SLEWING RING SELECTION

- SLEWING RINGS FUNCTIONS
- OTHER FEATURES

SLEWING RING FUNCTIONS

LOADS SELECTION

The slewing ring, being the link between a mobile element and a fixed base, must have the capacity to transmit the stresses of the mobile part towards the base. It is necessary to accurately define all the actual stresses applied so that a suitable slewing ring with adequate capacity can be selected. This should include the effects due to masses and inertias of the payloads and structures.

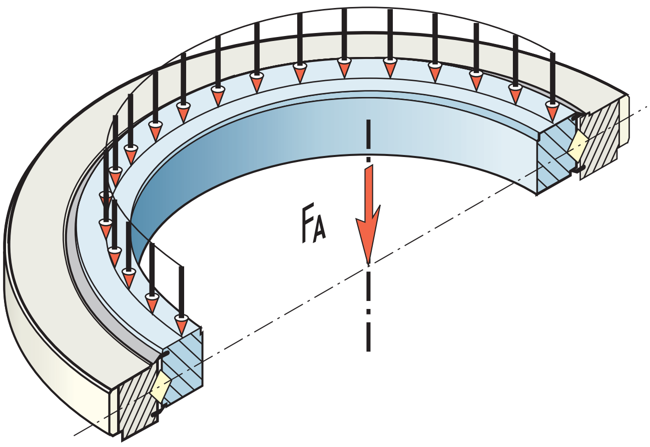
It is necessary to distinguish the fixed loads and the variable loads as well as the effects due to dynamic loads, the latter constituting "fatigue" stresses. The direction of forces affecting the slewing ring must be well defined so that the active tilting moment can be established.

We distinguish :

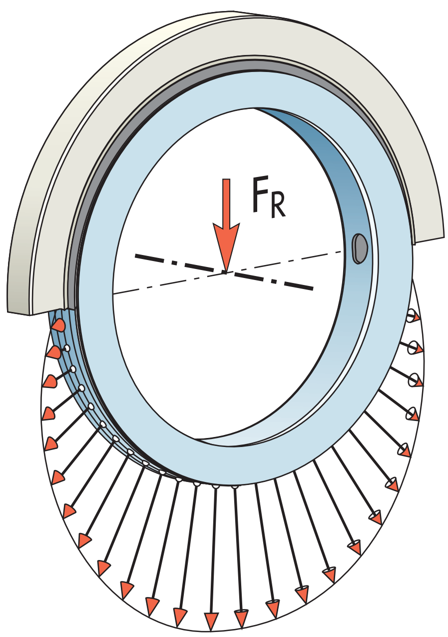
• **The AXIAL LOADS** whose direction is parallel to the slewing ring rotation axis. The resultant of these loads is called $\mathbf{F_A}$.

• **The RADIAL LOADS** contained in planes perpendicular to the rotation axis. The resultant of these loads is called $\mathbf{F_R}$.

AXIAL LOADS



RADIAL LOADS



SLEWING RING FUNCTIONS

LOADS SELECTION

• **TILTING MOMENTS** (bending) : in planes parallel to the rotation axis. The resulting moment working in the plane containing the rotation axis is called **M_T**.

• **SLEWING TORQUE C_D**
controls the slewing ring rotation.

CALCULATION OF THE EQUIVALENT LOAD
For the calculation, the resultant of the radial loads **F_R** is transposed into an equivalent axial load using a factor **K_R** as follows :

For **standard slewing ring :**

if : $\frac{F_R}{F_A} < 0,25$ →	$K_R = 0,5$
if : $0,25 < \frac{F_R}{F_A} < 1$ →	$K_R = 1,5$
if : $\frac{F_R}{F_A} > 1$ →	$K_R = 2,4$

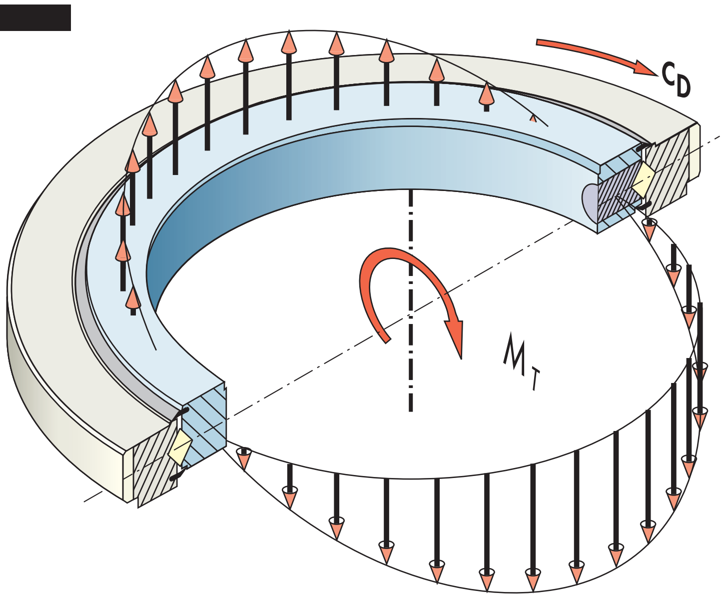
For **light series and solid sections :** → **$K_R = 3,225$**

The equivalent loads F_{eq} can be obtained by the following formula :

• For **the horizontally mounted slewing rings :** vertical rotation axis :
$F_{eq} = F_A + K_R \cdot F_R$

• For **the vertically mounted slewing rings :** horizontal rotation axis :
$F_{eq} = F_A + 1,2 \cdot K_R \cdot F_R$

TILTING MOMENT



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For ROLLIX	For the USER
N° Affaire :	Name :
Date :	Date :
AC ☐ VI ☐ RC ☐	
Visa :	Signature :

DESIGN DATA SHEET

1.1	COMPANY :	Tel. :
1.2	Address :	Fax :
1.3	Person in charge of the project :	Fonction :
2.1	DESCRIPTION OF THE APPLICATION + sketch (to be attached with loads applied)	
2.2	Machine / project reference :	New project : Yes ☐ No ☐
2.3	Slewing ring position : Horiz. ☐ Vert. ☐ Other ☐	Utilisation mode : Continu. ☐ Intermittent ☐ Other ☐
3.1	LOADS ON THE BEARING (including structural loads) :	Applied ☐ Suspended ☐

