TECHNICAL FEATURES

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1. GENERAL

The choice of a bearing depends on many factors that need to be examined in order to obtain the most successful results at the lowest cost.
In most cases the selection should be made when the overall design of the machine has been decided. Dimensional limits are then known, also the speeds and loads. At this stage the choice can be made from the many types of bearings offered from the standard ranges. The notes given in this section will generally permit one to select the most suitable bearing for each application.
As for all other types of bearing, the results obtained with needle bearing products depend to a large extent on the design and method of assembly, loading, and alignment between inner and outer rings.
Bearing alignment depends first of all on the geometry of the parts involved and secondly on the deflection of the shaft under load. The shaft diameter should therefore be sufficient to prevent large deflections. This is easier to achieve using needle bearings because they occupy a small radial area.

2. BEARING TYPE SELECTION

Bearing type selection is made after the general design concept of the mechanism has been established and the application requirements carefully evaluated.

The ability of a bearing to support radial or axial loads, tolerate misalignments, be suitable for high speeds or loads are the main criteria for guiding the selection in the correct way. To navigate the families of bearings in this catalogue an initial assessment can be made on the basis of the table below. Further details are specified in the relevant chapters.

	Radial needle roller cage	**Caged needle bushes**	**Full complement needle bushes**	**Caged needle bearings**	**Full complement needle bearings**	**Needle rollers**	**Thrust bearings**	**Combined bearings [1)]**
Radial load	High	Moderate	High	High	Very high	Very high	None	High
Axial load	None	None	None	None	None	None	Very high	Very high
Speed	Very high	High	Moderate	Very high	Moderate	Moderate	Moderate	Moderate
Misalignment tolerance	Moderate	Moderate	Low	Moderate	Moderate	Very low	Low	Low
Grease life	High	High	Moderate	High	Moderate	Moderate	Low	Low
Friction	Very low	Low	High	Very low	High	High	High	Moderate
Precision	Very high	Moderate	Moderate	High	High	Very high	High	Very high
Cross section	Very low	Low	Low	Moderate	Moderate	Very low	Moderate	Moderate
Cost	Low	Low	Low	Moderate	Moderate	Low	High	High

1) RAX 700 series not included

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3. CALCULATIONS FOR RADIAL AND THRUST BEARINGS

The details following enable one to evaluate lifetime of radial bearings and thrust bearings and also combined bearings which comprise a radial and a thrust component. These are calculated separately without transforming the axial load into an equivalent radial load.

The calculation for a radial or thrust bearing must take account of the following principal factors:

- actual supported loads and possible shock loads
- speed of rotation
- operating temperature
- hardness of the bearing raceways.

Other features such as lubrication, sealing and alignment must be considered in order to avoid introducing unfavourable factors.

The formulas for lifetime calculations here reported are considered valid under standard conditions, generally useful for first-sizing or product comparison.

For further details on correction factors for bearing lifetime in applications, please refer to ISO281 and ISO16281 standards and to Nadella Technical Service.

The life calculation of a radial bearing or a thrust bearing under rotation is established from the dynamic capacity C indicated in the tables of dimensions. The static capacity Co enables one to determine the maximum load under certain operating conditions (see table on page 8).

3.1. BEARING LIFETIME

3.1.1. Dynamic capacity C

The dynamic capacity of a bearing is the constant radial load which it can support during one million revolutions before the first signs of fatigue appear on a ring or rolling element. For a thrust bearing, the capacity for one million revolutions assumes a constant axial load centred in line with the axis of rotation.

The dynamic capacity is a reference value only; the base value of one million revolutions has been chosen for ease of calculation. Since applied loading as great as the dynamic capacity tends to cause local plastic deformations of the rolling surfaces that may affect their operations.

The dynamic capacity C for bearings shown in the tables of dimensions has been established in conformance with the ISO Standard 281.

3.1.2. Nominal life L_{10}

The life of a (or thrust bearing) is the number of revolutions (or the number of hours at constant speed) that it will maintain before showing the first signs of material fatigue.

The relationship between the life in millions of revolutions L_{10}, the dynamic capacity C and the supported load P, is given by the formula:

$$L_{10} = \left(\frac{C}{P}\right)^{p}$$

in this expression p is equal to 10/3 for needle or roller bearings. In order to assess the importance of the influence of load on the life expectancy, one should note for example that, if the load on a bearing is doubled, its life is reduced by a factor of 10. The formula above is independent of speed of rotation which must not exceed the recommended limit in respect of the radial bearing or the thrust bearing used an d the method of lubrication. If the speed of rotation n (r.p.m.) is constant, the life is given in hours by the function:

$$L_{10h} = \frac{L_{10} \times 10^{6}}{60\ n}$$

The above formula will ensure that 90% of the bearings operating under the same conditions will attain at least the calculated L_{10} life, known as the nominal life (the figure 10 being the percentage of bearings which may not attain this life). The formulae are based on the use of standard quality bearing steel and assume a satisfactory method of lubrication.

