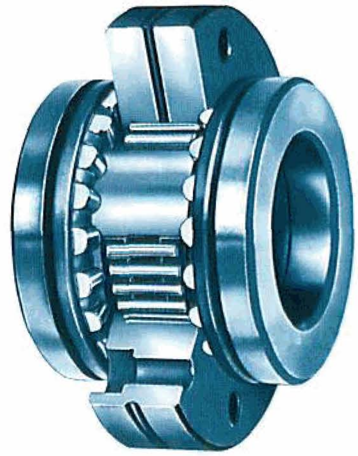
PRECISION COMBINED BEARINGS, WITH ADJUSTABLE AXIAL PRELOAD.



Types AXNA, AXNB and ARNB combined bearings and their derivatives consist of a needle bearing with or without a cage, in an outer race, with a high radial thickness, each face of which acts as a raceway for a needle or roller thrust bearing. The inner ring, secured laterally between the thrust plates, acts as the inner radial raceway.

These bearings which take up very little space, are particularly recommended for shafts requiring very precise axial positioning, operating under load, such as leading spindles, ball-screws for numerically-controlled machine tools, drive shafts on control apparatus, etc.

SERIES TYPE				
	With attachment holes	Radial caged bearing	Thrust bearing	
			needle	roller
AXNA			●	
AXNAT	●		●	
AXNB		●	●	
AXNBT	●	●	●	
ARNB		●		●
ARNBT	●	●		●

SELECTION OF BEARING TYPE

Subject to calculations made for each application, the following general classifications can be made:

AXNA, AXNAT and AXNB, AXNBT bearings for slow speed assemblies with low operating loads: the particularly high axial rigidity of needle thrust bearings, together with the advantages of preloading, ensure a very high axial precision and satisfactory working life.

For example: displacement drive shafts on control apparatus.

ARNB and ARNBT, series 1 and 2 bearings generally enable preloading to be chosen which suit the precision and working life required of production machine tools.

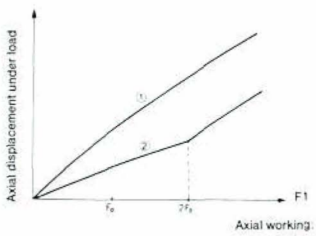
ARNB series 3 bearings for machine tools, machining units or special equipment requiring very high axial rigidity with high loads and slow speeds.

PRELOAD

This technique consists in subjecting the thrust bearings to controlled preload during assembly, using an adjusting nut, in order to eliminate play and reduce the axial displacement caused by the operating stress regardless of the direction or the axial load.

NADELLA has always made the inner ring slightly longer than the space between the thrust plates before adjustment. This means that when the nut is tightened, the inner ring is compressed between the thrust plates and exerts a stress, by reaction, on the internal thread of the screw. This prevents it from being loosened and loss of adjustment occurring.

In an assembly with an axial preload of Fo, an operating stress F1 overloads one of the thrust bearings and frees the other of a load approximately equal to F1/2. In an assembly without preload, the loaded thrust bearing must carry the entire stress F1.



(1) Thrust bearings without preload.
(2) Thrust bearings under a preload Fo.

In a preloaded assembly, the axial rigidity is therefore approximately twice that of an assembly without preload. This result is obtained as long as the operating stress F1 remains less than about twice the preload stress Fo. When F1 > 2 Fo, one of the thrust bearings is total freed and the other thrust bearing completely carries the load F1; in this case, the axial run-out remains less than it would have been for an assembly without preload (see figure).

DETERMINING OF PRELOAD

Preload should be determined according to the axial precision required under maximum load and the working life required.

The working life of the thrust bearing carrying the greater load depends on the resulting stress applied, i.e. Fo + F1/2 when F1 < 2 Fo or when F1 > 2 Fo. Since these two cases can both occur on the same machine according to the type of machining carried out, the calculations must take into account the running time ratios under the various loads and speeds.

For more usual assemblies, a preload stress Fo of 5 to 10% of the dynamical load carrying capacity C of the thrust bearing, is usually suitable.

For certain applications, with slow rotating speeds, for example, the preload stress can be increased to allow for a higher operating load while remaining within the limit of the preload effect, and achieving a satisfactory working life.