3.2			Static			Dynamic		
3.3			Nominal	Maximum	Test	Nominal	Maximum	Test
3.4	Utilisation	% Time						
3.5	Axial	kN						
3.6	Radial	kN						
3.7	Moment	kNm						

3.8	Rotating ring : Ext ☐ Int ☐ Speed (RPM)
3.9	Load factors applied : Excluded ☐ Included ☐ Values :
3.10	Required life :
4.1	GEAR External ☐ Internal ☐ Without gear ☐

	Geometry	S.R.	Pinion	Loads	S.R.	Pinion
4.2	Required Module			Tangentiel load (kN)		
4.3	Number of teeth required			Torque (kNm)		
4.4	Addendum modif. factor					

5.1	CRITICAL ITEMS Dimensions	Others :
5.2	Classification ☐ Commission ☐ FEM ☐ LLOYDS ☐ API ☐ BV ☐ DNV ☐ Other	
5.3	Specification relative to the application :	
	SPECIFIC REQUIREMENTS OF THE APPLICATION	
6.1	Environment :	Operating temperature :
6.2	Vibrations, shocks loads :	Storing temperature :

6.3	Acceleration :	Deceleration :	Inert. mom./Rotation axis :	Varied :

7.1	QUANTITY Yearly requirements :	Qty per delivery :
8.1	REQUIRED DELIVERY TIME :	

IT ETR 911

SKETCH OF YOUR APPLICATION SHOWING THE LOAD POSITIONS

SLEWING RING FUNCTIONS

BEARING FUNCTION

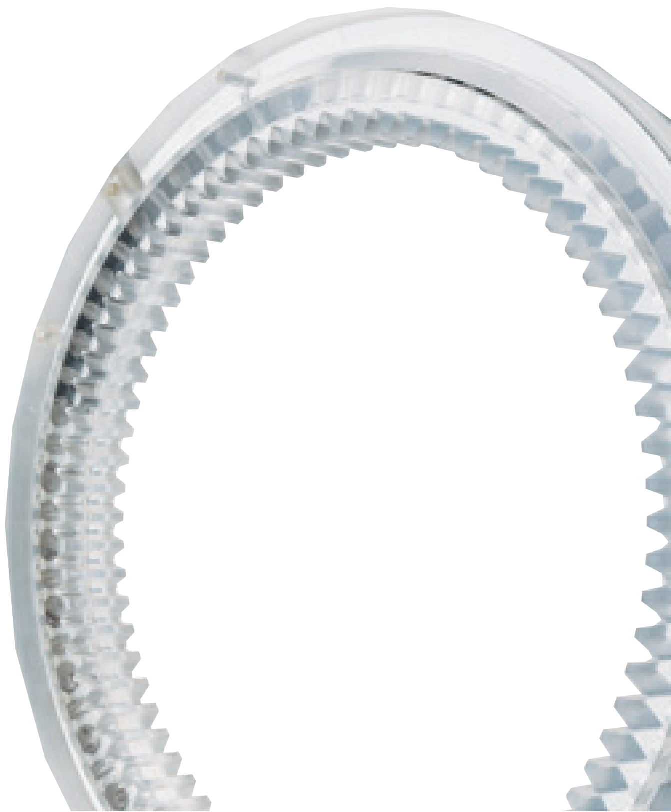
The knowledge of loads and working conditions is necessary to allow us to design and dimension the "BEARING" function of the slewing ring i.e : movement type, speed, accelerations, temperatures, environment, etc.

The **transmission of loads** from one raceway to another varies according to the nature of applied loads.
In order to calculate the ideal dimensions of the raceway, we define the load equivalent to all external efforts in the most stressed areas.
These loads are corrected by factors according to the application, the usage, etc.
We distinguish between :
- the application factor K_A
- the usage factor K_U
- the safety factor K_S

THE APPLICATION FACTOR K_A
is a coefficient taking into account the application specificity in relation to the slewing ring element.
This factor is established from ROLLIX experience. It is defined in the table "APPLICATION FACTORS".

THE USAGE FACTOR K_U
is defined according to the particular operating conditions : vibrations, shocks, occasional or accidental overloading, etc. If no other value is specified, then the nominal value is taken as 1.

THE SAFETY FACTOR K_S
is defined from standardized criteria for applications which must meet specific regulations such as : FEM, LLOYDS, API... This generally has the value 1, as the designer of the machine must include the regulation factors in the calculation of the loads applied on the bearing.



SLEWING RING FUNCTIONS

BEARING FUNCTION

APPLICATION FACTORS

2

MACHINES	Average speed Rpm	Applications factors K_A
Armament turret	1,5	1,5
Bucket	1,5	1,65
Cable shovel	2	1,65
Compacter	2,5	1,80
Concrete mixer	5	2,40
Concrete pump	1,5	1,65
Dragline	1,5	1,65
Dock crane	1	1,65
Fairlead	0,8	1,35
Fork-lift truck	1	1,35
Fork-lift wheel	1,5	1,50
Grabbing crane	1,5	1,80
Heavy winch	2	1,65
Hydraulic lift platform	1	1,50
Hydraulic shovel	2,5	2
Loading dock crane	1	1,65
Merry-go-round	5	2,40
Mine digging machinery	1,5	2
Mobile fixed boom crane	1	1,5
Mobile grapple crane	1	1,80

MACHINES	Average speed Rpm	Applications factors K_A
Mobile telecospic crane	1	1,65
Offshore crane	1	1,8
Post jib crane	1	1,35
Railway crane	1	1,50
Rapid rotation radar	5	2
Rapid rotation scanner	3,5	1,65
Robotics	3,5	1,65
Service deck crane	0,8	1,35
Settler (water and sewage treatment)	0,6	1,35
Slow rotation radar	1	1,35
Slow rotation radiology	1	1,35
Tower crane, slewing jib type	1	1,65
Tower crane, slewing tower type	1	1,80
Track hook crane	1,5	1,80
Truck crane	1	1,50
Turntable	1	1,35
Vibrating compacter	2,5	2
Welding positionner	0,8	1,35
Windturbine	0,8	1,65

These factors are determinated statistically and are based on a large number of observations for each type of application.

