CATALOGUE

Ball Bearings and Balls

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Lloyd's Register

Certificate of Approval

This is to certify that the Management System of:

Industrial Kinetic Lab Ltd.

2 Bialo more Str., 4004 Plovdiv, Bulgaria

has been approved by LRQA to the following standards:

ISO 9001:2015

Gilles Bessiere - Area Technical Manager

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Wholesale of bearings, components and accessories for them.

UKAS MANAGEMENT SYSTEMS

001

Ball Bearings and Balls

IKL

IKL je osnovan 10.7.1948. od vlade Jugoslavije kao «Industrija kugličnih i valjkastih ležaja». To je bilo prvo državno preduzeće za proizvodnju ležaja u zemlji. Preduzeće je osnovano u veoma teškoj ekonomskoj situaciji, nekoliko godina posle kraja 2. svetskog rata, u vreme ratom opustošene privrede i ekonomske blokade sa istoka i zapada.

Prve mašine su dobijene iz reparacije. Za prvih 6 meseci obučeno je 69 radnika za proizvodnju ležaja, koja je počela 1949. Prvi proizvedeni ležaji su bili 6204, 6306 and 6307.

Prva faza razvoja je trajala do 1955. kada se beleži veliki rast proizvodnje.

Od 1958. do 1962. se nabavljaju nove mašine i izgrađuje nova proizvodna hala koja omogućava dalji rast fabrike.

1968. menja ime u «Industrija kotrljajućih ležaja».

IKL je prema svojim proizvodnim kapacitetima prilagodio proizvodni program svim korisnicima kotrljajućih ležaja. IKL proizvodi, pored standardnih ležaja, i specijalne ležaje za automobile, poljoprivredne mašine, elektromotore,itd.

IKL sve tipove ležaja proizvodi u svim konstruktivnim izvođenjima i povišenim kvalitetima.

IKL, osim što proizvodi i prodaje ležaje, nudi tehničku pomoć i savete za ugradnju kao i predloge ili rešenja za konkretne probleme izbora ležaja.

IKL je 1949. proizveo 1.6 t ležaja što je tada pokrivalo samo 1% tadašnjih potreba, dok je 1979. proizvedeno 5500 t što je činilo 50% potreba (oko 1 kamion dnevno).

U izvoz je išlo 15 % proizvodnje.

IKL se specijalizovao u proizvodnji jednorednih kugličnih ležaja serija 60, 62 i 63 u svim modifikacijama i izvođenjima što se tiče funkcionalnosti, ugradnje i kvaliteta, jednorednih i dvorednih kugličnih specijalnih ležaja za automobile i poljoprivredne mašine.

IKL founded on 10.7.1948. by goverment of Yugoslavia as «Industry of ball and roller bearings». It was the first state-owned company for production of bearings in the country . The company founded in a very difficult economic situation, a few years after the end of the 2nd World War II, during the war-torn economy and the economic blockade of the east and west.

The first machines were obtained from reparations. For the first 6 months 69 workers are trained for bearings production, which began in 1949. First manufactured bearings were 6204, 6306 and 6307.

The first phase of development took up in 1955 when it recorded a growth of production. From 1958 to 1962 IKL purchased new equipment and built new production hall that allows further growth of the plant. In 1968 changing its name to «Industry of rolling bearings.

IKL has adjusted production program according to their production capacity to all users of rolling bearings. IKL produces, in addition to standard bearings, and special bearings for automobile, agricultural machinery, electric motors, etc. .

IKL produces all types of bearing in all constructive performance and high qualities. IKL, except that manufactures and sells bearings, provides technical assistance and advice for the installation as well as suggestions or solutions to specific problems of choice of bearings.

IKL produced in 1949. 1.6 t of bearings which then covered only 1 % needs, while in 1979. produced 5500 t, which covered for 50 % of the demand (about one truck per day). The export went 15 % of production . IKL has specialized in the production of single row ball bearings series 60, 62 and 63 in all modifications and types as far as functionality, installation and quality, one-row and two-row ball bearings for special vehicles and agricultural machinery .

Ball Bearings and Balls

1970. počinja saradnja sa SKF-om, a 1975. godine posle rekonstrukcije i proširivanja kapaciteta (fabrika ima 5000 kvadratnih metara), osvajanja konstrukcije i tehnologije SKF-a i proizvodnja novih tipova ležaja sa novim mašinama iz SKF-a. Novi tipovi ležaja su za elektroindustriju i eksploataciju, ležaji dobijaju Q6 nivo kvaliteta, proizvode se ležaji povišene tačnosti P4, P5 i P6, visokoturažni ležaji,itd.

Ovo je omogućilo da IKL snabdeva sve fabrike u Jugoslaviji i da izvozi (Italija, SAD, Poljska, SSSR, Španija, Grčka, Portugal, Egipat). IKL je bez problema prodavao sve što proizvede.

U junu 1978. IKL je sa 25 organizacija potpisao Sporazum o saradnji što je omogućilo potpuno sagledavanje potreba domaćeg tržišta, a već 1979. godine IKL ostvaruje 37 % veću proizvodnju kao rezultat sprovođenja Sporazuma.

Zbog većeg zadovoljenja potreba tržišta, IKL donosi odluku o izgradnji nove Fabrike ležaja u Barajevu, pored Beograda (22000 kvadratnih metara, počela da radi 1983.), koja je projektovana za proizvodnju 20.5 miliona ležaja manjih gabarita, Fabrike kuglica i kaveza u Bajinoj Bašti, i manje fabrike u Prijepolju.

U tom periodu je izvršena zamena stranih zaptivača za domaće, a cevi, šipke i žice su nabavljane od domaćih železara. Time je praktično IKL koristio sve kvalitetne sirovine domaćeg porekla, kao i iz Ukrajine.

IKL zapošljava 2500 radnika, od toga je oko 350 radnika zaposleno u Barajevu.

Od 1980. promet IKL nastavlja da raste po stopi od 2-3 % godišnje i dostiže cifru od 100 miliona dolara.

Proces restrukturiranja je počeo od 2008.godine usled raspada bivše Jugoslavije. Međutim, ključni ljudi su bili zainteresovani da održe IKL brend u životu i zbog toga je usvojena nova strategija razvoja IKL-a.

U postojećoj IKL Grupi tim vrhunskih profesionalaca je fokusiran na strategiju globalnog razvoja. Od 2000. godine pod imenom IKL-a su osnovane sledeće filijale: IKL Global FZE (UAE) – centar za snabdevanje proizvoda brenda IKL za Azijske zemlje (Vijetnam, Indija, Bangladeš, Irak), IKL Innovations Ltd – istraživačko-razvojni centar za ležaje, a naročito za elektro-izolovane i ležaje za visoke vibracije, IKL Bearings India Pvt Ltd – za promociju IKL ležaja u Indiji i budući centar za proizvodnju ležaja za visoke vibracije u Indiji, IKL International (SAD) – za koordinaciju globalnog širenja tržišta za IKL proizvode, IKL Plovdiv (Bugarska), za promociju IKL ležaja u Evropskoj Uniji i tehnička podrška za kupce za razvoj novih proizvoda za njihove potrebe, IKL Scalica (Slovačka) – proizvodni pogon. IKL Grupa je u prethodnim godinama prilagodila proizvodni program kako bi zadovoljila ogromnu tražnju za novim tehnologijama. Naročita pažnja je bila usmerena na razvoj opreme i tehnologije. Ove tehnologije nam omogućavaju da budemo mobilni prilikom obrade porudžbina. Firma zapošljava stručnjake sa dugogodišnjim iskustvom u industrijskim i naučnim poslovima. Opremljeni najnovijom CAD/CAM, kao i opremom za kalibraciju i testiranje, specijalnim postrojenjima za termičku obradu, najboljim inženjerskim rešenjima u industriji, u mogućnosti smo da proizvedemo veliki asortiman ležaja za zahtevne aplikacije. Radimo samo sa najboljim dobavljačima sirovina/komponenti koje su proverene i odabrane od naših stručnjaka i imaju najbolji standarde u proizvodnoj liniji.

IKL takođe obezbeđuje tehničku podršku na svim nivoima kooperacije: od detaljnog izučavanja potreba i pomoći u fazi odabira ispravnog dizajna ležaja do tehničke podrške prilikom montaže i praćenja životnog veka ležaja. Tim za održavanje, koji je sačinjen od visoko kvalifikovanih inženjera, je uvek spreman za posete kupcima radi rešavanja tehničkih pitanja i produženja veka trajanja ležaja.

Srećni smo što možemo da podelimo naše iskustvo i znanje o našim kvalitetnim proizvodima sa našim kupcima.

IKL – «Industrija kotrljajućih ležaja» – ležaji sa fer kombinacijom kvalitet – usluga – cena: nova filozofija u industriji ležaja. IKL znači Inovacija-znanje-vernost.

In 1970 begins cooperation with SKF, and in year 1975 after the reconstruction and expansion of capacity (factory has 5,000 square meters), the conquest of SKF design and technology and production of new types of bearings with new machines from SKF. New types of bearings are for electrical industry and exploitation, bearings receive Q6 level of quality, there are bearings of high accuracy P4 , P5 and P6, high-rev bearings, etc.

This allowed the IKL is supplying all factories in Yugoslavia and to export (Italy, USA, Poland, USSR, Spain, Greece, Portugal, Egypt). IKL has no problem to sell all it can produce.

In June 1978 IKL, with 25 organizations, signed a cooperation agreement which allowed a complete understanding of the needs of the domestic market, and already in 1979. IKL achieves 37 % higher production as a result of the implementation of the Agreement . Due to higher meet the needs of the market, IKL makes a decision on the construction of the new plant bearings in Barajevo near Belgrade (22000 square meters, started working in 1983.), which is designed to produce 20.5 million bearings smaller footprint, Factory for balls and cages in Bajina Basta, and small Factory in Prijepolje. In this period, the replacement of foreign seals for domestic is done, and tubes, rods and wires are procured from domestic steel mills. Practically, IKL used all the quality raw materials of domestic origin and Ukraine .

IKL employs 2500 workers, out of which about 350 workers employed in Barajevo. From 1980 IKL turnover continues to grow at a rate of 2-3 % per year reaching the figure of 100 million dollars. Since 2008 the restructuring process has started because of breaking down former Yugoslavia. But key personal was interested to keep the IKL brand alive therefore new strategy of development of IKL was adopted.

At present IKL Group is a team of high skilled professionals focusing on expanding globally. Since 2010 the following subsidiaries were established under IKL name: IKL Global FZE (UAE) – as hub for supply of IKL Brand bearings to Asian Countries (Vietnam, India, Bangladesh, Iraq), Industrial Kinetic Lab India Private Limited was incorporated in 2019 in India, R&D Technical Centre for study of the tribology and its effect on bearings for special critical application, technical support of end users in this segment, grease manufacturing and authorized importer and exporter of grease, bearings and its components, IKL Innovations Ltd – as research and development center for bearings and particular insulated bearings and vibrating screen bearings, IKL Bearings India Pvt Ltd – for marketing of IKL bearings In India and further manufacturing hub for vibrating screen bearings in India, IKL International (USA) – for co-ordination of global expanding of IKL products, IKL-Plovdiv (Bulgaria)– for marketing of IKL bearings in European Union and technical support of the customers for developing of new products as per customers need, IKL Scalica (Slovakia) – manufacturing facility.

IKL Group in recent years has adjusted production programm to satisfy the great demand for new technologies. Particular attention was paid to the development of equipment and technology. These technologies allow us to carry out orders mobilely. The company employs specialists with long experience in industrial and scientific work. Equipped with the latest CAD/CAM, calibration and testing equipment, special heat treatment facilities, with the best engineering solutions in the industry, we are able to produce bearings of a wide range for heavy duty application. We work only with the best suppliers of materials / components that were audited and approved by our specialists and has the highest standards in production line.

IKL also provides technical assistance at all stages of cooperation: from the detailed study of the demand and assistance at the stage of choosing a right design of bearings till technical support during the mounting and monitoring of the life of the bearings. Service team consisting of highly skilled engineers is ready for customers visit to resolve technical issues and to increase lifetime of the bearings.

We are happy to share our experience and knowledge about our quality products with our customers.

IKL - «Industrija kotrljajućih ležaja» – bearings with fair combination of quality – service – price: new phylosophy in bearing industry. IKL means Inovation- knowledge-loyalty.

Ball Bearings and Balls

ZERTIFIKAT ◆ CERTIFICATE ◆ 認証証書 ◆ СЕРТИФИКАТ ◆ CERTIFICADO ◆ CERTIFICAT

TÜV SÜD South Asia

CERTIFICATE

The Certification Body
of TÜV SÜD South Asia Private Limited

certifies that

IKL

Industrial Kinetic Lab India Pvt Ltd

Plot No.28, Phase III, IDA, Cherlapally,
Hyderabad - 500 051, Telangana, INDIA

has implemented a Quality Management System

in accordance with **ISO 9001:2015**

For Scope of

IMORT, EXPORT AND TRADING OF BEARINGS

The certificate is valid From **2019-11-07** until **2022-11-06**
Subject to successful completion of annual periodic audits
The present status of this Certificate can be obtained on www.tuv-sud.in
Further clarifications regarding the scope of this certificate may be obtained by consulting the certification body

Certificate Registration No. **99 100 20437**

Date of Initial certification : **2019-11-07**

Certification Body
of TÜV SÜD South Asia Private Limited, **Mumbai**
Member of TÜV SÜD Group

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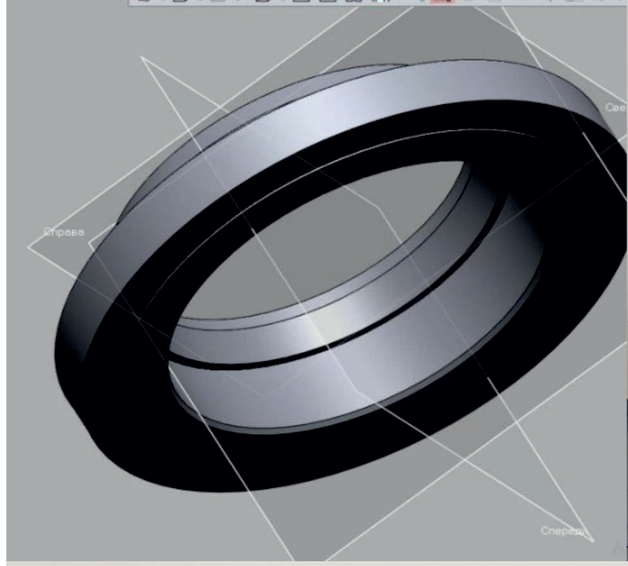



IKL

Ball Bearings and Balls


PRINCIPLES OF THE BALL BEARING

Ball Bearings and Balls

Load on a plain bearing is supported by a film of oil, the functioning of which dependent upon such variable factors as film thickness is permitted by bearing radial clearance and the effect of temperature changes upon lubricant viscosity. Maintenance of the oil film is influenced by other variables, such as speed of rotation, the method of lubricant supply, the design of oil-distributing grooves, and the chances in bearing clearance due to heat, as well as wear induced by sliding friction.

To secure more permanent and accurate positioning of shafts and reduce the power loss attributable of sliding friction, but in a manner not subject to the fluctuating thickness of oil film, nor progressively changing with the abrasive wear of the plain bearing, rollers, first of cylindrical, later of spherical or tapered form, were early employed.

Even in their crudest form, it was evident that, rolling bearings possessed very marked advantages which, with the advent of personal transportation in the form of bicycle, assumed an importance hitherto not realised. In the bicycle the saving of power became a matter of importance to millions. The use of ball bearings proved a necessity.

Just as a IKL and the bicycle coaster brake became synonymous, so at the same period began IKL experience with the ball bearing, in which field it has pioneered all of the present day forms of dual capacity (angular contact, single and double row type) bearings, and many other such as the self-sealed and lubricated-for-life bearings.

While it is concede that the steel ball and the ball bearing are American inventions, it remained to the German, Stribeck, financed by the Deutsche Waffen und Munitions-fabriken, to determine by research the fundamental principles underlying the theory of the modern ball bearing. These principles were employed in Germany by Conrad in designing the single row angular or radical ball bearing and by IKL engineers in their development of the double purpose, combined radical and thrust ball bearing.

The ball bearing has, therefore, been distinguished from other forms of rolling bearings, in having a sound background of scientific principle governing its design. The resultant load-carrying capacity has thus not been merely a matter of experiment, but is subject to as accurate a mathematical determination as is the strength on a steel bridge. Though, compared with the plain bearing, the ball bearing is a mere infant in age, its load-carrying ability, endurance life and friction loss is predictable with infinitely greater accuracy and assurance than that of any other form of shaft support.

The steel ball, due to its shape, possesses inherent advantages not equalled in any other form of rolling body, in that whatever the angle at which the load may be applied, it presents a uniform and calculable resistance.

To the engineer and scientist, therefore, the ball bearing appeals as a known quantity which may be brought into his computations with the same confidence as may such a factor as the moment if inertia of a beam. No other form of bearing presents as definite a mathematical certainty. No other roller, cylindrical or otherwise, but has two ends. The behaviour of stress coming to these abrupt inflection points is indeterminate. There are no ends to a ball; its axis of rotation of load need never be artificially fixed.

The ball, being so unique a roller, required only a determination of the mechanics of the race members in relation to it in order to establish the law of performance of an ideal anti-friction bearing.

Superfine Material and Extreme Accuracy Absolutely Essential

From the magnitude of the compression stresses occurring at the contact areas of balls and races (varying from 200,000 to 300,000 pounds per square inch under normal load conditions) it is clear that no ordinary steel or standards of accuracy could meet the requirements of successful ball bearings. Continual development and improvement has resulted, therefore, in alloy steel, such as used by IKL, that is much harder and tougher than the straight carbon steels and capable of acquiring the same degree of hardness all through, instead of merely in the surface layers. This steel also is of a much more refined composition, being freer from slag, voids and dirt than steels used for any other purpose.

Of importance equalling, if not surpassing, that of the steel, have been the vast improvements which have taken place in machines and been the vast improvements which have taken place in machines and manufacturing methods. Variables have been steadily eliminated, with machines and gauges so perfected that the uniform standard of accuracy achieved today would, but a few years ago, have been considered an utter impossibility commercially.

Thus, the exquisite finish of ball and surfaces, in modern bearings, is not achieved in the effort to further reduce friction losses, but is simply the direct and visible result of the extreme accuracy which is a characteristic of present day ball bearings and which is of the utmost importance in their resistance to fatigue under high stress.

It is felt that a discussion of some of the more important principles involved in the design of ball bearings may be desirable to the users of this Handbook. In the following pages of this section, therefore, those fundamentals of especially interest and value to the average reader have been set down in a more or less logical order, through brevity has been necessary in the general treatment.