ADJUSTMENT OF PRELOAD

For a given assembly, the shaft torque is defined first, which corresponds to the preload required. Series adjustments can then be made on each machine by simply checking the torque. If, as a result of assembly, this is not possible, the nut tightening torque needed to obtain preload is determined separately on the test assemblies. The torque must then be respected for series adjustments. The torque must be measured after starting up the thrust bearing, since it can be up to 50% higher at the beginning of rotation.

BEARING TOLERANCES

The outer and inner rings of the combined bearings are manufactured with class 5 tolerances according to ISO Standard 492 (class P6 of standard DIN 620).

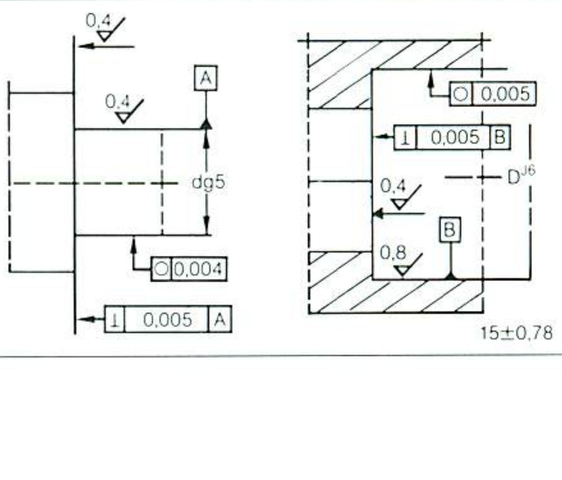
The radial play before assembly is kept within the limits of group 2 given for inner and outer paired rings according to ISO Standard 5753 (class C2 "paired rings" of standard DIN 620). See table page 45 (play C2ZS).

The axial run-out of the thrust bearing plates is in accordance with tolerance class 4 according to ISO Standard 199 (class P4 of standard DIN 620).

ASSEMBLY RECOMMENDATIONS

Shaft tolerance: g5 on dimension Di.
Tolerance of outer ring housing: J6 on dimension De.

The bearing parts of the thrust bearings must be rigid, with plane faces, perpendicular to the rotation axis and of very good surface quality to avoid caulking during use as this decreases the preloading. Their outside diameter must be at least equal to the average diameter of the race, dimension Dm.



The outer ring of the combined bearings should be blocked against a shoulder in order to avoid any axial displacement under load.

In type AXNA, AXNB and ARNB bearings, they are usually blocked by a spacer positioned lengthways during assembly. A flange attached by screws to the frame is located against the spacer.

The outer ring of type AXNAT, AXNBT and ARNBT bearings has three attachment screw holes for direct attachment to the frame.

Apart from watertight bearings (AXNBT.../2 or ARNBT.../2) or the use of long plates (AXNB (T).../1 or ARNB (T).../1), friction of joints on the outside diameter of the thrust bearing plates (dimension A) can be envisaged. In this case, please consult us for positioning.

NADELLA's technical services will supply any further information concerning the choice or assembly of these bearings, on request, together with calculation and adjustment of the axial preload.

LUBRICATION

The oil used to lubricate the other parts of the assembly is generally suitable for combined bearings whose outer ring has three 120° holes connected by a groove. Grease can generally be used if the rotating speed is in the order of 50% of the maximum speeds given in the dimensional tables. However, special top quality greases enable higher speeds to be reached. By way of information, oils with viscosities of 30 to 150 cSt are recommended.

EXAMPLES OF CALCULATIONS

► *Choice of bearing*

P: stress under which precision is needed.

$P < 2$ x Preloading.

In this field of preloading, the axial rigidity is equal to 2K.

The interference is $\frac{1}{2K} P$

Example: If P = 7000 N, ARNB 50 90 will be chosen, since the preloading value is 3800 N and

$2 \times 3800 = 7600 \text{ N} > P$.

Rigidity in this field $k = 2K = 3900$ N/µm.

Under P, the interference will be

$\frac{1}{3900} \times 7000 = 1{,}79$ µm.

► *Working life*

The hypotheses given in the table below enable the equivalent speed and an equivalent load to be determined according to the maximum load and maximum speed, which enables a rapid calculation of the theoretical working life to be made under average operating conditions.