The standard parameters retained are as follows :
- Theoretical service life : 6000 hours.
- Work under normal weather conditions.
- Conventional application (and not specific).

SLEWING RING FUNCTIONS

BEARING FUNCTION

Selection of the ring according to capacity

The load capacity of the slewing ring is calculated according to its performance in function of :

- its geometric envelope,
- the nature of the ring materials,
- the heat treatment carried out,
- the nature, the number and the dimension of the rolling elements,
- the contact parameters of the rolling elements.

The curve of the maximum permissible capacity is drawn on a graph whose Ox axis bears the equivalent axial load and the Oy axis bears the tilting moment. To simplify it, it is represented by a straight line called the "LIMIT CURVE".

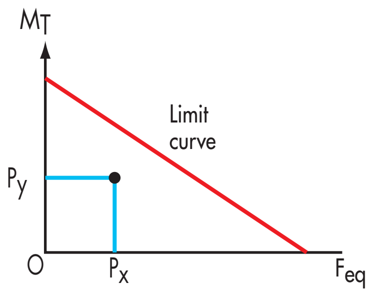
The ring size is determined by plotting the representative point of loads onto this curve. This point, called "appli-cation point" has the following coordinates :

- on the horizontal axis :

$P_x = F_{eq} . K_A . K_U . K_S$

- on the vertical axis :

$P_y = M_T . K_A . K_U . K_S$



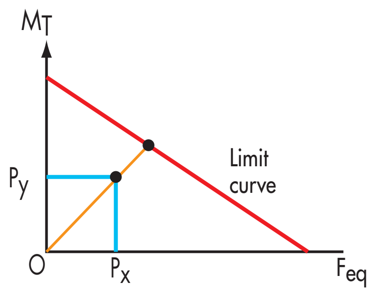
In any case, the application point P must be under the limit curve.

Service life

Many external factors have a very important influence on the service life of the bearing. Among others, we can cite :

- geometric quality of supports,
- structure deformation under load,
- climatic conditions and environment,
- quality of operating maintenance
- conditions of use :

repeated exposures to shocks, vibrations or sudden or intermittent movements can considerably reduce the theoretical service life.



$K_T = OL / OP$

An estimate of the theoretical service life can be obtained by comparing the application point to the limit curve : the ratio OL/OP is called K_T.

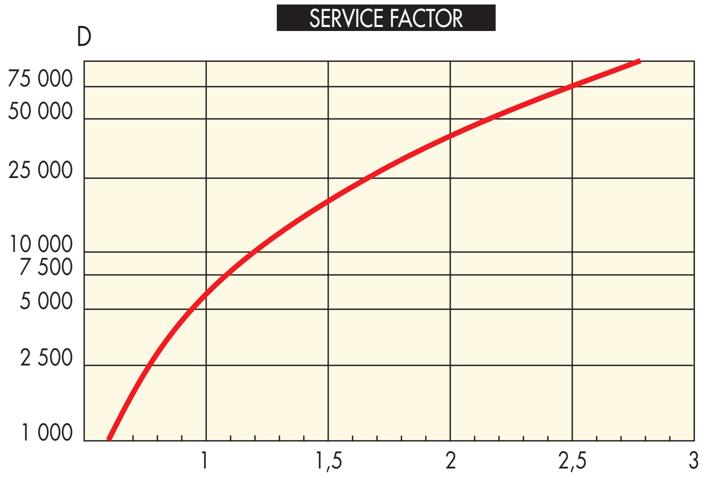
SLEWING RING FUNCTIONS

BEARING FUNCTION

Service Life

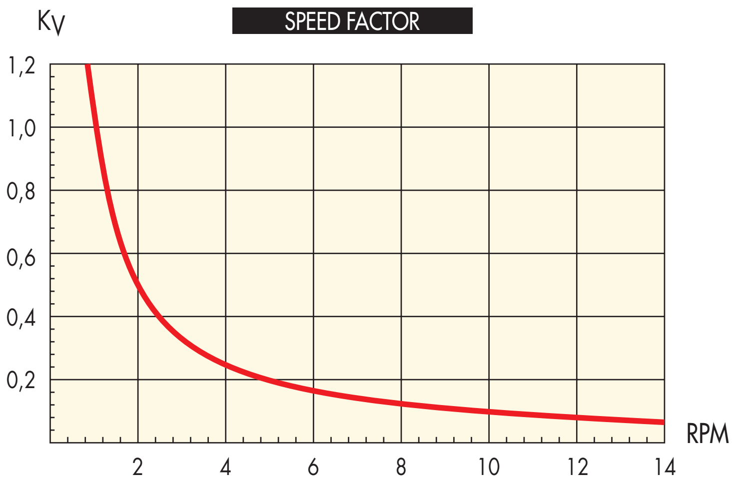
An estimate of the service life D can be obtained using the opposite graph :

• The curve indicates the estimated service life (hours) directly from the K_T value on the horizontal axis.



Rotational speed influence

The service life D, estimated on the graph is only valid for the applications having a low rotational speed : 1 RPM. The value obtained must be multiplied by the **speed factor K_V** indicated on the opposite graph when speeds differ from this.



$D(n) = K_V \times D$

For applications having oscillating movements, the following formula applies :

$n_{average} = 0{,}60 \times n_{real}$

SLEWING RING FUNCTIONS

FASTENING FUNCTION

In order to transmit the loads previously defined, it is necessary to realize an adequate mechanical fastening of the bearing on the associated frames, thus forming rigid connection of the ring and its supports.
While several fastening methods are feasible, the most efficient one remains the use of screws and nuts.
Welding operations are absolutely prohibited.
The slewing ring proper functioning and the application safety are dependent on the correct bolting definition and fastening method during installation, complying with our workmanship.

BOLTS QUALITY
The ISO 898-1 standards define the bolting grade adapted to structure assemblies such as slewing rings.
ROLLIX recommends the use of **HIGH TENSILE bolts grade 10.9** and exceptionally grade 8.8 or 12.9 with rolled threads after heat treat.

