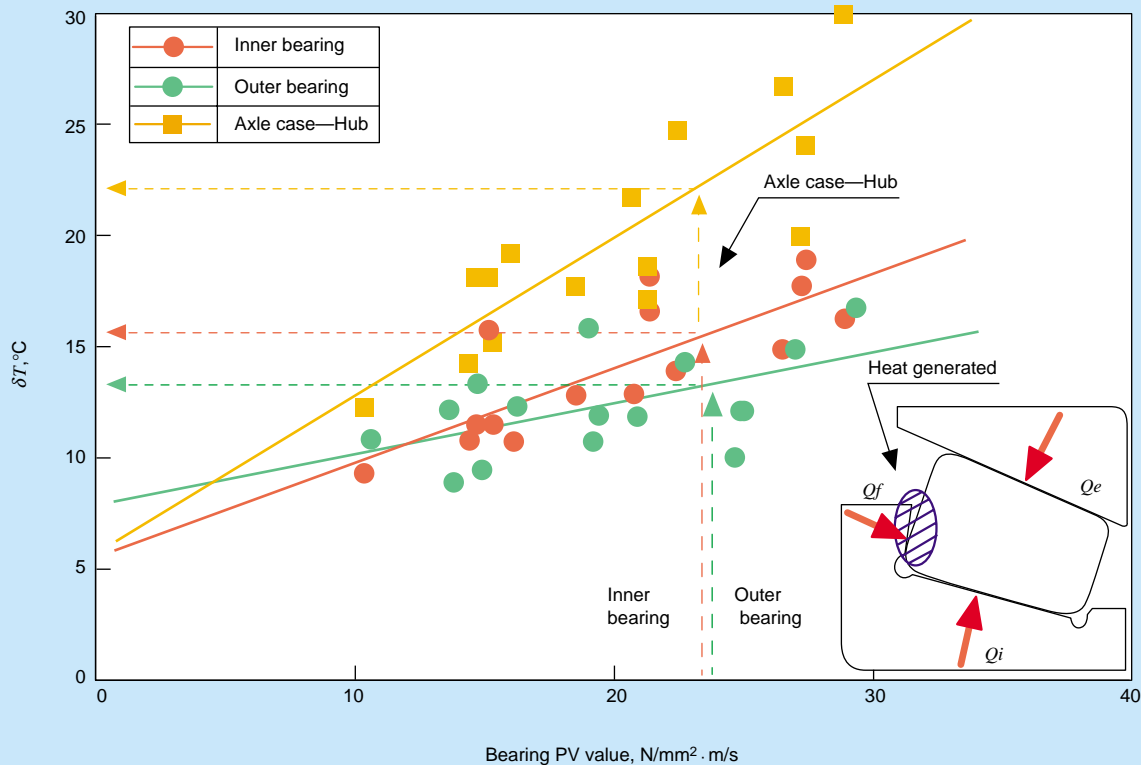


Fig. 12 Temperature difference δT with PV value

Table 3 Objectives and solutions of optimum bearing design

	Objective	Solution
A	Optimum balance between life of inner and outer bearings	<ul style="list-style-type: none"> • Outer-row center inside tire load center • Increased load capacity of outer bearing
B	Reduced sensitivity to preload	<ul style="list-style-type: none"> • Better preload control with inner clearance of double-row bearing • Increased load capacity of outer bearing
C	Lightweight and compact design	<ul style="list-style-type: none"> • Reduced axial length of axle shaft, case, and hub with double-row construction
D	Improved serviceability	<ul style="list-style-type: none"> • Separable for easy grease replenishment
E	Appropriate bearing rigidity	<ul style="list-style-type: none"> • Contact angle same as current bearing with successful record

(2) Bearing axial load, F_a : Calculated using the equation developed by Lundberg and Palmgren.³⁾

5.4 Calculation of bearing life

Bearing life can be calculated using this formula:⁴⁾

$$L = (C/P)^{10/3} \tag{10}$$

where **L**: Rating fatigue life of reliability factor 90%
C: Basic dynamic rating load
P: Bearing load

The bearing life of the 4-ton truck under an operating pattern of 90% straight running and 5% cornering in each direction is shown in Fig. 13.

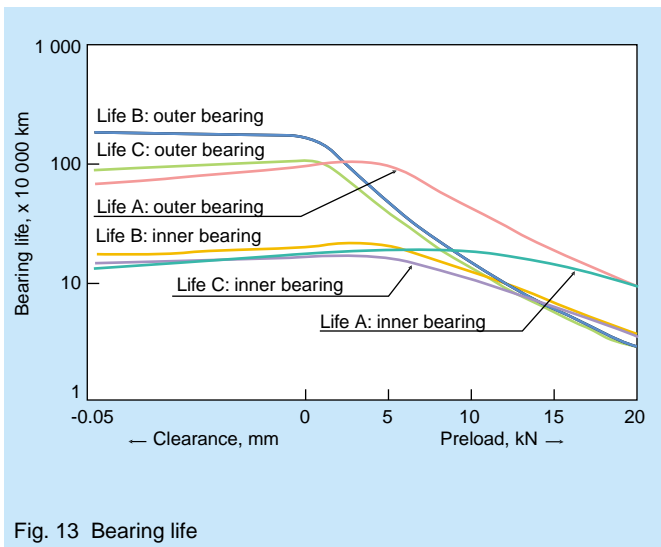


Fig. 13 Bearing life

The different bearing lives in Fig. 13 were calculated using the following three methods:

- Bearing life A - Calculated with the past calculation load value
- Bearing life B - Calculated with the measured load value
- Bearing life C - Calculated with the load value obtained by the method described in this study

The validity of the method described in this study is demonstrated by the fact that the bearing life obtained using it is very close to the bearing life obtained using the measured load values.

6. Optimum Design Procedure for Bearings in Fully Floating Systems

6.1 Review of the optimum design

6.1.1 Existing design and problems

Fig. 15b shows the existing bearing design for fully floating axle systems. In this design, the inner and outer bearings are located on opposite sides of the tire load center, and a specified preload value is set by tightening the hub nut. An operating pattern of 90% straight, 5% right turn at 0.35G, and 5% left turn at 0.35G is assumed. As damage to the outer bearing can cause the axle to break while damage to the inner bearing has less severe consequences, the life of the inner bearing is shorter than the outer one at the set preload under the assumed operating pattern. The shorter life of the inner bearing enables detection, in the form of abnormal vehicle vibration, of bearing deterioration before any serious damage can occur. However, excessive tightening of the hub nut increases the preload resulting in a sudden decrease in the life and load capacity of the outer bearing.

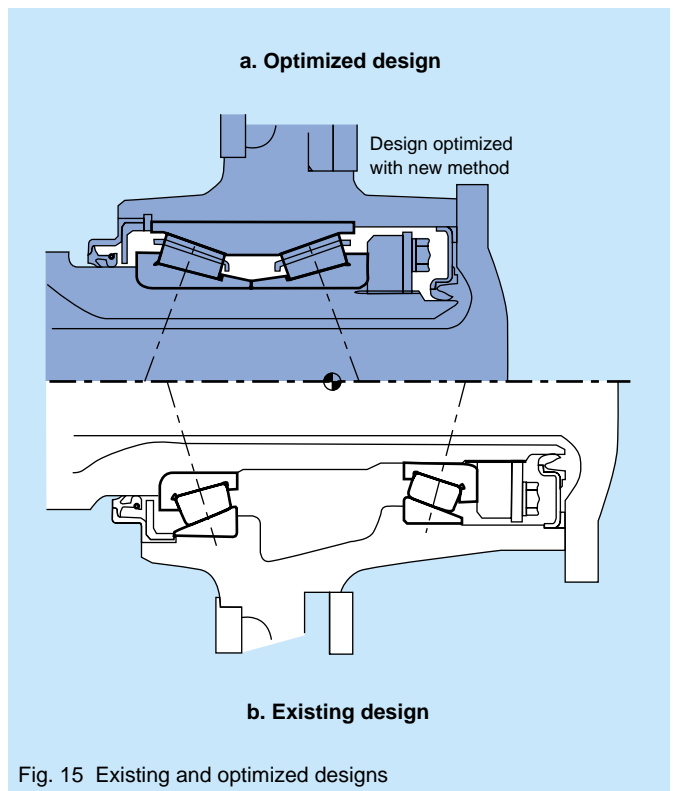


Fig. 15 Existing and optimized designs

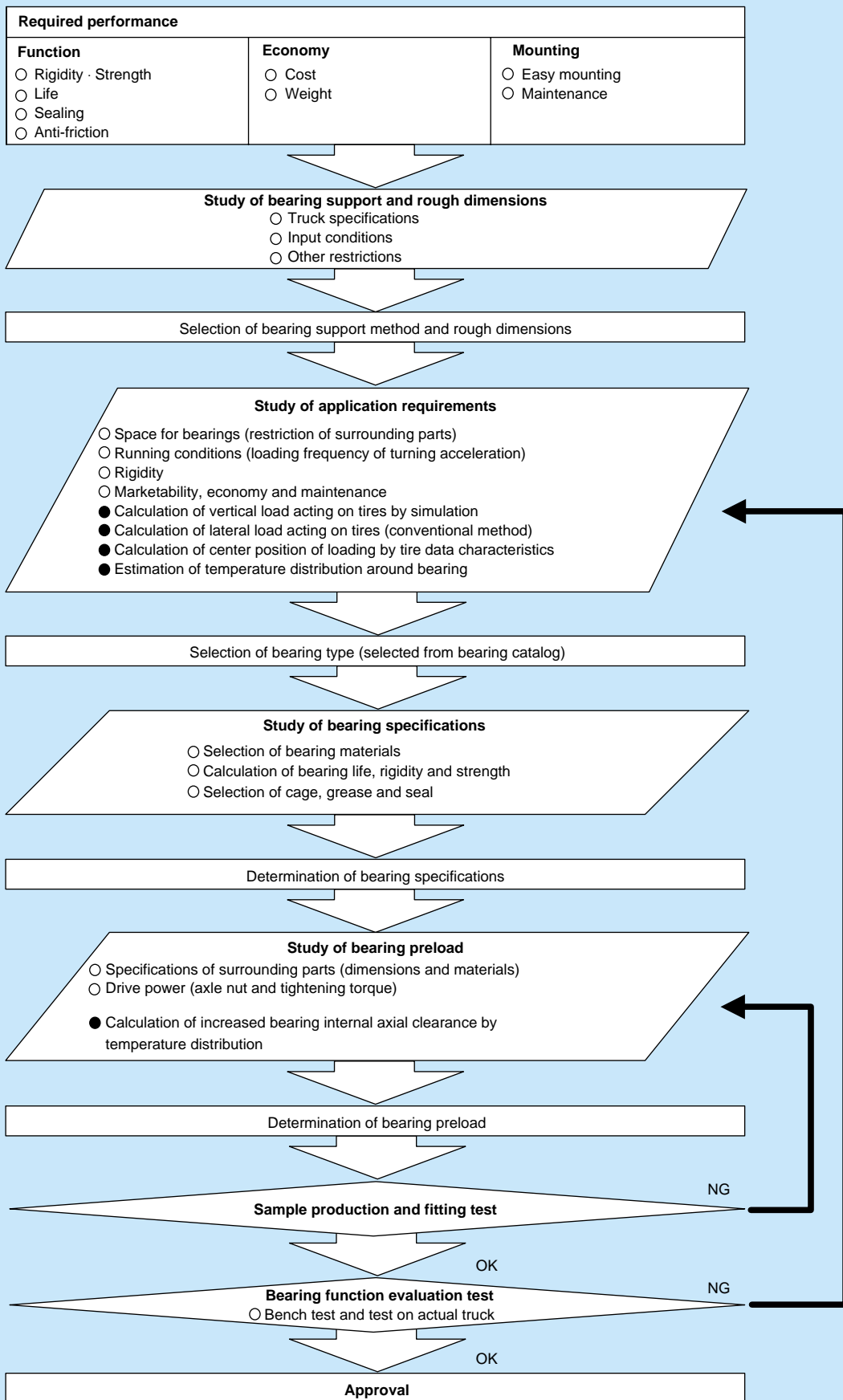


Fig. 14 Design flow chart: bearings for rear wheels of light trucks

Current problems are as follows:

- The outer bearing is sensitive to the preload.
- Under normal preload, the life of the inner bearing tends to be much shorter than that of the outer bearing.

6.1.2 Optimum bearing design flow and design concept

Based on the analysis results of the 4-ton and 2-ton trucks, we have developed the flow chart in Fig. 14 for the optimum design of bearings for the rear wheels (tandem-wheels) of small trucks. By calculating both the displacement of the center of gravity when running and the position of the center of the applied tire load, and estimating temperature distribution with the method described in this study, it becomes possible to determine bearing load more accurately. A corresponding improvement in the precision of the bearing design can be expected.

Table 3 presents objectives and solutions for creating optimum bearings for the 4-ton truck. The resulting optimized design (Fig. 15a) has a double-row bearing with the associated increase in load capacity of the outer row and the optimum bearing arrangement (i.e., the outer-row center is inside of the tire load center). The weight of the bearing arrangement is reduced by 1.2 kg, thereby improving serviceability, and the width of the wheels is shortened by 80 mm. Better control of the preload is ensured by the double-row bearing.

6.2 Life estimation of the optimum product

6.2.1 Estimation of bearing life

The life of the newly designed bearing is compared with that of the existing bearing in Fig. 16. As a result of optimizing the bearing arrangement, the shortest life of the inner row was increased by 30%. Moreover, the increased load capacity of the outer row resulting from the double-row construction makes the bearing arrangement less sensitive to variations in clearance and preload.

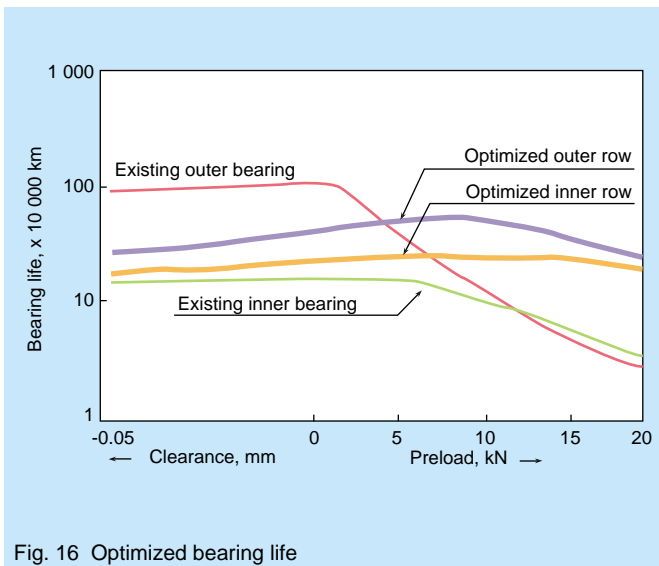


Fig. 16 Optimized bearing life

Measured load and calculated load on 4-ton truck	Straight			0.35G Cornering (outer two wheels)		
	0.5	1	1.5	0.5	1	1.5
Radial load F_r (kN)	Inner bearing			Outer bearing		
	Calculated 2.5	19.3	16.4	55.9	-22.7	-23.5
Axial load F_u (kN)	Inner bearing			Outer bearing		
	Measured 5.4	17.9	18.0	26.6	27.5	15.0
Vertical load acting on tires R_r (kN)	Inner bearing			Outer bearing		
	17.9	18.0	18.0	15.0	15.8	15.8
Transverse load acting on tires R_r (kN)	Inner bearing			Outer bearing		
	21.8	21.8	21.8	33.2	34.7	34.7
Vertical load acting on tires R_r (kN)	Inner bearing			Outer bearing		
	0	0	0	11.6	11.7	11.7
Transverse load acting on tires R_r (kN)	Inner bearing			Outer bearing		
	0	0	0	11.6	11.7	11.7

Fig. 17 Measured and calculated load of optimized bearings

6.2.2 Comparison between calculated and measured values

Fig. 17 shows calculated and measured values of the new bearing's load conditions during 90 km/h straight running and turning at 0.35G. Good agreement is evident between calculated and measured values. The radial load of the inner row during straight running is more than double that when turning, but this difference may actually be smaller because the values are all absolute.