	1	2	3	4
Loads	Pmax	0,8 x Pmax	0,5 x Pmax	0,2 x Pmax
Speeds	0,05 x Vmax	0,2 x Vmax	0,5 x Vmax	Vmax
Fraction of time	0,15	0,40	0,30	0,15

► *Calculation of equivalent speed:*

$V_{éq.} = (0{,}15 \times 0{,}05 + 0{,}40 \times 0{,}2 + 0{,}30 \times 0{,}5 + 0{,}15)\ V_{max} \approx 0{,}39 \times V_{max}$

Calculation of equivalent load:

$$P_{eq} \approx \sqrt[P]{\frac{P^P_{max} \times n_{max}\ (0{,}0075 + 0{,}08 \times 0{,}8^P + 0{,}15 \times 0{,}5^P + 0{,}15 \times 0{,}2^P)}{0{,}39 \times V_{max}}}$$

$P_{éq} \approx 0{,}575 \times P_{max}$

$p = 10/3$

NADELLA

This comparative method can be used for traverse mechanisms on conventional machine tools.

For special machines and control apparatus, the breakdown of loads and speeds can be different and the formula must be applied with caution.

Note: in this rapid calculation, preload is not taken into consideration; its influence on the working life of the bearings is actually very low for most applications if the adjustment conditions given in the literature are respected: preload between 5 and 10% of the dynamic capacity of the thrust bearings.

Example: for a maximum load P of 14 000 N and a maximum speed of 1000 r.p.m.

Equivalent speed: 0.39 x 1000 = 390 r.p.m.

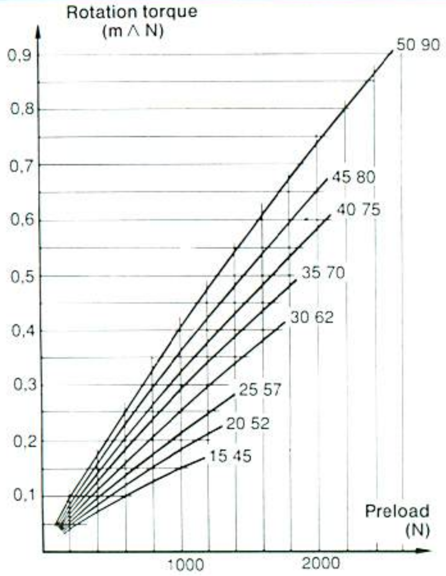
Equivalent load: 0.575 x 14 000 = 8050 N.

Theoretical working life of ARNB 50 90:

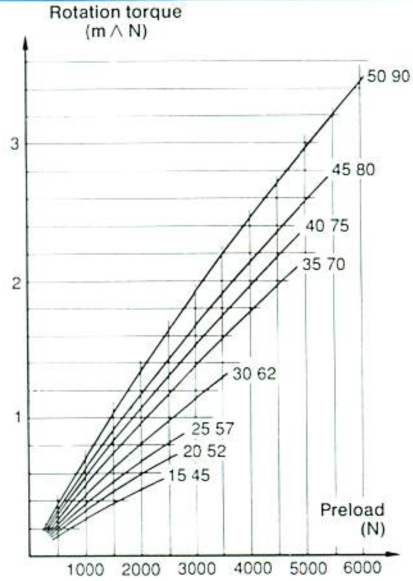
$$\frac{\left(\frac{C}{P}\right)^{\frac{10}{3}} \times 10^6}{60n} = \frac{\left(\frac{60.000}{8050}\right)^{\frac{10}{3}} \times 10^6}{60 \times 390} = 34\,600 \text{ hours}$$

In this example, it is assumed that the time fraction n° 2 is a time fraction when precision machining is not required.

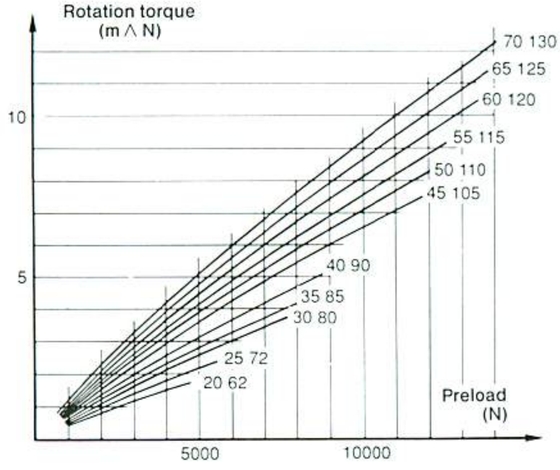
ROTATION TORQUE AS A FUNCTION OF THE PRELOAD