The formulas for life calculation are effective for an applied load smaller than 0.5 C.

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3.1.3. Modified life L_{na}

In conditions different from the mentioned above, a modified life L_{na} can be determined (in millions of revolutions) following the general formula:

$$L_{na} = a_1 \,.\, a_{ISO} \,.\, L_{10}$$

in which a_1 and a_{ISO} are correction factors linked respectively to reliability, contamination and lubrication.

Reliability correction factor a_1

A reliability factor in excess of 90% may be required in certain industries fields, such as aviation, for reasons of security and to reduce the risk of a very costly immobilisation. The table below indicates the values of the correction factor a_1 as a function of reliability:

Reliability %	Factor a_1	Modified life L_{na_1}
90	1	L_{10}
95	0,64	L_5
96	0,55	L_4
97	0,47	L_3
98	0,37	L_2
99	0,25	L_1
99,5	0,175	$L_{0,5}$
99,9	0,093	$L_{0,1}$

In order to select as an example a bearing of life L_4 (reliability 96%) it is necessary to estimate life L_{10} with the formula $L_{10} = (C/P)_{10/3}$ starting from the dynamic capacity C given in this catalogue:

$$L_4 = 0.55 \,.\, L_{10}$$

Correction factor a_{ISO}

The factors that affect bearing life are numerous, and their analysis is not one in this catalogue. The effects of temperature, misalignment, bearing clearance, cleaning and lubrication conditions, which require a detailed discussion is beyond the scope of the product catalogue. For a more detailed discussion, please refer to Standards:
ISO 281:2007 introducing the coefficient a_{ISO} to take into account the effects of lubrication and cleanliness of the lubricant.
ISO 16281, which introduces in the calculation the effect of clearance and misalignments in the bearing.
Nadella technical service is available for advice on the choices to be made in special cases.

3.1.4. Variable loads and speeds

When the loads and speeds are variable, the life calculation can only be made by first establishing an assumed constant load and constant speed equivalent in their effect on the fatigue life.
This type of operating condition is frequently met and the possible variations although cyclical are numerous. One encounters this feature in particular, in variable speed drives on some supports, but constant on each support for an interval of time referring to the total operating time (example: change of speed). The equivalent load P and the equivalent speed n are obtained from the following formulae:

$$P = \sqrt[p]{\frac{m_1 n_1 P_1^p + m_2 n_2 P_2^p + \ldots + m_n n_n P_n^p}{m_1 n_1 + m_2 n_2 + \ldots + m_n n_n}}$$

$$n = m_1 n_1 + m_2 n_2 + \ldots + m_n n_n$$

in which:

$m_1, m_2 \ldots, m_n$: interval of operating time under constant load and speed (by definition: $m_1 + m_2 + \ldots + m_n = 1$).

$n_1, n_2, \ldots, n_n$: constant speed corresponding respectively to intervals of time $m_1, m_2, \ldots, m_n$.

P_1, P_2, P_n: constant loads corresponding respectively to intervals of time $m_1, m_2, \ldots, m_n$.

For needles and rollers bearings and thrust bearings, p is equal to 10/3.

Whilst at constant speed, the load varies linearly during a given time, between a minimum Pmin and a maximum Pmax. the equivalent load is given by:

$$P = \frac{P_{min} + 2\,P_{max}}{3}$$

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3.1.5. Oscillating motion
In order to calculate the life during oscillating motion it is necessary to determine an equivalent speed n in revolutions per minute from the formula:

$$n = \frac{n_{osc}\alpha}{180}$$

n_{osc}: number of oscillations "Forward and Return" per minute
α: amplitude of oscillation "Forward" in degrees.

However, this formula risks being in error and giving inaccurate lives for oscillations at small amplitudes. It is therefore recommended not to apply it for angles of oscillation below 15°.
When the angle of oscillation is very small fretting corrosion is likely to be produced and a suitable lubricant must be chosen in consequence. Experience confirms that full complement needle bearings provide better results under this phenomenon in view of their better load sharing capability.

3.1.6. Application criteria
The life calculation may be unreliable when values for speed and load reach the ultimate limits. A low speed and/or load can yield an extremely long calculated life but this will be limited in practice by other operating factors such as sealing, lubrication and maintenance, all of which have a decisive influence on the life of the product in such cases.