6.3 Evaluation of new bearing design

The new bearing design meets the necessary quality requirements as shown in the evaluation presented in Table 4.

7. Conclusion

In this report, we have presented analysis of bearing load, proposed a design flow for optimum bearings and introduced a new bearing design that solves the problems of existing bearings, namely, the imbalance between inner and outer bearing life and the outer bearing being very sensitive to preload. Our conclusions can be summarized as follows:

1. We have developed lightweight, compact bearings with 30% longer life and improved serviceability.
2. To solve the problems of existing bearings, the outer and inner bearings were combined into a double-row type (hub unit). In the new arrangement, the outer-row center is inside the tire load center. This has improved the life balance between the inner and outer rows. The unit construction of the bearing suppresses increases in preload when the hub nut is excessively tightened.
3. Measured load conditions of the new bearing agreed well with calculated results. Original quality and performance objectives were achieved.
4. The new bearing is 1.2 kg lighter and the width of the bearing arrangement has been reduced by 80 mm for 4-ton wheels.

Table 4 Evaluation of new bearing design

		Target	Evaluated item	Result	Judgment
Performance	Axial load and starting torque	Understanding of variation of axial load during tightening	Fitting test	Sudden preload increase due to excessive tightening of hub nut is limited	○
	Temperature and bearing load measurement	Increase in axial load due to internal temperature difference during straight running	Measurement of temperature during straight running of area surrounding bearing	Equivalent to present product	○
		Thermal effect of engaging brake	Bearing temperature during braking	Equivalent to present product	○
Durability	Operating pattern durability	Comparison with standard life	Straight: 90% Right/left turn at 0.35G: 5% ea.	Exceeds customer specifications	○
	Seizure resistance	Check for seizure under excessive tightening of hub nut	No seizure at twice the proper preload	Exceeds customer specifications	○

8. Final Acknowledgment

In our four years of studying bearings for the rear wheels of light trucks, we clarified various previously ignored factors that affect the kinetic behavior and performance of the bearings. As a result, review of more appropriate bearing construction is now possible. In the future, we will proceed with product development utilizing what we have achieved up to now.

The major part of this study was presented jointly with Isuzu Automobiles in four parts at Automobile Technology Meetings from 1995 - 1998.⁵⁾⁻⁸⁾ This paper was also presented at the 1998 meeting of the Society of Automotive Engineers.⁹⁾



Satoshi Hiraki

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Rolling Bearings for Hard Disk Spindle Motors

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ABSTRACT

For many years, NSK has been producing rolling bearings for that most vital of tools for the modern world, the computer. With the unremitting growth in personal computers, minimizing the size of hard disk drives while maximizing their capacity has been and will continue to be a primary focus. Rather than their usual role of transferring force, the function of ball bearings in computers is supporting the accurate transfer of information. The unique requirements of this function have necessitated specialized bearing designs, greases, seals and cages. As a result of continuous improvement of ball bearings for computers, their performance can now be discussed on the order of nanometers (a nanometer, nm, is 1/1000 of a micrometer, μm). This report describes the current situation of NSK ball bearings for applications in computers.

1. Introduction

The ongoing popularization of computers is truly amazing. One computer per person is now the norm in offices and the development of the Internet has accelerated the spread of computers into households. Partly because software has steadily become more sophisticated and partly because processing audio and video data requires greater capacity, current hard-disk drives have a few gigabytes (GB) of memory, compared to just several megabytes (MB) only a few years ago. An HDD and bearings are shown in Photo 1.

In spite of the increasing memory capacity of HDDs, their size remains unchanged or, as in the case of 2.5" HDDs, has actually gotten smaller. The memory capacity of 3.5" magnetic disks is generally about 2 or 3 GB at present, but is expected to increase to about 5 to 7 GB by the year 2000 as a result of increasing packing density of the recording surface, improved magnetic heads, and the steady reduction of NRRO (non-repetitive runout: runout components that are not synchronous with rotation).

As mentioned above, bearings for spindle motors are required to support the accurate transfer of information rather than force. For this purpose, primary importance is placed not on load-carrying capability, but on accurate rotation without irregular motion throughout the entire service life of the bearing.



Photo 1 HDD and bearings

Based on the higher-than-10% growth rate in HDD production over the past few years, it is projected that over 140 million units will be produced throughout the world in 1998 (see Fig. 1). More than fifty percent of these will contain NSK bearings. To meet this strong demand and fulfill the requirements of spindle motor manufacturers, NSK will continue to improve the performance of its spindle motor bearings.

The following sections describe high-accuracy NSK rolling bearings for hard-disk spindle motors.

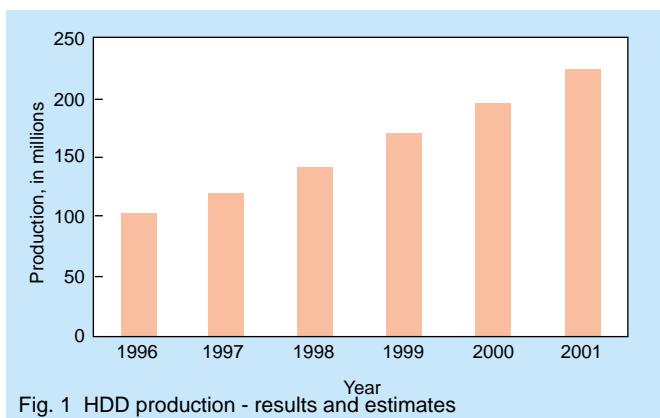


Fig. 1 HDD production - results and estimates

Table 1 Requirements of HDDs, spindle motors and the bearings they contain

HDD	HDD spindle motor	Rolling bearing
Small size	Thin motors	Small width
High density, High capacity	Low NRRO, High speed	Low NRRO, High speed
High reliability	Low contamination	Low outparticling
Low noise	Low noise	Low noise
Low power consumption	Low power consumption	Low torque
Low cost	Low cost	Low cost

2. Bearing Requirements

Table 1 illustrates how requirements for HDDs translate to requirements for HDD spindle motors and finally to requirements for spindle motor bearings. Thus, customer requirements for HDDs and related requirements for the motors they contain decide the requirements for bearings.

2.1 Dimensions of HDD spindle motor bearings

For small-size HDDs and thin HDD spindle motors, rolling bearings are required to have a small width and be available at low cost. For this type of application, we recommend choosing from the mass-produced, low-cost bearings listed in Table 2.

Some bearings in Table 2 have the same inside and outside diameters but have different ball diameters and ball quantities. They are provided as options that can be used to avoid resonance between the motor or HDD and the rolling vibration of the bearing. Regarding specific applications of the bearings in the table: 6-mm-inside diameter (I.D.) bearings are often used in high-end HDD spindle motors, 5-mm-I.D. bearings in desk-top 3.5-inch HDD spindle motors, and 4-mm-I.D. bearings in 2.5-inch HDD spindle motors.

2.2 NRRO of bearings

As the capacity of HDDs increases, the rolling bearings they contain are designed to have lower NRRO. Fig. 2 illustrates a method for measuring the NRRO of a single ball bearing. The ball bearing is mounted with an air bearing and driven by a DD motor. Using this method, the NRRO of a typical bearing for HDD spindles (the 5-mm-I.D. bearing 695A2 in Table 1) was measured. The distribution of the results is presented in Fig. 3. The average NRRO of the bearing was approximately 0.05 μm . Indicating the speed with which performance improvement has progressed, this value is nearly half the level of NRRO only four or five years ago. Considering that the molecular size of tobacco smoke is on the order of

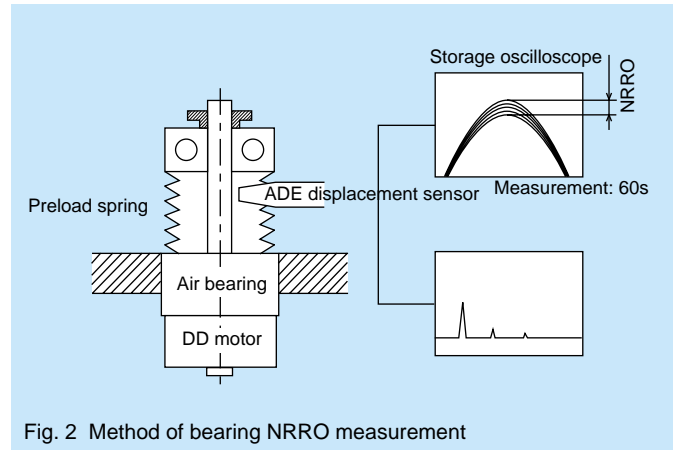


Fig. 2 Method of bearing NRRO measurement

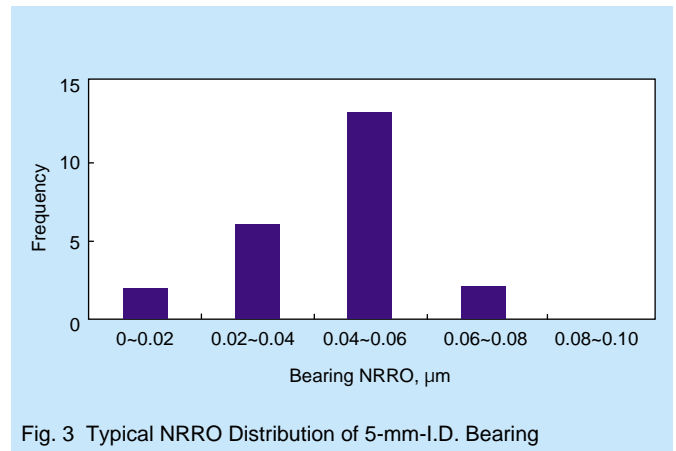


Fig. 3 Typical NRRO Distribution of 5-mm-I.D. Bearing

0.1 μm , the accuracy of this bearing is truly remarkable.

There are three main components of NRRO. Presented in order from largest to smallest as determined by frequency analysis, they are:

- ① vibration from rotation of the balls and cage around the bearing axis
 - ② vibration from rotation of the outer and inner rings
 - ③ vibration from the balls rotating about their axes
- Component ① is known to be caused by differences

Table 2 Dimensions of HDD spindle motor bearings

NSK Brg No.	Rubber seal	Mounted metal shield	d Bore dia.	D Outer dia.	B Width	BALL		Cr (N)	Cor (N)
						Dia.	Q'ty.		
696	2	2	6	15	5	7/64"	7	1,730	670
B6-84	2	2	6	15	4	3/32"	8	1,470	605
B6-91	2	2	6	15	4	2 mm	10	1,260	570
695	2	2	5	13	4	2 mm	8	1,080	430
695A2	2	2	5	13	4	1/16"	10	830	365
B5-39	1	1	5	13	3	2 mm	8	1,080	430
B5-43	2	2	5	13	3	1/16"	10	830	365
B5-44	1	1	5	13	3	2 mm	7	980	365
MR115B	2	2	5	11	4	1/16"	8	715	276
684A-A-1	2	2	4	9	4	1/16"	7	640	225
B4-50A	1	1	4	9	2.6	1/16"	7	640	225
B4-55		1	4	9	2.6	1.2 mm	10	500	209
B4-60		1	4	8	2	1 mm	10	360	148

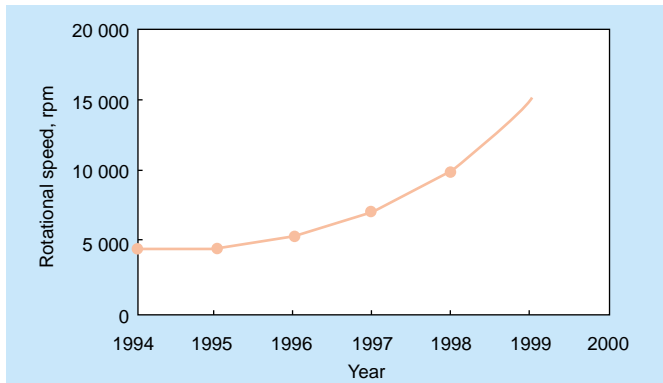


Fig. 4 Increase of HDD spindle motor speed

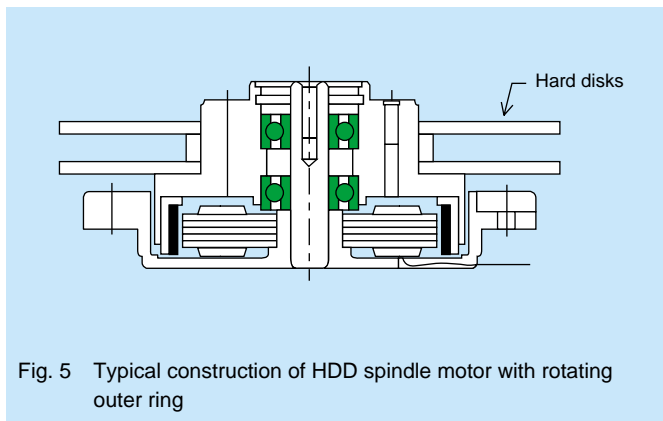


Fig. 5 Typical construction of HDD spindle motor with rotating outer ring

between ball diameters and the amount of play between the cage and balls. To further reduce NRRO, we are working on improving outer and inner raceway accuracy at the nanometer level, increasing ball accuracy, and developing optimal cage designs.

2.3 Higher-speed design and selection of bearing seal types

To help achieve faster HDD data transmission, the running speed of HDD spindle motors has been improved to 10 000 rpm (see Fig. 4).

Moreover, concurrent with downsizing of HDDs and the development of thinner spindle motors, rolling bearings with rotating outer rings have been increasingly used. Fig. 5 is an illustration of the typical construction of an HDD spindle motor with rotating outer ring.

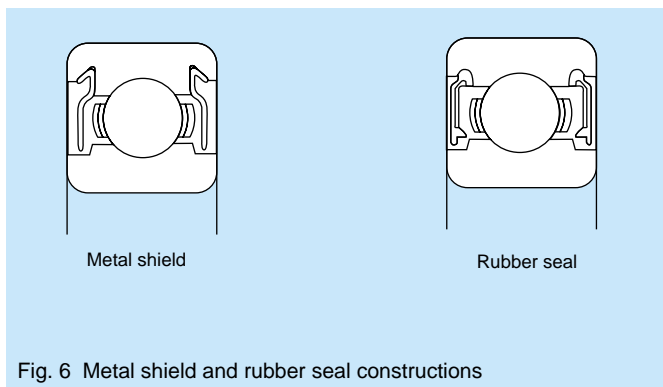


Fig. 6 Metal shield and rubber seal constructions

For motor bearings with rotating inner rings (mounted on a rotating motor spindle), metal shields have generally been used because they provide sufficient sealing at low cost. However, when the outer ring is rotated at high speeds, the tightness between the metal shield and groove in the outer ring is not sufficient to prevent grease leakage. As a result, the switch to outer-ring rotation for higher-speed operation has led to the use of rubber seals, whose greater flexibility and resilience ensure adequate sealing and minimization of grease leakage. In addition to their excellent sealing performance, rubber seals can contribute to limiting outgassing (particle emission) problems if they are made of an appropriate type of rubber. In fact, HDD bearings with rubber seals permit much less outgassing than general bearings.

Fig. 6 illustrates structural differences between the metal shield and rubber seal. In Fig. 7, the weight loss of bearings with metal shields is compared with that of those with rubber seals. The inside diameter of the tested bearings was 6 mm and their outer rings were rotated at 10 000 rpm. Figs. 8 and 9 show results of bearing noise endurance tests. These figures clearly indicate that decreasing bearing weight corresponds to reduced noise performance (i.e., more noise), and that rubber seals are superior to metal shields. It should be noted here that the difference in bearing noise performance is practically negligible for general-use bearings.