Ball Bearings and Balls

Load-Carrying Ability of Steel Balls and Importance of Shape of Contacting Races

Since any load supported by a ball bearing must be transmitted through the balls and their contact with the races can only be a fraction of the surface area of these members, it is obvious that not only is the number of balls great importance, but so, likewise, is any change in the shape of the supporting rings which may affect the area of the contacting surface.

In order to properly carry heavy loads without permanent change of shape, the balls must be extremely hard. However hard a steel may be, it is elastic. If a hardened steel ball is dropped up on hard steel plate, it rebounds, and this rebound is due to elastic deformation. When the ball strikes the plate, both undergo a certain instantaneous deformation, but due to the inherent resiliency of the steel, it immediately recovers its shape, thus throwing the ball upward. Modern alloy, ball bearing steels possess, to an extreme degree, this property of elasticity, and the ability to withstand repeated deformation, enabling present day ball bearings to support heavy loads for very long periods.

When Stribeck* embarked upon his fundamental researches, he first repeated and enlarged upon the previous general researches of the earlier scientist, Hertz, whose work related to the elastic properties of bodies of various shapes when pressed against each other under known loads. Stribeck confirmed the deductions of Hertz and applied them to the conditions existing in ball bearings, reaching the following conclusions.

1. The approach "δ" of the raceways, or the total compression of a ball and its races, varies as the 2/3 power of the normal load "p".

Expressed mathematically,

$$\Delta = K_1 p^{2/3}$$

Where K1 is a constant depending upon the material and the shape of the supporting races.

2. From limits, determined by the load which produces a sudden increase in permanent, deformation, the permissible static loading, p0, of the ball of diameter, "d," in given raceways, varies as the square of the ball diameter, or

$$P_0 = K_2 d^2$$

Where K_2 is a constant for the material and shape of raceways in question.

When a ball of any homogenous material is pressed against a flat plate of the same material so that compression takes place, the flattening of the ball and the indentation in the plate will be of the same magnitude, the resulting curvature being a mean between that of the original surfaces. The material displaced in both bodies increases the density of the mass supporting the contact area, causing a certain flow to take place progressively extending the area of deformation until equilibrium is established.

It is clear that the form of ball support is of very great importance. Hertz formulae indicate that for the same average stress in the contact area, a ball resting upon a flat plate will support four times load that it will when resting between two other balls of the same diameter. If in place of these two balls there are substituted two members shaped as in fig 1, with grooves having radii of curvature equal to twice the ball diameter, the carrying capacity will be increased sixteen fold.

It is necessary therefore, that the curvature of the raceways of a ball bearing be accurately controlled, and in order to lessen stresses in the contact area these curvatures should closely confine the ball.

Stress in the ball bearing varies as the cube root of the load. For example, with load increases of 100%, the stress increases but 26%. This unique characteristic of the ball bearing is responsible for its extreme carrying capacity under adverse conditions.

The relation between the curvature of the raceway and the ball is called "conformity", and from the foregoing, it is evident that this factor has a very decided influence upon the performance of the bearing and upon the formulae used to compute the capacity.

In Stribeck's time it was feared that race grooves made with radii less than two-thirds of the diameter of the ball would produce contact ellipse with too great a major axis, causing slip and loss of efficiency, due to variance in surface speeds. Better methods of race curve generation, together with better materials and metallurgy, have allowed skilled bearing makers to employ much closer conformity. In some cases, however, where a shaft is required to flex under load, more open curvature is needed, else the bearing becomes too rigid, attempting, often unsuccessfully, to act as a shaft straightener.

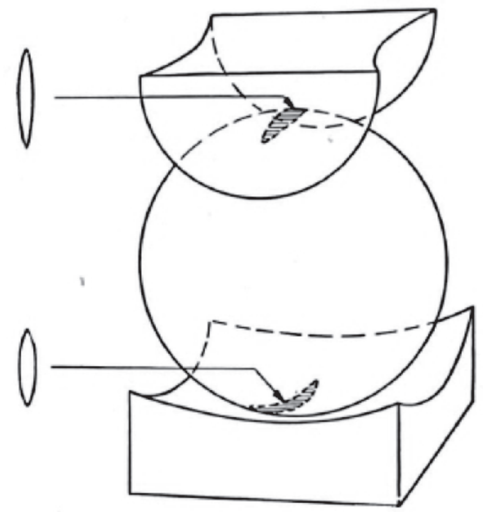


Figure 1.

Close conformity has not proven detrimental to the efficiency of the ball bearing as was feared, since, in the first place, it materially reduced the deformation and hence the molecular work, but, in the second, the length of the major axis of the contact ellipse does not become so great under any safe load as to produce a variance in velocity sufficient to overbalance the gain from the lessened deformation.

Ball Bearings and Balls

Distribution of Radical Load in a Ball Bearing

Pure radial load, applied to ball bearing, is not evenly distributed in all the balls. To illustrate, let it be assumed that such a load is applied vertically to the outer ring and that the position of the balls is as shown in figure 2. The top most ball, "0", will then be under the heaviest load, and the two adjoining it, one on each side, "1", will carry somewhat less but equal amounts. The next pair, "2", will be equally loaded, but to a still lesser extent. Under this load, therefore, because of the deformation the balls and rings in the contact areas, the outer ring will move downward slightly so that all of the balls in the lower half of the bearing are relieved of load.

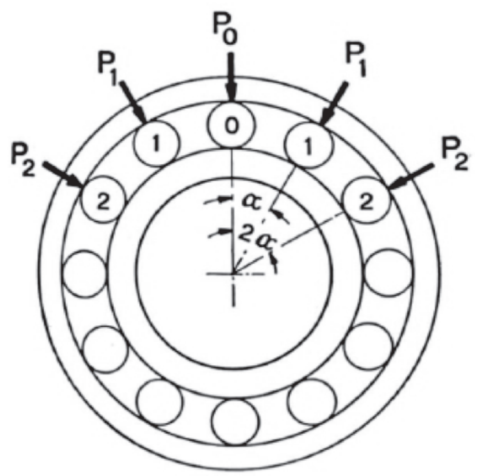


Figure 2.

Designating the total load applied to the row of balls, "P", and the vertical approach between inner and outer races, "δ", then

$$P=P_0+2P_1COS\alpha+2P_2COS2\alpha+\dots\dots\dots+2P_zCOS\alpha$$

Where z is less than one-fourth the total number of balls, and α = the angle between the balls.

Therefore, zα<900. The approach, δ0 corresponds to the ball load, p_0, δ_1 to the load P_1, etc.

Then $\delta_1=\delta_0COS\alpha$, $\delta_2=\delta_0COS2\alpha$, etc.

As previously stated, Hertz and Stribeck agreed that the total compression, δ_0 varies as $P_0^{2/3}$. Conversely, P_0 varies as $\delta_0^{2/3}$.

Thus, $P_1/P_0=\delta_1^{3/2}/\delta_0^{3/2}$, $P_2/P_0=\delta_2^{3/2}/\delta_0^{3/2}$ etc.

But, $\delta_1=\delta_0COS\alpha$, and $\delta_2=\delta_0COS2\alpha$, etc.

Therefore, by the substitution and simplification,

$P_1=P_0cos^{2/3}\alpha$ and $P_2=P_0cos2\alpha$, etc.

And $P=P_0\,(1+2cos^{5/2}\alpha+2cos^{5/2}\alpha+\dots\dots\dots2cos^{5/2}\alpha)$

Which is the load distribution equation

Bearing Section

Since the load-carrying ability of the balls is proportional to the square of their diameters, it is evident that the larger the balls, the greater is the proportion of the total load that each one can carry.

As the principal dimensions of ball bearings--that is, the bores, outside diameters and widths-have been standardized. It is the chief duty of the ball bearing designer to so proportion the non-standardised dimensions of the components that a Large and many balls as possible may be used consistent with safe race structure.

While the load-carrying ability of a ball bearing depends primarily upon the size and number of balls and their physical characteristics, these are, of course, only the media whereby the load is transmitted from one raceway to the other. Although the latter are supported and carried by shaft and housing, they are subjected to very heavy stresses. It has been shown that for proper functioning they must be provided with grooves in which the balls may roll, so that under Load, contact areas of such size are formed as to greatly reduce the unit stress.

Nevertheless, these stresses are of such magnitude that the height of the race shoulders exerts but little influence on the carrying capacity of the bearing. The thickness of the section of the race directly opposed to the ball is of extreme importance; its relationship to the diameter of the complete ring must be carefully studied and for balanced performance, can seldom be less than one-third of the ball diameter.

Opinions vary in regard to the proportioning of the metal between the two rings. On the one hand, the inner ring, in most applications (shaft revolving), has a very firm seat on the shaft, which may be more accurately shaped and finished than the housing bore. Shaft and ring thus more closely approach the stress resistance of a solid piece of metal. Moreover, the number of ball impacts per revolution, when the shaft revolves, is greater on the inner race than on the outer, thus requiring more metal to absorb the stress causing fatigue.

Though large ball size is desirable, it is possible to affect bearing endurance quite adversely if the percentage of bearing section occupied by the ball is increased to the point where the thickness of metal in the ball races is too small to properly absorb the stresses. Also, bearings, in which the ball race sections are sacrificed for the sake of extra-large balls, are much more sensitive to mounting fits and accuracy of shafts and housings, in that they must, at all times, be solidly and very accurately supported. The manufacture of a precious steel ball of great strength is a difficult art. Often the use of extra-large balls, unbalancing design, is evidence of weakness rather than strength.

Ball Bearings and Balls

Fatigue.

A continuous repetition of stress at a given place in an elastic material with gradually cause a loss of molecular strength called «fatigue». If continued length of time, this will result in a breaking down of the structure, called «fatigue failure.»

When a load is applied to a ball bearing at rest, so that deformation takes place in the balls and raceways, points within the contact areas are, as has previously been shown, principally under compressive stress, but beyond their confines the stress is largely tensile, the surface metal surrounding the contact areas being in pure tension.

It is apparent, then, that when the bearing is put into motion and the balls roll in the races, each point over which they pass is first under tension, then compression, followed by tension again.

Assuming, for example, that both races were driven in opposite directions, but with the same surface speed for both, so that the axes of rotation of the balls remained fixed, it is clear that any given point in the ball races, within the loaded zone of the bearing, would go through the cycle-tension-compression-tension-once for every ball passed during one revolution.

The magnitude of the stresses would vary, of course, depending upon the position of the balls in the loaded zone and maximum values would be reached only once per revolution.

The useful life of a bearing is normally limited only by the length of time the races and balls will resist fatigue failure due to this cycle of stresses. If the load, which determines the intensity of the stress, is either reduced or increased, the fatigue life of a bearing will also be reduced or increased according to the change in stress.

IKL engineers insist that fatigue in a properly designed and manufactured ball bearing must be evidenced by a surface flaking of the bearing members and never by actual fracture of the parts.

Under these conditions, the fatigue life of a ball bearing of reputable manufacture, that is, made by an organisation acquainted by research with the requirements, and mentally capable of controlling the most minute details to the ultimate of precision and uniformity, is equivalent to a certain number of revolutions, irrespective of the speed at which it may be operated. Bearing life in length of time will vary inversely with the speed.

The capacity rating for IKL ball bearings are founded upon the average life under known speeds and loads of large numbers of test bearings. Having, therefore, a definite relationship between the load ratings, the fatigue life, the imposed load, the speed and general service conditions, it is possible to select bearings for any application upon an accurate and economic basis.

Ball bearings of many makes may now be purchased in similar types, to the uninitiated, all apparently comparable to one another. Real worth, however, depends upon the uniformity of long Life, under adverse conditions. Where uniformity is obtained by the maintenance of sub-microscopic dimensions, ultimate quality is only assured by the mental and physical resources of the manufacturer.

Often bearings are given very high load ratings in order to appear unusually desirable to the buyer. Under such conditions, the length of life assured is seldom mentioned, and it is necessary to resort to the use of larger factor of safety in order to secure normal length of service.

Principal Requisites of a Capacity Formula

From static consideration only Stribeck concluded that the capacity, or permissible journal load, P. may be expressed in terms of the normal load, P0, which the heaviest loaded ball can safely withstand.

For the average bearing containing « n» number of balls (10 to 20) he found from the distribution equation that the relation

$P/P_0=n/4.37$

But $P_0 = K_2d^2$ (see page 3)

Therefore, $P = K_2nd^2/4.37 = Knd^2$,

Where K is a general constant determined from compression tests. This equation furnished a simple basis of comparison for bearings of different sizes but of the same transverse sectional characteristics and material

However, in extending the Stribeck capacity formula from one size of bearing to another, no rational factor was introduced which compensated for changes in the curvatures in the central plane normal to the axis of the bearing. Obviously, these variations materially affect the respective shapes of the contact ellipses, hence influence the intensity of the contact stress, although the computed maximum allowable ball load is not exceeded.

The problem, therefore, of developing a rational bearing capacity rating must include a more comprehensive analysis of all the factors entering into the shape of the contact ellipses so that the average unit, stress, S=P0/=ball load/contact areaα is held within a safe value. Thus, the rational expression for the static capacity should read:

$P = KCnd^2$,

where C is the «conformity factor,» a function of the curvature of the ball relative to both the transverse curvature and the curvature in the plane normal to the axis of the bearing. Since the value of C for the contact in the inner race may differ from that of the outer, the lesser value should be employed to obtain the safe rating.

The rational formula for safe loading during running conditions must include some factor based upon the endurance of the bearing under high stress slowly repeated, or under low stress rapidly repeated; in other words, a speed factor, so that regardless of the speed at which the bearing is running, the rated load will be such that the endurance life of the bearing will be the same.

Ball Bearings and Balls

Running Capacity

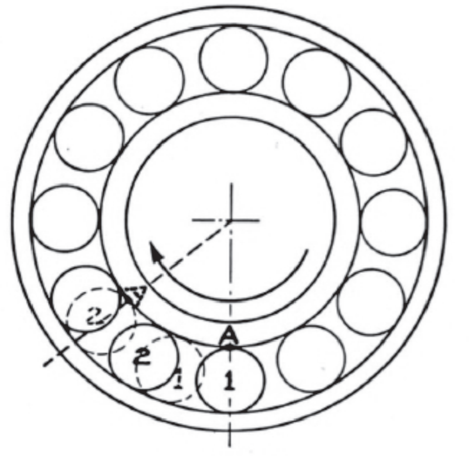


Figure 3.

From the preceding discussion, it may be seen that the static capacity of a ball bearing is determined in a manner similar to that of any static structure, such as a bridge. The safe load is found, that is, one which does not cause a sudden, permanent deformation. But for running conditions, safe loads must be considered from another view point. When a load is repeated a considerable number of times on the same part of a structure, it will cause the material in that region to ultimately fail through fatigue. What really happens is that even though the stress is within the elastic limit, it will eventually cause a microscopic parting or change in the molecular structure which, after sufficient repetition of the stress, will gradually be magnified until it becomes great enough to cause a breakdown in the material. Extensive experiments with fatigue, in various kinds of metal, show that for every material and type of loading, there exists a definite relation between the stresses imposed and the number of times the material can withstand them. The mathematical expression for this relation is given by

$i = ks^n$,

where s is a stress at the point in question;

i is the number of times this stress is repeated, and

k and n are constants which depend on the nature of stress and the material used.

It may be seen from the above expression that the rate at which the stress may be applied does not affect the total number of stress repetitions. This general fatigue law is applicable to ball bearings, as can be seen by following a stress point on one of its members, for example, such as point A on the inner race, as shown on figure 3.

If the inner race is rotated, point A is relieved of load imposed by ball 1 but since the velocity of the inner ball race is greater than the linear speed of the balls, point A is again stressed by ball 2 ahead of ball 1 at some angular position, as shown in dotted lines on the diagram

This repetition of stress is kept up until point A leaves the loaded zone and is repeated as it enters the loaded part of the bearing.

Here, therefore, a surface stress on the inner ball race is repeated and if the bearing is run long enough, either this point or others similarly loaded, will be fatigued. The constants in the expressioni = kn have been established for IKL ball bearings through extensive tests, covering a considerable number of sizes and types. From the geometry of each bearing, i may be given in terms of the total number of revolutions, and s in terms of the externally applied load, and conformity of the race grooves.

It is clear that when bearing life is expressed in hours and r.p.m. instead of total number of revolutions, the life in revolutions is used up faster at higher speeds than at lower ones. Hence, if it is desired to obtain the same life for all speeds, the load must be reduced by a corresponding amount. This explains why, in the IKL bearing ratings, the load diminishes as the speed increases.

Functions of the Separator

In the very early types of ball bearings, it was customary to proportion the components so that the space between the two rings could be completely filled with balls. This construction, however, gave rise to considerable trouble, since the rubbing velocity of ball surfaces in contact and rotating in opposite directions is twice that of the velocity of the driven ball race. Also, rubbing pressures between the balls, particularly under radial load and at the higher speeds, would be considerable, since with the «full» type bearing radially loaded, the balls must force themselves from the unloaded into the loaded zone with pressures exerted at the ball equators or points of highest surface speed. Full type bearings have, in certain instances, been employed for comparatively high speeds, but only where conditions of loading impose sufficient thrust upon each ball to maintain spacing.

From the foregoing, it is evident that the first duty of the ball separator or cage is not only to space the balls equally so as to insure proper load distribution and balance within the bearing, but also to maintain control of the balls in such a manner as to produce the least possible friction through sliding contact.

Though it is not proposed here to enter into the details of separator design, a very brief analysis may be given of the principal separator stresses, which will serve to indicate the need for strength and accuracy in construction.

It may be noted that in the radial plane, when the balls enter the loaded zone and take on a gradually increasing load with consequent increase in deformation (experienced also by the raceways) the relative proportion between the diameters of the driving and driven members, that is, the inner race and balls, also undergoes a slight variation, tending to slow up the movement of the ball centre until the heaviest loaded position is reached. From this point on, the process is reversed. The movement into the loaded zone tends to compress the separator between the balls, each preceding ball retarding its movement more than the following and, leaving the loaded zone, each preceding ball tends to pull the separator.

In order to avoid serious pressures in the tendency to compress or stretch the separator, it is necessary for the balls to have a certain freedom of movement in the direction parallel with the race groove and it is also necessary that the contact of the separator with the ball poles be large enough to maintain ball positions without undue stresses in the pockets.

Possible misalignment is another factor which necessitates a certain clearance around the balls in the separator pockets. When operating in a misaligned condition, the balls in diametrically opposite parts of the bearing must roll higher up on the raceways, that is, away from the centre of the grooves, but on opposite sides. Thus, during each revolution, the balls must rapidly change from one side of the raceway to the other. Under such conditions, too small a clearance in the separator pockets would cause binding and ultimate failure of the separator.