3.2. MINIMUM LOAD
Slippage can occur if loads are too light and, if accompanied by inadequate lubrication, cause damage to the bearings. The minimum load for bearings with cage must be

For radial bearings
- $F_{r\,min}$ = 0,04 C
(C is the Dynamic Capacity for lifetime calculation)

For thrust bearings are correct the formulas
- Needle bearings Fa_{min} = 0,005 Co
- Roller bearings Fa_{min} = 0,001 Co
(Co is the Static Capacity)

3.3. STATIC CAPACITY Co AND LIMIT LOAD Po
The static capacity Co given in the tables of dimensions has been established in conformance with ISO Specification 76. This takes into consideration the maximum admissible contact stress (Hertzian stress). The value currently being adopted in 4000 MPa.
Since permanent deformation is produced as readily in a bearing rotating as in one that is stationary, the static capacity Co determines the limit load Po which depends on the type of bearing and the operating conditions. When the limit load Po is given within the "min-max" range, the load applied may attain the indicated maximum provided it is applied continuously without sudden repeated variations. Alternatively, in the case of shock loads and vibrations, the load applied should not exceed the minimum value of limit load Po. The relationship between the static capacity and the limit load defines the safety static factor fo:

$$fo = Co/Po$$

The suggested values for the safety factor, depend on the type of application and product

Solid rail bearings

fo = 1,5 … 2,5	Important requirements for smoothness of function, silent operation or accuracy of rotation
fo = 1 … 1,5	General applications
fo = 0,7 … 1	Slow rotation or oscillatory motion.

Drawn bearings

fo > 4	Important requirements for smoothness of function, silent operation or accuracy of rotation
fo > 3	General applications and oscillatory motion

Cam followers: the allowable load for cam followers depends on the static load of the bearing and from the strength of the stud and of the outer ring. Authorised values are listed in the tables of dimensions.

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3.4. COEFFICIENT OF FRICTION

The resistance torque M of a bearing supporting a load P is given by the following relationships:

• Radial bearing: $M = f \cdot P \cdot \frac{Fw}{2}$

(with Fw is the diameter of the inner raceway of the bearing)

• Thrust bearing: $M = f \cdot P \cdot \frac{dm}{2}$ with $dm = \frac{E_b + E_a}{2}$

(E_b and E_a being the internal and external raceway diameters given in table of dimensions).
The coefficient of friction f depends on a number of factors, amongst which are:

• type of mechanism
• applied load
• speed of rotation
• lubrication
• surface finish and alignment of raceways.

The mean values shown below are for oil lubrication

f = 0,002 ÷ 0,003 for caged needle bearings
f = 0,003 ÷ 0,004 for full complement bearings and needle thrust bearings
f = 0,004 ÷ 0,005 for roller thrust bearings.

These coefficients are applicable for values of C/P between 2 and 6 approximately. For values less than or in excess of these limits the coefficient of friction f can be increased by 10 to 50%. Under starting conditions from rest, the values of f may be up to 1.5 times higher than those shown above.
To evaluate the losses of the entire bearing assembly, account must also be taken of the friction due to the seals which can be significant, especially during "running-in".

3.5. LIMITING SPEED

The tabular pages list the limiting speed values calculated under normal operating conditions, properly mounting tolerances and clearance, absence of misalignments, low loads. For speed calculated with oil lubrication it is considered a normal flow of lubricant. A bearing may operate at a speed higher than the listed limiting speed with use of a clean, with good quality oil and correct flow to remove the heat generated in the table. Consult Nadella Technical Service for further details.
In case of high speed and acceleration to avoid internal slippage between the rolling elements and the raceways the relationship between the applied load P and the base load of the bearing C must be at least $P/C > 0.02$.
The wheels are supplied normally lubricated with grease suitable for general use, so the limit speed given in the dimension tables take account of such lubrication. For wheels without seals, lubricated with oil, the indicated speed limit may be increased by about 30% for continuous rotation (about 50% for intermittent rotation).

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4. MOUNTING

4.1. SHAFT FOR BEARINGS WITHOUT INNER RING

4.1.1. Heat treatment of raceways

The minimum hardness of 58-64 HRC required to apply the calculations without reducing the basic capacities may be obtained with a through-hardened bearing steel or with a case-hardened and tempered steel. In the latter case, the hardened case must be homogeneous and regular over the entire surface of the raceway: the case depth is the thickness between the surface and the core having a hardness value of Vickers HV1 of 550 (see Standard NF A 04 202).

The minimum effective case depth of hardening depends on the applied load, the size of the rolling elements and the core strength of the steel used. To calculate the approximate case depth minimum depth can be used the following formula

Minimum case depth = (0,07÷0,12) x Dw
Dw = diameter of the rolling element

In any case the minimum suggested case depth is of 0.4 mm.

The load capacities shown in the tables of dimensions apply to raceways with a hardness of between 58 and 64 HRC.