For HDD bearings, selecting a seal that ensures sealing on both sides is another important consideration. While conventional small-width bearings for thin spindle motors have a seal designed for sealing on only one side, a seal capable of sealing on both sides is recommended for HDD bearings whose outer rings are rotated at high speeds. Since the number of revolutions of the balls (i.e., running speed of the cage) in a bearing whose outer ring is rotated is higher than that of one with inner-ring rotation, the centrifugal force acting on the grease is higher with outer-ring rotation, especially when motor speed is very high. With this higher centrifugal force, not only the grease base oil but also the grease itself may leak out from between the rings.

As HDD spindle motor bearings are required to perform with very high reliability, the gradual loss of grease can be a disastrous problem. For this reason, bearings whose outer ring is rotated at high speeds require a rubber seal capable of two-way sealing.

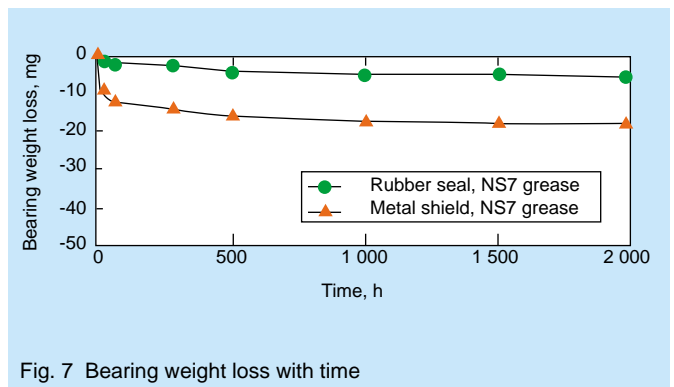


Fig. 7 Bearing weight loss with time

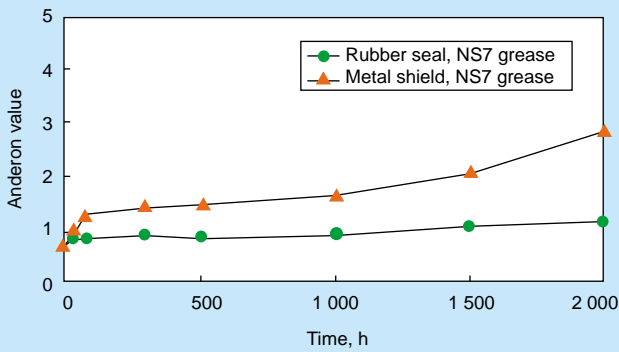


Fig. 8 Bearing noise endurance test (M.B.)

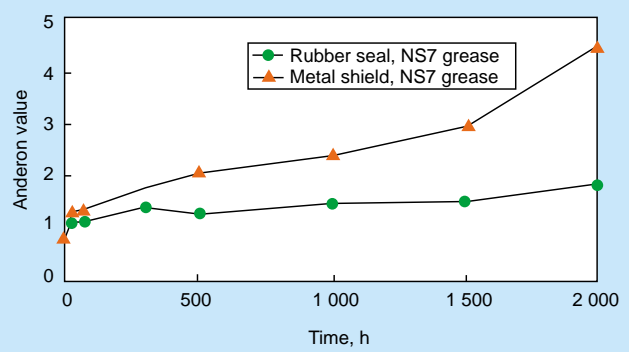


Fig. 9 Bearing noise endurance test (H.B.)

Test conditions:
 Test bearings: Bearing with rubber seal 696T12AVV
 Bearing with metal shield 696T12AZZ1
 Quantity: 8 each
 Running speed: Outer ring running speed of 10 000 rpm
 Temperature: 70°C, without humidity control

Note: The Anderson value represents axial vibration of the outer ring when the inner ring is operated at 1 800 rpm. The M.B. (medium band) covers frequencies from 300 to 1 800 Hz, and the H.B. (high band) from 1 800 to 10 000 Hz.

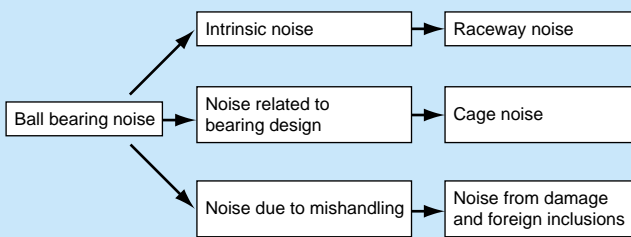


Fig. 10 Types of ball bearing noise

2.4 Low bearing noise

HDD spindle motors, and therefore the bearings they contain, are required to produce as little noise as possible. Noise from a bearing can be generally classified into the types listed in Fig. 10.

To minimize raceway noise, which is intrinsic to bearings and can adversely affect the noise performance of HDDs, the only area for improvement is in the surface roughness and waviness of the balls and outer ring raceway, and this requires very advanced technology. Fig. 11 shows how the noise level of Bearing 695 has been lowered in recent years.

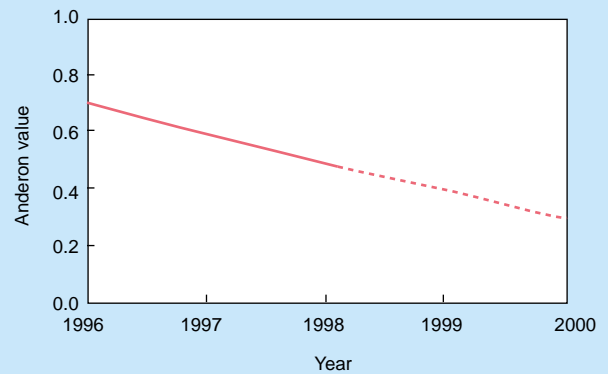


Fig. 11 Bearing noise level improvement (Bearing 695)

2.5 Use of plastic cages

Table 3 lists the mechanical properties of the polyamide plastic (Nylon 66) material used for cages. Table 4 compares the dynamic friction coefficients of Nylon 66 and steel. Fig. 12 shows the structure of plastic cages and corrugated pressed-steel cages with fasteners. Plastic polyamide cages are often used in rolling bearings because they offer the following features and corresponding benefits:

Table 3 Mechanical properties of plastic cage material

		When dry	In actual use*	Note
Flexural modulus	(MPa)	4 540	2 580	
Tensile modulus	(MPa)	3 430	2 550	
Impact strength	(J/cm ²)	37.2	99.0	With Izot notch
Heat deformation temperature	(°C)	233	215	1 818.9 kPa
Coefficient of linear expansion		5 x 10 ⁻⁵		
Melting point	(°C)	253~263		
Specific gravity		1.21		

* In actual use means when containing 2.5% moisture.

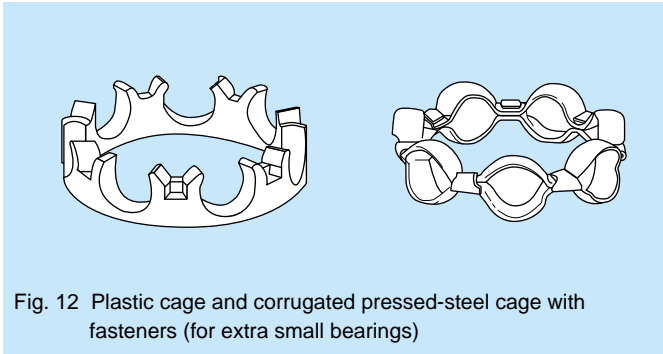


Fig. 12 Plastic cage and corrugated pressed-steel cage with fasteners (for extra small bearings)

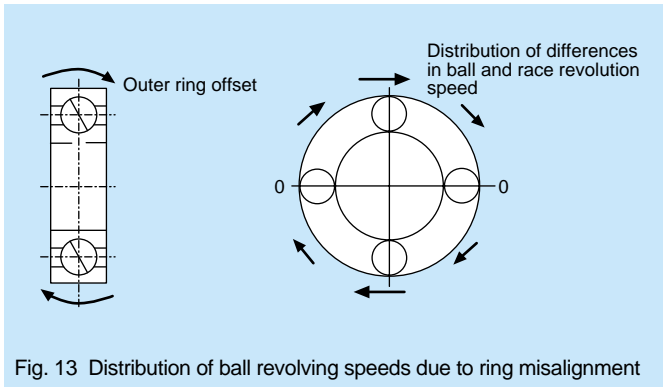


Fig. 13 Distribution of ball revolving speeds due to ring misalignment

Feature	Benefit
1. Highly self-lubricating	Wear-resistant and less likely to produce noise from self-exciting cage vibration
2. Resiliency	Able to accommodate misalignment
3. Low mass (lightweight)	Suitable for high-speed operation
4. Low friction coefficient	Conducive to low torque

As HDD spindle motors operate at higher and higher speeds, plastic cages are increasingly indispensable for the bearings they contain.

In ordinary grease-lubricated bearings, grease occupies about 20 to 30 percent of a bearing's internal space. However, in HDD spindle motor bearings, grease occupies only about 10 to 20 percent of internal space. While this low grease content is necessary to meet requirements for low torque, outparticling and NRRO, it is clearly not conducive to good lubrication. The self-lubricating property of plastic cages helps to mitigate the negative effect of low grease content.

A bearing cage is intended to separate and hold the steel bearing balls at even intervals, but is in perfect sliding contact with the balls. If sufficient lubrication is not maintained between the cage and the balls, the sliding contact may lead to abrasion. This abrasion is particularly apt to occur when the outer ring and the inner ring of the bearing are misaligned (i.e., offset from each other) or when the bearing is under moment load. In both cases, differences in contact angles between the balls cause differences in their revolving speeds that result in the balls and cage colliding repeatedly (cf. Fig. 13).

Table 4 Dynamic friction coefficients of Nylon 66 and steel

Fixed \ Rotating	Nylon 66	Steel
	Nylon 66	0.08
Steel	0.19	0.45

Conditions
Load: 81.3 kPa, Speed: 62 mm/s

Table 5 Properties of tested greases

Grease	NS7	NSC	NSG
Thickener	Lithium soap	Lithium soap	Lithium soap
Base oil	Polyol ester Diester	Polyol ester Diphenyleter	Polyol ester
Dynamic viscosity (40°C)	26.0	53.0	33.2
of base oil mm ² /s (100°C)	5.1	8.3	6.0

If a pressed-steel cage should be abraded, metal dust produced by the abrasion may not only further accelerate the abrasion, but also cause damage to the raceway as it is taken in between the rolling surfaces (between outer and inner ring raceway surfaces and steel ball surfaces) of the bearing, resulting in eventual increase in bearing and motor noise.

It should be noted, however, that this raceway surface damage and increase in bearing and motor noise are of very small magnitudes completely negligible for general motors and bearings. That they matter for HDD applications indicates the severity of HDD requirements. It is no exaggeration to say that there can be cases where the very appearance of ball traces in the raceway surfaces is a problem for HDD spindle motor bearings.

Plastic cages have many advantages including their resiliency, which mitigates the negative effects of steel ball collisions when bearing rings are misaligned, and their self-lubricating capacity, which helps to minimize wear between contact surfaces.

Figs. 14 and 15 compare noise levels over time of bearings with pressed-steel cages and those with plastic cages. The inside diameter of the bearings was 6 mm. As is apparent from the test results, the deterioration in terms of noise performance over time is smaller in the bearing with the plastic cage than in the bearing with the pressed-steel cage.

Cage noise includes the slight noise produced when steel balls collide with the cage as mentioned above and unusual noise due to frictional vibration generated when steel balls are pressed against the pockets of the cage. The latter noise, in particular, which may occur when cage pocket surfaces are not well lubricated, such as when a stiff grease or a grease of low lubrication performance is used or when grease fluidity is low at low temperatures, is often taken as abnormal motor noise.

Plastic cages are known to produce less noise than pressed-steel cages because the surfaces of the balls are kept better lubricated.

Thus, plastic cages have many advantages that contribute to improvement in the reliability of bearings, motors and HDDs. They are highly recommended for HDD bearings.

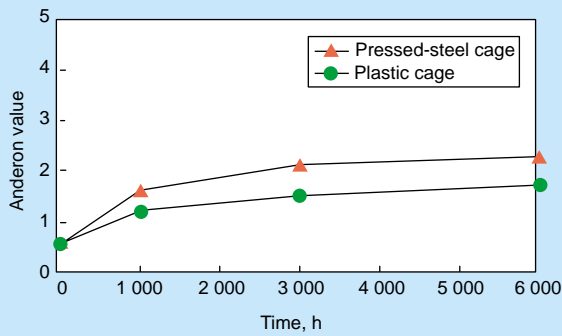


Fig. 14 Cage and bearing noise (M.B.)

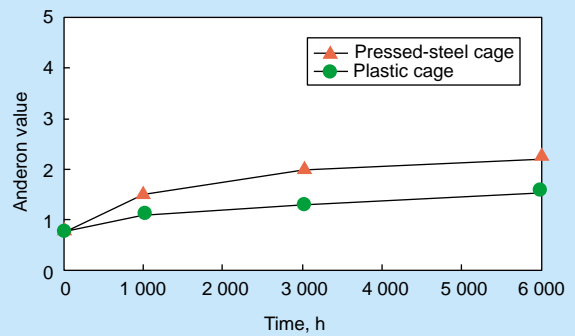


Fig. 15 Cage and bearing noise (H.B.)

Test conditions:

Test bearing: 696
 Environment: 60°C, 70% R.H.
 Running speed: 1800 rpm
 Preload: 19.6 N
 Lubricant: VTG grease

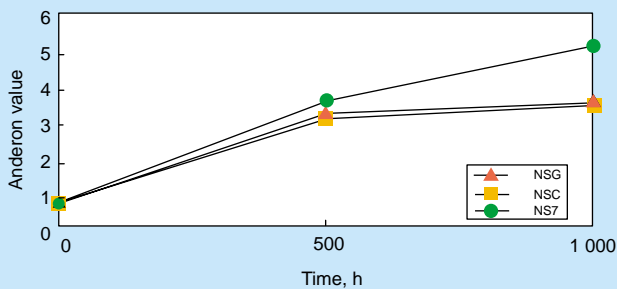


Fig. 16 Bearing noise endurance (M.B.)

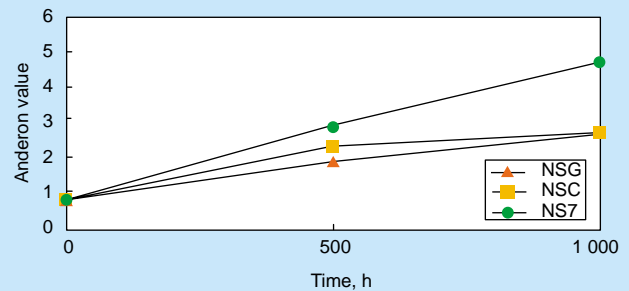


Fig. 17 Bearing noise endurance (H.B.)

Test conditions:

Tested bearing: 695T12VV
 Quantity: 18 each
 Running speed: 12 000 rpm, outer-ring rotation
 Temperature: 90°C, without humidity control
 Preload: 19.6 N

2.6 Grease selection

Greases suitable for the lubrication of HDD spindle motor rolling bearings are required to have the following attributes:

- Long life
- Low noise
- Low torque
- Low outparticle generation

Many greases have been evaluated for selection of reliable grades that meet requirements for HDD spindle motor bearings. At present, NS7 grease is most often used. NS7 grease was developed as a low-noise, low-torque and long-life grease for motors in general and has been extensively used in a broad range of applications, including HDD spindle motors.

The development of new grades of grease to serve at

high speeds and high temperatures in HDD motors has been progressing steadily. Figs. 16 and 17 present results on a few greases tested at high speed and high temperature.