Although the bearing manufacturer can thus anticipate most of the separator trouble that might occur due to improperly mounted or misaligned bearings, excessive clearances in the ball pockets must be avoided if the separator is to retain its properly concentric position in the bearing.

For this reason, the separators of bearings for general service are designed with a certain degree of flexibility, but combined with considerable strength, so that they are enabled to yield within reasonable limits, yet have the strength to continually resume their normal shape.

Ball Bearings and Balls

PRINCIPLES OF THE BALL BEARING

Ball Bearings and Balls

In high speed bearings, lightness and strength are particularly desirable, since the stresses increase with the speed, but may be great ly minimized by reduction of separator weight.

Under pure thrust, all of the balls share equally in the load and there is less need of spacing assistance. However, absolute thrust loads are comparatively rare, the most common load being made up of radial and thrust components in which the duties imposed upon the separator may lie anywhere between the conditions of the purely radial bearing or the one used for pure thrust.

In very large bearings, it is sometimes the practice to carry the separator upon smoothly ground shoulders of the inner ring. Although, by this method, there is sliding between the separator and inner ring, the speeds involved are seldom high, as the separator speed is approx -imately only half that of the shaft. Again, the weight of such a separator, as compared with the large amount of film area supporting it in operation, is very small.

Double Row bearings of any type require individual separators for each row of balls, since even the most perfect construction and accuracy in the various parts may permit a slight difference in ball speeds between the two rows, particularly under the influence of vary ing mounting and load conditions.

Proper lubrication plays a most important part in the functioning of a ball bearing separator, and this subject may be referred to in Section IV.

Loading Groove and Non-Loading Groove Bearings

Since, as has been shown, the number of balls is an important factor in the capacity of a ball bearing, the majority of types have been designed to contain the greatest number possible, consistent with ball diameter and separator space. However, in a radial bearing having ball race grooves of adequate depth together with proper thickness of metal at the thinnest part of the race section, the sides of the grooves, or shoulders, definitely limit the opening between the rings.

In order to introduce the last few balls (usually three or four in number) excluded by the race shoulders, a so-called loading groove, or filling notch, is cut in the shoulders on the same side of both rings. Extremely accurate gauging in manufacture assures that this notch does not approach the bottom of the ball raceway, so that under load, the contact areas of the balls with the raceways do not impinge upon it.

Assembly is accomplished by insertion of as many balls as possible with the rings eccentrically positioned. The rings are then brought into concentric relation and rotated until the filling notches align. The remaining balls may then be inserted into the bearings, their entry usually being facilitated by judicious expansion of the outer ring.

By virtue, therefore, of the large complement of balls, the loading groove, or as it is sometimes called, the maximum capacity type bearing, provides the greatest radial capacity for a bearing having one row of balls. This bearing is not suitable for pure thrust, and com -bined loads may be applied only where the axial component is not great enough to cause the contact areas of balls and races to extend over the filling notch, due to the relative displacement of the two rings. For combined loads, therefore, the amount of thrust is limited to not more than one-half of the applied radial load.

Non-loading groove bearings are identical with the type just described, except that they contain only as many balls as can be introduced without the filling notch. Owing to the absence of this notch, they are less sensitive to variations in the load angle (the angle between the plane of the bearing and the direction of the load) and are, therefore, capable of withstanding combined loads in which the thrust component is relatively high, but are not intended for pure thrust loads.

The radial capacity, as compared with loading groove bearings, stands approximately as the relation of the number of balls in the two types.

Ball Bearings and Balls

PRINCIPLES OF THE BALL BEARING

Ball Bearings and Balls

Angular Contact Bearings

The loads which must be resisted by ball bearings in general service vary to such an extent, particularly as to the direction in which they are applied, that a number of types are required to meet all conditions to the best advantage. By far the greater percentage of loads are neither purely radial nor axial, but some combination of the two.

A radial bearing, particularly the non-loading groove type, is capable of withstanding a certain amount of thrust by reason of the change in angularity, with respect to the balls and races, that takes place when the load is applied. Aside from such factors as the initial degree of loose ness, or radial play, with which the bearing may be assembled, this change is due to the difference between the radii of curvature of the balls and their raceways.

Thus, when a thrust load is applied, the rotating, or driven ring is permitted a certain amount of axial movement which is not brought to a stop until the balls have rolled in their grooves, in the direction of the load, far enough to cause a compression of races and balls sufficient to balance the load or bring it to a state of equilibrium. This condition is illustrated in the greatly exaggerated diagram, figure 4. Although thereafter the addition of more load would produce only a relatively small increase in axial de flection, the initial movement permitted would be undesirable in certain mechanisms where extremely close axial location of parts is essential.

The shoulders in radial bearings are made of such height as to give ample ball support under maximum combined loads or a reasonable amount of misalignment. To make them higher would not only render it difficult or impossible to insert the proper number or size of balls, but would also occupy space required by a correctly proportioned separator.

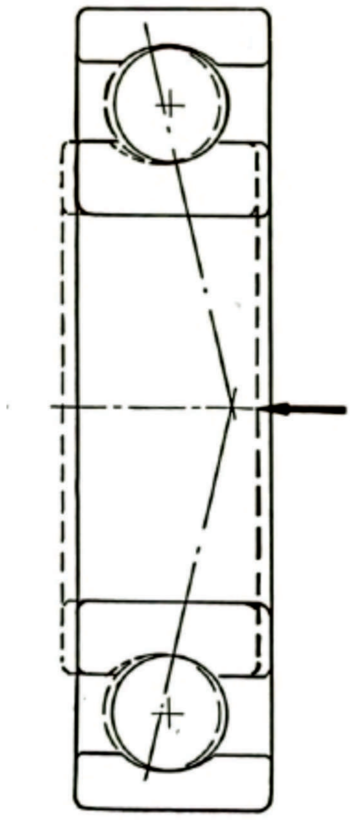


Figure 4.

In order to better resist thrust or combined loads, ball bearings have been designed by IKL in which the balls and raceways normally contact at a pre-determined angle with the plane of the bearing. These are made with high shoulders on the thrust sides of the rings and low shoulders opposite, as in figure 5, thus correctly sup porting the balls under angular loads and permitting introduction of the maximum number and size of balls. With such bearings, particularly when made as are the Radax bearings, in three types, having different contact angles, it is possible to select a bearing whose angle of contact would so nearly coincide with the resultant angle of the combined loads as to very definitely restrict any axial movement.

Since a ball bearing can support load with the least deflection when the load is applied in a direction corresponding to the centre plane of the contact area, and de flection is greatest when a load is applied at a right angle to this plane, it is obvious that a Single Row Angular Contact bearing is neither suited nor intended for pure radial load. Unless used for pure thrust load in one direction, they must be applied either in pairs (duplex) or one at each end of the shaft, opposed.

The Double Row Angular Contact bearing is, in effect, a combination of two Single Row Angular Contact bearings, incorporating such an angle as to give the greatest possible rigidity, both radially and axially, under general combined load conditions. Preloading, as described shortly hereafter in this section, is utilized in both Single Row and Double Row Angular Contact bearings to decrease deflection under loads of any nature.

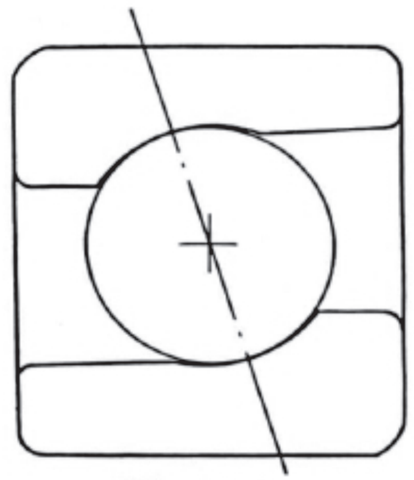


Figure 5.

As the principles underlying Angular Contact bearing construction are derived from the manner in which the applied load acts and as the load is nearly always radial and thrust in combination, a short definition may be desirable. A combined load may be either a radial load plus a certain thrust load, or a load applied to the bearing at any angle lying between 0° and 90° from the plane of the bearing and acting with a force which can be split up into radial and thrust components. Geometrically, a combined load is the square root of the sum of the squared radial and thrust loads,

$$\text{combined load} = \text{Sqrt}(\text{load}^2 + \text{thrust}^2)$$

Therefore, a combined load is always an angular load and its angularity is such that the thrust load is the tangent of the load angle.

Since a combined load may thus vary anywhere between the practically pure radial to pure thrust, and since the capacities of ball bearings cannot very well be given for such an infinite number or load angles, it is necessary to convert combined loads into «equivalent radial load» through the use of angularity factors.

Ball Bearings and Balls

PRINCIPLES OF THE BALL BEARING

Friction

Friction in a ball bearing has been the subject of numerous tests, none of which absolutely agree, except to indicate that the subject deals with an extremely minute quantity and that the results obtained may be altered by differences in bearing construction, such as ball race curvature and separator design, as well as variations in bearing lubrication. It is shown, however, that both static and kinetic friction are less than in other bearing forms.

In the ball bearing, friction may be divided into four sources: first, the sliding friction due to sliding contact between the balls and separator pockets; second, to rolling friction or internal friction of the steel due to deformation of the balls and races under load; third, the slip (negligible insofar as practical functioning of the bearing is concerned), which may occur in the contact areas, due to difference in radii length at the centre of the contact area and at the ends of the ellipse, and fourth, friction due to windage at the higher speeds.

The United States Bureau of Standards has endeavoured to determine a coefficient of static friction for balls placed between grooved surfaces or plain hardened steel plates, and subjected to loads of varying magnitudes. It was found that for balls of respectively 1», $1^{1/4}$ and $1^{1/2}$ diameters, it was practically constant under growing loads, and equal to .00055, until critical loads of respectively 1300, 1700 and 2200 pounds had been reached, when it began to increase.

The coefficient of friction for ball bearings (of use principally for purposes of comparison) has been found, by tests, to remain within relatively close limits under different loads and speeds, varying according to operating conditions from .0005 to .003, with a general average around .001

Since the frictional resistance, in point of power consumed, is so very small a quantity with which to deal, instruments capable of accurately indicating the relative values of the different factors contributing to friction in ball bearings have not been found. However, very careful tests made to determine the total frictional torque of ball bearings indicate that under constant speed, it varies approximately as the square of the load.

Under normal conditions, this would mean that the resistance would be very small. As a specific illustration of this, the frictional torque of a 3202 bearing, operating under a load of 150 lbs., at 1140 r.p.m., has been found to be .00062 h.p. With the load increased to 250 lbs. (which would be greater than the shaft for this size of bearing would normally be required to withstand) the frictional torque would be .00171 h.p. With a y2-h.p. motor driving the shaft, the frictional resistance, due to the ball bearing under the heaviest load, would still leave .4983 available h.p.

Bearing Deflection and Bearing Design

From the shape and proportions of the contacting parts in a ball bearing and the laws governing their elastic deformation, already covered, it is understood that deflection, or relative movement, of the bearing components must have some influence upon the parts to be supported by the bearing.

Resistance to deflection in the bearings, with consequent reduction in the radial or axial movement permitted in the machine parts supported, may be achieved in two ways. One of these is through the modification of bearing design and the other by means of preloading- The former may best be illustrated by the Double row Angular Contact bearing.

In this bearing, rigidity is made a fundamental characteristic primarily through the disposition of the contact angles, which are made to converge outside of the outer race, as in figure 6a. Had the contact angles been reversed, however, so as to converge inside the bearing, as in figure 6b, a misaligning load applied at a given point on the shaft would result in much higher ball loads and consequently, greater de flection. This is made sufficiently clear by the diagrams, in which the arrows indicate the direction and magnitude of the forces involved. The relative deflections resulting are also indicated.

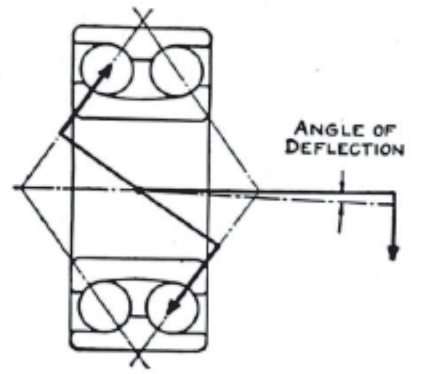


Figure 6a.

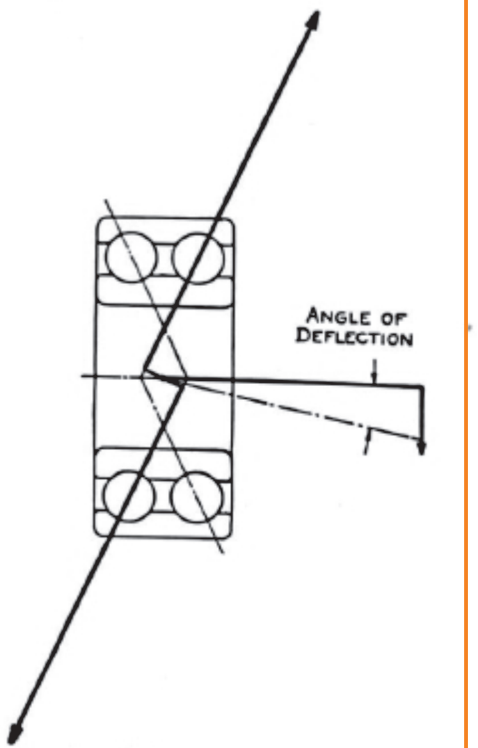


Figure 6b.

Preloading— Its Influence on Shaft Positioning

It is becoming increasingly realized by designers and machine builders that the life of their machines and the accuracy of the work they produce depends upon the fixity, precision and permanence with which shafts, spindles and other moving parts are cantered. Grinder spindles, for instance, must be mounted with very close fits. Wear of a plain journal or variation in the thickness of an oil film produces chattered work.

To overcome these defects the ball bearing is the only recourse, yet it has already been shown that even the extremely hard elements of such a bearing are subject to some deflection under load.

The provision of the most friction-free, yet rigid, accurate and permanent means of locating spindles and shafts thus far devised has been the accomplishment of the IKL research organization. First, through their pioneering of the Angular Contact ball bearing in both Single and Double Row types, and later, by their discovery of the possibility of installing such bearings under an initial or preload, this organization may be said to have made the modem high production, high precision machine an economic success.

It must be recognized that in a purely radial hall bearing, no increase in load can possibly serve to increase the accuracy of centring of the supported shaft. An angular contact ball bearing, however, may be so designed that increase of load increases the firmness of centring.

In figure 7 is given a typical deflection curve of a non-preloaded Angular Contact bearing.
From this curve, it is seen that at first the deflection increases rapidly with the load. But if means are provided for eliminating this low load deflection part of the curve in the design or installation of a ball bearing, further loading will cause exceedingly slight displacement. This is the principle applied in preloading.

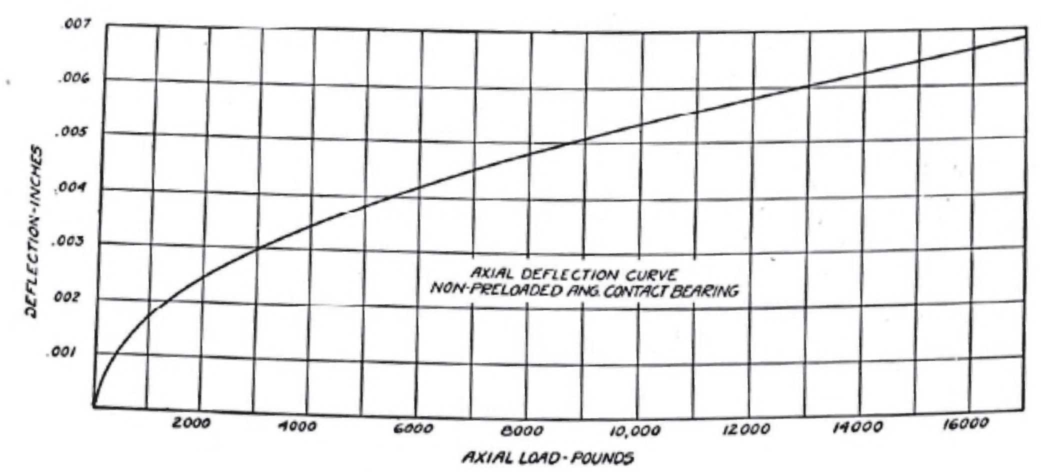


Figure 7.

Of course, it is recognized that only in the ball bearing is the inherent friction loss so small that advantage can be taken of the absolute precision of shaft centring which is gained when preloading is applied.

The principal effects of preloading upon the rigidity of a shaft may be shown by means of an installation employing bearings of the type and size for which a deflection curve has been given. Where two bearings are mounted opposed under a preload, as in figure 8, and an external or work load is then imposed, the front bearing does not, of course, carry the working load plus the original preload.

Referring to figure 8, let it be assumed that sleeve «B» is moved toward the right by the preloading nut, «N,» to such an extent that an internal load of 3000 lbs. is imposed on the bearings. This means that the shaft between the bearings is in tension by a similar amount and that the bearings have each been deflected a definite distance.

Now let an external thrust load of 2500 lbs. be applied to the spindle. This additional load causes a slight increase in deflection of the front bearing, but at the same time, decreases the spindle tension, thus partially relieving the preload on both bearings. Therefore, the front bearing is carrying less than the sum of the external and internal loads and the rear bearing is carrying less than the original preload.

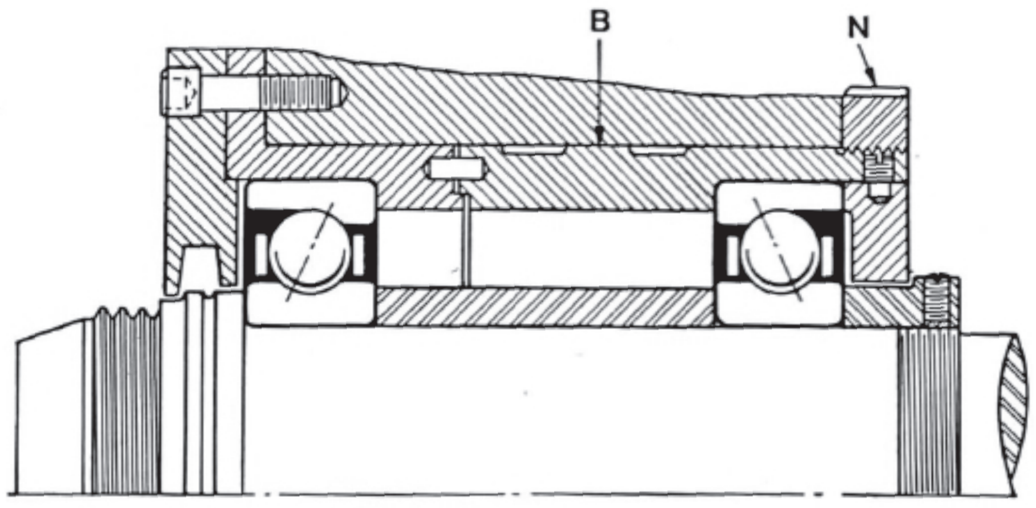


Figure 8.