The dynamic and static capacities are reduced when hardness values are lower than 58 and 54 HRC respectively according to the following table:

Hardness	HRC	60	58	56	54	52	50	48	45	40	35	30	25
	HV*	697	653	613	577	545	512	485	447	392	346	302	267
Coefficients for load reduction	Dyn.	1	1	0.93	0,84	0,73	0,63	0,52	0,43	0,31	0,23	0,15	0,11
	Stat.	1	1	1	1	0,96	0,86	0,77	0,65	0,50	0,39	0,30	0,25

4.1.2. Surface finish

The shafts or housing used directly as raceways for needles must have a surface finish acceptable for the operating conditions and the precision requirements:

- applications with high speeds and loads: Ra = 0,2 µm
- general applications: Ra = 0,35 µm

4.1.3. Tolerances and form deviations

The suggested tolerances for the mean shaft diameter are indicated in the appropriate chapters specific for every product.

The suggested tolerance for deviation from the cylindrical raceways form (radial bearings).

- Variation of mean shaft diameter within the length of the bearing raceway should not exceed 0.008 mm or one-half the diameter tolerance. The profile should never be concave (the core diameter must protrude to the diameter at the ends)
- Deviation from circular form: the minimum between 0.0025 mm and one quarter of diameter tolerance

For thrust bearings and combined bearings refer to the specific chapter prescriptions.

4.1.4. End chamfer

For the most effective assembly and preventing damage to the roller complements or needles, provide a chamfer to the ends of the raceway.

4.1.5. Surface in contact with seals

The surface in contact with the sealing lips must be finished with plunge cut grinding. The propeller subsequent to the grinding process without centers can create a pumping effect of the lubricant through the seal.

4.2. SHAFT FOR BEARINGS WITH INNER RING

4.2.1. Surface finish of the shaft

Maximum roughness suggested: Ra = 1,6 µm

4.2.2. Tolerances and form deviations

The suggested tolerances for the mean shaft diameter are indicated in the appropriate chapters specific for every product.

The suggested tolerance for deviation from the cylindrical raceways form (radial bearings)

- Variation of mean shaft diameter within the length of the bearing raceway: one-half of the diameter tolerance
- Deviation from circular form: one-half of the diameter tolerance

4.2.3. End chamfer

For the most effective assembly provide a chamfer to the ends of the shaft on which the inner ring must be inserted.

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4.3. HOUSING FOR BEARINGS WITH OUTER RING

4.3.1. Surface finish of the shaft
Maximum roughness suggested: Ra = 1,6 μm

4.3.2. Tolerances and form deviations
The suggested tolerances for the housing is indicated in the appropriate chapters specific for every product.
The suggested tolerance for deviation of form is
- Variation of mean housing diameter within the length in contact with needle: 0.013 mm
- Deviation from circular form: one-half of the diameter tolerance of the housing

4.3.3. End chamfer
For the most effective assembly provide a chamfer to the ends of the shaft on which the inner ring must be inserted.

4.3.4. Alignment between hole housing
When possible ream the housing of the same shaft with a single placement on the machine tool.

4.4. HOUSING FOR CAGES AND NEEDLES

4.4.1 Requirements for materials, processing and finishing
Observe the rules for the shafts, paragraph 4.1.

4.4.2. Alignment between hole housing
When possible ream the housing of the same shaft with a single placement on the machine tool.

5. LUBRICATION
Bearings are protected against oxidation with a corrosion protection, but normally supplied unlubricated. Please don't forget to lubricate them when mounting.

5.1. LUBRICANT FEATURES
Lubrication of a bearing provides a viscous film between the rolling elements in order to reduce heat and wear caused by friction. The lubricant can also assist in preventing corrosion and help to seal the bearing from the introduction of dirt and impurities; it reduces friction between the shaft and seals and lowers the noise level generated within the bearing.
Wherever the operating conditions permit, grease should be chosen in preference to oil, as it is more convenient to use and more economic. Furthermore, it acts as an efficient seal against the effects of dust and humidity. On account of its consistency, grease can improve the effectiveness of sealing rings and can be used on its own as a seal, when it is used to fill grooves or labyrinths provided for this purpose.
Alternatively, oil is necessary for high rotational speeds in excess of the limits advised for grease lubrication and in cases where there is a problem of heat dissipation. Oil can also remove moisture and impurities from the bearing and is usually easily controlled to monitor the state of lubrication. Oil lubrication is also necessary where it is used already in the function of the equipment, such as hydraulic motors and pumps, speed variators and gear boxes etc.
Oil and grease lubricants must be free of all impurities which could cause premature failure of the bearing and removal from service. Sand and metal particles are particularly injurious to bearings. Every precaution must be taken to assure the cleanliness of gear casings, pipes, grease nipples, couplings, as well as lubricant containers.
The efficiency of a lubricant decreases in service both by age and by the continuous mixing to which it is submitted. Therefore replenishment must take place at regular intervals, taking account of operating and environmental conditions (humidity, dirt, temperature) except for applications where the bearing has been lubricated for life with a suitable grease.