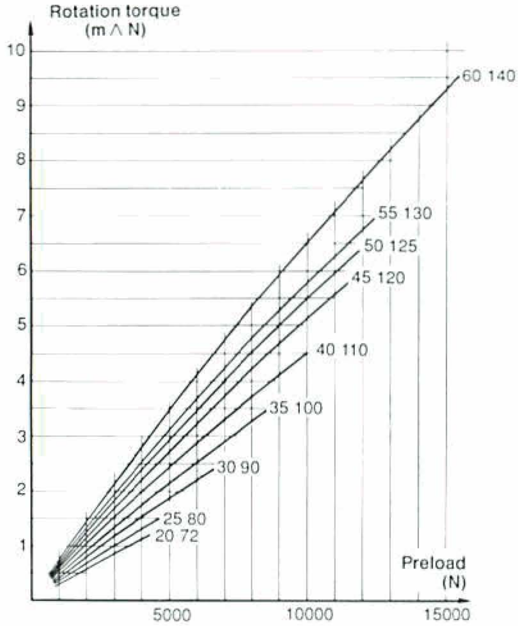
TYPE AXNB



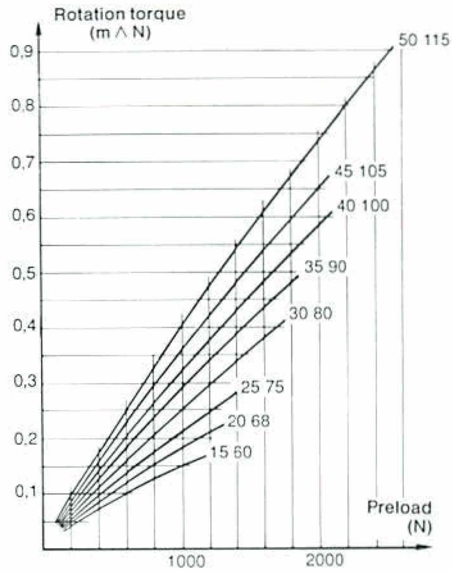
TYPE ARNB Series 1



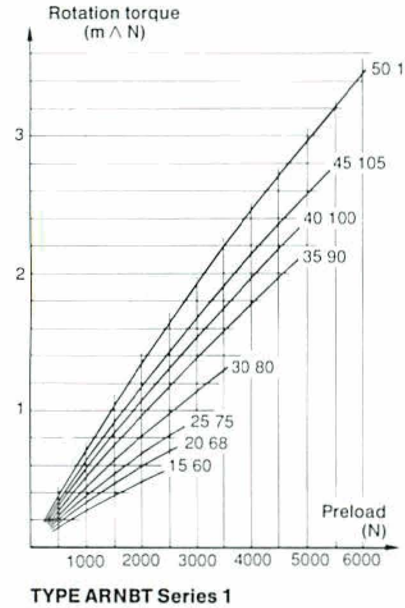
TYPE ARNB Series 2



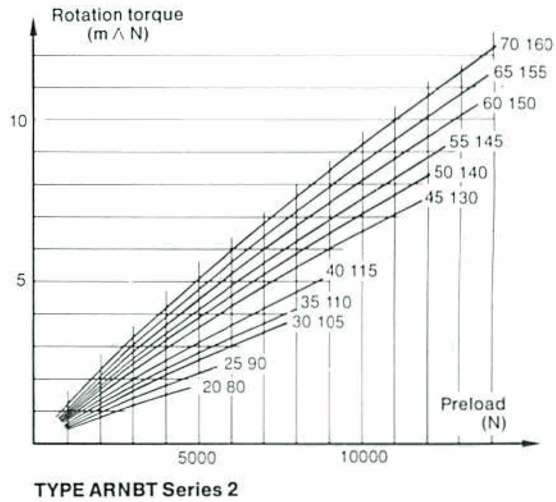
TYPE ARNB Series 3



TYPE AXNBT



TYPE ARNBT Series 1



TYPE ARNBT Series 2