Graphical Determination of Bearing Deflection

This load distribution and the total axial deflection, due to the external load, may be graphically determined by use of the deflection curve previously illustrated and reproduced in figure 9

Since the rear bearing has been relieved of some preload by deflection of the front bearing (due to application of the external load), it follows that the rear bearing must have partially resumed its original shape, and substantially at the same rate at which it was deflected. Thus, if the sum of the internal and external loads (5500 lbs.) is located on the chart, and that section of the deflection curve up to 3000 lbs. (the preload) is inverted and moved to this point (inverted because when relieved the bearing must resume its shape at the rate indicated by its deflection curve, and moved because the rear bearing relief starts at 3000 lbs., but is caused by a momentary load of 5500 lbs. on the front bearing) the point of stability is reached at the intersection with the original curve, being found in this case to correspond to a load of 4400 lbs. for the front bearing.

Then, since the preload relief is the same for both bearings, the relief, «R,» must be subtracted from the original load on the rear bearing, giving for this position a total of 1900 lbs. In all cases, the difference between the loads on the front and rear bearings will equal the external load.

It will be observed that the total axial deflection under the work load mentioned amounts to .00057», as compared to .0026» deflection, which would have occurred were the preload not employed.

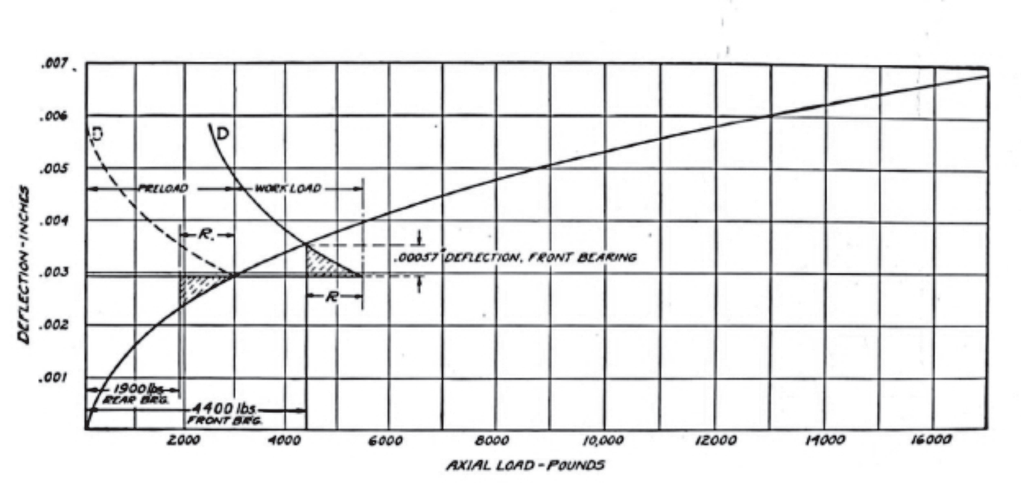


Figure 9.

External Load Required to Relieve Preload

When the preload of 3000 lbs. was imposed in the case just cited, both bearings were equally loaded and were, therefore, compressed, or deflected, an equal amount. Reference to the curve, figure 9, shows that the deflection in each bearing, due to this initial load, amounted to .0029». Thus, in order to render the preload entirely ineffective, it would be necessary to subject the front bearing to a thrust load great enough to cause an axial deflection in that bearing of twice .0029», or .0058». Again referring to figure 9, it will be found that it requires a load of 11,750 lbs. to produce this deflection. In other words, the relieving load, in this case, would have to be approximately four times as great as the preload.

In practically all cases where a preload may be utilized to increase the rigidity of a bearing, the amount of preload, in comparison to the maximum thrust capacity of the bearing, is very small. Also, as has been shown, when the external load is applied to the bearing, the pre load is reduced according to the amount of the external load and the effect on bearing endurance is negligible.

Ball Bearings and Balls

Preload in Double Row Bearings

The objective of preloading in Double Row bearings is, of course, the reduction of deflection under any kind of external load. This is well illustrated by comparison of the axial deflection curves, figure 10, one of which is for a single row of balls (equivalent to a non-preloaded bearing) and the second for a preloaded Double Row bearing.

In this Double Row bearing, it requires an external load of 5000 lbs. to relieve the preload and this point in the curve is indicated by a change in the deflection characteristics. After this point has been passed, the curve for the preloaded bearing is exactly parallel to that of the other, the distance between the curves being .0013».

Therefore, after the preload has been entirely relieved in a Double Row baring, its axial deflection under any load will always be less than a similar non-preloaded bearing.

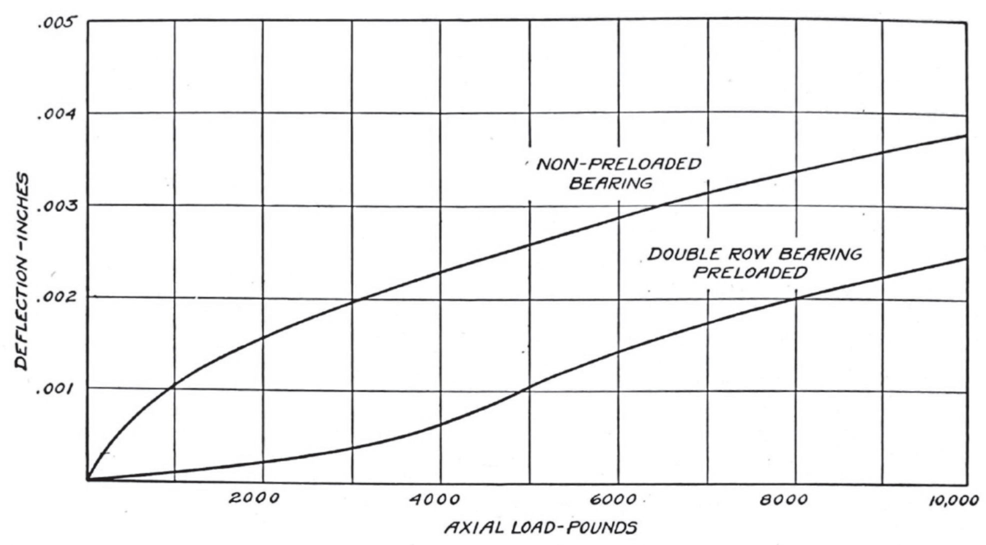
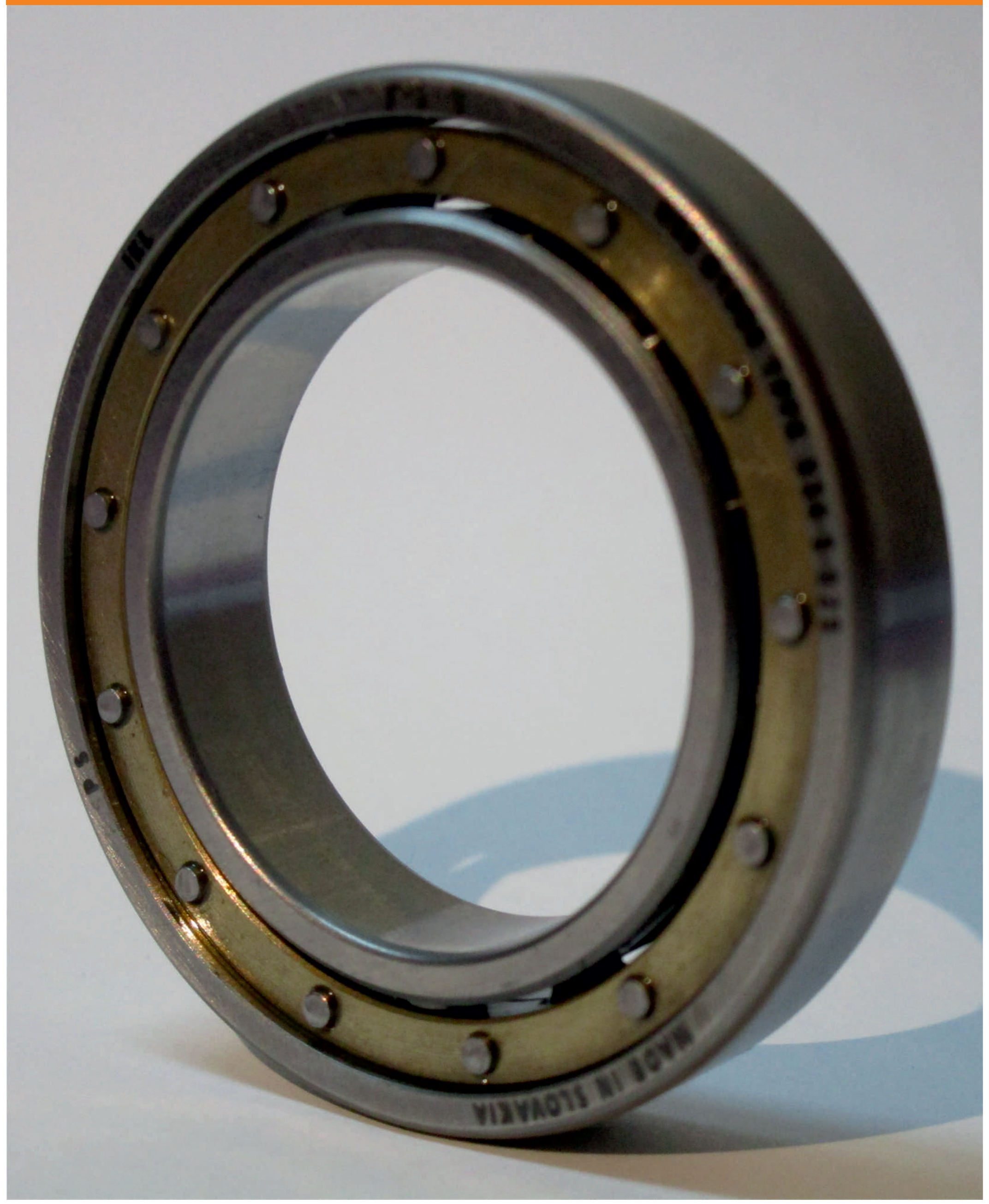


Figure 10.



Ball Bearings and Balls

BEARING SECTION

The selection of a ball bearing for a specific installation involves the consideration of three major points:

The bearing must be of the series best suited to the installation, both from the standpoint of physical proportions and capacity.

The type of bearing employed must have either the radial, the thrust or the combined load characteristics required by the loads to be imposed.

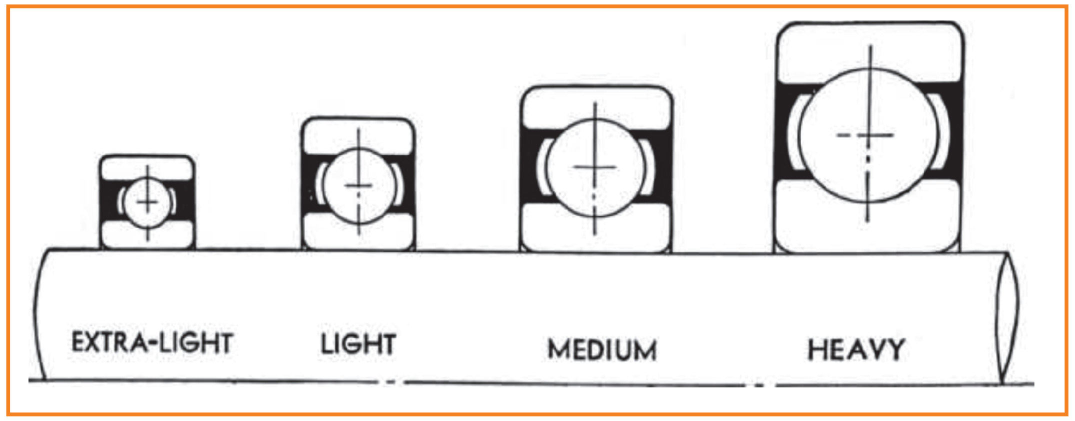
The size of bearing must be such that the term of service it will deliver will be sufficiently close to the length of service required as to make it an economical purchase. Thus, if too large, the bearing would possess capacity and life, a considerable part of which would never be used. The cost of this excess capacity would be needless expense, hence uneconomical.

In general, therefore, the first step in the selection of a bearing is to determine which series will best meet such physical aspects of the machine as shaft size and housing space available.

Determination of Bearing Series

Ball bearings have been standardized in four different proportions or series for each bore size made. That is, most standardized bearings may, with the exception of a few types designed for a special or limited variety of service, such as Magneto or Front Wheel bearings, be obtained in four sizes as regards width, outside diameter and capacity, but made to fit the same shaft.

Comparative proportions of these four series, known as Extra light, Light, Medium and Heavy, are shown in figure 1.



Extra-Light and Light Series

In determining the proper series of bearing to use, it is important to remember that while two Single Row Radial or Single Row Angular Contact bearings applied at the ends of a shaft will provide rigid support at the bearing positions against radial or combined loads, they do not prevent bending or deflection of the shaft itself at points between the bearings. For this reason, where shafts are long, and loads are imposed midway between bearings, it is frequently necessary to use a larger diameter shaft, while, at the same time, an increase in bearing capacity is not required.

For such installations, Extra-Light or Light series bearings would be employed, since by their use an increase in shaft size would involve minimum increase in bearing capacity or change in housing size.

In some applications, where shafts are hollow, and are of comparatively large diameter in proportion to the load to permit adequate size of parts passing through the bore, Extra-Light series bearings are usually most suitable.

Therefore, as a general rule

Extra-light or Light series bearings are the logical choice where loads are moderate, and shaft sizes comparatively large; Where housing space requires the smallest bearing width and outside diameter available for a given bore size.

Ball Bearings and Balls

Ball Bearings and Balls

Medium Series

Medium series bearings provide a capacity increase over the light series of approximately 30 to 40%, but being proportionately wider, and of larger outside diameter, they occupy the greater space on the shaft and in the housing, necessary to support the heavier loads.

In the majority of cases, where loads are very heavy and considerable bearing capacity must be employed in proportion to bearing bore or desired shaft size, medium series bearings are desirable.

Heavy Series

Heavy series bearings provide a capacity increase of approximately 20 to 30% over medium series, but since bearings of the medium series have as much capacity as ordinary steel shafts require, the heavy series is usually applied to specially proportioned shafts, and finds less use than either of the other two series.

Bearing Fits*

Ball bearings are usually applied with the rotating ring a press fit and the stationary ring a push fit, the tightness or looseness depending upon the service intended. This rule is founded upon the following facts:

Under normal conditions a press-fitted ring will not slip or turn on or in a rotating shaft or housing.

A ring fitted with the correct looseness to a stationary shaft or housing, is allowed to creep very slowly, thus avoiding prolonged stressing of any one part of the raceway.

Ball Bearings and Balls

BEARING INSTALLATION

Bearing Creep

The very slow creep of a bearing ring, mounted with a close push fit in a stationary part, should, under no circumstances, be confused with the comparatively rapid turning of a ring (improperly) mounted loose in or on a rotating part.

The former represents a relatively light force, the creep being controlled by the degree of fit used or by normal clamping of the bearing rings where axial location must be maintained. Any wear normally caused by such creep is too slight to be of consequence.

The latter creep, however, is due to a definite and powerful rotative force, which, in the case of an inner ring, figure 1, may be likened to an internal gear drive with the shaft driving a bearing ring.

It would appear that practically pure rolling motion takes place between the shaft and ring, but actually considerable sliding occurs. Although various forces, difficult to determine with exactitude, contribute to this slip, a major cause may be given as follows:

At that point (L) within the area of contact where load is greatest and where both shaft and ring are deformed to the maximum extent, the radii of the surfaces in contact are identical. However, on both sides of this point, the radii begin to return to their normal values (S and R). These being different according to the looseness of the fit, a difference in velocities is obtained which produces sliding and wear, so that slippage increases with the looseness of fit, with the load and with the speed.

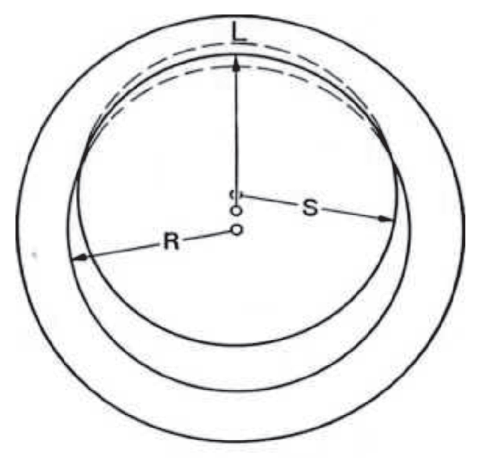


Figure 1.

Experience has proven that the evil effects of a loose fit of a bearing ring on a rotating shaft cannot be counteracted by any practical form of keying. Not only do keyways seriously weaken the ring due to the interruption of continuity of the section, but the forces existing are so great as to shear any ordinarily practicable key.

Accuracy and Proportions of Mounting Parts

In order to achieve maximum results with ball bearings, both as to length of service and accuracy of support obtained, it is necessary that all machine parts closely associated with the bearings should conform to a reasonably high standard in workmanship and finish. Also, they should be adequately proportioned so as to resist the various stresses involved; that is, so as to assure a minimum of deflection.

Corner Fillets

For maximum strength, both shafts and housings (particularly the former) require definite fillets at the junction of bearing seat and locating shoulder. However, it is very important that such fillets be of smaller radii than the bearing comers (figure 2) so that there is no danger of the bearing axial location being accomplished through interference at the corner radius rather than the proper face to shoulder contact. To insure proper mounting, the S.A.E. has given as a standard the maximum shaft and housing fillet radii which the bearing corners will clear. These radii are given in the bearing dimension tables, in Volume I.

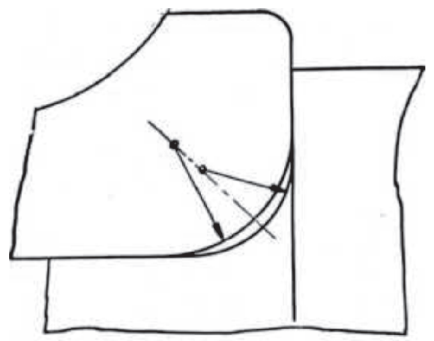


Figure 2.