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5.1.1. Base oil

It is the main constituent of a lubricant, being it an oil (obtained by adding base oil to chemical additives) or a grease (which is obtained by adding the thickener to the oil). Technically base oils differ between them for their chemical/physical properties and for their ability to work in particular conditions such as high temperatures or low temperatures or even in oxidizing environments, and so on.
The following table shows the main base oils and their main physical features distinguishing its capabilities.

Parameter	Mineral oil	Ester based oil	Polyglycol oil	Silicone oil	Fluoro-carbon oil
Density [g/ml]	0.9	0.9	0.9-1.1	0.9-1.05	1.9
Viscosity index VI (1)	100	150	>200	200/500	50/150
Pour Point [°C] (2)	-10/-40	-30/-70	-20/-50	-30/80	-30/-70
Flash point [°C] (3)	200/250	230/300	150/300	150/300	No one
Oxidation resistance	Sufficient	Good	Good	Excellent	Excellent
Temperature stability	Sufficient	Good	Good	Excellent	Excellent
Lubricating ability (4)	Good	Good	Excellent	Low	Good
Compatibility with seals	Good	Low	Sufficient	Good	Good

(1) The viscosity index represents the ability of the lubricant to maintain constant its viscosity with changes in temperature; An high value of index VI means good ability to maintain a constant viscosity (key parameter for oils).
(2) The pour point is the lowest temperature at which the lubricant loses the ability to scroll (solidification), so it is an index for the utilization of the lubricant at low temperatures.
(3) Minimum temperature at which the air / gas mixture above the lubricant will ignite if it gets too close to a heat source.
(4) The lubricating ability indicates the ability of the lubricant to withstand large loads applied.

The mineral oils are used in most applications. Synthetic oils (such as esters, polyglycols, silicon) and finally the fluorocarbon that are special oils as chemically inert (due to the presence of fluoride) in the case of specific needs.

It is important to note the general rules on the viscosity of the oils:
- fluid oil = excellent refrigerant;
- thick oil = excellent lubricant;
never use a lubricant with a viscosity greater than necessary.

5.1.2. Additives

The addition of additives to the base oil, allows to obtain an oil with performance features clearly higher than the base oil itself. The additives allow to reduce some negative sides of base oils, although a silicone oil (particularly weak to support applied loads) suitabley additiveted (eg with EP additives) will never be as a synthetic oil or polyglycol .
The following table shows the main technological characteristics related with additives.

Additives	Features
Anti-oxidants	They slow down the oxidation that creates deposits on the surfaces in contact with detriment to the lubricating fluid that deteriorates
Anti-corrosion	Slow chemical reactions with materials such as copper, aluminum and sulfur
Anti-rust	Slow down the chemical reactions with ferrous materials that give life to rust
Anti-wear	Slow down the wear phenomena of materials in contact with the lubricant
EP	Extreme Pressure it allows to increase the ability of the lubricant to withstand the applied load thereby reducing the danger of seizure
Detergents	Clean the metal surfaces from debris or oxidation products by emulsion
Dispersants	Maintain the oxidation and emulsion products in suspension, preventing their deposit on metal surfaces
Pour Point	Lower the flow temperature of a lubricant allowing its use at low temperatures
Enhancers of VI	Increase the viscosity index allowing to obtain a lubricant constant in a wide range of temperature. Used mainly to the extreme temperatures temperature
Anti-foaming	Reduce the danger of the formation of foam in the lubricant
Adhesiveness enhancers	Increase the adhesion of the lubricant to the surface with which it is in contact
Compatibility with seals	Good

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5.2. GREASE LUBRICATION

Greases for bearings must possess high lubricity power, good mechanical stability, an effective oxidation resistance and good anti-rust features, especially for parts operating in humid environment or subjected to splashing water. Their consistency, generally of grade 1, 2 or 3 of the NLGI scale, must remain as stable as possible within the temperature limits allowed by their composition.

5.2.1. Main types of grease

The grease is a thick lubricant, it consists of the base oil, plus additives and a thickener which is very often composed of a soap.
Greases based on lithium soap are particularly suitable for the lubrication of needle and rollers bearings and thrust bearings. They can be used at operating temperatures between -30 and +120°C, and even up to 150°C if they are of good quality. They are generally fitted with anti-rust additives and offer a good protection against corrosion.
Greases based on sodium soap are suitable for the lubrication of the bearings up to approximately 100°C (minimum temperature -30°C) and ensure a good seal against dust. They can absorb small amounts of water without losing their lubricating properties, but high amounts of water will dissolve and cancel all their effectiveness.
Greases based on calcium soap are stable to water and can be used only up to 50 or 60°C. Their mechanical stability and their power anti-rust are weak. Their use as lubricants for bearings is therefore not recommended, but may be used in labyrinth seals. However, some grease calcium based, with increased mechanical stability and anti-rust power, can be used up to 100°C to lubricate bearings in a humid atmosphere.