In addition to the heating of HDD spindle motors operated at higher speeds, heating due to windage loss of the magnetic disk may become considerable when its effect on the temperature of bearings used in such motors is considered. Bearings for such motors have to be able to withstand temperatures higher than 80°C.

NSG grease, listed in Table 5, was developed by improving the disadvantageous high-torque property of NSC grease, while maintaining the advantages of NSC as a grease for high-temperature applications. NSG has a longer life than NS7 in high temperatures and is recommended for high-temperature, high-speed HDD motor bearings.

3. Conclusion

State-of-the-art product and manufacturing technology have been used for the enhancement of the reliability and functions of rolling bearings for HDD spindle motors. Further advances are expected as we approach the 21st century. These technologies are also widely applied in the production of miniature precision bearings. We hope that you will find the information presented in this report useful when considering HDD spindle motor bearings and other miniature precision bearings.



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Development of the NSK K1 Seal™ for Linear Guides

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ABSTRACT

New NSK K1 Seals™ for linear guides fulfill two functions: sealing and supplying lubricant. They are made of “Molded Oil,” a special lubricant-containing plastic material developed by NSK. Linear guides employing K1 Seals have excellent durability, especially in environments where they are exposed to dust and/or water.

This report describes Molded Oil and K1 Seals and reports on the performance of linear guides with K1 Seals under various operating conditions in different environments.

1. Introduction

Until recently, there have been basically two methods of lubrication for linear guides: grease and intermittent oil supply. Both of these methods are known to have their own merits and demerits. With grease lubrication, losses from outflow and deterioration from aging necessitate periodic replenishment. With intermittent oil supply, while a steady supply of fresh oil is ensured, complicated oil supply lines and periodic replenishment of the oil tank are required. Needless to say, neither of these methods is completely maintenance-free. In an effort to satisfy persistent market demand for the maintenance-free lubrication of linear guides, NSK has developed and marketed the NSK K1 Seal™. Compact lubricant-supplying K1 Seals are made of “Molded Oil,” NSK’s unique oil-impregnated plastic resin material. While impregnating resins with oil or grease and using the resulting material for lubrication had been done before, using such material to lubricate linear rolling guides was an industry-first for NSK. Initially, two types of maintenance-free linear guides with K1 Seals were developed: light-load ones for conveyor systems, and long-life ones for industrial machinery that mainly included woodworking machines, in which linear guides are subject to lubricant loss. Encouraged by the success of these products, NSK turned its attention to creating linear guides with K1 Seals for higher-load applications such as machine tools where exposure to metal dust and chips, coolant and other contaminating substances is an important consideration. As a result of these development efforts, NSK now has a range of widely acclaimed linear guides that meets market requirements for maintenance-free performance, saving energy, and eliminating the environmental contamination and putrid smell that occurs when lubricating oil mixes with coolant.

This report first describes Molded Oil and the basic properties and characteristics of K1 Seals, and then discusses the durability and performance, under various operating conditions in different environments, of linear guides with K1 Seals.

2. Molded Oil

Molded Oil is a unique material consisting of lubricating oil and polyolefin resin with an affinity for oil. It becomes fluid when heated to the melting point of the polyolefin resin and can therefore be easily molded into any desired shape.

Table 1 compares the basic mechanical properties of Molded Oil, nylon 66 (PA 66), and nitrile rubber (NBR). As the method (scale) for determining the hardness of these three materials is not the same (i.e. Rockwell R scale for plastics and Shore A scale for rubber), the hardness values in the table are not directly comparable. For this reason, their flexural modulus values are also listed. The tensile strength of Molded Oil is lower than the other two materials because of the lubricating oil it contains, but its hardness, or flexural modulus of elasticity, is between that of the plastic and rubber. The flow of lubricating oil from Molded Oil is a result of its residual stress, which constantly acts to contract the polyolefin resin molecules

Table 1 Comparison of basic physical properties

Basic Physical Property	Molded Oil	PA66	NBR
Tensile yield strength, MPa	4.2 ~ 4.7	89	14 ~ 21
Tensile yield elongation, MPa	13 ~ 14	5.4	300 ~ 600
Hardness	HD _A 91	H _R R114	HD _A 65
Flexural modulus of elasticity, MPa	300	3 000	100

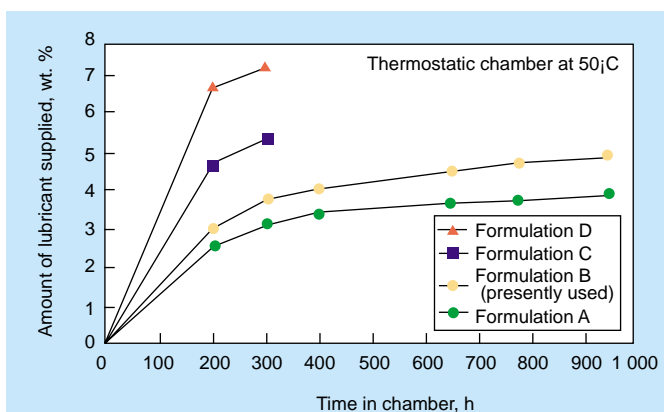


Fig. 1 Lubricating oil quantity by resin formulation

that are elongated during molding. As shown in Fig. 1, the rate of the lubricating oil supply from Molded Oil can be controlled by changing the formulation and grade of the polyolefin resin.

3. Basic Properties of K1 Seals for Linear Guides

3.1 Appearance and installation

Photo 1 shows various K1 Seals, and Fig. 2 illustrates the installation of a K1 Seal on a linear guide. As shown in Fig. 2, K1 Seals are mounted between the end seal and protector outside the end cap of a linear guide. The tightening ring, by virtue of its diameter being larger than the hole into which it is inserted, forces the K1 Seal to maintain tight contact with the rail. Additionally, the length of the tightening ring is made slightly longer than the thickness of the K1 Seal so that the side seal and protector do not interfere with its remaining in tight contact with the rail. Once mounted, K1 Seals provide lubricating oil to the raceways and steel balls for extended periods.

3.2 Lubricating oil-supplying characteristics

Different ratios of lubricating oil to resin yield Molded Oil with different characteristics. K1 Seals for linear guides described herein contain 70 wt% lubricating oil.

The amount of lubricating oil released by Molded Oil is known to vary with temperature. As temperature rises, the increased stress of the polyolefin resin squeezes out greater quantities of lubricating oil. Fig. 3 shows results of an evaluation of the amount of oil released by K1 Seals operating at different temperatures. The results clearly illustrate the strong relationship between the quantity of oil released and ambient temperature.

In earlier K1 Seal applications, the maximum operating temperature was limited to 50°C (80°C for short periods) in order to ensure high performance over a long period. Results of recent experiments, however, indicate that K1 Seals can perform well while operating continuously for extended periods in atmospheres of 70 to 80°C.

Also analyzed was the oil supply quantity of K1 Seals when mounted on linear guides and used as the sole form

of lubrication. Some linear guides were simply operated continuously while others were stopped periodically to wipe off oil which had been released by the K1 Seals. The

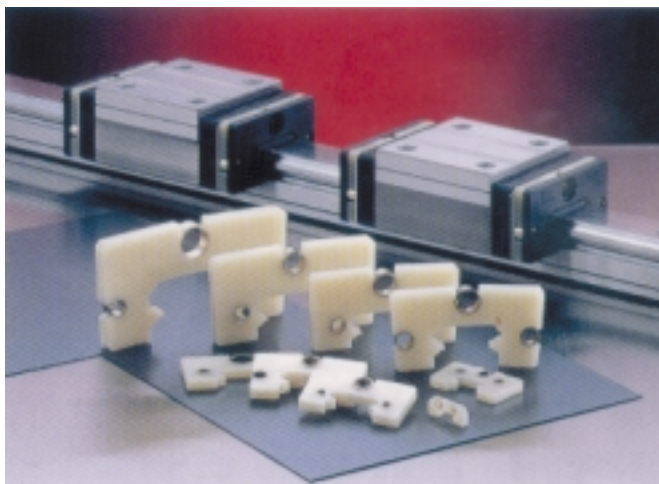


Photo 1 NSK K1 Seals™

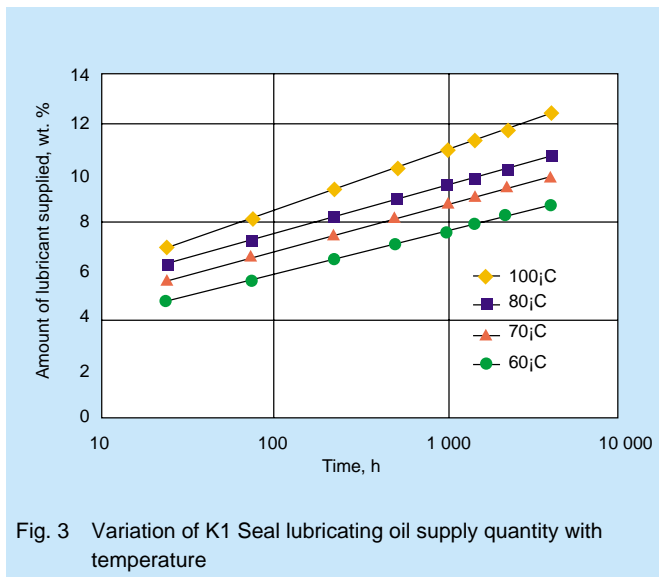


Fig. 3 Variation of K1 Seal lubricating oil supply quantity with temperature

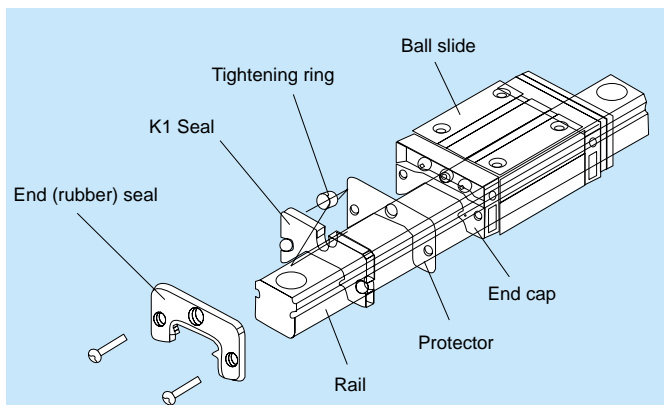


Fig. 2 Installation of a K1 Seal

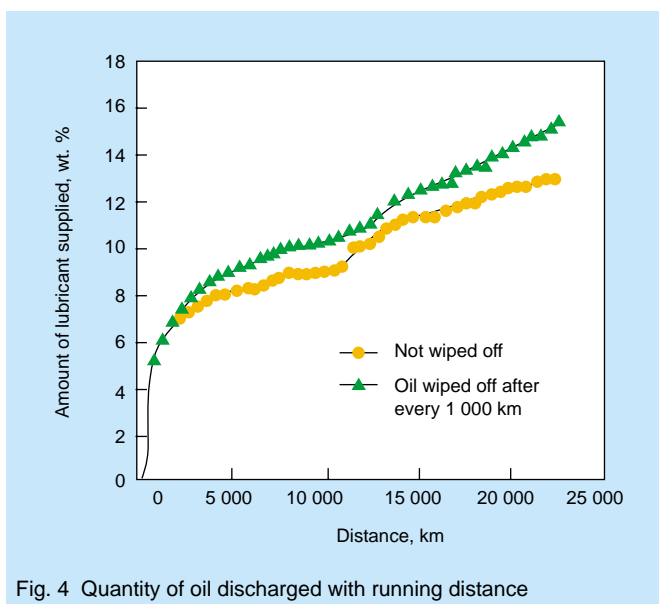


Fig. 4 Quantity of oil discharged with running distance

results, which are presented in Fig. 4, showed that the oil supply from the K1 Seals was greater when the linear guide was wiped off than when it was operated continuously. This means that oil is released from K1 Seals at a higher rate when the rail is dry than when it is oily. In other words, lubricating oil tends to remain in K1 Seals as long as an adequate amount of grease or other lubricant stays on the linear guide.

Resistance to chemicals is another important characteristic of K1 Seals. In order to evaluate their performance in special environments such as those containing inorganic acids or alkalis, K1 Seals were left immersed for 10 days in three different solutions: hydrochloric acid, nitric acid (approximately pH1) and caustic soda (approximately pH13), all at a concentration of 1 mol/liter. As shown in Table 2, compared with the K1 Seals exposed to only air, no difference was observed in the appearance and lubricating oil-supply percentage of those exposed to the acids and alkali. Additionally, exposure of K1 Seals to ordinary coolants, lubricating oils and greases results in the color of the material being transferred to the seals but causes no significant problems. However, being exposed to paint thinner or other organic solvents, kerosene, or rust-preventive oil that contains kerosene components for extended periods may eventually deprive the seals of their capacity to function properly and necessitate their replacement.

Table 2 Chemical resistance of K1 Seals

Exposure to	Air	Hydrochloric acid	Nitric acid	Caustic soda
Appearance	No change	No change	No change	No change
Change in weight of K1 Seal (%)	0.5 ~ 1.6	1.2	0.8	1.1

3.3 Lubricating oil-supply life estimation

Through linear guide operation tests conducted with an ambient temperature of 40°C and then, for the purpose of expediting the test process, 60°C, we have been able to

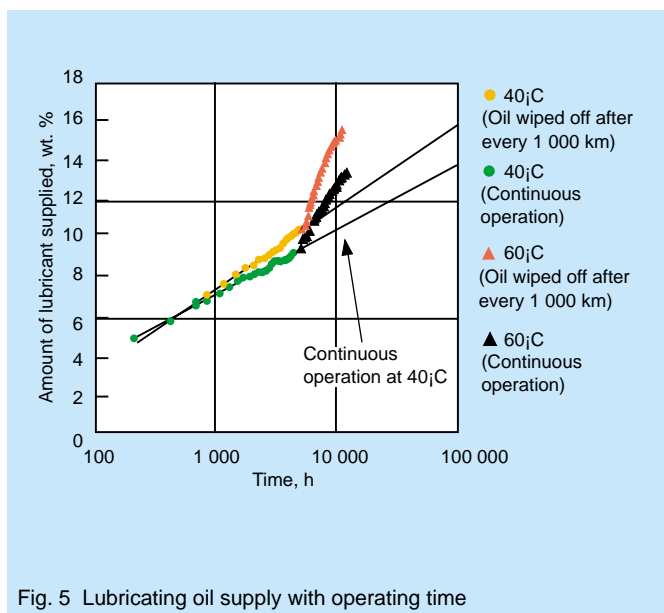


Fig. 5 Lubricating oil supply with operating time

estimate the lubricating oil supply life of K1 Seals. Fig. 5 shows the lubricating oil supply in weight percent vs. operating time in hours. Raising the ambient temperature in the latter part of the test period increased the lubricating oil supply, but the K1 Seals remained functional and continued to operate for some time even after releasing 10% of their originally contained oil. Based on the assumption that 40°C is the ambient temperature typical of most machines, we used the least squares method to estimate the lubricating oil supply life of K1 Seals operating at this temperature. Based on currently available actual results, the maximum lubricating oil supply of K1 Seals in continuous operation was determined to be 13% of the oil they originally contain and that of K1 Seals whose oil outflow is wiped off after every 1 000 km of operation was conservatively estimated to be 15%. From these figures, a life of approximately 100 000 hours was derived. This is equivalent to more than 16 years for a machine operated 250 days a year, 24 hours a day. Thus, K1 Seals are considered to have sufficient durability for applications in linear guides.