BEARING INSTALLATION

Shafts

As a rule, there is little trouble experienced with inaccuracies in shafts. Bearing seats and locating shoulders are turned and ground to size with the shaft held on centers and, with ordinary care, there is small chance for serious out-of-roundness or taper. Shaft shoulders should be made, so far as possible, to the recommended heights given in Volume I, but in any case should present sufficient surface in contact with the bearing face to assure positive and accurate location.

Where an undercut must be made for wheel run out in grinding a bearing seat, care should be exercised that no sharp corners are left, for it is at such points that fatigue is most likely to result in shaft breakage. It is best to undercut as little as possible and to have the undercut end in a fillet instead of a sharp corner.

Where clamping nuts are to be used, it is important to cut the threads as true and square as possible in order to insure even pressure at all points on the bearing inner ring faces when the nuts are set up tight. It is also important not to cut threads so far into the bearing seat as to leave part of the inner ring unsupported or carried on the threads.

Excessive deflection is usually the result of improperly designed or undersized machine parts. With a weak shaft, it is possible to seriously affect bearing operation through misalignment due to shaft deflection. Where shafts are comparatively long, the diameter between bearings must be great enough to properly resist bending. In general, the use of more than two bearings on a single shaft should be avoided, owing to the difficulty of securing accurate alignment. With bearings mounted close to each other, this can result in extremely heavy bearing loads.

The use particularly of two Double Row bearings of the Type 5000 on the same shaft, either free in the housing or clamped, should not be undertaken unless the application has received the full approval of IKL engineers.

Housings

Design is as important as careful machining in construction of accurate bearing housings. There should be plenty of metal in the wall sections and large, thin areas should be avoided as much as possible, since they are likely to permit deflection of the boring tool when the housing is being finish-machined.

Wherever possible, it is best to design a housing so that the radial load placed on the bearing is transmitted as directly as possible to the wall or rib supporting the housing. Diaphram walls connecting an offset housing to the main wall or side of a machine are apt to deflect unless made thick and well braced.

When two bearings are to be mounted opposed, but in separate housings, as is often the case with Radax bearings, the housings should be so reinforced with fins or webs as to prevent deflection due to the axial load under which the bearings are opposed.

Where housings are deep and considerable overhang of the boring tool is required, there is a tendency to produce out-of-roundness and taper, unless the tool is very rigid and light finishing cuts are taken. In a too roughly bored housing there is a possibility for the ridges of metal to peen down under load, thus eventually resulting in too loose a fit for the bearing outer ring.

Spacers and Clamping Nuts

Opportunities for error increase with the number of separate parts used in a bearing application and it is always a good rule to employ as few spacers, washers and similar parts as possible.

All spacers, clamping nuts and washers should be faced square and true in relation to their bores, so that when the assembly is locked up, there is a minimum tendency toward misalignment or bending of the shaft. Where relatively long spacers are used to separate the bearings, they should be made heavy enough to properly resist the compression load imposed when the bearings are clamped tight. A spacer having walls thick enough to withstand the clamping load without serious distortion will also be thick enough to have ample clamping face for bearing location.

Clamping and Retaining Methods

Where ball bearing rings are properly press fitted, they will not creep excessively or turn on the shaft or in the housing under normal conditions. Therefore, where suitable shoulders can be provided in a mounting design to definitely locate the bearings against endwise movement, it is frequently unnecessary to employ clamping members.

However, in most applications where thrust is present, it is usually desirable to resist the axial load with a single bearing leaving the remaining bearing or bearings on the same shaft to perform radial duty only. Obviously, under such conditions, particularly where the thrust is subject to reversal of direction, it is necessary to clamp the thrust-resisting bearing in place.

Ball Bearings and Balls

Clamping Inner rings

Where bearing rings must be clamped, the type of device employed is determined principally by the nature of the service to which the bearing will be subjected. Thus, where the loads are heavy or the mounting is to be operated under a steady pounding or vibration, the device used must be of a kind which can be securely locked in position after being tightly set up. Since the nut which can be positively locked is required to a greater extent than any other, the type shown in figure 3 has been standardized by the principal ball bearing manufacturers.

Here a lock washer, interposed between the bearing and the nut, is keyed to a slot in the shaft. Six wings or ears on the washer are so spaced that one of the four slots in the nut will so nearly line up with a wing when the nut is set up tightly that only a slight movement of the nut one way or the other is necessary before the wing can be bent into the slot, thus interlocking the assembly. This device is quickly applied or removed and is positive, but requires the machining of a keyway in the shaft which, in some installations, is not convenient.

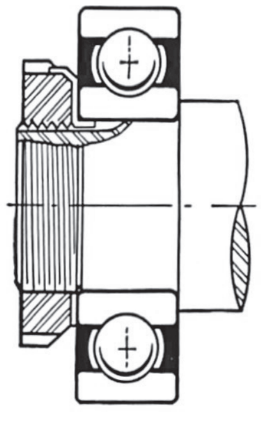


Figure 3.

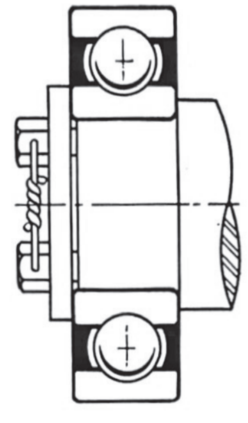


Figure 4.

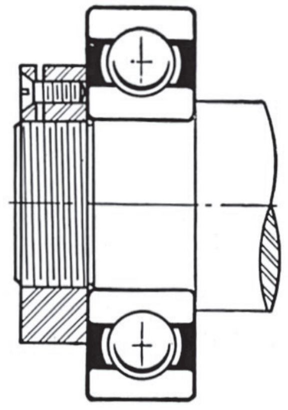


Figure 5.

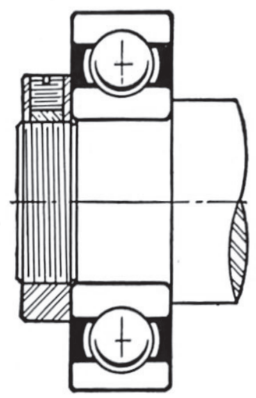


Figure 6.

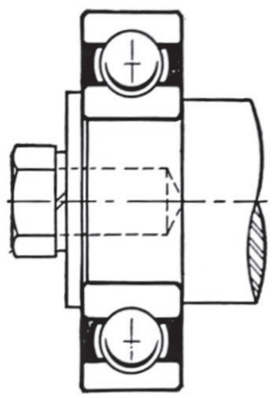


Figure 7.

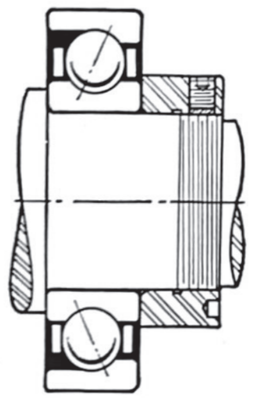


Figure 8.

Under some conditions, sufficient shaft extension to allow threading for a standard locknut is not possible and yet a positive clamping device is necessary. A method frequently adopted in such cases is that shown in figure 4. In this, a heavy washer is clamped against the bearing face by means of two cap screws which are threaded into the shaft. The screws have holes drilled in their heads so that a wire passed through and twisted, as shown, prevents any possibility of loosening in service.

Another type which, though not positively locked when set up, is capable of being more securely held in place than is possible with a plain nut, is shown in figure 5. This is the split nut using a screw to cramp the thin split section against the shaft threads. To be most effective, this nut must be so cut that the slot extends past the center of the bore of the nut and near enough to one side so that the desired cramping can be obtained with the size of screw used. A screw applied exactly as shown will lock more effectively than will one which forces the split portions apart and will also give a more uniform clamping pressure at all points on the bearing race, thus minimizing the possibility of misalignment. It is advisable to use a screw of the largest diameter, consistent with the nut size. This screw can be either as shown or a standard hexagon or fillister head.

Another device in which the locking is obtained by means of a setscrew is shown in figure 6. Either a round nut with slots or spanner holes, or a standard hexagonal nut may be used, the setscrew being as large as can be introduced without seriously weakening the nut. The screw should fit tightly so as not to easily lose its locking pressure, and in order that the threads on the shaft may not be damaged, a small slug of soft metal should be placed between the screw and the shaft.

The clamping method illustrated in figure 7 may be used in certain instances where it is not desirable to thread the shaft, and a device as positive as that shown in figure 4 is not required. The washer should be of such thickness as not to spring too much at the center when the screw is set up tightly, since this would reduce the contact surface between the washer and the bearing race. A lock washer placed under the head of the screw will assist in keeping the assembly tight. In such a mounting, if the direction of rotation remains constant, it is an additional security against loosening if the clamping screw is threaded against the direction of rotation; that is, so as to tighten rather than loosen, should the bearing turn upon the shaft.

Where extreme precision is essential, the clamping face of the nut must be square with the axis of the shaft. In such cases it is often advisable to utilize a nut of the type shown in figure 8. This nut should be an easy fit on the threads so as to prevent binding where the ground piloting bore of the nut fits over the shaft. The length of the piloting surface of the nut should at least equal the length devoted to the threads.

Ball Bearings and Balls

Locating Shoulders for Clamping Bearings

When a bearing inner ring is to be clamped, it is essential to provide a sufficiently high shoulder on the shaft to locate the bearing positively and with sufficient accuracy to meet the requirements of the installation.

There are applications where the difference between the bearing bore and the maximum shaft diameter gives a low shoulder which would enter the comer radius of the bearing. To use such a shoulder would be very poor practice, and if the shaft diameter cannot be increased, it is best to use a shoulder ring, as in figure 9. This ring should be a tight fit and should extend well into the shaft corner, the shaft having the smallest corner radius, consistent with strength. This is very important, since the ring must receive support from all available shaft shoulder, if it is to present a firm locating face to the bearing.

In some instances it is desired to mount a ball bearing on a shaft, the diameter of which is either identical with the bore of the bearing, or so nearly the same as to admit of no integral locating shoulder. Unless an extreme degree of accuracy is required, a shoulder ring may be applied to overcome this obstacle, as illustrated in figure 10. This method requires that a groove be cut in the shaft to receive a snap wire. The shoulder ring should be tight on the shaft and should likewise be a snug fit over the wire, thus locking it firmly in place. This makes a satisfactory assembly, especially if the bearing is positively clamped against the ring. This would not, however, be recommended where loads are extremely heavy, since the snap ring groove, to a certain extent, weakens the shaft.

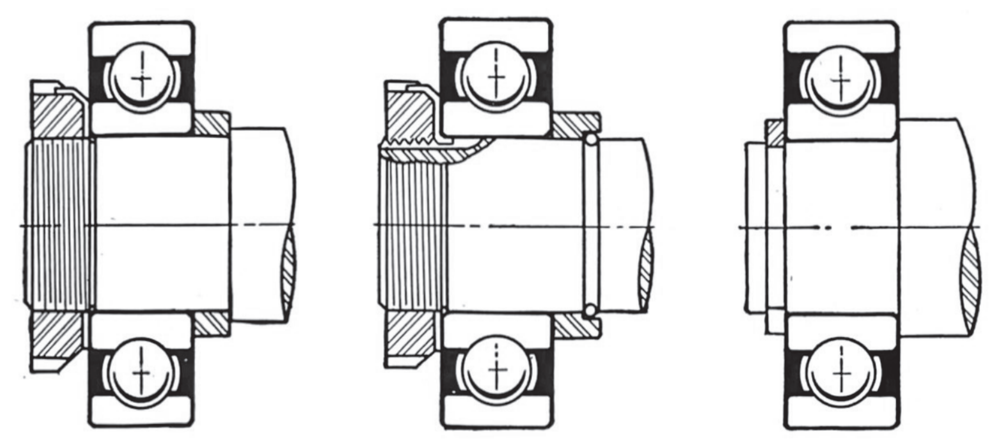


Figure 9. Figure 10. Figure 11.

Use of Retaining Ring

Although the inner ring of a bearing which has been properly press fitted on a shaft will not creep or move from its seat under normal radial load, it is sometimes advisable to utilize some form of retaining member to prevent axial displacement of the bearing due to a possible change in load conditions.

The device most used is a snap ring fitted into a groove in the shaft, as in figure 11. Though not intended to exert any clamping pressure against a bearing, such a ring should be located close against the bearing face so as to permit minimum end movement, should the bearing be forced away from its locating shoulder. This retainer ought not to be used at any point where a cut in the shaft surface can lead to fatigue failure of that member. It can best be used at the end of a shaft, as illustrated

Clamping Outer Rings

Where it is possible to machine a locating shoulder in a bearing housing, the clamping of the bearing is a relatively simple matter. The clamping member is best made with a narrow flange fitting into the housing bore, as shown in figure 12. This flange is of most importance when the clamping piece must also act as a closure about the shaft extension, for in such a case, it centers the closure with respect to the shaft and prevents possible rubbing or interference between the two.

In many housing designs where one bearing must be clamped, it is a decided advantage if the housing can be bored straight through without a shoulder. If this is done, it becomes necessary to incorporate some sort of separate shoulder against which the bearing outer race may be located. One method by which this is frequently accomplished is through the use of an adapter sleeve, as shown in figure 13. This sleeve may be either a relatively thin machined casting, or, in cases where a large number of parts are to be produced, may be a heavy sheet metal pressing. Whichever kind is employed, the fit of the sleeve in the housing should always be a light tap or press.

Where load conditions are not severe, and particularly where thrust loads are light, the shoulder ring in figure 14 may be satisfactorily employed.

This is simply an adaptation of the device illustrated in figure 10 and the same precautions should be observed if used in a housing design; that is, the ring should fit tightly in the housing bore and should also fit very close over the snap wire to prevent any chance of its loosening in the groove.

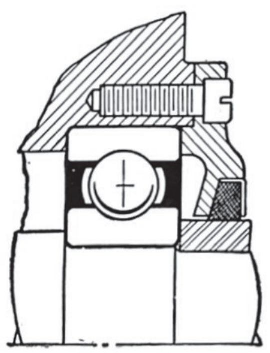


Figure 12.

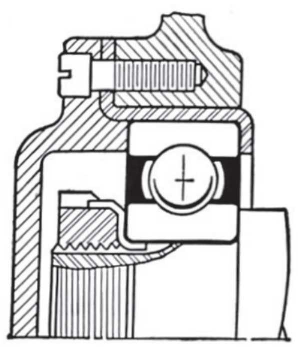


Figure 13.

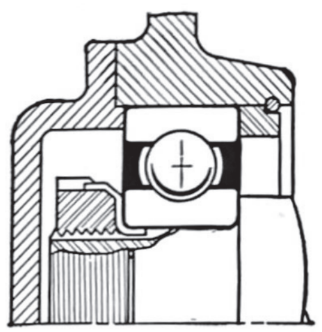


Figure 14.

Use of Adapter Sleeves on Shafts

It is desirable, wherever possible, to mount ball bearings directly upon the shaft; however, there are instances more frequently encountered in changing machines already built than in new designs, where it is not feasible to adhere strictly to this rule. In certain cases this may necessitate the use of some form of adapter sleeve. Figure 15 illustrates a mounting used where it is necessary to employ a bearing, the bore of which is too large to be fitted directly upon the existing shaft. This type of sleeve mounting is also used to some extent in new applications, the chief reason in such a case being the relative ease with which the bearing may be removed from the shaft without materially affecting the mounting fit. However used, the bearing should be applied to the sleeve with a firm press fit, and the sleeve to the shaft with a tap fit, a key being used to prevent rotation about the shaft.

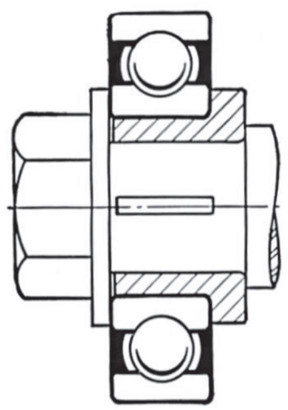


Figure 15.

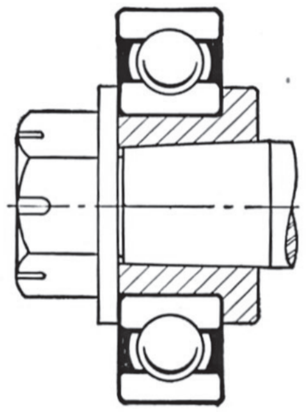


Figure 16.

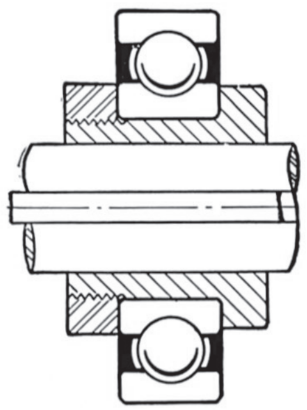


Figure 17.

A variation of this type of adapter is shown in figure 16, where the sleeve is mounted on a tapered shaft. In this instance, the use of a key is not essential, since a moderate pressure exerted by the clamping nut would be sufficient to prevent movement between the sleeve and the shaft. It should be emphasized that wherever such a mounting is used, the nut must not be set up tighter than is necessary to hold the sleeve securely in place; otherwise, excessive pressure is quite apt to expand the sleeve and the bearing inner race as well. It is also very important that the clamping nut used in such an installation be provided with some form of positive locking device.

In certain instances it is desired to apply a steadying bearing to a shaft which must be axially movable through the boring bore. While such an application is not recommended for heavy loads or more than ordinary accuracy, the bearing may be applied as in figure 17. Four points require especial observance:

The bearing should be a press fit on the sleeve; the sleeve should be not closer than .0003» loose on the shaft; the sleeve should be of sufficient length to give a relatively low unit pressure between shaft and sleeve, and the key should be of substantial cross section to minimize wear in restraining the sleeve against the natural creep due to the loose fit.

Ball Bearings and Balls

The mounting shown in figure 18 may be used for line shafting, where the speeds are not excessive. It consists of a tapered inner sleeve, slotted lengthwise to allow for expansion or contraction, and a solid outer sleeve having a taper corresponding to that of the inner member. An ear washer, with a key fitting into a slot in the inner sleeve, locks the clamping nut when the assembly has been tightened. Such a mounting is not suitable where thrust loads are present which would tend to move the adapter from its original position

The vertical mounting shown in figure 19 is employed in varying forms, the use of the adapter here being directly due to the type of bearing closure desired. That is to say, the bearing is mounted upon the adapter sleeve only for the reason that it is in this way possible to introduce a stationary quill or tube between the bearing and the shaft, as indicated by the dotted lines. This is done so that the level of the lubricant may be brought up into the path of the balls.