	Lithium soap	Sodium soap	Calcium soap	Polyurea	Lithium aluminium complex soap
Temperature range	120	110	60	160	160
Drop point	190	260	100	230	260
Water resistance	Good	Low	Excellent	Excellent	Good
EP capacity	Good	Good	Good	Low	Excellent

5.2.2. Consistency

The parameter that determines the softness or hardness of the grease is the consistency, that is, the penetration of the lubricant. It is defined by the NLGI consistency scale of measurement, according to eight levels which corresponds to a range of values of the Worked Penetration, expressed in tenths of millimeter.

The following table shows the classes defined by the NLGI consistency.

NLGI class	Worked Penetration	Texture
000	445 – 475	Liquid
00	400 – 430	Semi-liquid
0	355 – 385	Very very soft
1	310 – 340	Very soft
2	265 – 295	Soft
3	220 – 250	Medium
4	175 – 205	Hard
5	130 – 160	Very hard
6	85 - 115	Extremely hard (as softwood)

5.2.3. Special grease

Greases with **EP additives** (high pressure) can be useful when bearings or thrust bearings must work with heavy loads. These greases generally offer a good lubricating power and have good anti-rust properties even in the presence of moisture. EP additives are used in the case of bearings with high load and low rotation speed, insufficient to create a meatus of lubricant sufficient to separate the metal parts.
Greases for **low temperatures**. The starting torque at low temperatures can be problematic. Suitable acids are commercially available.
Greases for **high temperatures**. The stability and duration of the grease is strongly influenced by temperature. In general the standard greases can be used up to 120°C-150°C. Further should be provide specific products. For high temperatures can be used lubricating pastes.

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	Al Complex	Ba Complex	Ca Stearate	Ca 12Hydroxy	Ca Complex	Ca Sulfonate	Clay Non-Soap	Li Stearate	Li 12 Hydroxy	Li Complex	Polyurea	Polyurea S S
Aluminum Complex	Best Choice	Incompatible	Incompatible	Compatible	Incompatible	Borderline	Incompatible	Incompatible	Incompatible	Compatible	Incompatible	Compatible
Barium Complex	Incompatible	Best Choice	Incompatible	Compatible	Incompatible	Compatible	Incompatible	Incompatible	Incompatible	Incompatible	Incompatible	Borderline
Calcium Stearate	Incompatible	Incompatible	Best Choice	Compatible	Incompatible	Compatible	Compatible	Compatible	Borderline	Compatible	Incompatible	Compatible
Calcium 12 Hydroxy	Compatible	Compatible	Compatible	Best Choice	Borderline	Borderline	Compatible	Compatible	Compatible	Compatible	Incompatible	Compatible
Calcium Complex	Incompatible	Incompatible	Incompatible	Borderline	Best Choice	Incompatible	Incompatible	Incompatible	Incompatible	Compatible	Borderline	Compatible
Calcium Sulfonate	Borderline	Compatible	Compatible	Borderline	Incompatible	Best Choice	Incompatible	Borderline	Borderline	Compatible	Incompatible	Compatible
Clay Non-Soap	Incompatible	Incompatible	Compatible	Compatible	Incompatible	Incompatible	Best Choice	Incompatible	Incompatible	Incompatible	Incompatible	Borderline
Lithium Stearate	Incompatible	Incompatible	Compatible	Compatible	Incompatible	Borderline	Incompatible	Best Choice	Compatible	Compatible	Incompatible	Compatible
Lithium 12 Hydroxy	Incompatible	Incompatible	Borderline	Compatible	Incompatible	Borderline	Incompatible	Compatible	Best Choice	Compatible	Incompatible	Compatible
Lithium Complex	Compatible	Incompatible	Compatible	Compatible	Compatible	Compatible	Incompatible	Compatible	Compatible	Best Choice	Incompatible	Compatible
Polyurea Conventional	Incompatible	Incompatible	Incompatible	Incompatible	Compatible	Incompatible	Incompatible	Incompatible	Incompatible	Incompatible	Best Choice	Compatible
Polyurea Shear Stable	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Borderline	Compatible	Compatible	Compatible	Compatible	Best Choice

5.2.4. Compatibility of greases

Certain greases are incompatible with others and, if they are mixed, their function will be impaired.
With greases considered as compatible, account should be taken of the reduction in their consistency when mixed and the maximum permissible temperature should be reduced accordingly.