4. Durability of Linear Guides with K1 Seals

4.1 Durability under light-load conditions

K1 Seals were originally intended for use with linear guides pre-packed with grease. However, as cases occur in which the pre-packed grease is used up and the K1 Seals become the sole source of lubricant, we tested the durability of K1 Seals in lubricating greaseless LH30 linear guides operated under light load at running speeds higher than 200 m/min, conditions often encountered in transporting systems. As is apparent from the results in Fig. 6, the linear guides continued functioning over long distances while being lubricated exclusively by oil from the K1 seal. Linear guides from whose raceways discharged oil was removed with a solvent at certain distances, continued functioning properly for over 10 000 km while being lubricated solely by the K1 Seals. While wear of the K1 Seals during high-speed operation was considered a possibility prior to the test, the results indicated that sufficient lubrication was maintained to prevent this.

Fig. 7 shows endurance test results on the same LH30-model linear guides with K1 Seals but operated at a moderate feeding speed of 60 m/min under light load. Similarly, the results indicated that K1 Seals are capable of providing sufficient lubrication over extended periods. Even after the grease deposited in the raceway grooves of the linear guides was removed midway through the test, sufficient lubrication was provided by the K1 Seals.

4.2 Durability under heavy load

Typical high-load applications for linear guides include machine tools. The endurance tests described in this section were conducted with a view toward developing a grease-lubricated linear guide with K1 Seals for long-term

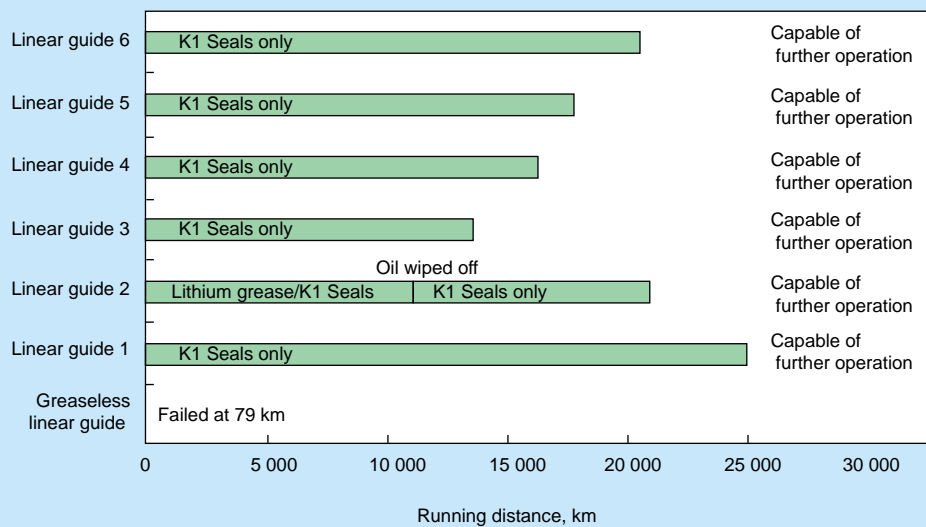


Fig. 6 Results of high-speed, light-load endurance test

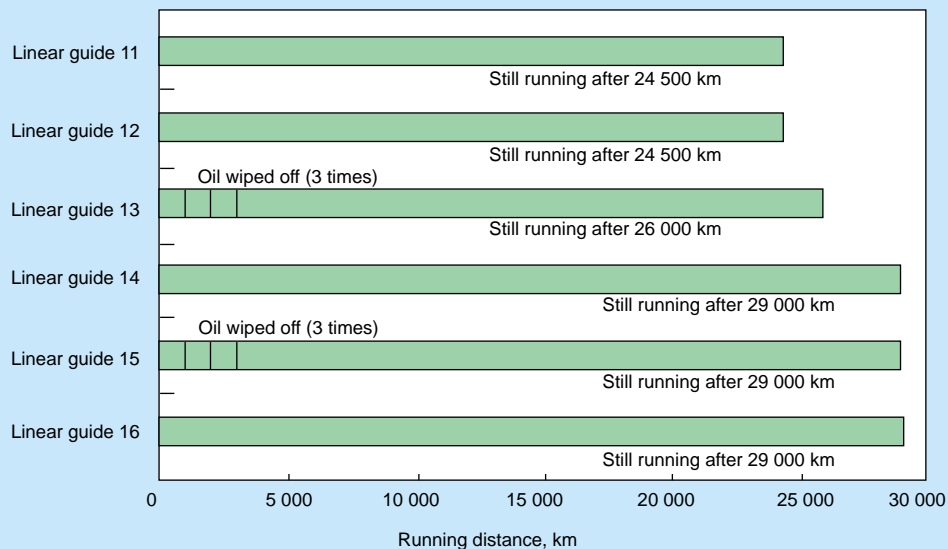


Fig. 7 Endurance test under medium speed with light load conditions

maintenance-free operation in machine tools. In the tests, the maximum contact surface pressure in the grooves of the linear guides was set at approximately 2 000 MPa, generally the highest contact surface pressure found in linear guides operating under high loads. The endurance of linear guides was evaluated in the following four situations:

- (1) lubrication by K1 Seals only (the situation that would develop in the event of deterioration of pre-packed grease);
- (2) lubrication by K1 Seals in their presumed condition after a few years of operation (artificially created by leaving K1 Seals to stand at an elevated temperature until a certain percent of the oil they contained was released);
- (3) simulated actual conditions with grease lubrication and K1 Seals; and
- (4) the same conditions as (3), with the addition of

contaminating substances.

In most cases, two K1 Seals were used on each side of a linear guide as in a standard high-load application, but the condition of four K1 Seals on each side was also tested. In view of the recent trend for high-speed operation, the target running distance was initially 3 000 km and then increased to 5 000 km.

In the test of situation (1), an LY35 linear guide consisting of two rails and two bearings mounted on each rail was operated under the following conditions:

Lubrication:	Only by K1 Seals
Seals:	2 K1 Seals + standard end seal on one side
Preload:	1 760 N (medium preload)
External load:	6 080 N/bearing
Maximum contact surface pressure:	2 060 MPa
Stroke:	400 mm

Table 3 Results after 5 000 km of operation under high load

	Results of lubrication with K1 Seal only	Results of intermittent oil lubrication
Appearance	Running trace and very slight roughness observed on surface of grooves. No abnormality was observed.	Same
Rail groove roughness	Only slight running trace	Same
Percent residual rigidity	90 to 94%	87 to 90%

Avg. feed speed: 24 m/min
(varied between 0 and 37.7m/min by crank operation)

Table 3 lists the test results obtained after a running distance of 5 000 km. Included for reference are test results obtained using a different model which was intermittently oil-lubricated under similar conditions. The degree of change in rigidity indicates that lubrication by oil from K1 Seals only is almost the same as intermittent oil lubrication. Even though the test was stopped, the linear guide lubricated with the K1 Seals was capable of further operation.

In the tests of situations (2) and (3), the linear guides are all still successfully operating in excess of the test target of 5 000 km. Thus, all of the tests support the use of K1 Seals in high-load applications such as in machine tools.

4.3 Durability in environments containing contaminants

4.3.1 Exposure to cutting coolant containing cast iron dust

This test evaluated the performance of linear guides with K1 Seals when exposed to cutting coolant that contains cast iron dust, as often occurs in machine tools. Photo 2 shows the test rig.

The test conditions were as follows:

Tested guide: LY45BN (heavily preloaded)
A total of four rails and eight bearings
(two per rail)

Four different rails with differing seal configurations:
A- four K1 Seals and a standard end seal on each side
B- two K1 Seals and a standard end seal on each side

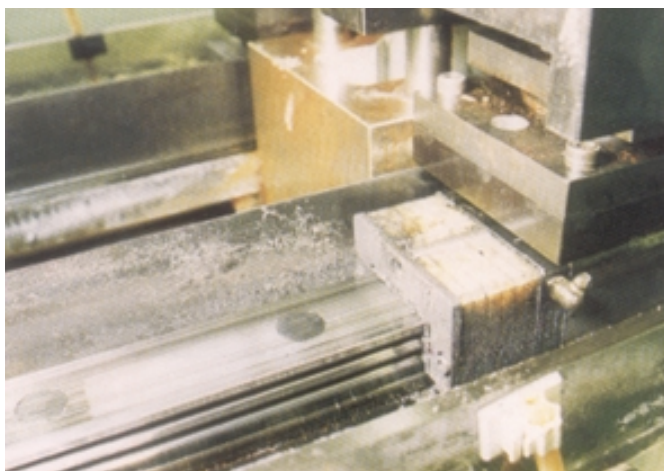


Photo 2 Endurance test of a linear guide exposed to cutting coolant contaminated with cast iron dust

C- standard double seal (without K1 Seals)

D- standard double seal (intermittent oil lubrication)

In the arrangement with four K1 Seals on each side (eight per bearing), there were six K1 Seals on the outside and two on the inside so that the testing machine could more easily accommodate the linear guide.

Lubricant: Alvania No.2 grease (pre-packed, except with intermittent oil lubrication)

Preload: 4 120 N

External load: 9 800 N/bearing

Stroke: 400 mm

Avg. feed speed: 24 m/min (varied between 0 and 37.7 m/min by crank operation)

Coolant contaminant: 115-mesh FCD45 cast iron dust was mixed into cutting coolant MICRO CUT 3850LH (30 × — diluted).

A cycle in which the linear guide was operated for two days with contaminated coolant filled up to the rail upper groove and then operated for five days with the coolant removed was repeated.

First, Rail A (four K1 Seals on each side) and Rail B (two K1 Seals on each side) were tested. The end cap on Rail B broke after 3 000 km of operation. Rail B was then replaced with Rail C, while Rail A continued operating. The end cap on Rail C broke after 600 km. With Rail A still capable of operation, the test was discontinued for analysis after a total running distance of 3 600 km. Table 4 shows the test results of the three rails and Rail D, which was operated for 3 000 km with intermittent oil lubrication.

As noted above, without any K1 Seals (Rail C) and with two on either side (Rail B), the end cap broke after 600 and 3 000 km, respectively. In contrast, as some preload remained in Rail A (four K1 Seals per side) after the test, it was clear that the linear guide was capable of further operation.

Throughout the test, the frictional force of the linear guides was measured using precision spring balances on the testing machine. Fig. 8 shows the changes in frictional

Table 4 Results of endurance test under exposure to cutting coolant contaminated with cast iron dust

	K1 Seals per side	Lubrication	Running distance, km	Clearance (lost preload), m
Rail A	4	Grease	3 600	0 0
Rail B	2	Grease	3 000	>27 >26
Rail C	0	Grease	600	43 >30
Rail D	0	Intermittent oil	3 000	20 15

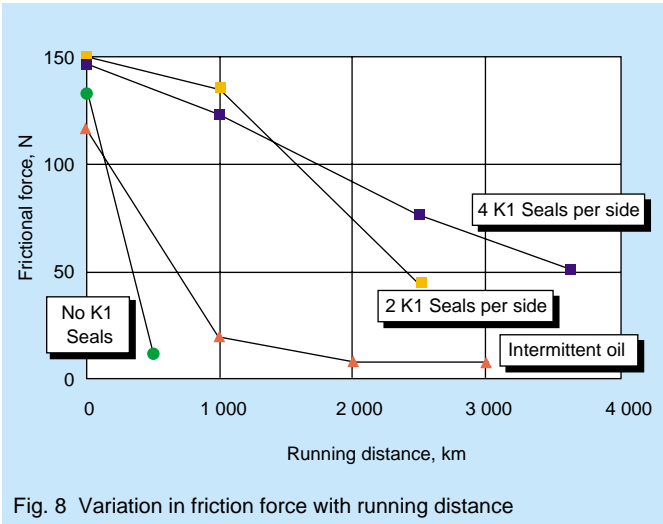


Fig. 8 Variation in friction force with running distance

force that occurred with running distance.

The sharp initial declines in frictional force of the linear guides without K1 Seals indicated that wear began at an early stage. Conversely, the frictional force declined gradually over a much longer period in the linear guides with K1 Seals.

Based on these test results, it is believed that the life of linear guides for heavy-load applications in severe environments containing contaminants can be greatly increased through the addition of one or two K1 Seals to the already standard two seals on either side of the ball slide.

4.3.2 In atmospheres containing wood swarf

A typical example of a severe operating environment in which lubricant is liable to be absorbed by foreign inclusions is in woodworking machines where wood swarf is present. Common problems that occur as a result of the presence of wood swarf include the absorption and reduction in lubricity of ball slide grease as well as abrasion of standard seals. Standard seals, once abraded, more readily permit the ingress of swarf leading to

decreased lubricant circulation to rolling elements. K1 Seals are effective not only in protecting ball slides against lubrication loss but in preventing standard seals from being abraded as well.

Linear guides with the standard double seal (two standard seals combined) and those with K1 Seals were tested in an atmosphere containing wood swarf under the following conditions:

- Tested guide: LH30
- Preload: Light preload
- Average feed speed: 24 m/min
- Stroke: 400 mm
- External load: 490 N/bearing
- Lubricant and seals: AV2 grease + standard double seal
- AV2 grease + K1 Seal + standard double seal

Fig. 9 shows the test results.

The linear guides were operated for different distances depending on the concentration of wood swarf. Among those operated while exposed to the same concentration of wood swarf, the linear guides with K1 Seals were able to continue operating for distances two to four times longer than those without K1 Seals.

4.3.3 In atmospheres containing cast iron dust

This section describes a test conducted in another severe environment, one containing cast iron dust. The conditions of the test were as follows:

- Tested guide: LH30
- Preload: Clearance insert
- Lubricant: AV2 grease (pre-packed grease only)
- Quantity of cast iron dust: 300 g per rail
- Feed speed: 45 m/min
- Stroke: 400 mm

Initially, cast iron dust was sprinkled over both ends of the ball slide and covers were then mounted on the ends. Thereafter, approximately once every day, dust which had accumulated at the stroke ends was put back into the

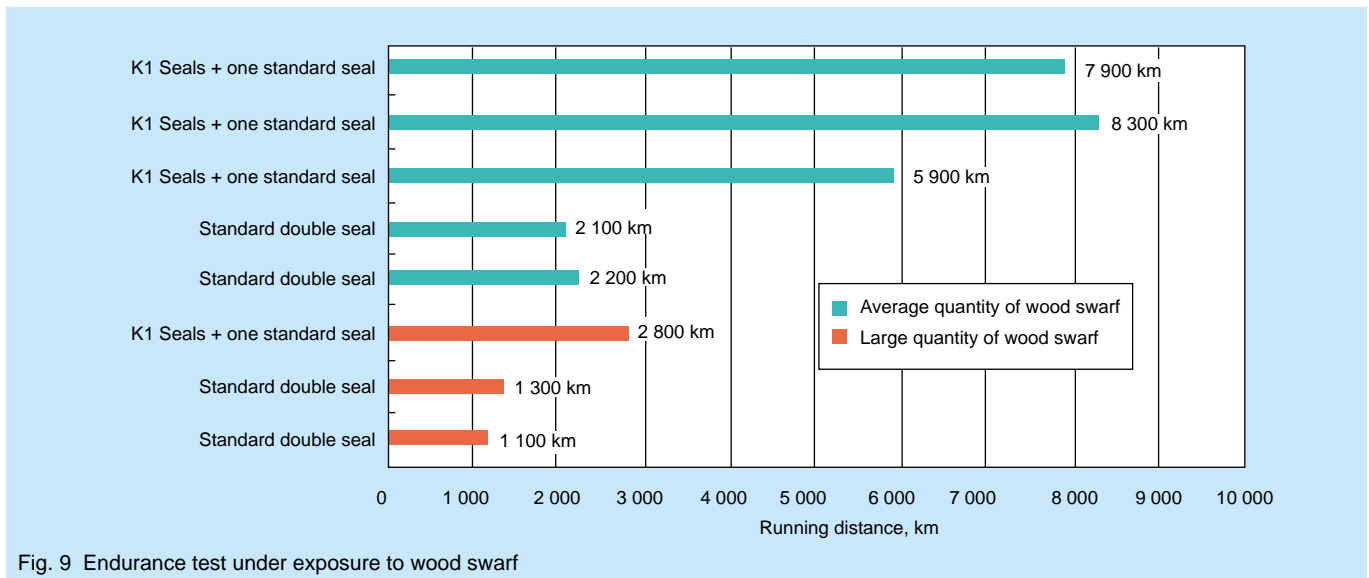


Fig. 9 Endurance test under exposure to wood swarf

Table 5 Results of endurance test under exposure to cast iron dust

Seal Configuration	Std. double seal + protector	K1 Seal + std. double seal + protector
Groove appearance	Nothing abnormal	Nothing abnormal
End seal appearance	Approx. 75% of the entire seal was worn or roughened.	Approx. 15% of the entire seal was worn or roughened.
K1 Seal condition		Lubricant oil still being discharged
K1 Seal weight loss		Reduced by 4.0 to 4.2%
Quantity of dust ingress	0.84 to 2.0 g	0.30 to 0.40 g
Assembly clearance	Increased by 1 m	No change
Frictional force loss	3.9 to 4.9 N (seal resistance)	0.5 to 1.0 N (seal resistance)
Abrasion of grooves	Only slight running trace	Only slight running trace
Wear of steel balls	None	None

covers. Additionally, an arrangement was made so that the cast iron dust on the rail was collected by the covers as the ball slide moved. Photo 3 shows the appearance of a linear guide after operating in these conditions for over 2 000 km.