The use of adapters for mounting ball bearings upon shafts is to be avoided, if possible, in the design of new machines, not only because of the extra machining and fitting involved, but because of the possible errors in alignment introduced. Their use in any mechanism requires special attention to the accuracy obtained in machining or grinding the bore of the adapter in relation to the outside or bearing seat to make sure that these two surfaces are finished parallel and that the eccentricity is held to as low a figure as possible.

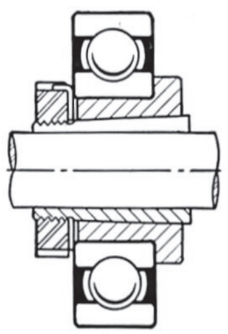


Figure 18.

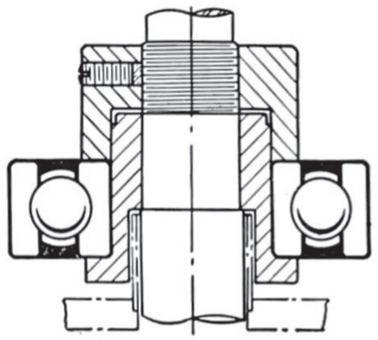


Figure 19.

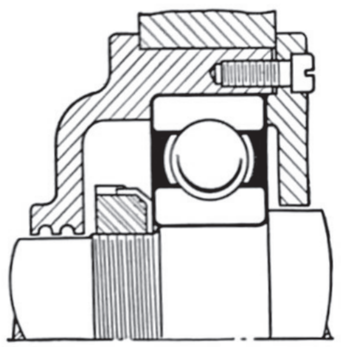


Figure 20.

Use of Adapter Sleeves in Housings

There are certain bearing applications where the use of an adapter sleeve in the housing not only justifies the somewhat greater expense of such Construction, but is really essential to success of the design.

Two-Piece Housings

Where it is necessary to resort to a split housing, the bearings should always be fitted to a sleeve which may be located axially in the housing by means best suited to the particular case. In no instance, however, should location be obtained by tight clamping of the sleeve between the housing halves, for the purpose of the adapter is to prevent cramping or pinching of the bearing, and even where the sleeve is of relatively thick section, an out-of-round condition of the housing may still affect the bearing if the fits are too tight.

Figures 20 and 21 illustrate commonly used adapters, in either of which the bearing may be positively located or «floated,» as required.

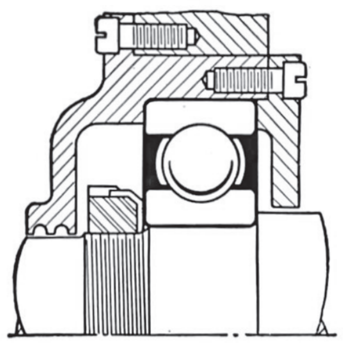


Figure 21.

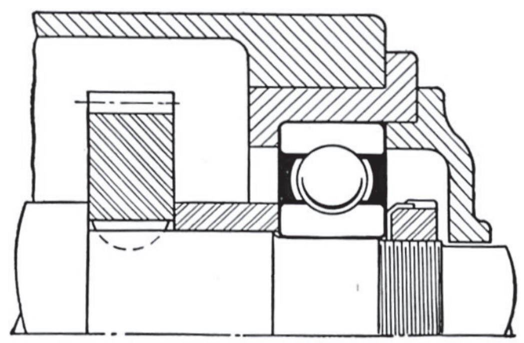


Figure 22.

Sleeves Required for Assembly Reasons

Instances are sometimes encountered where parts carried upon the shaft must be assembled by passing through the bore of the bearing housing, but where the bore, to accommodate a bearing of adequate size, is too small to allow of such assembly. In many such cases, a slight enlargement of the housing would solve the problem, and this may be accomplished by the use of a simple adapter sleeve applied as in figure 22. When an adapter is utilized for this purpose, it should be made a light tap fit in the main casting and should, of course, have the bore and O.D. finished as parallel and concentric as possible.

Ball Bearings and Balls

Sleeves for Precision Mountings

Although the initial cost of precision applications, such as machine tool spindles, is higher where sleeves are employed, their use may be such as to greatly facilitate assembly and their value from other standpoints usually outweighs the item of expense. Where two bearings are mounted in sleeves at the front end of a precision spindle, figure 23, the bearing outer rings are press fitted with their eccentricity «high points» diametrically opposite similar points in the sleeves, so as to compensate for the eccentricity of both parts, thus minimizing spindle runout. Where precision bearings must be so mounted in the housing as to permit a slight longitudinal movement due to spindle expansion, the use of a sleeve, as in figure 24, not only assures sufficient surface in contact with the housing to prevent excessive radial play, but it allows the bearing outer rings to be press fitted so as to maintain the original location of bearing and sleeve eccentricity high points

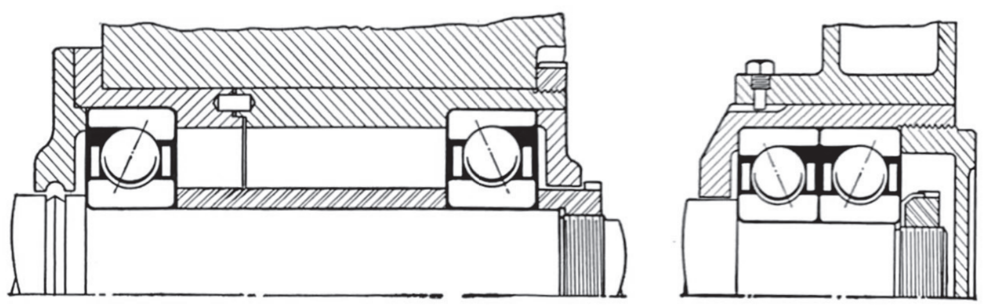


Figure 23. Figure 24.

Eccentric Sleeves

It is occasionally necessary to do mount ball bearings as to allow a certain eccentric movement of the shaft, either as a means of adjustments for parts carried, belt tightening or other reasons. This may be accomplished, where the shaft is supported in one housing only, by means of a sleeve whose outside diameter is sufficiently eccentric in relation to the bore to give the required movement, as in figure 25.

This type of sleeve is not practicable where separate housings and sleeves are desired at each end of a shaft, since unless the sleeves are adjusted very carefully and in unison, the shaft may be very badly misaligned.

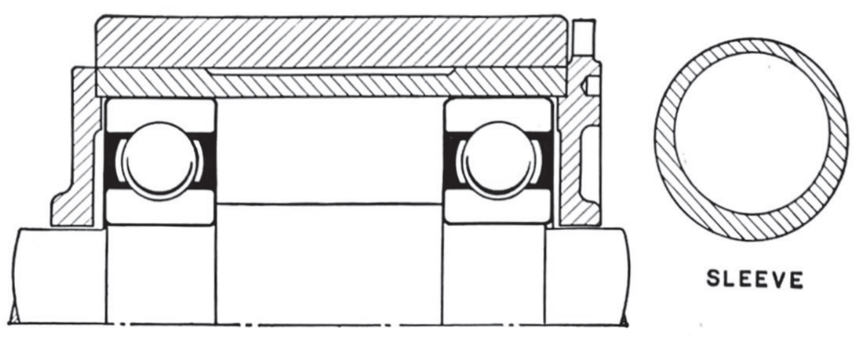
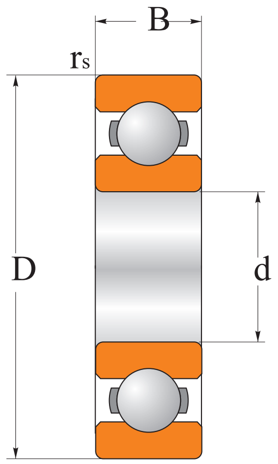


Figure 25.

Ball Bearings and Balls

Ball Bearings

1. Main dimensions and mass of bearings should correspond to drawing and table 1-7



d - nominal diameter of hole of internal ring
D - nominal diameter of external cylindrical surface of outer ring
B - nominal width of bearings
r - nominal co-ordinate of assembly chamfer

Superlight series of diameter 8,
Standard series of range 1
and broad series of range(width) 2

Dimensions in mm — Table 1

Bearing No	d	D	B	r	Weight
1000083	3	7	2	0.3	0.0003
2000083	3	7	2.5	0.3	0.0004
1000084	4	9	2.5	0.3	0.0007
1000085	5	11	3	0.3	0.0012
1000086	6	13	3.5	0.3	0.0020
1000087	7	14	3.5	0.3	0.0022
1000088	8	16	4	0.4	0.0030
1000089	9	17	4	0.4	0.0034
1000800	10	19	5	0.5	0.0055
1000801	12	21	5	0.5	0.007
1000802	15	24	5	0.5	0.008
1000803	17	26	5	0.5	0.009
1000804	20	32	7	0.5	0.020
1000805	25	37	7	0.5	0.022
1000806	30	42	7	0.5	0.027
1000807	35	47	7	0.5	0.031
1000808	40	52	7	0.5	0.035
1000809	45	58	7	0.5	0.043
1000810	50	65	7	0.5	0.057
1000811	55	72	9	0.5	0.091
1000812	60	78	10	0.5	0.12
1000813	65	85	10	1	0.13
1000814	70	90	10	1	0.18
1000815	75	95	10	1	0.19
1000816	80	100	10	1	0.22
1000817	85	110	13	1.5	0.29
1000818	90	115	13	1.5	0.3
1000819	95	120	13	1.5	0.32
1000820	100	125	13	1.5	0.34
1000821	105	130	13	1.5	0.45
1000822	110	140	16	1.5	0.6
1000824	120	150	16	1.5	0.65
1000826	130	165	18	2	0.93
1000828	140	175	18	2	1.08
1000830	150	190	20	2	1.43
1000832	160	200	20	2	1.49
1000834	170	215	22	2	2
1000836	180	225	22	2	2.03
1000838	190	240	24	2.5	2.6
1000840	200	250	24	2.5	2.7
1000844	220	270	24	2.5	3
1000848	240	300	28	3	4.5
1000852	260	320	28	3	4.8
1000856	280	350	33	3	7.4
1000860	300	380	38	3.5	10.5
1000864	320	400	38	3.5	11.8
1000868	340	420	38	3.5	12
1000876	380	480	46	3.5	20
1000892	460	580	56	4	36.3

Superlight series of diameter 9,
Standard series of range (width) 1

Dimensions in mm — Table 2

Bearing No	d	D	B	r	Weight
1000091	1	4	1.6	0.2	0.0001
100009/1.5	1.5	5	2	0.3	0.0002
1000092	2	6	2.3	0.3	0.0004
100009/2.5	2.5	7	25	0.3	0.0006
1000093	3	8	3	0.3	0.0007
1000094	4	11	4	0.3	0.0020
1000095	5	13	4	0.4	0.0025
1000096	6	15	5	0.4	0.004
1000097	7	17	5	0.5	0.005
1000098	8	19	6	0.5	0.007
1000099	9	20	6	0.5	0.008
1000900	10	22	6	0.5	0.009
1000901	12	24	6	0.5	0.010
1000902	15	28	7	0.5	0.017
1000903	17	30	7	0.5	0.018
1000904	20	37	9	0.5	0.035
1000905	25	42	9	0.5	0.042
1000906	30	47	9	0.5	0.049
1000907	35	55	10	1.0	0.086
1000908	40	62	12	1.0	0.11
1000909	45	68	12	1.0	0.15
1000910	50	72	12	1.0	0.18
1000911	55	80	13	1.5	0.19
1000912	60	85	13	1.5	0.26
1000913	65	90	13	1.5	0.30
1000914	70	100	16	1.5	0.32
1000915	75	105	16	1.5	0.38
1000916	80	110	16	1.5	0.43
1000917	85	120	18	2	0.7
1000918	90	125	18	2	0.73
1000919	95	130	18	2	0.76
1000920	100	140	20	2	1.02
1000921	105	145	20	2	1.05
1000922	110	150	20	2	1.1
1000924	120	165	22	2	1.4
1000926	130	180	24	2.5	1.9
1000928	140	190	24	2.5	2.1
1000930	150	210	28	3	3.5
1000932	160	220	28	3	3.7
1000934	170	230	28	3	4
1000936	180	250	33	3	4.9
1000938	190	260	33	3	5.2
1000940	200	280	38	3.5	7.7
1000944	220	300	38	3.5	8.1
1000948	240	320	38	3.5	9.6
1000952	260	360	46	3.5	14.5
1000956	280	380	46	3.5	15
1000960	300	420	56	4	24
1000964	320	440	56	4	25.5
1000968	340	460	56	4	27

Ball Bearings and Balls

Ball Bearings

Extra light series of diameter 1, Narrow series of range (width) 7

Dimensions in mm **Table 3**

Bearing No	d	D	B	r	Weight
7000101	12	28	7	0.5	0.020
7000102	15	32	8	0.5	0.027
7000103	17	35	8	0.5	0.032
7000104	20	42	8	0.5	0.050
7000105	25	47	8	0.5	0.053
7000106	30	55	9	0.5	0.087
7000107	35	62	9	0.5	0.111
7000108	40	68	9	0.5	0.125
7000109	45	75	10	1	0.170
7000110	50	80	10	1	0.188
7000111	55	90	11	1	0.260
7000112	60	95	11	1	0.280
7000113	65	100	11	1	0.300
7000114	70	110	13	1	0.433
7000115	75	115	13	1	0.457
7000116	80	125	14	1	0.597
7000117	85	130	14	1	0.626
7000118	90	140	16	1.5	0.848
7000119	95	145	16	1.5	0.885
7000120	100	150	16	1.5	0.91
7000121	105	160	18	1.5	1.2
7000122	110	170	19	1.5	1.46
7000124	120	180	19	1.5	1.8
7000126	130	200	22	2	2.69
7000128	140	210	22	2	2.86
7000130	150	225	24	2	3.58
7000132	160	240	25	2.5	3.6
7000134	170	260	28	2.5	5.77
7000136	180	280	31	3.0	7.6
7000138	190	290	31	3	7.89
7000140	200	310	34	3.5	10.1
7000144	220	340	37	3.5	13.5
7000148	240	360	37	3.5	14.5
7000152	260	400	44	4	21.5
7000156	280	420	44	4	23

Extra light series of diameters 1, standard series of range (width)0

Dimensions in mm **Table 4**

Bearing No	d	D	B	r	Weight
16	6	17	6	0.5	0.008
17	7	19	6	0.5	0.009
18	8	22	7	0.5	0.015
19	9	24	7	0.5	0.018
100	10	26	8	0.5	0.019
101	12	28	8	0.5	0.022
102	15	32	9	0.5	0.03
103	17	35	10	0.5	0.04
104	20	42	12	1	0.07
105	25	47	12	1	0.082
106	30	55	13	1.5	0.119
107	35	62	14	1.5	0.154
108	40	68	15	1.5	0.191
109	45	75	16	1.5	0.241
110	50	80	16	1.5	0.26
111	55	90	18	2	0.383
112	60	95	18	2	0.411
113	65	100	18	2	0.437
114	70	110	20	2	0.604
115	75	115	20	2	0.638
116	80	125	22	2	0.845
117	85	130	22	2	0.892
118	90	140	24	2.5	1.167
119	95	145	24	2.5	1.224
120	100	150	24	2.5	1.271
121	105	160	26	3	1.591
122	110	170	28	3	1.953
124	120	180	28	3	2.098
126	130	200	33	3	3.257
128	140	210	33	3	3.388
130	150	225	35	3.5	4.157
132	160	240	38	3.5	5.056
134	170	260	42	3.5	6.910
136	180	280	46	3.5	8.876
138	190	290	46	3.5	9.31
140	200	310	51	3.5	11.93
144	220	340	56	4	18.4
148	240	360	56	4	19.6
152	260	400	65	5	29.3
156	280	420	65	5	31.0
160	300	460	74	5	43.8
164	320	480	74	5	46.1
168	340	520	82	6	62.0
172	360	540	82	6	65.0

Light series of diameters 2, Narrow series of range(width) 0

Dimensions in mm **Table 5**

Bearing No	d	D	B	r	Weight
23	3	10	10	0.3	0.0015
24	4	13	13	0.4	0.0032
25	5	16	16	0.5	0.0047
26	6	19	19	0.5	0.008
27	7	22	22	0.5	0.0123
28k	8	24	24	0.5	0.019
29	9	26	26	1	0.02
200	10	30	30	1	0.031
201	12	32	32	1	0.037
202	15	35	35	1	0.046
203	17	40	40	1	0.073
204	20	47	47	1.5	0.0108
205	25	52	52	1.5	0.129
206	30	62	62	1.5	0.2
207	35	72	72	2	0.284
208	40	80	80	2	0.349
209	45	85	85	2	0.404
210	50	90	90	2	0.46
211	55	100	100	2.5	0.597
212	60	110	110	2.5	0.771
213	65	120	120	2.5	0.997
214	70	125	125	2.5	1.072
215	75	130	25	2.5	1.179
216	80	140	26	3	1.402
217	85	150	28	3	1.799
218	90	160	30	3	2.159
219	95	170	32	3.5	2.606
220	100	180	34	3.5	3.130
221	105	190	36	3.5	3.740
222	110	200	38	3.5	4.370
224	120	215	40	3.5	5.150
226	130	230	40	4	6.200
228	140	250	42	4	7.560
230	150	270	45	4	9.850
232	160	290	48	4	15
234	170	310	52	5	16.5
236	180	320	52	5	17.5
238	190	340	55	5	23.3
240	200	360	58	5	28
244	220	400	65	5	32.4
248	240	440	72	5	51.0
252	260	480	80	6	65.5
256	280	500	80	6	71.0

Middle series of diameters 3, Narrow series of range(width) 0

Dimensions in mm **Table 6**

Bearing No	d	D	B	r	Weight
34	4	16	5	0.5	0.005
35	5	19	6	0.5	0.009
300	10	35	11	1	0.054
301	12	37	12	1.5	0.061
302	15	42	13	1.5	0.085
303	17	47	14	1.5	0.115
304	20	52	15	2	0.145
305	25	62	17	2	0.23
306	30	72	19	2	0.331
307	35	80	21	2.5	0.447
308	40	90	23	2.5	0.625
309	45	100	25	2.5	0.828
310	50	110	27	3	1.062
311	55	120	29	3	1.375
312	60	130	31	3.5	1.717
313	65	140	33	3.5	2.098
314	70	150	35	3.5	2.543
315	75	160	37	3.5	3.055
316	80	170	39	3.5	3.622
317	85	180	41	4	4.201
318	90	190	43	4	4.954
319	95	200	45	4	5.728
320	100	215	47	4	7.068
321	105	225	49	4	7.992
322	110	240	50	4	9.592
324	120	260	55	4	12.22
326	130	280	58	5	15
328	140	300	62	5	18.32
330	150	320	65	5	21.75

Ball Bearings and Balls

Ball Bearings

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Heavy series of diameters 4,
Narrow series of range(width)0