The **nuts** must be of a same or higher grade as the associated screw. For a screw diameter d, a nut height of 1.d is recommended.

For rings in normalized steel Z or N, the use of **hardened flat washers** is required.
Minimal properties should be :
- a yield strength greater than or equal to 600 Mpa,
- a diameter : $D_R = 2$ d,
- a thickness : $h \geq 0{,}3$ d

External hexagon head screws must be preferred to cap screws (internal hexagon) whenever possible.
ROLLIX recommendation : screws and nuts, with guaranted mechanical properties, matched, prelubricated so as to obtain a **known** and permanent screw/nut friction factor. The surface coating on the bolts must not generate any embrittlement.

Minimal mechanical characteristics (according to ISO)

GRADE	TENSILE (MPa)	YIELD (MPa)	FATIGUE (MPa)	
8.8	800	640	40	**Exceptional**
10.9	**1040**	**940**	**40**	**Recommended**
12.9	1220	1100	**40**	**Exceptional**

SLEWING RING FUNCTIONS

FASTENING FUNCTION

BOLTING CALCULATION

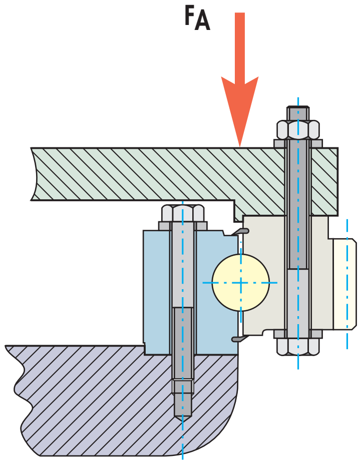
The ROLLIX calculation formulae take into account the current standards and regulations as well as the many research and experimental findings. These calculations are mainly inspired by the AFNOR FD E 25.030, the recommendation VDI 2230 (1988) and the standard API 2C (1995).

Supported loads must be distinguished from hanging (suspended) loads acting in tension.
Consult ROLLIX, in the case of hanging (suspended) loads .

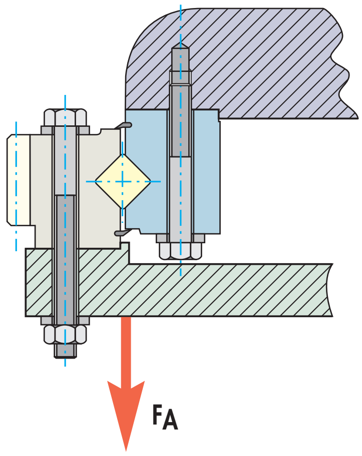
Standard calculation hypotheses

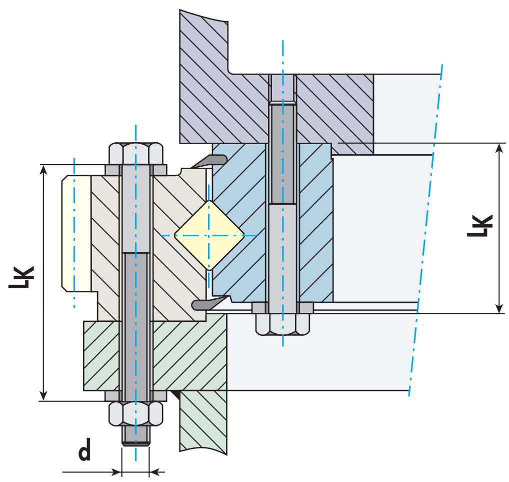
- Supported loads acting in compression.
- Equispaced bolts ; i.e equally positioned on the pitch circles.
- Steel rings and supports.
- Supports complying with our instructions : thickness, stiffness, surface eveness *(see chapter STRUCTURES page 42).*
- Rings bolted directly onto its supports.
- In cases of heavy radial loads, we recommend to use pilots or to glue so that bolts will not be subjected to shear stresses.
- **The clamping length must be at least equal to five times the diameter : $L_K \geq 5.d$.**

SUPPORTED LOAD



HANGING LOAD





SLEWING RING FUNCTIONS

FASTENING FUNCTIONS

Calculation of the number of bolts
When the ring has been previously selected according to its utilization and its load capacity, the bolting is then determined to correspond to the bearing capacity. The calculation of the minimum number of fasteners is carried out according to the following formula for the **most unfavourable** load case. In any case, a sufficient number of bolts ensuring an effective connection between ring and support frames must be kept, in order to avoid any ring deformation.

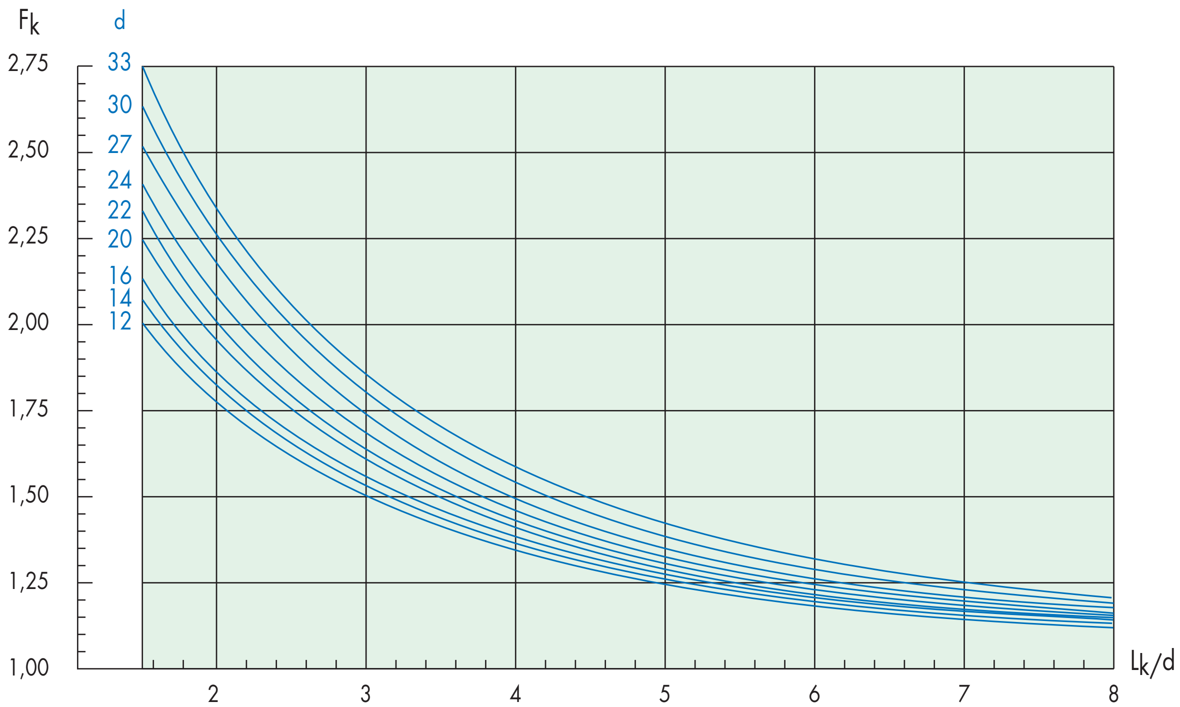
$$N = \frac{1{,}6 \,.\, F_K \,(\,4 \,.\, M_T - F_A \,.\, D_f\,)}{D_f\,(\,T_s - F_{pc}\,)}$$