The test results shown in Table 5 illustrate the effectiveness of K1 Seals. Particularly of note is their ability to reduce the abrasion and roughening of the standard seals as evidenced by the low reduction in frictional force (seal resistance). In addition to this lubricant-supplying capability, the sealing attribute of



Photo 3 Linear guide after 2 000 km of operation while exposed to cast iron dust

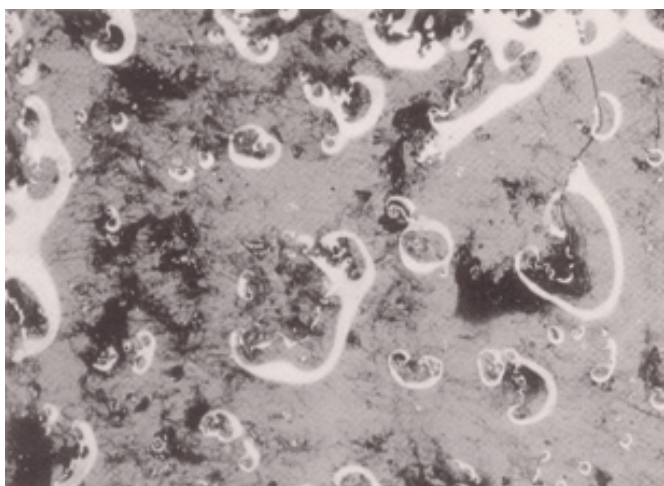


Photo 4 Tested K1 Seal after 24 hours at room temperature (x 10)

K1 Seals was confirmed by the low quantity of cast iron dust found in the ball slide after the test.

The test was discontinued after a running distance of 2 000 km. Had the test been continued, a greater difference in the extent of deterioration of the test pieces probably would have been observed due to differing levels of seal abrasion.

Immediately after the test, one of the tested K1 Seals was left to stand at room temperature for 24 hours so that its residual lubricating capacity could be observed. Photo 4 is a magnified image of the seal after 24 hours. Cast iron dust and droplets of lubricating oil discharged by the seal can be clearly seen. The oil droplets indicate that the K1 Seal is capable of continued operation.

5. Performance of K1 Seals in Linear Guides on Welding Lines

While the test results presented in the foregoing sections were from bench tests, this section presents an example of K1 Seals used in the field. The performance of linear guides, both with and without K1 Seals, in a welding machine on an automobile production line was evaluated. Table 6 describes the linear guides that were analyzed. In the same location of the same machine, Linear Guide 1 (without K1 Seals) was first used for 10.5 months, followed by Linear Guide 2 (with K1 Seals) for 13 months. Attributable to the K1 Seals, there were marked differences in the condition of the linear guides after their respective periods of operation (Photos 5 through 8). The raceways of the rails, the ball slide and the balls of Linear Guide 2 were obviously less worn than those of Linear Guide 1. And, while rust was observed on the rails, ball slide raceways and balls of Linear Guide 1, none was present on Linear Guide 2. The quantity of contaminants that entered the ball slide of Linear Guide 2 was also slightly lower than Linear Guide 1.

Table 6 Linear guides used in a welding machine

	Linear Guide 1	Linear Guide 2
Double seal + protector	Yes	Yes
K1 Seals	None	Yes
Capped fixing holes	Yes	Yes
Duration of use (months)	10.5	13
Cr fluororesin coating	Yes	Yes

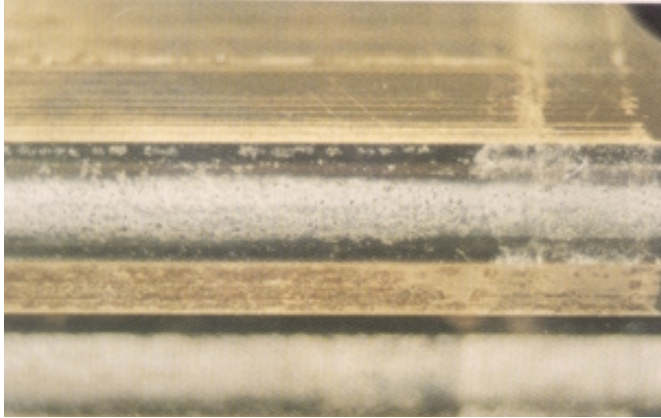


Photo 5 Raceway of Linear Guide 1 (no K1 Seals)

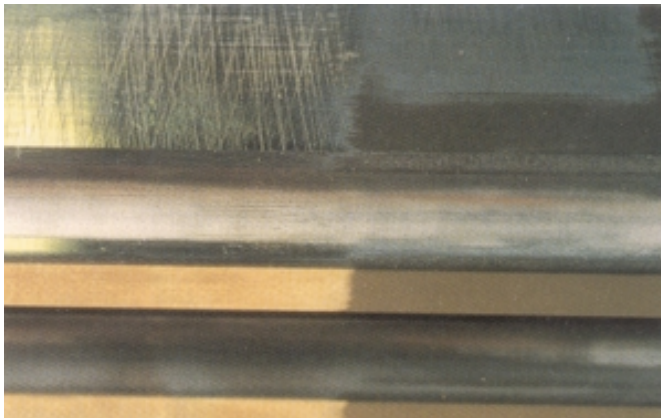


Photo 6 Raceway of Linear Guide 2 (with K1 Seals)



Photo 7 Balls of Linear Guide 1 (no K1 Seals)

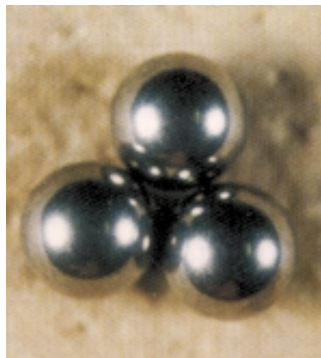


Photo 8 Balls of Linear Guide 2 (with K1 Seals)

6. Recent Trends in K1 Seals

6.1 K1 Seals for food processing machines

Lubricating grease used in food processing machines is subject to frequent exposure to water during cleaning of the machinery for hygienic control. The K1 Seals described above for standard linear guides are well suited for use in such machines if there is no possibility of their direct exposure to the food being processed. For cases in which such a possibility does exist, however, the composition of the lubricating oil in the K1 Seal needed to be changed. Using materials conforming with guidelines issued by the FDA (US Food and Drug Administration), we developed a K1 Seal safe for use in food processing machines. The



Photo 9 Endurance test simulating a food processing machine (water-immersion test)

newly developed K1 Seal has been undergoing general operating endurance tests and water-immersion tests to assess its suitability for use in food processing machines (Photo 9). At present, the K1 Seals in the general endurance tests have exceeded 25 000 km of operation, and those in the water-immersion tests (immersed in water for one day/week) have successfully operated for over 4 000 km. These results demonstrate performance comparable to that of the standard K1 seal.

6.2 K1 Seals for ball screws

When machinery-feed systems in general are considered, maintenance-free ball screws also need to be developed. Employing the same concept as for linear guides, NSK has successfully developed K1 Seals for ball screws used in heavy-load applications such as in machine tools. (Please see "Recent Technical Trends in Ball Screws," Motion and Control No. 4.)

7. Conclusion

The test results indicate that NSK K1 Seals can ensure lubrication not inferior to oil lubrication and lead to extended maintenance-free performance of linear guides and ball screws. K1 Seals have outstanding potential as lubricating oil-supplying units, particularly in environments containing contaminants. More data on linear guides and ball screws with K1 Seals will be accumulated in order to improve their maintenance-free operational reliability.



Soichiro Kato

Insulated Bearings for Traction Motors

The flow of electric current through bearings used in rolling stock can result in damage rather quickly. The very high temperatures generated by arcs between electric contacts inside the bearing result in small craters being formed on the raceway and rolling element surfaces (Photo 1). In some cases, bearing internal surfaces may take on the appearance of a washboard (Photo 2). While this type of damage has existed for a long time, recent designs and extended maintenance intervals of rolling stock have increased the importance of preventing it. In response, NSK has developed insulated bearings for traction motors (Fig. 1 and Photo 3).

1. Bearing Design

Employing plasma spraying technology, the outside and end faces of the outer ring of the bearing are coated with a ceramic material consisting mostly of alumina (Al_2O_3). A thin metal layer is applied prior to the application of the ceramic coating in order to improve its adhesion. The thickness of the ceramic coating may vary from 0.05 mm to 0.5 mm. The standard thickness in Japan is 0.5 mm to prevent damage caused by the high-frequency current leaked by AC motors. As the width and outside dimensions of coated outer rings are the same as conventional bearings without the coating, it is not necessary to adjust bearing housings to accommodate special dimensions.

2. Performance

Because pores compose about 5% of the sprayed ceramics layer, bearings coated only with ceramics will suffer excessive deterioration of electrical resistance when exposed to humidity. NSK's insulated bearings are therefore coated with a special resin that seals the pores in the surface and thus eliminates the harmful effect of humidity. The special resin coating also prevents significant deterioration of electrical resistance when the bearings are exposed to the strong alkaline washing liquid (pH 12.7) at high temperatures (80°C) during disassembly and washing in some rolling stock maintenance shops.

NSK's insulated bearings were subjected to a variety of tests to assess their insulating capacity and rotational performance. Table 1 describes the tests and presents the results.

3. Typical Applications

The use of NSK's insulated bearings in Bullet Trains began in the traction motors of the 300 Series "Nozomi," which started operation in 1992. The bearings are now used in all new Bullet Trains operated at the maximum speed of 300 km/h. Large insulated bearings for the traction motors of electric locomotives are employed not

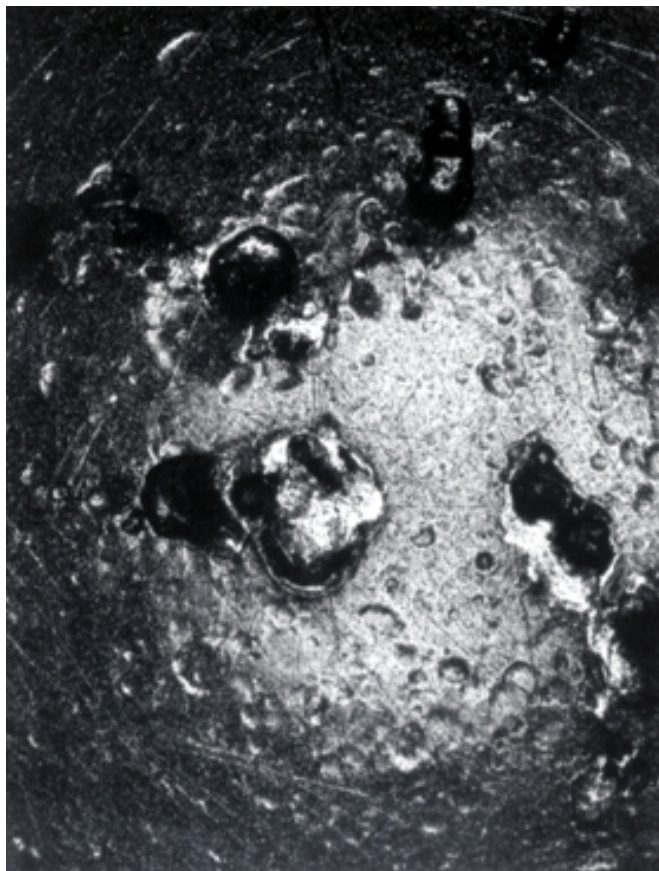


Photo 1 Craters caused by electric current flow

only in Japan, but in countries around the world. The sales history of NSK's insulated bearings is shown in Table 2.

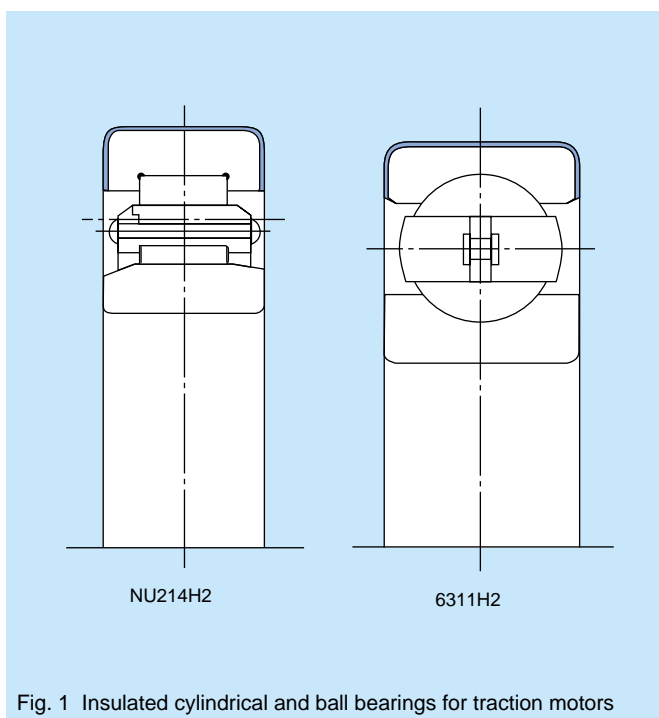


Fig. 1 Insulated cylindrical and ball bearings for traction motors

Table 1 Various performance tests and results

Test	Description	Results
Electrical resistance	Measure electrical resistance during application of 500 VDC	10 MΩ or more
Mounting and dismounting	Evaluate change in strength and electrical resistance of coating after mounting and dismounting	No change in both electrical resistance and strength of coating after mounting and dismounting five times
Effect of temperature	Assess relationship between bearing temperature and insulation resistance	No deterioration of insulation resistance up to 110°C Even above 110°C, 10 MΩ of resistance was maintained (Fig. 2)
Thermal deterioration	Evaluate effect of heating cycles on strength and electrical resistance of coating	Insulation resistance remained higher than 10 MΩ No abnormality was observed in the coating
Liquid immersion	Check coating and measure electrical resistance after immersion in kerosene, alkaline detergent, and grease	Insulation resistance remained higher than 10 MΩ No abnormality was observed in the coating
Vibration test	Check coating and measure the change in electrical resistance after exposure to vibration	No change in the insulation resistance and film
Humidity	Measure change in insulation resistance due to humidity	Insulation resistance of 10 MΩ or more and no abnormality in the film
High-speed rotation	Compare temperature rise of insulated and conventional bearings during high-speed rotation	Temperature rise from room temperature was equivalent to that of conventional bearing (Fig. 3)
Dielectric breakdown	Measure dielectric breakdown strength	Sparks occurred between end faces and housing at 2 400 V with insulation resistance deteriorated