Dimensions in mm — Table 7

Bearing No	d	D	B	r	Weight
403	17	62	17	2	0.265
404	20	72	19	2	0.398
405	25	80	21	2.5	0.53
406	30	90	23	2.5	0.725
407	35	100	25	2.5	0.954
408	40	110	27	3	1.227
409	45	120	29	3	1.54
410	50	130	31	3.5	1.89
411	55	140	33	3.5	2.29
412	60	150	35	3.5	2.76
413	65	160	37	3.5	3.28
414	70	180	42	4	4.85
415	75	190	45	4	5.74
416	80	200	48	4	6.72
417	85	210	52	5	7.88
418	90	225	54	5	11.4

Mass of Bearings is calculated with cage, pressed of steel sheet, steel denisity 7.85kg/dm2
Example of conventional code of radial ball bearings of extra light series of diameter 1,
Range(width) series 0 d=50mm,D=80mm,B=8mm
Bearing 110

Ball Bearings and Balls

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Ball Bearings

Ball Bearings and Balls

Reference static(Co) and dynamic (C) load capacity Superlight series of diameters 8,

Dimensions in mm — Table 1

Bearing No	d	load ratings C	load ratings C_0
1000083	3	392	127
2000083	3	450	147
1000084	4	540	186
1000085	5	635	280
1000086	6	884	325
1000087	7	956	360
1000088	8	1330	510
1000089	9	1430	585
1000800	10	1480	630
1000801	12	1430	650
1000802	15	1560	830
1000803	17	1680	930
1000804	20	2700	1500
1000805	25	3120	1980
1000806	30	3420	2350
1000807	35	4030	3000
1000808	40	4160	3350
1000809	45	6050	3800
1000810	50	6240	4250
1000811	55	8320	5600
1000812	60	8710	7350
1000813	65	11700	8300
1000814	70	12100	9150
1000815	75	12500	9800
1000816	80	12400	9800
1000817	85	19000	15000
1000818	90	19500	15600
1000819	95	19700	17400
1000820	100	19900	17000
1000821	105	20800	18000
1000822	110	28100	23500
1000824	120	29100	25500
1000826	130	37700	32500
1000828	140	38000	35500
1000830	150	48800	43000
1000832	160	49400	45500
1000834	170	61800	56000
1000836	180	62400	57000
1000838	190	74100	69500
1000840	200	76100	72000
1000844	220	78000	78000
1000848	240	108000	106000
1000852	260	111000	114000
1000856	280	138000	140000
1000860	300	172000	173000
1000864	320	174000	182000
1000868	340	178000	196000
1000876	380	247000	280000
1000892	460	319000	409000

Superlight series of diameters 9

Dimensions in mm — Table 2

Bearing No	d	load ratings C	load ratings C_0
1000091	1	125	34
100009/1.5	1.5	125	34
1000092	2	280	86
100009/2.5	2.5	280	86
1000093	3	560	186
1000094	4	950	340
1000095	5	1080	390
1000096	6	1470	555
1000097	7	2020	770
1000098	8	2240	880
1000099	9	2680	1050
1000900	10	3340	1350
1000901	12	3390	1350
1000902	15	3480	1480
1000903	17	3640	1650
1000904	20	6550	3040
1000905	25	7320	3689
1000906	30	7590	3990
1000907	35	10400	5650
1000908	40	12200	6920
1000909	45	14300	8130
1000910	50	14500	9700
1000911	55	16000	10000
1000912	60	16400	10600
1000913	65	17400	11900
1000914	70	23730	17300
1000915	75	24300	16800
1000916	80	27500	18900
1000917	85	31900	22200
1000918	90	32900	23500
1000919	95	32900	23500
1000920	100	44900	32000
1000921	105	46500	33500
1000922	110	46500	33500
1000924	120	53300	40000
1000926	130	65300	50000
1000928	140	66600	53000
1000930	150	85000	67000
1000932	160	85000	67000
1000934	170	88900	75000
1000936	180	114000	95000
1000938	190	117000	100000
1000940	200	148000	125000
1000944	220	153000	132000
1000948	240	157000	146000
1000952	260	212000	200000
1000956	280	216000	212000
1000960	300	270000	280000
1000964	320	277000	294000
1000968	340	293000	320000

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Extra light series of diameters 1

Dimensions in mm — Table 3

Bearing No	d	load ratings C	load ratings C_0
7000101	12	5070	2240
7000102	15	5590	2500
7000103	17	6050	2800
7000104	20	7020	3400
7000105	25	7610	4000
7000106	30	11200	5850
7000107	35	12400	6950
7000108	40	13300	7800
7000109	45	15600	9300
7000110	50	16300	10000
7000111	55	17000	11700
7000112	60	18600	12400
7000113	65	19000	13100
7000114	70	22200	15300
7000115	75	28600	20000
7000116	80	33200	23600
7000117	85	33800	25000
7000118	90	41600	29000
7000119	95	42300	31500
7000120	100	44200	32500
7000121	105	52000	38000
7000122	110	57200	42500
7000124	120	61800	47500
7000126	130	79300	61000
7000128	140	80600	64000
7000130	150	92300	73500
7000132	160	99500	80000
7000134	170	119000	96500
7000136	180	138000	112000
7000138	190	148000	125000
7000140	200	168000	143000
7000144	220	174000	153000
7000148	240	178000	160000
7000152	260	238000	232000
7000156	280	242000	245000

Extra light series of diameters 1

Dimensions in mm — Table 4

Bearing No	d	load ratings C	load ratings C_0
16	6	2200	860
17	7	2200	1160
18	8	3250	1340
19	9	3710	1530
100	10	4620	1960
101	12	5070	2240
102	15	5590	2500
103	17	6050	2800
104	20	9360	4500
105	25	11200	5600
106	30	13300	6800
107	35	15900	8500
108	40	16800	9300
109	45	21200	12200
110	50	21600	13200
111	55	28100	17000
112	60	29600	18300
113	65	30700	19600
114	70	37700	24500
115	75	39700	26000
116	80	47700	31500
117	85	49400	33500
118	90	57200	39000
119	95	60500	41500
120	100	60500	41500
121	105	72800	51000
122	110	81900	57000
124	120	85000	61000
126	130	106000	78000
128	140	111000	83000
130	150	125000	96500
132	160	143000	112000
134	170	168000	134000
136	180	190000	156000
138	190	195000	166000
140	200	216000	190000
144	220	247000	228000
148	240	255000	245000
152	260	291000	290000
156	280	302000	315000
160	300	358000	390000
164	320	371000	415000
168	340	442000	540000
172	360	462000	570000

Light series of diameters 2

Dimensions in mm — Table 5

Bearing No	d	load ratings C	load ratings C_0
23	3	490	217
24	4	900	415
25	5	1480	740
26	6	2170	1160
27	7	3260	1350
28k	8	3330	1360
29	9	4620	1960
200	10	5900	2650
201	12	6890	3100
202	15	7800	3550
203	17	9560	4500
204	20	12700	6200
205	25	14000	6950
206	30	19500	10000
207	35	25500	13700
208	40	32000	17800
209	45	33200	18600
210	50	35100	19800
211	55	43600	25000
212	60	52000	31000
213	65	56000	34000
214	70	61800	37500
215	75	66300	41000
216	80	70200	45000
217	85	83200	53000
218	90	95600	62000
219	95	108000	69500
220	100	124000	79000
221	105	133000	90000
222	110	146000	100000
224	120	156000	112000
226	130	156000	112000
228	140	165000	122000
230	150	189000	150000
232	160	200000	165000
234	170	240000	209000
236	180	229000	196000
238	190	255000	232000
240	200	270000	250000
244	220	296000	290000
248	240	358000	380000
252	260	390000	430000
256	280	410000	480000

Midle series of diameters 3

Dimensions in mm — Table 6

Bearing No	d	load ratings C	load ratings C_0
34	4	1450	740
35	5	2190	1160
300	10	8060	3750
301	12	9750	4650
302	15	11400	5400
303	17	13500	6650
304	20	15900	7800
305	25	22500	11400
306	30	28100	14600
307	35	33200	18000
308	40	41000	22400
309	45	52700	30000
310	50	61800	36000
311	55	71500	41500
312	60	81900	48000
313	65	92300	56000
314	70	104000	63000
315	75	112000	72500
316	80	124000	80000
317	85	133000	90000
318	90	143000	99000
319	95	153000	110000
320	100	174000	132000
321	105	182000	143000
322	110	203000	166000
324	120	217000	180000
326	130	229000	193000
328	140	255000	224000
330	150	276000	250000

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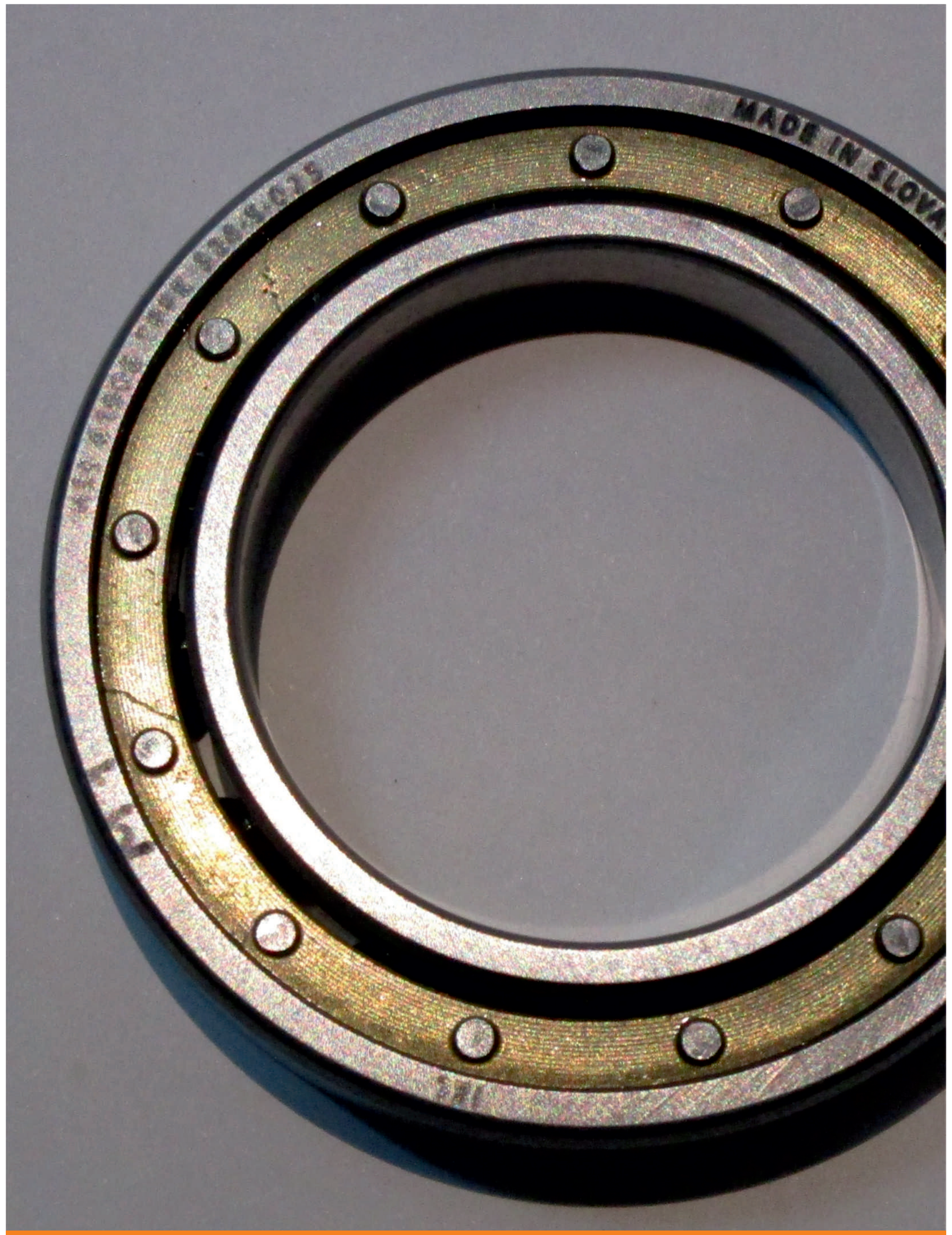
Ball Bearings and Balls

Heavy series of diameters 4

Dimensions in mm Table 7

Bearing No	d	load ratings C	load ratings C_0
403	17	22900	11800
404	20	30700	16600
405	25	36400	20400
406	30	47000	26700
407	35	55300	31000
408	40	63700	36500
409	45	76100	45500
410	50	87100	52000
411	55	100000	63000
412	60	108000	70000
413	65	119000	78000
414	70	143000	105000
415	75	153000	114000
416	80	163000	125000
417	85	174000	135000
418	90	186000	146000



Ball Bearings and Balls

Dimension and weight of the balls should meet the value, given in table 1

Dimensions in mm — **Table 1**

NOMINAL DIAMETER OF THE BALL Dw		Mass 1000 Pcs kgs
MM	INCH	
0.250	-	0.00008
0.300	-	0.00011
0.360	-	0.00016
(0.370)	1/64	0.00025
0.400	-	0.00026
0.500	-	0.00051
0.508	-	0.00054
0.600	-	0.00089
0.635	-	0.00105
0.680	-	0.00129
0.700	-	0.00141
(0.794)	1/32	0.00206
0.800	-	0.00210
0.840		0.00243
0.850	-	0.00252
1000		0.00414
(1191)	3/64	0.00694
1200	-	0.00710
1300	-	0.00903
1500	-	0.0139
1588	1/16	0.0164
1984	5/64	0.0321
2000	-	0.0329
2381	3/32	0.0554
2500	-	0.0642
2778	7/64	0.0881
3000	-	0.111
3175	1/8	0.132
3500	-	0.176
3572	9/64	0.187
(3969)	5/32	0.257
4000	-	0.263
4366	11/64	0.342
4500	-	0.374
4763	3/16	0.444
5000	-	0.514
5159	13/64	0.564
5500	-	0.684
5556	7/32	0.705
5800	-	
(5953)	15/64	0.867

Dimensions in mm — **Table 1**

NOMINAL DIAMETER OF THE BALL Dw		Mass 1000 Pcs kgs
MM	INCH	
6000	-	0.887
6350	1/4	1.05
6500	-	1.13
6747	17/64	0.26
7000	-	1.41
7144	8/32	1.50
7500	-	1.73
(7541)	19/64	1.76
7938	5/16	2.06
8000	-	2.10
8334	-	2.38
8500	-	2.52
8731	11/32	2.73
9000	-	3.00
9.128	23/64	3.12
9.525	3/3	3.55
9.922	27/64	4.01
10.000	-	4.11
10.319	7/16	4.51
10.716	27/64	5.06
11.000	-	5.47
11.112	7/16	5.64
11.500	-	6.25
11.509	29/64	6.26
11.906	15/32	6.93
12.000	-	7.1
12.303	31/64	7.65
12.700	1/2	8.42
13.000	-	9.03
13.494	17/32	10.1
14.000	-	11.3
14.288	9/16	12
15.000	-	13.9
(15.081)	19/32	14.1
15.875	5/8	16.4
16.000	-	16.8
16.669	21/32	19
17.000	-	20.2
17462	11/16	21.9
18.000	-	24
18.256	23/32	25

Dimensions in mm — **Table 1**

NOMINAL DIAMETER OF THE BALL Dw		Mass 1000 Pcs kgs
MM	INCH	
19.000	-	28.2
19.050	3/4	28.4
19.544	25/32	32.1
20.000	-	32.9
20.638	13/16	36.1
21.000	-	38
21.431	27/32	40.4
22.000	-	43.8
22.225	7/8	45.1
23.000	-	50
(23019)	29/32	50.1
23.812	15/16	55.5
24.000	-	56.8
24.606	21/32	61.2
25.000	-	64.2
25.400	1	67.3
26.000	-	72.2
26.194	1 1/32	73.8
26.988	1 1/16	80.8
27.781	1 3/32	88.1
28.000	-	90.2
28.575	1 1/8	95.8
30.000	-	111
(30162)	1 1/16	113
34.750	1 1/4	132
32.000	-	135
32.544	1 9/32	142
33.338	1 5/16	152
34.000	-	162
(34925)	1 3/8	175
35.000	-	176
35.719	1 13/32	187
36.000	-	192
36.512	1 7/16	200
38.000	-	225
(38100)	1 1/2	227
(39688)	1 9/16	257
40.000	-	263
41.725	1 5/8	289
42.862	1 11/16	324
44.450	1 3/4	361

Dimensions in mm — **Table 1**

NOMINAL DIAMETER OF THE BALL Dw		Mass 1000 Pcs kgs
MM	INCH	
45.000	-	374
46.038	1 13/16	401
47.625	1 7/8	444
49.212	1 13/16	490
50.000	-	514
50.800	2	530
52.388	2 1/16	591
53.975	2 1/8	646
55.000	-	684
57.160	2 1/4	767
60.000	-	887
60.325	2 3/8	902
61.912	2 7/16	975
63.500	2 1/2	1052
65.000	-	1128
66.675	2 5/8	1218
69.850	2 3/4	1400
73.025	2 7/8	1600
75.000	-	1733
76.200	3	1818
79.395	3 1/8	2054
80.000	-	2103
82.550	3 1/4	2311
85.725	3 3/8	2588
88.900	3 1/2	2886
90.000	-	2995
92.075	3 5/8	3207
95.250	3 3/4	3550
98.425	3 7/8	3917
100.000	-	4108
101.600	4	4308
104.775	4 1/8	4725
107.950	4 1/4	5168
108.000	-	5175
110.000	-	5468
111.125	4 3/8	5637
114.309	4 1/2	6134
120.000	-	7100
127.000	-	8415
150.000	-	13865

Ball Bearings and Balls

Ball Bearings and Balls

Deviation of mean diameter, difference in dimension of balls as per diameter in lot, variability of unit diameter, deviation from spherical shape(without considering waviness) and surface finish of the surface should not exceed the value mentioned in the table-2

Table 2

Degree of Accuracy	Nominal diameter of the ball Dw MM	Deviation of Mean Diameter of Balls used as separate parts DWM MKM, Max	Size variationof Balls as per diameter in the lot DWL MKM, Max	Variable unit of diameter vs DWS MKM, Max	Deviation from Spherical shape Δ MKM, Max	Roughness of surface Ra	 Rz
3	> 0.25 > 12	± 5	0.13	0.08	0.08	-	0.100
5	> 0.25 > 12	± 5	0.25	0.13	0.13	0.020	0.100
10	> 0.25 > 25	± 9	0.50	0.25	0.25	0.020	0.100
16	> 0.25 > 25	± 10	0.80	0.40	0.40	0.032	0.160
20	> 0.25 > 38	± 10	1.00	0.50	0.50	0.040	0.200
28	> 0.25 > 38	± 12	1.40	0.70	0.70	0.050	0.250
40	> 0.25 > 50	± 16	2.00	1.00	1.00	0.080	0.400
60	> 0.25 > 80	± 30	3.00	1.50	1.50	0.100	0.500
100	> 0.25 > 120	± 40	5.00	2.50	2.50	0.125	0.600
200	> 0.25 > 150	± 60	10.00	5.00	5.00	0.200	0.800

Deviation of average diameter of balls for all degrees of accuracy, applicable in rolling bearings, is given in table 3

Table 3

NOMINAL DIAMETER OF THE BALL Dw MM	Deviation of Mean Diameter of Balls used as separate parts DWM mm,Max
> 0.25 > 1.5	± 0.010
> 1.5 > 3	± 0.010
> 3 > 6	± 0.020
> 6 > 10	± 0.025
> 10 > 18	± 0.050
> 18 > 30	± 0.100
> 30 >100	± 0.200

1.1The balls of diameter from 3 to 45 mm should withhold the test for permissible load.