where :
N = Number of bolts theoretically necessary.
1,6 = Tightening factor (assembly error factor) for torque wrench Grade B according to FD E 25-030.
F_k = Bolt stretch factor, see sketch.
M_T = Total tilting moment applied to the ring in kNm.
F_A = Axial load in kN.
D_f = Fastener pitch circle diameter in m.
T_s = Tightening tension.
d = Bolt diameter in mm.
Ø m = Raceway mean diameter in m.

F_{pc} = Loss of tension due to embedding in kN, see graph.
L_k = Clamping length in mm.

Bolt stretch factor F_k
This factor takes the assembly geometry into account. It is based on the bolt diameter and the ratio of clamping length to diameter.
The best fastening is obtained with through-holes in the ring and the supports : by using screws and nuts, the clamping length is long, bolt stiffness is satisfactory and tension losses are minimised.
In case of screw fastening into tapped holes, the setting depth must not be less than 1.25.d.

BOLT STRETCH FACTOR



SLEWING RING FUNCTIONS

FASTENING FUNCTIONS

Tightening tension : T_s
Tightening tension of fastening bolts must be sufficient to warrant the absence of looseness which is essential to ensure the resistance of the assembly fatigue.

Calculation of the minimum fastener preload :
It is useful to check that the standar-dized preload of the chosen bolt diameter is sufficient compared to the dynamic stresses imposed when operating.

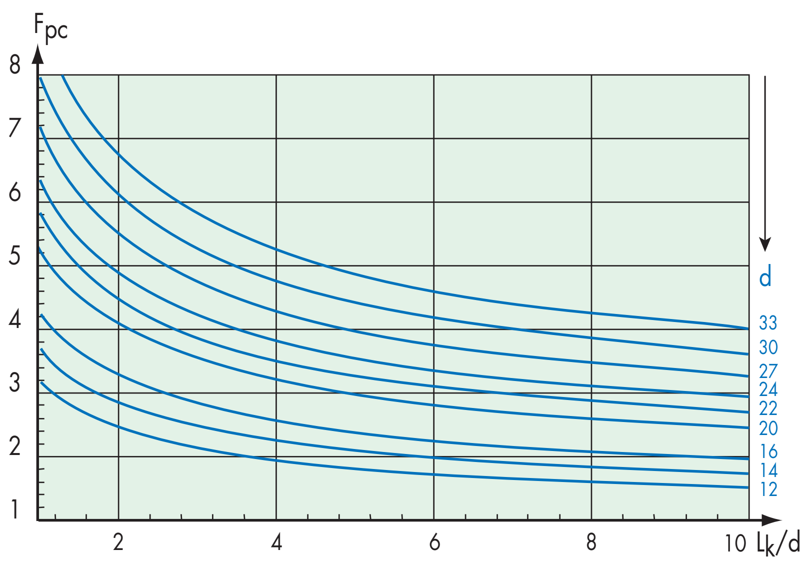
$$T_s > \left(\frac{2,25}{N}\right)\left[\left(\frac{4.M_T}{\varnothing m}\right)-F_A+80\ N.d.10^{-3}\right]$$

The standardized tightening tension at 80% of Re must be selected from the following table according to the chosen bolt diameter : bolting grade 10.9.

Diameter (mm)	12	14	16	20	22	24	27	30	33
Tension (kN)	56	77	106	166	208	239	315	385	480

Loss of tension :
During tightening and under external loads, peening of the surface roughness of the contact parts occurs, reducing the initial bolts elongation and thus producing a loss of tension, which decreases the preload in the assembly. This loss of tension has been tabulated on the following graph which shows values in function of diameter **d** and the ratio **L_k/d**.

LOSS OF TENSION



Calculation of under head contact pressure :
Usually, this calculation is not required when treated flat washers are used. It is however recommended when cap screws are used. (Chc).

We must have :

$$\frac{F_{B\,max}}{A_c} < P\ adm$$

with

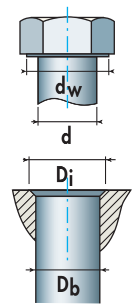
$$F_{B\,max} = T_s + 0,13.F_E$$

with

$$F_E = \left(\frac{1}{N}\right)\left[\left(\frac{4.M_T}{\varnothing m}\right) - F_A\right]$$

and

$$A_c = \left(\frac{\pi}{4}\right)\left(d_w^2 - D_i^2\right)$$



Allowable pressure :

for steels N and Z	400 MPa
for steels D and X	620 MPa
for steels type E36	270 MPa

IMPORTANT REMARK
The use of **elastic washers** whatever the type or model is **absolutely prohibited** and will void all warranty.