5.2.5. Application

Grease can be introduced into the bearings at the time of assembly, care being taken to distribute it around the crown of the needles (see below "Quantity of grease").
The free space found in the bearing which is filled with grease, constitutes a reservoir and a reinforced seal. This method is possible if replenishments of grease are necessary at regular maintenance periods, during the course of which one can dismount the bearings, clean and examine them. Otherwise one has to use a hand pump which forces grease into the bearing by means of valves and replenishes the adjacent reservoir and also the channels and labyrinth seals.
The entry passage for the grease must directly abut the bearing or be in close proximity to it, in order that new fresh grease pushes out the used grease through the seals. For this reason the lip of the sealing ring must be oriented towards the outside of the bearing for it to rise under the force of the grease being ejected. This method has the advantage of removing impurities which could be introduced into the seals, particularly in the case of a highly contaminated atmosphere.

5.2.6. Quantity of grease

The amount of grease that should be contained in a bearing can be established by considering the relationship of the limiting speed permissible for the grease n_G to the speed of rotation n:

- $n_G/n < 1{,}25$ minimum quantity; bearing must be lubricated with a small quantity of grease and the adjacent parts packed with grease
- $1{,}25 < n_G/n < 5$ 1/3 to 2/3 of the available volume
- $n_G/n > 5$ bearing must totally filled with grease.

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5.2.7. Re-lubrication

The frequency of grease re-lubrication depends on a number of factors, amongst which are the type of bearing and its dimensions, the speed and load, the temperature and ambient atmospheric conditions (humidity, acidity, pollution), the type of grease and sealing.

Only after controlled trials can the re-lubrication period be defined exactly and particular importance should be given to the effects of temperature, speed and humidity. Under normal conditions of function without unfavourable factors using an appropriate grease with a maximum temperature of 70°C, the re-lubrication interval T_G in hours can be determined approximately from the formula:

$$T_G = \frac{K \times 10^6}{n \times \sqrt{Fw} \times \sqrt[4]{\frac{n}{n_G}}}$$

n: speed of rotation

n_G: permissible speed limit for grease lubrication (see page 14)

Fw: diameter of inner raceway of bearing in mm

K: coefficient according to the type of bearing:
K = 32 for caged needle bearings K = 28 for full complement needle bearings K = 15 for needle or roller thrust bearings.

For the bearings below, the diameter Fw is replaced by the following dimensions, given in the table of dimensions:

Cam followers type FG and derivatives: dimension d_A
Needle or roller thrust bearings: dimension E_b

Cam followers type GC and derivatives:
average dimension $\frac{d+d_A}{2}$

If the operating temperature exceeds 70°C, the interval T_G determined from the formula above should, for each increase of 10°C, be reduced by 50%. However, this adjustment is not applicable beyond 115°C; for temperatures above this level trials should be made to determine the acceptable re-lubrication interval.

In the case of very slow speed rotation, which would give interval T_G in excess of 35000 hours corresponding to 8 years operation at a rate of 12 hours per day, it is recommended to limit the period to a maximum of 3 years.

For oscillating motion, the speed to be considered is the equivalent speed given by the formula on page 11. For very small amplitudes of oscillation it is recommended to reduce by half the calculated re-lubrication period T_G.

5.3. OIL LUBRICATION

5.3.1. Viscosity

The essential characteristic of an oil is its basic kinematic viscosity in mm²/sec. at a reference temperature of 40°C according to ISO 3448.

The base viscosity V_{40} should be increased proportionately as the operating temperature increases but decreased as the speed increases, without however reaching a lower limit below which the film strength of the oil is impaired. For applications under moderate load without shocks up to about 1/5 of the dynamic capacity of the bearing, the viscosity V_F at the operating temperature should not be lower than 12 mm²/sec. For higher loads greater than 1/5 of the dynamic capacity the min. viscosity V_F can be about 18 mm²/sec. The variation in viscosity of an oil as a function of temperature is reduced as the number measuring its index of viscosity is increased. A viscosity index of 85 to 95 is generally satisfactory for the lubrication of bearings.

Diagram 1 below gives the viscosity V_F required at the operating temperature from the ratio n_H/n (n_H: permitted speed limit for oil lubrication - n: speed rotation) and of the applied load (ratio C/P).

For the viscosity V_F required in operation and from operating temperature, diagram 2 gives the base viscosity V_{40} at the reference temperature of 40°C.

Example: A bearing supporting a load P>C/5 and having a speed limit for oil lubrication of 10000 r.p.m., must rotate at 2000 r.p.m. at temperature up to 60°C.

The ratio $\frac{n_H}{n} = \frac{10.000}{2.000} = 5$

indicates a viscosity in operation V_F = 60 mm²/sec. (diagram 1). For an operating temperature of 60°C, the horizontal V_F = 60 cuts the vertical of 60°C (diagram 2) in the 150 zone, which is therefore the base viscosity required at 40°C.