Permissible loads while testing of balls should be not less than mentioned in table 4 for given diameters; for balls with other nominal diameters the permissible load should be not less than as established in mandatory

Table 4

NOMINAL DIAMETER OF THE BALL Dw MM	BREAKING LOAD N(kgf) Minimum	NOMINAL DIAMETER OF THE BALL Dw MM	BREAKING LOAD N(kgf) Minimum
3.175	5394 (550)	18,256	168674 (17200)
3.969	8434 (860)	19,050	183384 (18700)
4.763	12062 (1230)	19,844	199075 (20300)
5.556	16279 (1660)	20,638	214766 (21900)
5.953	18142 (1850)	21,431	220650 (22500)
6.350	21280 (2170)	22,225	247128 (25200)
7.144	26968 (2750)	23,019	257915 (26300)
7.938	32852 (3350)	23,812	281451 (28700)
8.701	39717 (4050)	25,400	318716 (32500)
9.128	43149 (4400)	26,194	333426 (34000)
9.525	47071 (4800)	26,988	357943 (36500)
9.922	51875 (5300)	27,781	374614 (38200)
10.319	54917 (5600)	28,575	397169 (40500)
10.716	59820 (6100)	30,162	441299 (45000)
11.112	63743 (6500)	31,750	487390 (49700)
11.509	69646 (7000)	33,338	534462 (54500)
11.906	73549 (7500)	34,925	582515 (69400)
12.303	78453 (8000)	35,719	603109 (61500)
12.700	83356 (8500)	36,512	632529 (64500)
13.494	94143 (9600)	38,100	686465 (70000)
14.288	104931 (10700)	39,688	735499 (75000)
15.081	116699 (11900)	41,275	799242 (81500)
16.875	128467 (13100)	42,862	853179 (87000)
16.669	142196 (14500)	44,450	912018 (93000)
17.462	154954 (15800)		

Ball Bearings and Balls

TECHNICAL REQUIREMENTS

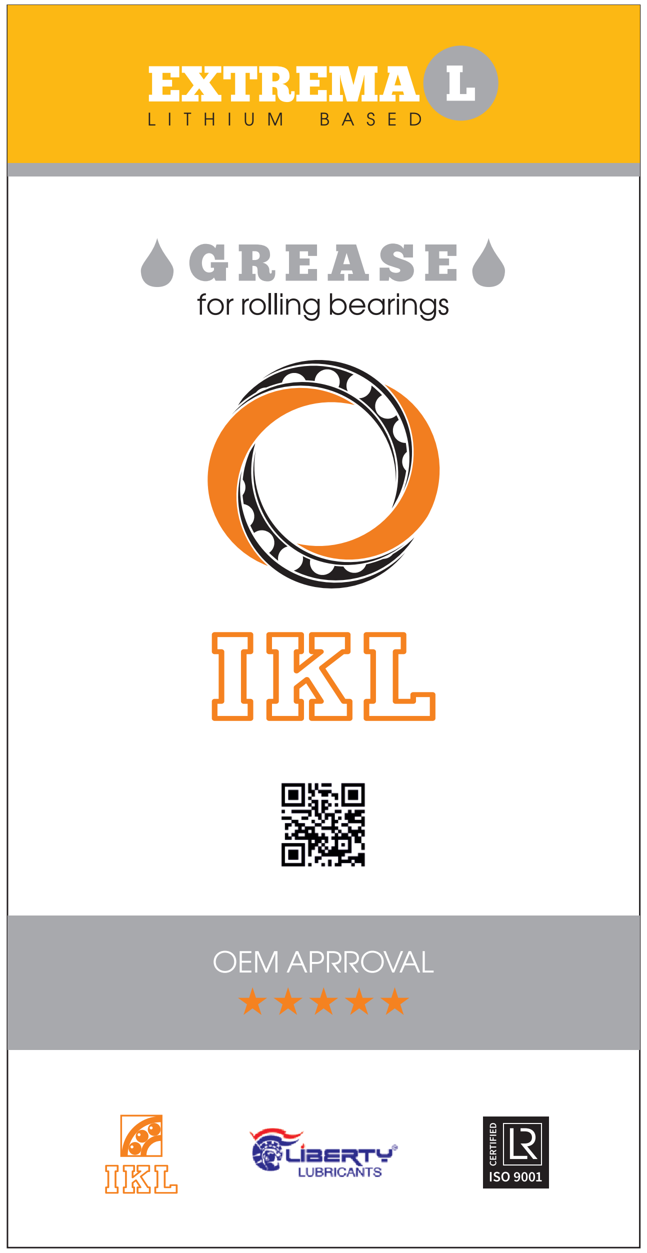
1.1Balls with diameter more than 45 should maintain the test for compression testing load while compressing and change of diameter of ball due to the effect of this load, depending upon the dimension of ball should correspond the value, mentioned in table 5.

Table 5

NOMINAL DIAMETER OF THE BALL Dw MM	46.038	50.8	60	76.2	100	101.5	150
LOAD KN (TS)	63.7 (6.5)	78.45 (8)	98.07 (10)	147.1 (15)	245.17 (25)	245.17 (25)	490.33 (50)
DIFFERENCE IN DIMENSIONS OF DIAMETERS MEASURED BEFORE AND AFTER PRESSING MKM MAX	2.5	3	3	3	4	4	6

Ball Bearings and Balls

In technical collaboration with world leader of grease manufacturer Liberty Lubricants IKL introduces 3 lines of products which service engineers of IKL recommend to use for heavy duty application to extend the lifetime of the machinery: EXTREMAL, EXTREMUM and EXTRIMITY.



EXTREMAL

EXTREMAL - Premium quality Lithium based multi purpose grease manufactured from superior quality base oil and soap meeting ISO 7623:1993.

EXTREMAL is designed for the lubrication of various applications in all types of operating conditions, but particularly where the loads and operation temperature are high and the use of conventional lithium soap greases is limited.

These lithium complex grease are formulated to provide extra protection against wear, rusting and water washout.

EXTREMAL has better water resistance properties compared to sodium soap greases, better high-temperature properties compared to calcium soap greases, and excellent mechanical properties (both resistance to shearing and good ability to be pumped).

The recommended operating temperature range is from -20ºC to 130ºC but they may be used at higher temperatures if the lubrication frequency is increased accordingly

EXTREMAL Benefits

- First-class economy grade for multi-functional use in rolling and sliding bearings
- Reduced wear under heavy or shock loading and vibration for good equipment reliability and availability
- Protection against rust and corrosion and resistance to water washout for equipment protection and good lubrication even in presence of water
- Extended bearing life potential in wet environments for reduced bearing costs and unanticipated downtime

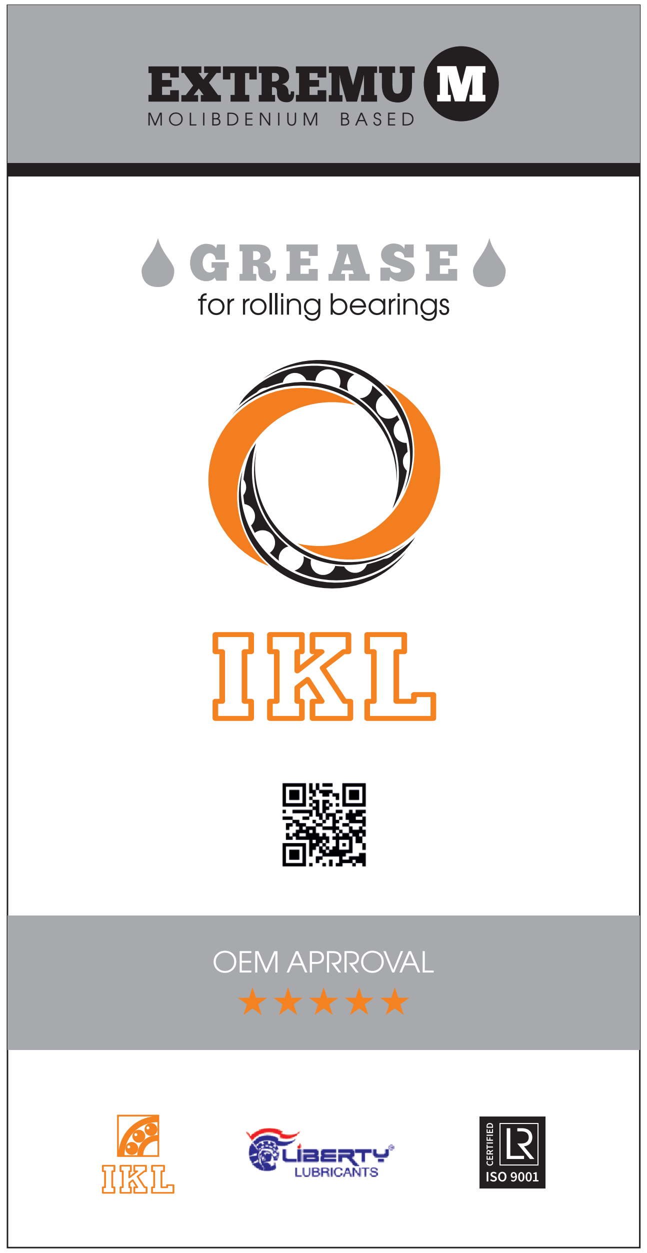
EXTREMAL has improved mechanical stability which characterize the ability of grease to maintain consistency when subjected to mechanical shear forces.

EXTREMAL is the best grease for the professionals servicing the equipment operating with heavy loads and dust environment.

The quality of EXTREMAL Grease is equivalent to EU regulations
DIN 51 502: KP 2 K-30
ISO 7623:1993

EXTREMAL Properties and Specifications

Property	Method	Unit	Value
Color	visuell	-	Braun/brown
Thickener type	-	-	Lithium
Classification	DIN 51 502	-	KP 2 K-30
NLGI-class	DIN 51 818	-	2
Worked penetration	DIN ISO 2137	0,1 mm	265-295
Dropping point	DIN ISO 2176	°C	> 185
Usage temperature	-	°C	-30 bis + 120
Usage temperature	-	°F	-22 bis + 248
VKA welding force	DIN 51 350-4	N	3000
Corrosion protection (Emcor)	DIN 51 802	Korr.-Grad	0-0
Resistance to water	DIN 51 807-1	Bew.-Stufe	1-90
Base oil viscosity at 40 °C	ASTM D-7042	mm2/s	130
Corrosion effect on copper	DIN 51 811	Korr.-Grad	1-100



EXTREMUM

EXTREMUM - Premium quality Molybdenum additive Grease is multi purpose grease manufactured based on highly refined base oils with lithium soap as a thickening agent, supplemented with molybdenum disulfide (3%) and other additives to achieve the following properties: STRONG ADHESION TO METAL.

EXTREMUM - is designed for the lubrication of various applications in all types of operating conditions, but particularly where the loads and operation temperature are high and the use of conventional lithium soap greases is limited. It is fortified with molybdenum disulfide (3%), known for its excellent lubricating properties and its ability to coat metal surfaces as a protection against wear.

EXTREMUM Benefits

- Positive lubrication and wear protection because tacky and adhesive lubricant film stays in place, resisting wipe off or squeeze out at high speeds and loads
- Effective extreme pressure (EP) and antiwear protection prevents scoring, scuffing, and seizure of moving surfaces under heavy loads, shock loads, or overloads
- Molybdenum disulfide coats metal surfaces for additional protection against destructivesliding friction and wear in especially severe applications.
- Strongly adherent lubricant film resists wash-off and throw-off.
- Provides a better rust protection and longer oxidation protection
- Retain their structure and consistency even after severe mechanical working
- Does not change hardness after repeated heating and cooling cycles.
- Extended bearing life potential in wet environments for reduced bearing costs and unanticipated downtime

EXTREMUM is used for high pressure metal sliding against metal situations.

EXTREMUM is recommended for roller bearings subjected to very heavy loads and shock loading, especially in slow, or oscillating motion such as found Vibration Screen, Crushers and Heavy applications where radial load and axial loads are applied on Rolling Bearings.

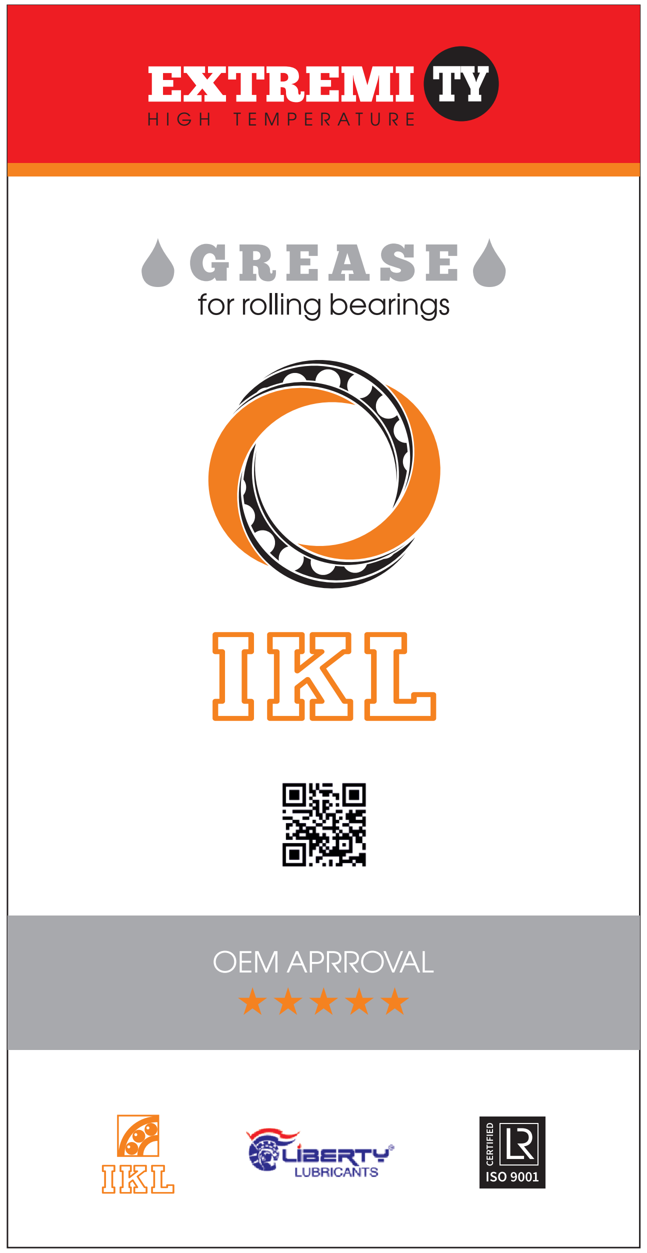
EXTREMUM proves its possibility to increase the lifetime of IKL Rolling Bearings by minimum 30% in vibrating screen application and our team will be happy to share this experience with you, our esteemed Customer.

The quality of EXTREMUM Grease is equivalent to EU regulations
DIN 51 502: KPF K-2 30
ISO 7623:1993

Properties and Specifications

Property	Method	Unit	Value
Color	visuell	-	Anthrazit/anthracite
Thickener type	-	-	Lithium
Classification	DIN 51 502	-	KPF 2 K-30
NLGI-class	DIN 51 818	-	2
Worked penetration	DIN ISO 2137	0,1 mm	265-295
Dropping point	DIN ISO 2176	°C	> 185
Usage temperature	-	°C	-30 bis + 130
Usage temperature	-	°F	-22 bis + 266
VKA welding force	DIN 51 350-4	N	4000
Resistance to water	DIN 51 807-1	Bew.-Stufe	1-90
Base oil viscosity at 40 °C	ASTM D-7042	mm2/s	130
Corrosion effect on copper	DIN 51 811	Korr.-Grad	1-100

Greases of IKL
Greases of IKL



EXTREMITY

EXTREMITY – PREMIUM HITEMP GREASE is a clay thickened non-soap base smooth structure grease.

EXTREMITY - It offers outstanding performance over a wide temperature range. The wax-free nature of the synthetic base fluid, together with its high viscosity index compared to mineral oils, provide excellent low temperature pumpability, very low starting and running torque, and can help reduce operating temperatures in the load zone of rolling element bearings. The clay thickener gives EXTRIMITY a high dropping point value of around 280ºC, which provides excellent stability at high temperatures. EXTRIMITY resists water washing, provides superior load-carrying ability, reduces frictional drag, and prevents excessive wear.

EXTREMITY Benefits

- Allows wide operating temperature range - outstanding high and low temperature performance
- Provides thicker fluid films protecting against wear of equipment parts operating at high temperature: Reduces wear by minimizing metal-to-metal friction, even under heavy loads
- Protects metal from rust and corrosion
- Causes low resistance during start-up at very low temperatures
- Superb bearing protection and helps extend bearing life and reduce bearing replacement costs
- Avoids excessive wear, even under shock load
- Long relubrication intervals

EXTREMITY proves its possibility to increase the lifetime of IKL Rolling Bearings by effectively lubricating rolling element bearings under conditions of high speeds and temperatures. EXTRIMITY has also shown excellent ability to lubricate heavily loaded sliding mechanisms.

Properties and Specifications

Property	Method	Unit	Value
Color	visual	-	Brown Grease
Thickener type	-	-	Clay thickened non-soap base
NLGI-class	DIN 51 818	-	2
Worked penetration	DIN ISO 2137	0,1 mm	290
Dropping point	ASTM D2265	℃	> 280
Base oil viscosity at 100 ℃	ASTM 1700	mm^2/s	5,7
Base oil viscosity at 40 ℃	ASTM 1697	mm^2/s	29,3
Copper strip corrosion test @ 1000 ℃ for 24 hrs	ASTM D 4048	Korr.-Grad	Negative

Greases of IKL