SLEWING RING FUNCTIONS

SLEWING FUNCTION

ROLLIX slewing rings generally incorporate a SLEWING mechanism to control rotation of the mobile part. This function can be achieved by various means :

1. gear drive (the most frequent case)
2. belt drive
3. chain drive
4. direct drive

GEAR DRIVE

Involute teeth, spur or helical are directly cut into the outer or inner ring (spur gear only).

GEOMETRY

Most of the ROLLIX bearings have a gear improved by **positive addendum modification** which notably decreases pressures as well as by **a truncation** avoiding teeth root interference at the pinion.

It is also essential to make a positive addendum modification on the pinion teeth, in order to avoid the geometric interference which appears under 18 teeth.

Furthermore, the drive stresses induce shaft bending and gear deformation which are harmful to good meshing.

To prevent these faults, we recommend profile corrections be carried out on the pinions : i.e.crowning and tip relief.

Our Engineering Department will assist upon request.

RESISTANCE

Our rating graphs indicate the values of allowable **maximum tangential force** in fatigue T.

The continuous operating capacity is obtained by the application of a suitable load moderating factor.

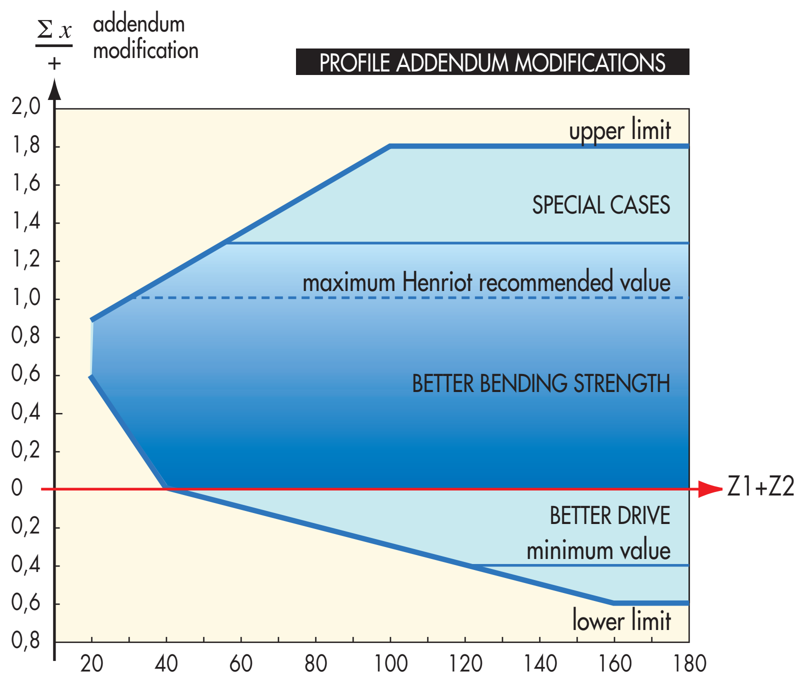
$$T = 2\ \frac{C_D}{D_{ref}}$$

C_d = Torque on gear

D_{ref} = Reference diameter

Unless otherwise stated, the indicated values are valid for geared rings made of normalized steel XC45 : code Z.

When these values are not sufficient, ROLLIX can proceed with contour hardening which considerably improves the resistance to tooth root bending and the resistance to contact pressure. For very heavy loading conditions, ROLLIX carries out complete hardening of the tooth and of its root in the wheel rim. When only a better wear resistance is required, surface hardening of tooth flanks only is possible.



SLEWING RING FUNCTIONS

SLEWING FUNCTION

GEAR QUALITY

Unless stated otherwise, ROLLIX manufactures slewing ring gears according to AFNOR or DIN standards which meet the following criteria :

WITHOUT SUPERFICIAL HEAT TREATMENT

DIN class	AFNOR class	Maximum diameter	Maximum module	Options
12	12	all sizes	25	Module 45 with special tooling
10-11-12	11-10-9	all sizes	20	
9-10	8-9	3100	22	Specific equipment necessary
7-8	7	2500	20	• made upon request

When a higher gear quality such as grade 5 or 6 is needed, gear grinding becomes necessary (ask our Engineering department).

SUPERFICIALLY HARDENED GEAR

• Generally, by contour hardening to 55 HRc (± 5).

• The gear classes stated above are offset and ROLLIX can meet AFNOR or DIN standards for grades 11-12.

IMPORTANT

ROLLIX considers that the relevant gear characterizing parameters for each quality class defined by AFNOR, DIN or ISO must be met. In case a customer does not require all of the parameters and needs to meet only one or two of them, ROLLIX can achieve higher qualities.

OTHER FEATURES

PRECISION - TOLERANCES

Standard ring tolerances are generally defined according to ISO 286-1 and 2 standards.
For applications requiring a higher precision level : robotics, radars, etc. a better grade is considered. The tolerance values are then indicated on the bearing drawing. For bearings of large diameter with thin cross sections where radial stiffness is low, the tolerance values must be considered when bearing is assembled to its supports, these ensuring proper circularity.

GEOMETRY
The selected criteria are :
- For the diameters : **Js 13**
- For the centerings

Bores : **H9**
Shafts : **f9**
- For the overall height : ± **1 mm**

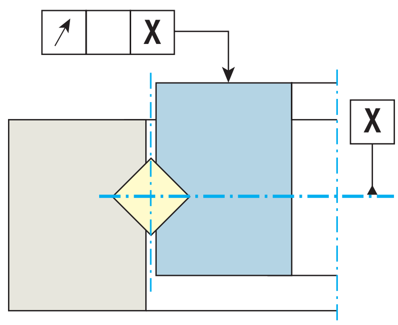
FASTENING
The bolt circle diameters are machined to tolerance **Js10** with a minimum of ± **0,2 mm**.