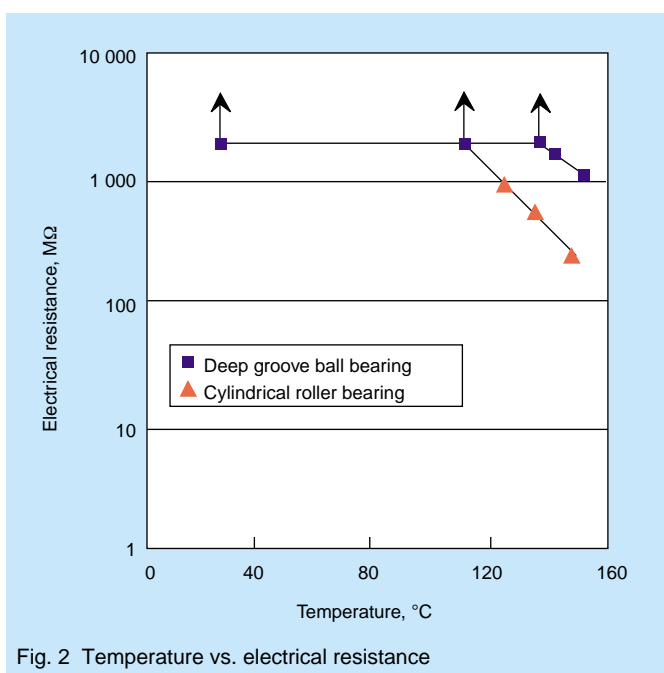


Fig. 2 Temperature vs. electrical resistance

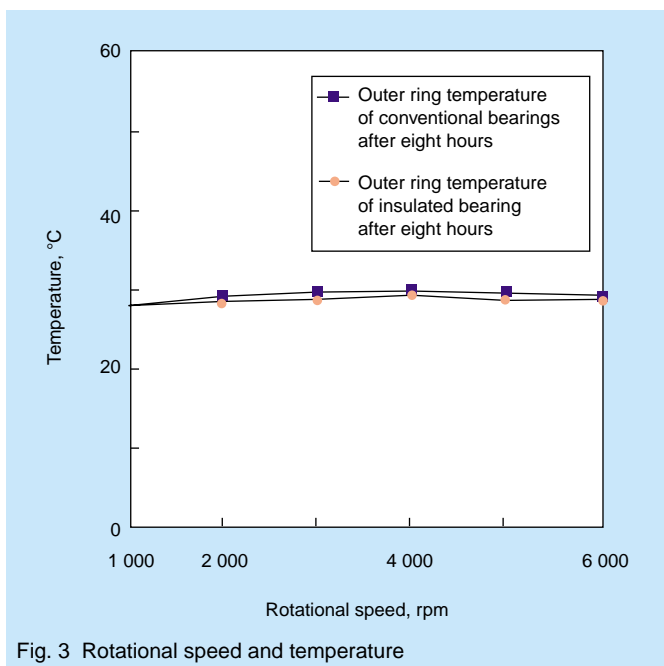


Fig. 3 Rotational speed and temperature

Table 2 Sales history of NSK's insulated bearings for traction motors

Bearing number	Year	Quantity	Application
NU315MH3C4EP6	1984	8	Electric Car
6312H2C4EP6	1984	34	Electric Car
NU219aH2MC4EP6	1987	34	Electric Car
6219aH2C4EP6	1987	34	Electric Car
NU315a1H2C4EP6	1990	120	Electric Car
6314a1H2C4EP6	1990	120	Electric Car
NU214aH2MR1C4EP6U5	1991	200	Bullet Train
6311a4H2C4EP6	1991	200	Bullet Train
NU315a1H2MRC4EP6U31	1991	24	Electric Car
6312a1H2C4EP6	1991	24	Electric Car
NU215H2MRC4EP6U31	1992	36	Electric Car
6311H2S8C4EP6	1992	36	Electric Car
NU214H2MR1C4EP6U5	1992	200	Bullet Train
6311H2S8C4EP6	1992	200	Bullet Train
NU316H2C4EP6U5	1993	48	Electric Car
6312H2C4EP6	1993	48	Electric Car
NU214H2MR1C4EP6U5	1993	652	Bullet Train
6311H2S8C4EP6	1993	689	Bullet Train
NU214H2MR1C4EP6U5	1994	290	Bullet Train
6311H2S8C4EP6	1994	389	Bullet Train
NU214H2MR1C4EP6U5	1995	1 389	Bullet Train
6311H2S8C4EP6	1995	361	Bullet Train
NU314H2MR1C4EP6	1995	70	Electric Car
NU214H2MR1C4EP6U5	1996	429	Bullet Train
6311H2S8C4EP6	1996	826	Bullet Train
NU314H2MR1C4EP6	1996	200	Electric Car
NU215H2MR2C4EP6	1996	75	Electric Car
NU214H2MR1C4EP6U5	1997	563	Bullet Train
6311H2S8C4EP6	1997	504	Bullet Train
NU314H2MR1C4EP6	1997	44	Electric Car
NU326H2MRC4EP6X265	1997	118	Electric Locomotive
NH318H2MRC4EP6X265	1997	118	Electric Locomotive
NU332EH2—TM0101	1997	100	Electric Locomotive

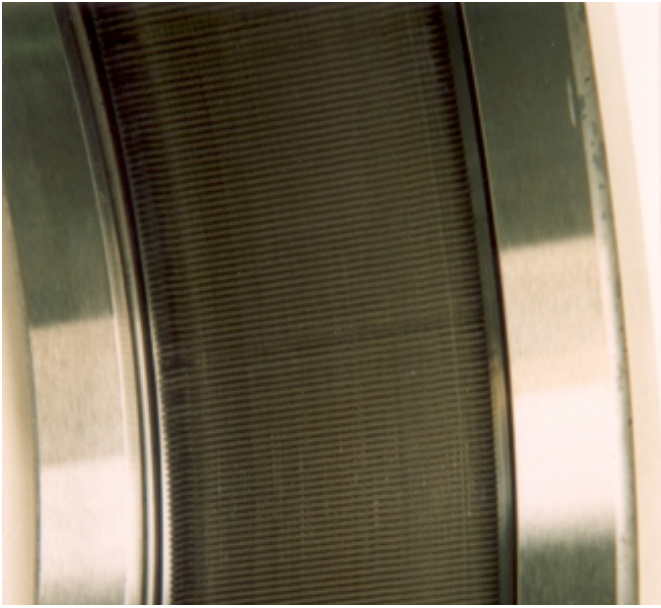


Photo 2 Washboard-like damage from electric current flow

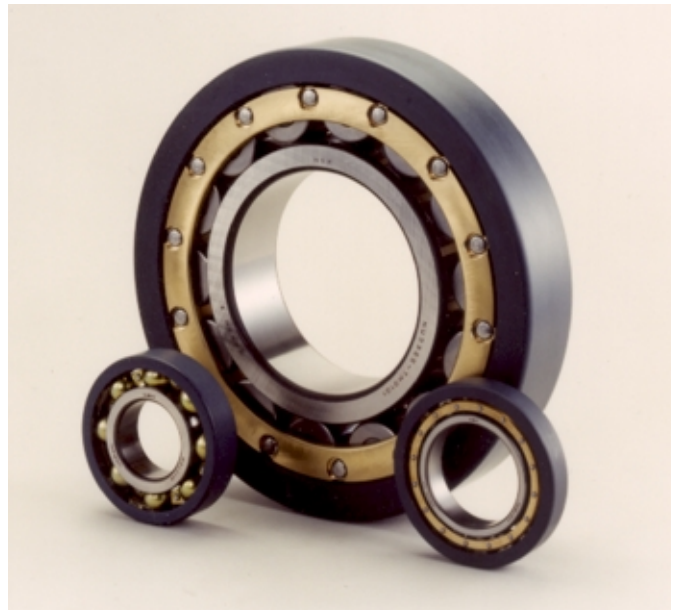


Photo 3 Insulated bearings for traction motors (small bearings for Bullet Train traction motors; large bearing for electric locomotives)

Lightweight, Maintenance-free Ball Bearing Units

Standardized NSK ball bearing units are compact, easy-to-use machine components that are suitable for a wide range of applications. Consisting of a bearing, housing and seals, they simplify construction, installation, maintenance and adjustment. Additionally, their self-aligning characteristic eliminates most misalignment problems resulting from mounting errors.

Due to the convenient features described above, bearing units are being used more widely in all areas of industry. To meet the strong demand for products facilitating lightweight, compact and maintenance-free machines and equipment, NSK offers a wide selection of carefully designed bearing units that contain deep groove ball bearings of the highest quality and meet diverse performance requirements.

Further broadening NSK's range of ball bearing units, we have recently started selling a new series. The new series is presently available only in the pillow-type 2XX, but other housing types and bearing numbers will be added in the near future.

Features

Lightweight

By improving both the shape of the housing and the cast iron used to make it, the housing is stronger and the mass of the unit is reduced by 15%.

Maintenance-free

A long service life without replenishing the grease results from using NSK-developed bearing steel, grease and seals.

Interchangeable

The mounting dimensions conform to ISO and JIS standards and are the same as previous NSK bearing units.

The ball bearing units in the new series are listed in Table 1.

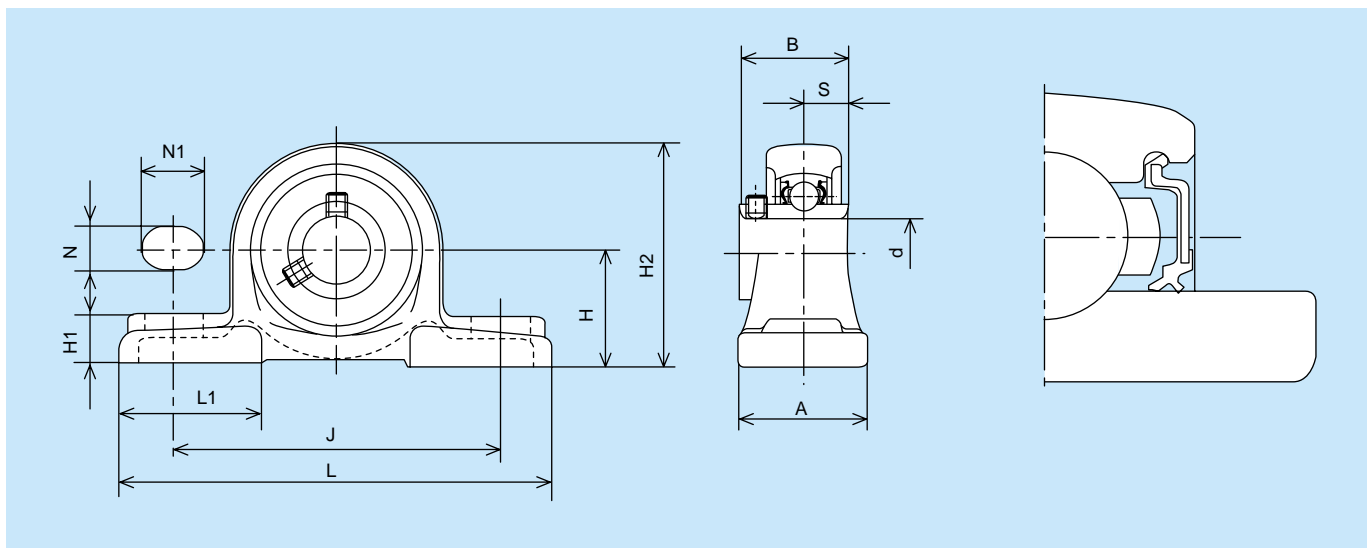


Table 1 New series of ball bearing units

d	L	A	J	H	N	N1	H1	H2	L1	B	S	Bolt size	Unit number	Bearing number	Housing number	Cr	Cor
20	125	37	95	33.3	13	17	14	65	41	31	12.7	M10	UCP204MFU	UC204MF	P204U	12.8	6.6
25	138	37	105	36.5	13	17	15	70	41	34.1	14.3	M10	UCP205MFU	UC205MF	P205U	14.0	7.85
30	163	46	121	42.9	17	21	17	83	54	38.1	15.9	M14	UCP206MFU	UC206MF	P206U	19.5	11.3
35	165	46	127	47.6	17	21	18	94	55	42.9	17.5	M14	UCP207MFU	UC207MF	P207U	25.7	15.3
40	182	52	137	49.2	17	22	18	100	56	49.2	19	M14	UCP208MFU	UC208MF	P208U	29.1	17.9
45	188	52	146	54	17	23	19	108	61	49.2	19	M14	UCP209MFU	UC209MF	P209U	31.5	20.4
50	204	58	159	57.2	20	25	21	114	64	51.6	19	M16	UCP210MFU	UC210MF	P210U	35.0	23.2
55	217	58	171	63.5	20	25	22	126	71	55.6	22.2	M16	UCP211MFU	UC211MF	P211U	43.5	29.3
60	239	67	184	69.8	20	25	24	138	72	65.1	25.4	M16	UCP212MFU	UC212MF	P212U	52.5	36.0
65	263	67	203	76.2	25	29	26	150	78	65.1	25.4	M20	UCP213MFU	UC213MF	P213U	57.5	40.0

VFA Series—Low-priced, Standard-stock Ball Screws



Photo 1 VFA Series Ball Screws

NSK's A Series of ball screws for factory automation (FA) has enjoyed an outstanding reputation. Recently, to complement the A Series, NSK has developed its VFA Series (Photo 1). We are pleased to present it in this article.

1. Description of the VFA Series

Ball screws in NSK's standard-stock VFA Series are inexpensive and available in three combinations of

shaft diameter and lead that are popular among ball screws in FA applications: 12 (shaft diameter, mm) × 10 (lead, mm), 15 × 10 and 15 × 20. The dimensions and specifications of the VFA-series are shown in Fig. 1 and Table 1 respectively. The accuracy of the series is equivalent to JIS CT7, and the ball grooves in the screw shaft and the nut are finished by precision thread grinding. The maximum available axial clearance is 0.010 mm.