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5.3.2. Application of the lubricant
Oil must be supplied to the bearings regularly and in sufficient quantity but not abundantly, otherwise an abnormal increase in temperature can occur. According to the speed of rotation, the following general lubrication methods can be applied:

Lubrication by oil bath: is suitable for assemblies with the shaft horizontal and average speeds up to about half the values shown in the tables of dimensions. The level of oil in the bath at rest must reach the lowest point of the inner raceway of the bearing, though the movement of oil caused by the immersion of parts in the oil bath may be sufficient to feed bearings situated above this level, providing there are pipes and collectors to ensure sufficient oil reserve when starting.

Forced lubrication: the circuit is typically composed of the tank, the circulation pump, hoses and fittings, filter, possibly the radiator. Allows to effectively lubricate the bearings even in case of high speed, remove dirt and moisture from the bearing, if necessary to remove the heat generated in the bearing.
For the thrust bearing, the arrival of the oil must be made, if possible, from the shaft to use the effect of centrifugation in the sense of movement.

Oil mist lubrication: consists of applying to the bearings oil finely atomised in suspension in a current of clean compressed air. The pressure created within the bearing effectively protects it from the introduction of dust, humid vapours and noxious gases. This procedure, which allows a substantial flow from a small quantity of oil, is used particularly for ultra-high speed applications in excess of speed limits given in the tables of dimensions.

6. BEARINGS STORAGE
With the exception of cam followers which are delivered lubricated with grease, all other needle or roller bearing products are supplied without grease, though protected against oxydation by an oil film compatible with most greases and mineral oil lubricants. Bearings should be stocked in a clean dry environment and retained in their original wrapping until the last moment before assembly. Even when assembling the bearing, care should be taken to prevent contamination from dirt or metallic particles and humidity.
In case of doubt concerning cleanliness of the bearing, it may be necessary to wash it in filtered petroleum. In so doing the bearing must be rotated and then suitably drained and dried. Smear the bearing with a suitable oil or grease to protect it against oxydation at the time of assembly.
Avoid the use of compressed air to clean or dry the bearing.
And to avoid the risk that a needle roller can be removed from its place and launched (danger for the operator and the people close to him), and because the air introduces moisture into the component.

Diagram 1

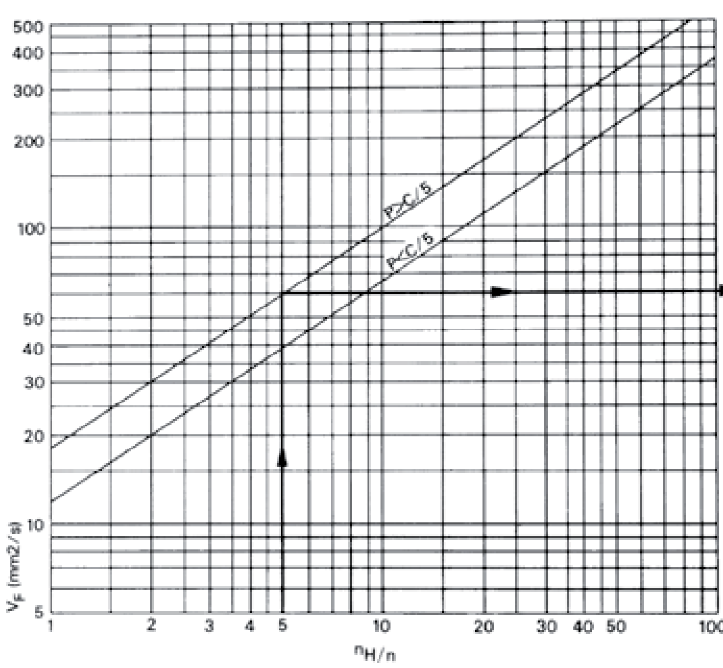
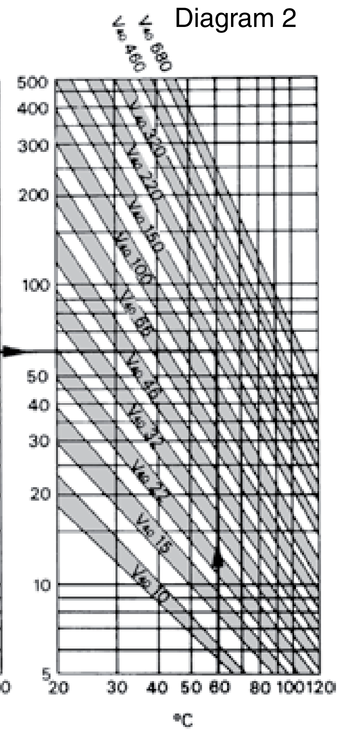


Diagram 2



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