GENERAL TOLERANCES TABLE (ACCORDING TO ISO 286-2)

Diameter (in mm)	from 180 to 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000	2000 2500	2500 3150
Pilot H9 (in µm)	+115	+ 130	+140	+155	+175	+200	+230	+260	+310	+370	+440	+540
Spigot f9 (in µm)	- 50 - 165	- 56 - 185	- 62 - 202	- 68 - 223	- 76 - 251	- 80 - 280	- 86 - 316	- 98 - 358	- 110 - 420	- 120 - 490	- 130 - 570	- 145 - 685
Diameter Js10 (in µm)	± 92	± 105	± 115	± 125	± 140	± 160	± 180	± 210	± 250	± 300	± 350	± 430
Diameter Js13 (in mm)	± 0,36	± 0,405	± 0,445	± 0,485	± 0,55	± 0,625	± 0,70	± 0,825	± 0,975	± 1,15	± 1,4	± 1,65

GEAR
The value of the total run-out is indicated in the drawing title block. The measurement and tolerance over K teeth are also indicated on the drawing.
This dimension includes the contribution of the ring to the meshing backlash.

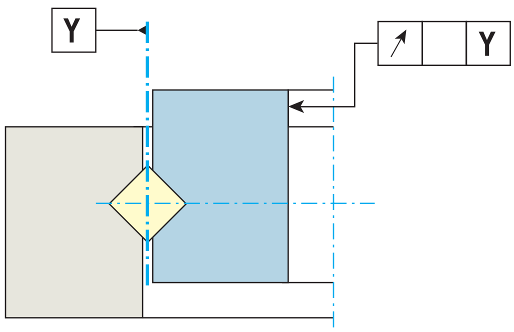
BEARING
- The AXIAL RUN-OUT of the supporting faces is measured using a dial gauge over one full revolution (see opposite sketch).



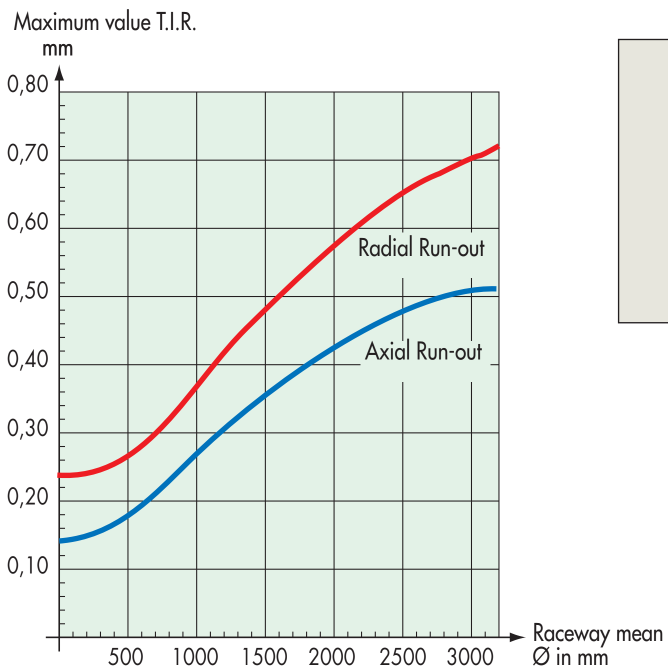
OTHER FEATURES

PRECISION - TOLERANCES

• THE RADIAL RUN-OUT of the spigots is also measured by rotation (following sketch).

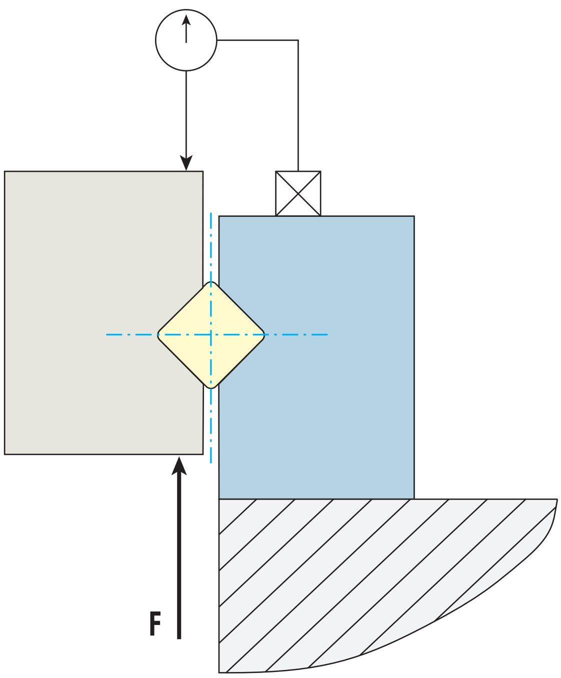


These measurements are carried out by placing the magnetic base of a dial gauge on the fixed ring, while the filler is taken in contact with the element to be measured. The reading will be obtained during one rotation of the turning ring (value T.I.R).



• THE DEFLECTION UNDER MASTER LOAD is measured as follows :
One of the bearing ring is fastened to a rigid support and a master load **F** is applied axially on the other ring to a point closely located to the raceway.
The deflection under load F is measured using a dial gauge whose base is attached to the other ring.
The deflection under standard load of every slewing ring is measured at the factory.
This value is compared to the allowable limits, and is registered.

DEFLECTION UNDER MASTER LOAD



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OTHER FEATURES

SLEWING TORQUE

Calculation of the necessary torque to enable a rotation of the whole assembly takes into account :
- loads on the machine,
- rotating masses,
- distance of these masses to the rotation axis,
- speeds and accelerations,
- resisting torques.

Two types of torques are distinguished :

Start up slewing torque : **Cd = Crv + Crc**
Acceleration slewing torque : **Cg = Crv + Crc + Ca**
Crv = Friction torque of unloaded bearing
Crc = Rotating torque due to loads
Ca = Acceleration torque
Cd = Starting torque

All these torques are expressed in kNm.

Crc : ROTATION TORQUE DUE TO LOADS

The starting torque required takes into consideration loads on the bearing and friction of the components.