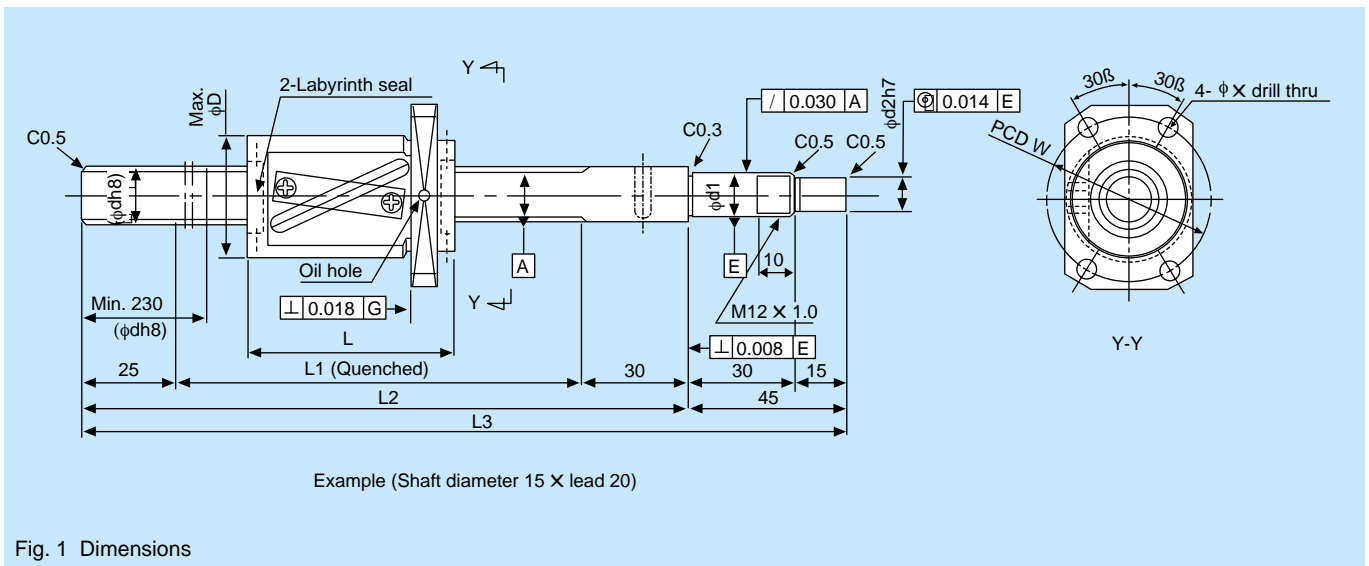


Fig. 1 Dimensions

Table 1 Specifications of VFA series

Shaft diameter (mm) <i>d</i>	Lead (mm) <i>l</i>	Ball screw No.	Stroke (mm)	Basic dynamic load rating (N) <i>C_a</i>	Dimensions (mm)									
					<i>D</i>	<i>W</i>	<i>X</i>	<i>L</i>	<i>d1</i>	<i>d2</i>	<i>L1</i>	<i>L2</i>	<i>L3</i>	
12	10	VFA1210C7S- 410	250 [-100]	3 750	30	40	4.5	50	10	8	310	365	410	
		VFA1210C7S- 610	450 [-300]								510	565	610	
15	10	VFA1510C7S- 500	300 [-100]	7 070	34	45	6	52	12	10	400	455	500	
		VFA1510C7S- 700	500 [-300]								600	655	700	
		VFA1510C7S-1 000	800 [-600]								900	955	1 000	
15	20	VFA1520C7S- 500	300 [-100]	4 560	34	45	6	57	12	10	400	455	500	
		VFA1520C7S- 700	500 [-300]								600	655	700	
		VFA1520C7S-1 000	800 [-600]								900	955	1 000	

2. Features

High-speed feed

The high helix lead of VFA ball screws facilitates high-speed feed.

Easy installation

The shape and dimensions of the fixed-side shaft end are compatible with existing support units for compact ball screws. The free-side shaft end is straight and step-less for use with a special support unit for the VFA Series (see Fig. 2). This allows easy adjustment of the span between the support bearings within the stroke ranges shown in the brackets ([]) in Table 1. If the screw shaft length needs to be adjusted, the screw shaft end can simply be cut. No machining of steps is required.

Low price

By limiting the available clearances, thread lengths and shaft diameter/lead combinations, and utilizing cost-effective specialized production processes, NSK is able to offer VFA ball screws at low prices.

Interchangeable with existing series

As the nut diameter and mounting dimensions of the VFA Series are the same as those of the A Series, the A and VFA Series are interchangeable.

3. Summary

Building on the solid reputation of the A Series and offering the excellent features described above, ball screws in the VFA Series are intended for a wide range of FA applications including Cartesian-type robots, single-axis linear actuators, and transporting and semiconductor manufacturing equipment.

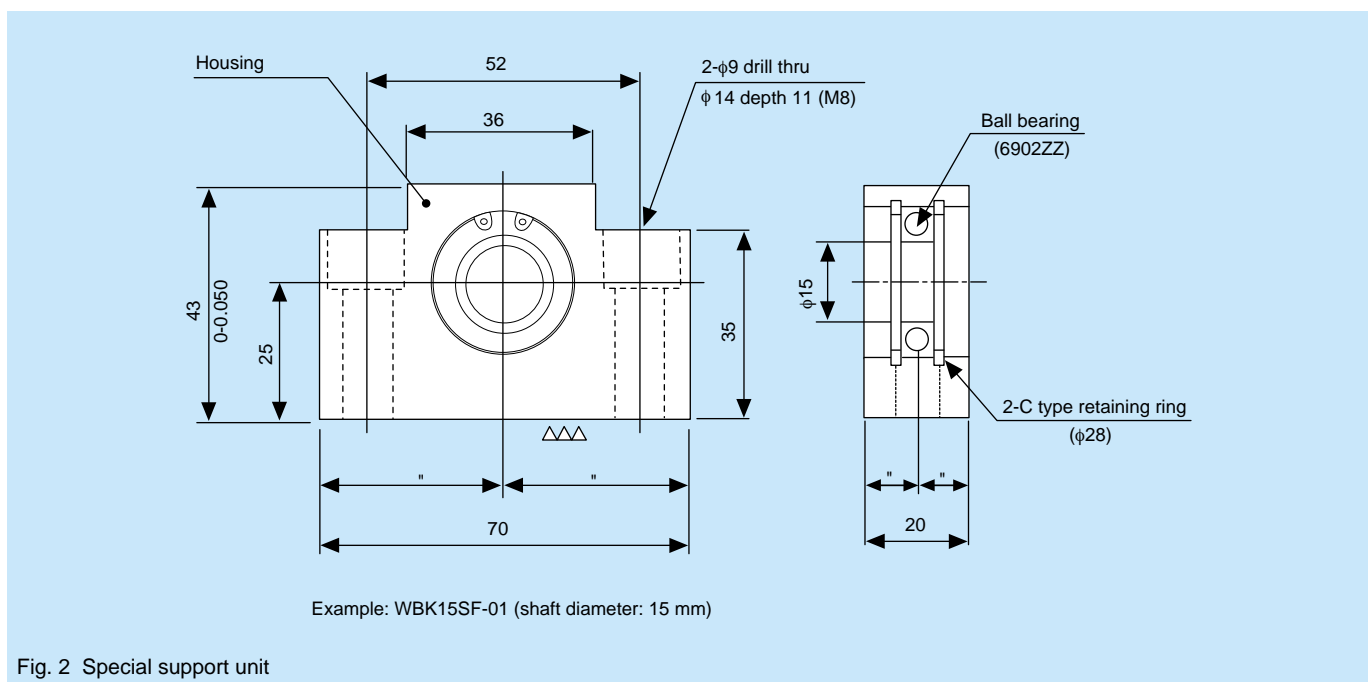


Fig. 2 Special support unit

Wedge Roller Traction Drive Units

Traction drive has been used in various industries because of its quiet and smooth operation. Recent attempts at using it in automobiles and electrically assisted bicycles have drawn attention to its potential as a future power transmission method.

In traction drive, power is transmitted between separate rotating bodies via EHL oil film. The basic equation is a simple friction equation: $F_t = \mu \cdot F_c$. In this equation, F_c is the contact force and can be generated by various methods.

NSK has recently developed Wedge Roller Traction Drive Units (Photo 1). They have a simple construction and use wedge action to obtain contact force in proportion to transmitted torque.

1. Design

The fundamentals of NSK's Wedge Roller Traction Drive Unit are illustrated in Fig. 1 and its structure is presented in Fig. 2. The input shaft and output ring are offset. When the input shaft rotates in the normal direction, the wedge roller causes contact force (F_c) to be generated between the input shaft and the output ring. This force generates traction force, and torque is transmitted. On the other hand, when the rotation of the input shaft is reversed, the wedge roller moves out of contact with the input shaft and output ring, and the contact force (F_c) becomes zero. The input shaft rotates under no load and does not transmit any torque.

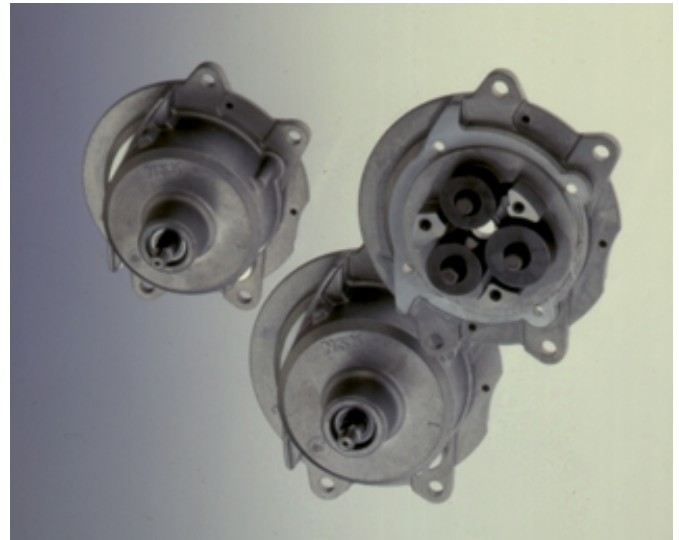


Photo 1 Wedge Roller Traction Drive Units

2. Features

High efficiency

As the compression force generated through the wedge effect is proportional to the transmission torque, high efficiency, even under low load, is ensured. Fig. 3 shows measurement results on input torque and efficiency.

Clutch function is assumed by wedge roller

As the wedge roller assumes the functions of a clutch, space is saved.

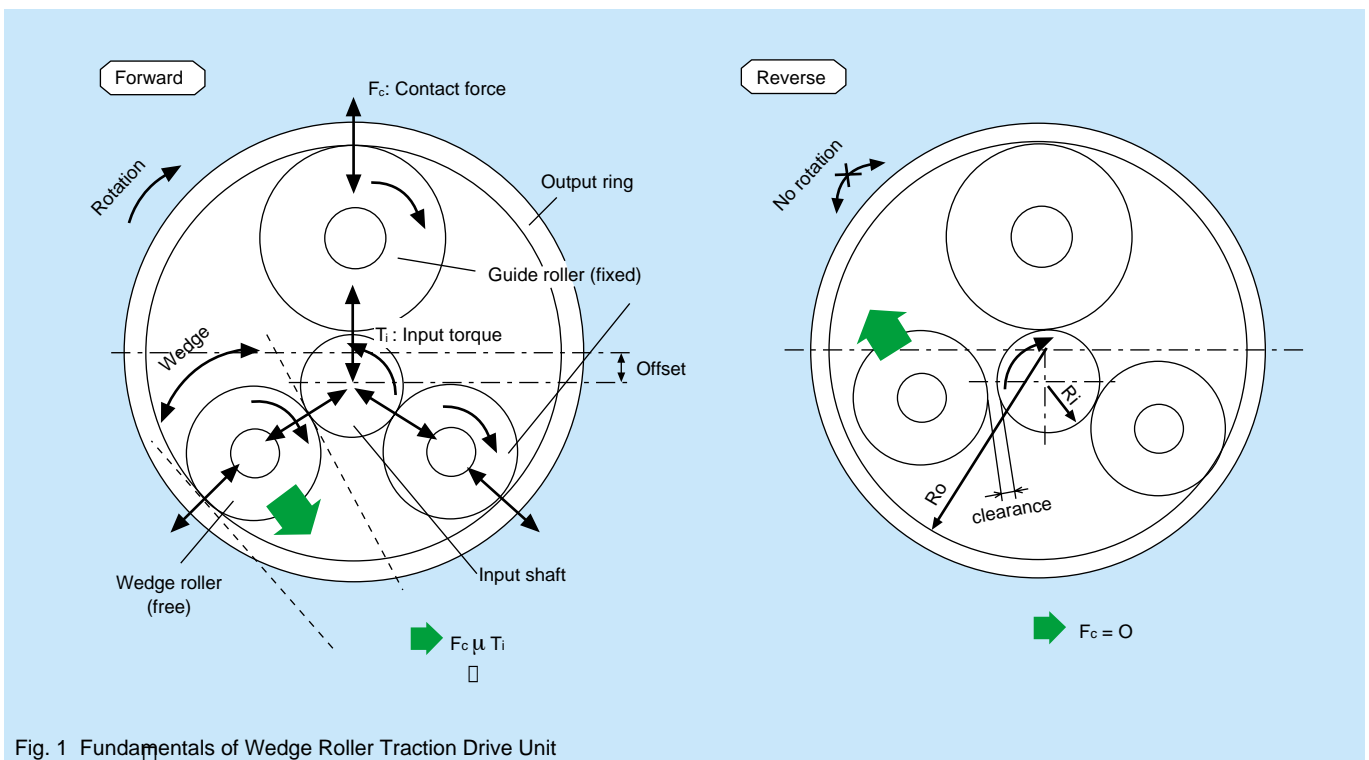
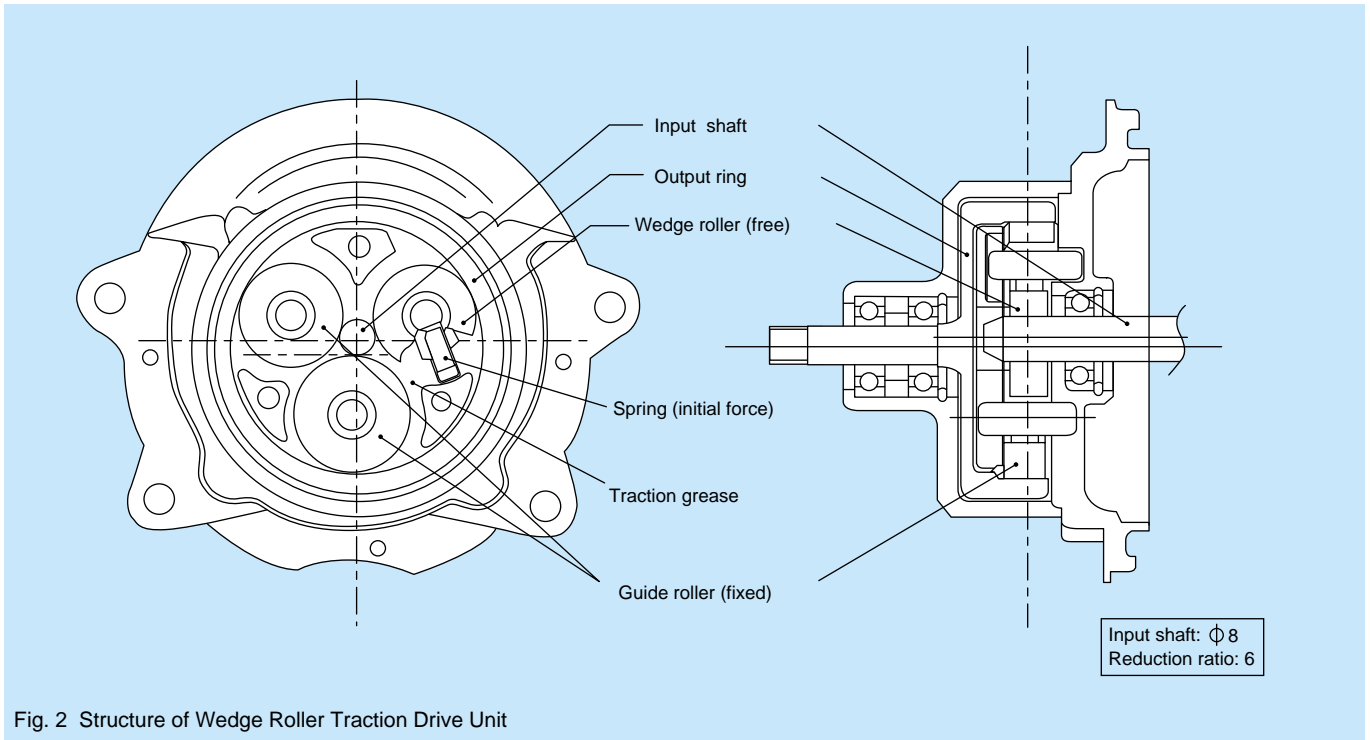


Fig. 1 Fundamentals of Wedge Roller Traction Drive Unit



Improved installation

The wedge roller is movable and can be quickly affixed to the motor shaft.

3. Summary

NSK's Wedge Roller Traction Drive Unit has both the quiet and smooth performance of traction drive and, due to the wedge roller, the outstanding features described above. Its excellent performance has been confirmed in an electrically assisted bicycle. In the future, this reduction unit will be widely used in small motors.