Balls type slewing ring

$$Crc = \left[\frac{13{,}11\ M_T}{\varnothing m} + 3\ F_A + 11{,}34\ F_R \right] \varnothing m \ .\ 10^{-3}$$

Crossed rollers type slewing ring :

$$Crc = \left[\frac{15{,}3\ M_T}{\varnothing m} + 3{,}75\ F_A + 8{,}19\ F_R \right] \varnothing m \ .\ 10^{-3}$$

M_T = Resulting moment in kNm
Ø m = Raceway mean Ø in meters
F_A = Axial load in kN
F_R = Radial load in kN

Ca : ACCELERATION TORQUE

The torque needed to accelerate the loads from the initial speed up to the final speed, during time (t) is defined by :

$$Ca = \frac{\pi\ .\ n\ .\ I}{30\ .\ t}\ .\ 10^{-3}$$

t = Acceleration time in sec.

n = Speed variation in RPM
(Final speed - Initial speed)

I = Moment of inertia of the machine in Kg . m^2

$I = I_1 + I_2 + I_3 + \ldots.. I_n$

where I_1 à I_n = moments of inertia of the moving loads with regard to the rotation axis expressed in Kg . m^2.

Generally we have :

$I_1 = G_1 \times r_1^{\,2}$

$I_n = G_n \times r_n^{\,2}$

G_1 to G_n = Mass of various rotating components expressed in Kg.

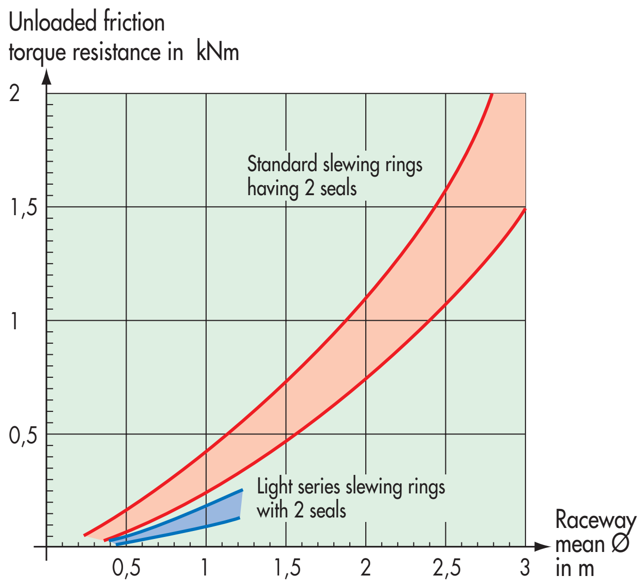
r_1 to r_n = Distances between the loads centre of gravity and the ring rotation axis expressed in meters.

Note : *The resisting torque depends on the support surface flatness and lubrication.*

OTHER FEATURES

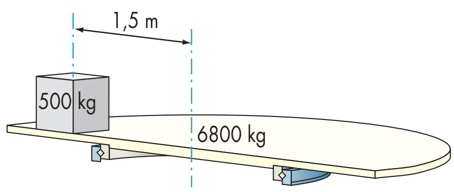
SLEWING TORQUE

The friction torque of standard slewing rings is defined in the following graph. ROLLIX, upon request, can supply slewing rings with lower or higher torque values.



APPLICATION EXAMPLE

Platform diameter : 4 m.
Platform mass : 6800 kg
Cube mass : 500 kg
Ball type slewing ring
raceway mean Ø : 2 m.
Distance from the cube to the rotation axis : 1,5 m.
Initial speed : 2 RPM
Final speed : 6 RPM
Acceleration time : 20 sec.



LOADS APPLIED ON THE RING

Axial F_A : 68 kN + 5 kN = 73 kN
Radial F_R : 0,29 kN, negligible
Moment M_T : 5 kN x 1,5 m = 7,5 kNm

SLEWING TORQUE : Raceway mean Ø = 2 meters
Crv : according to the graph : 1 kNm

$$Crc = \left[\frac{13,11 \times 7,5}{2} + (73 \times 3) + (11,34 \times 0)\right] 2.10^{-3}$$

Crc = 0,536 kNm

Slewing torque at start up
Cd = 1 + 0,536 = 1,536 kNm

Platform moment of inertia :

$$\frac{MR^2}{2} = \frac{6800 \times 2^2}{2} = 13600 \text{ Kg m}^2$$

Cube moment of inertia :
$Mr^2 = 500 \times 1,5^2 = 1125 \text{ Kg m}^2$

Total moment of inertia :
$13600 + 1125 = 14725 \text{ Kg m}^2$

Acceleration torque :
n = 6 - 2 = 4 RPM

Acceleration time : 20 sec

$$Ca = \frac{14725 \times \pi \times 4}{30 \times 20} 10^{-3} = 0,3084 \text{ kNm}$$

Slewing torque during acceleration
Cg = 1 + 0,536 + 0,3084 = 1,845 kNm

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OTHER FEATURES

PROTECTION

SEALS

ROLLIX slewing rings are generally equipped with protecting seals on both sides of the raceway.
These seals have the function of :
• protecting raceways against small-sized contaminating agents,
• retaining the lubricant in the raceway.
For specific requirements, ROLLIX designs suitable protective devices, such as :
• standard and specific seals,
• lip seals
• combination of two or more seals.

SHROUDS

For severe application conditions and in order to limit the effects of aggressive agents such as :
• swarf and metal chips
• welding grains
• abrasives
• mud
• sand
• water and heavy sea splashes
• cutting fluids...

ROLLIX strongly recommends to install effective protective shields.
Furthermore, during cleaning with solvents or pressurised water, it is advisable to avoid working in the protective device area.

SURVEY

If, during regreasing, large grease discharges are noticed, it is advisable to check :
• that seal is still correctly positioned,
• that seal is not damaged (cut, torn, worn),
• that seals are still capable for proper bearing operation.
Therefore, this seal can either be reinstalled or replaced.

SEAL RANGE TABLE

Operating conditions	Ranges
"Normal" - 30° C à + 70° C	NITRILE-BASED elastomer
"Extreme" $\theta < -30°C$; $70°C < \theta < 200°C$	FLUOR-BASED elastomer
"Special" : Various physical or chemical aggressive agents.	NITRILE-BASED elastomer modified or others

PROTECTIVE SURFACE COATING

For particular applications, ROLLIX suggests a protective surface treatment, such as :

1. Zinc plating (+ chromate coating)
2. Phosphate coating
3. Chemical nickel plating
4. Paint

5 to 9 Miscellaneous treatments, for example, chrome plating, Schoop's metal spraying process, anodic oxidization, etc.

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